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Compensation of Wave-Induced Motion for Marine Crane Operations

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Problem Description

The candidate will consider the problem of compensation of wave-induced motion for marine crane operations. In particular, excitation in the heave, roll and pitch modes are relevant, both for inter-vessel and water-entry operations. The following elements must be considered:

1. Review existing work on compensation of wave-induced motion for marine crane operations.
2. Suggest how existing heave-based designs can be modified to also take roll and pitch motions into account.
3. Simulate relevant motion compensation scenarios in MATLAB/Simulink.

Assignment given: 07. January 2008
Supervisor: Morten Breivik, ITK

Preface

The primary aim of this thesis is to look at some aspects of wave-induced motion compensation for ship-mounted cranes, including operational concerns, vessel behaviour, crane designs and control schemes, as well as simulations. I was so fortunate as to get a summer job at VTT Maritime in Bergen in 2007, and as a result I was suggested to work on this topic. A main motivation has been to evaluate the effect of existing motion compensation systems, which are mostly moon pool based, and also transition them to cranes working over the side of the vessel, which means that the roll and pitch motions also have to be considered in addition to the heave motion.

There has been many people helping me in writing this thesis. First, I would like to thank my advisor Morten Breivik for his excellence in guiding me through this thesis, his contribution has been invaluable. Secondly, I want to mention the company which gave me the opportunity to write this report, VTT Maritime, and my contact person there, Torgeir Evjen, for his contribution on the practical and operational concerns that need to be taken into account when considering offshore motion-compensated crane systems. I would also like to thank my wife and daughter for keeping my spirits up when I was struggling to get things right.

Summary

Most of the systems considering wave synchronization in the literature are concerned with moon pool operations. The objective of this thesis is to transition the existing knowledge from some of that work into marine cranes working over the side of the vessel. This aim is mainly motivated by the fact that cranes working over the side of the vessel are more versatile. They can be used for more comprehensive and differently shaped objects. In addition, vessels with cranes mounted this way are more flexible since the moon pool configuration consumes a considerable amount of deck space.

The main contribution of this thesis is a consideration of the transition from cranes lifting through a moon pool to cranes lifting over the side of the vessel. This includes consideration of the roll and pitch motion of the vessel, as well as simulating the complete system with wave synchronization control. The system is also simulated without any motion compensation for comparison. The degrees of freedom (DOFs) available for motion compensation control in different crane configurations are discussed, and some comments are made on how the actuators may cooperate in order to achieve better motion compensation.

It is found that nothing obstructs the use of control algorithms originally developed for moon pool cranes for cranes working over the side of a vessel. The behaviour of the systems is roughly equal in the two cases, and the physical aspects are the same regarding wave elevation, still water level, and wire behaviour. There is however, one aspect to consider when designing overboard cranes, and that is the response time needed for following the wave elevation with the payload. When the payload is lifted over the side of the vessel, the roll and pitch motions affect the movement of the crane tip. In the worst case scenario, a wave can have a velocity directed upward, while the crane tip has a velocity directed downward. If this is the case, a faster motion compensation system is needed than for the moon pool applications.

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1 Introduction

1.1 Motivation

Motion compensation for offshore cranes is used to reduce the motion of a payload induced by vessel motion. When a moon pool is used in the lifting operation, it is only necessary to consider heave motion, as the roll and pitch motion is very small compared to that of heave. However, when cranes are operating at the side of the vessel, it might be necessary to additionally consider roll and pitch to get accurate behaviour, depending on sea-state and how far off the side of the vessel the payload is lowered. Good compensation of vessel motion in offshore environments become more and more sought for, as there are great economical, environmental and safety benefits. While surge and sway motion is well handled by dynamic positioning (DP) systems [1], heave motion still remains an issue. It is virtually impossible for DP systems to compensate heave, roll and pitch motion, and such systems only address surge, sway and yaw motion. This means that other compensation methods are needed, such as slackening and tightening of the wire holding the payload by use of the winch. With motion compensation systems installed, vessels may operate in higher sea states and operate significantly longer than without such systems. The risk of damage to payloads and subsea installations are reduced as well as the risk of polluting emissions. Risk of personnel injuries is also reduced. All these factors make it a worthwhile effort to design better motion compensation systems.

1.2 Contribution

The contribution of this report is to look at what solutions are available inside the motion compensation area, and how these systems may work together in order to perform better. In this thesis, a transition of control algorithms from cranes lifting through a moon pool to cranes lifting over the vessel side is conducted. This includes adding the roll and pitch motion of the vessel to find the crane tip movement, as well as simulating the complete system with and without motion compensation. How many of degrees of freedom (DOFs) there are in different crane configurations are also discussed, and some comments on how the actuators may work together in order to achieve better motion compensation are made.

1.3 Outline

Chapter 2 descibes some typical crane designs used in marine lifting applications, and how these cranes may be used to achieve motion compensation. Both vertical and pendulum motion are discussed. In Chapter 3 some operational concerns are given, before moving on to the wave and vessel model used in this thesis. The kinematics of the vessel with a crane is given in Section 3.4, and the behaviour of the wire is given in Section 3.5. The latter includes both vertical deformation and pendulum motion of the payload. In Section 3.6, both static and hydrodynamic forces acting on the payload are described. The chapter is rounded off with Section 3.7 where it is described how the metacentric height affects the wave-induced vessel motion. Chapter 4 includes the motion compensation control. Feed-forward control is used in both heave compensation and wave synchronization. Chapter 5 contains simulations of the motion compensation systems. Finally, the report is rounded off by a discussion of simulation results in Section 5.7, followed by conclusions in Chapter 6.

1.4 Notation and Abbreviations

The notation used when describing the vessel and crane tip motion is mainly adopted from the Society of Naval Architects and Marine Engineers (SNAME) [2] from 1950, where the motion variables is defined as shown by Figure 1.1 and in Table 1.1. This notation is expressed conveniently by vectorial terms [1] as shown in Table 1.2. The coordinates used when describing the dynamics of the total system, is defined in Figure 1.2, with the corresponding variables given in Table 1.3. The abbreviations used in this report are given in Table 1.4.

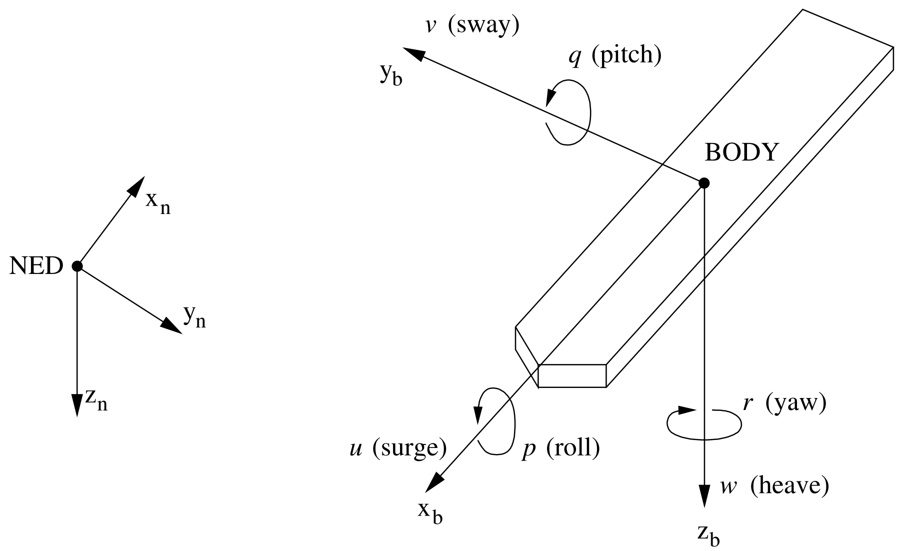

Figure 1.1: Illustration of SNAME notation for describing the motion of a marine vessel. The illustration is inspired by [1].

DOF		Linear and angular speeds	Position and Euler angles
1	Motion in the x-direction (surge)	u	x
2	Motion in the y-direction (sway)	v	y
3	Motion in the z-direction (heave)	w	z
4	Rotation about the x-direction (roll)	p	ϕ
5	Rotation about the y-direction (pitch)	q	θ
6	Rotation about the z-direction (yaw)	r	ψ

Table 1.1: Overview of notation used in Figure 1.1, inspired by [1].

NED position	$\mathbf{p}^n = \begin{bmatrix} x \\ y \\ z \end{bmatrix}$	Attitude (Euler angles)	$\boldsymbol{\Theta} = \begin{bmatrix} \phi \\ \theta \\ \psi \end{bmatrix}$
BODY-fixed linear velocity	$\mathbf{v}_b^o = \begin{bmatrix} u \\ v \\ w \end{bmatrix}$	BODY-fixed angular velocity	$\boldsymbol{\omega}_{nb}^b = \begin{bmatrix} p \\ q \\ r \end{bmatrix}$

Table 1.2: SNAME notation in vectorial form, inspired by [1].

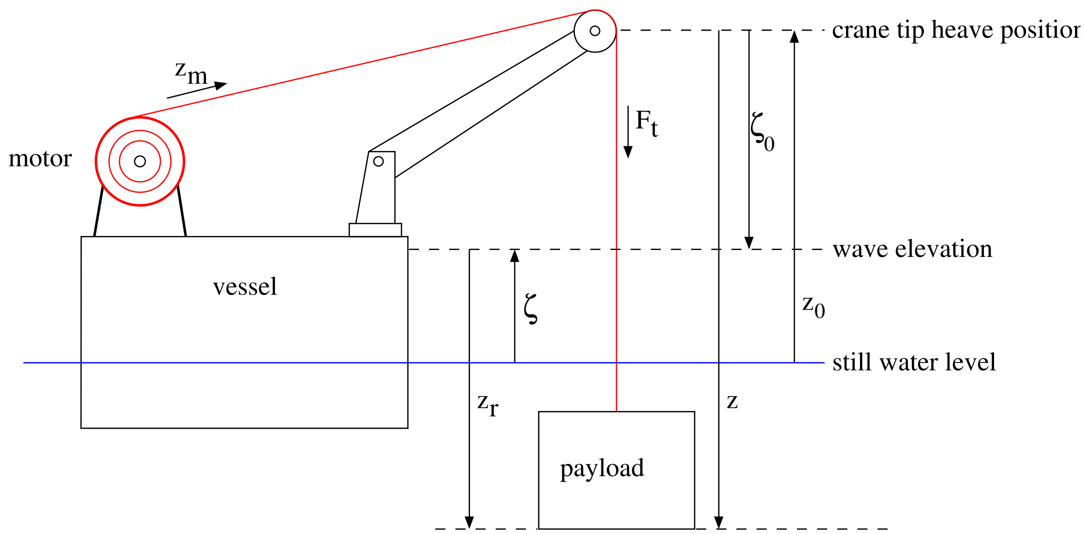

Figure 1.2: Definition of the coordinate system used when deriving the dynamic model. The reference is still water level. The illustration is inspired by [3].

z_0	crane tip position in heave
z	payload position
z_m	motor position
z_r	payload position relative to wave eleavation
ζ	wave elevation below/above payload, relative to still water level
ζ_0	wave elevation below/above payload, relative to crane tip position in heave
F_t	wire tension

Table 1.3: Notation used when describing the dynamics of the vessel, crane and wire, as shown in Figure 1.2.

DP	:	Dynamic Positioning
NED	:	North East Down
DOF	:	Degree Of freedom
SNAME	:	The Society of Naval Architects and Marine Engineers
GM	:	Metacentric height
RAO	:	Response Amplitude Operator
MARINTEK	:	The Norwegian Marine Technology Research Institute
MSS	:	Marine Systems Simulator

Table 1.4: Abbreviations used in this report.

2

Crane Designs

When choosing what crane design and motion compensation method to use, there are many aspects to consider; how the vessel is designed, what kind of applications the system will be used for, how the weather conditions are, whether there is sufficient deck space available, and so on. To get an idea of the different systems used, an overview of some common crane designs and motion compensation methods are given in the following chapter. The degrees of freedom (DOFs) in the designs are also considered, as this gives an indication of additional control possibilities. All illustrations are inspired by [4].

Knuckle-Boom Crane

As the name suggests, this is a crane constructed much like a human finger, see Figure 2.1. It has a rotating base and two or more "knuckles", controlled by hydraulic cylinders. It is also possible to have a telescopic link at the end, making the range longer. One of the greatest advantages with this crane is its flexibility for what tasks it may be used for. It may turn in all directions, and if it is built and placed properly it may reach an entire deck. When not in use, it take up little deck space. However, if heavy lifting is required at a long distance, this crane may be to weak. The crane in Figure 2.1 has 3 DOFs; one for the rotating base, one for the first knuckle, and one for the second knuckle. In the general case, 1 DOF will be added for each extra knuckle, and if the crane also include a telescopic boom, an extra DOF is added.

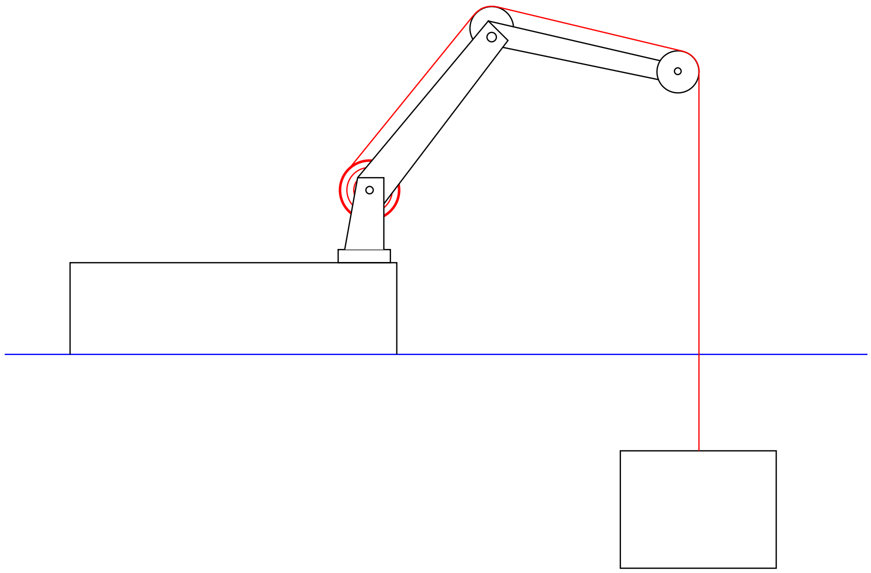

Figure 2.1: Knuckle-boom crane. This crane consists of two or more links, controlled by hydraulic cylinders.

Nodding-Boom Crane

This crane consist of a single boom fastened to a base. The boom is pivoting on top of the base, making it "nod", see Figure 2.2. This design does not use very much deck space, as the winch may be placed inside the crane base. This design is a 2 DOF system; one DOF for the rotating base, and one for the pitching boom.

A-frame

Here, the crane consists of bars forming an A, see Figure 2.3. This frame may be raised and lowered using hydraulic cylinders. The frame is usually fitted as close to the gunwale as possible. It may be placed on either side of the vessel, or at the stern. This makes a very solid crane, and it is capable of doing heavy lifts. However, the deck space required is large. This design only has 1 DOF, which consist of the frame moving in an arch over its base. This leaves very little space for optional control helping the winch in the motion compensation.

Sub-A-frame

This is a modified version of the A-frame, where another A-frame is fitted on the main frame. This allows the sub-frame to be an active part in the motion compensation, see Figure 2.4. The deck space consumed is the same as for the A-frame. The DOFs in this design are one more than for the A-frame, resulting in a 2 DOFs system. Here, the sub frame is connected

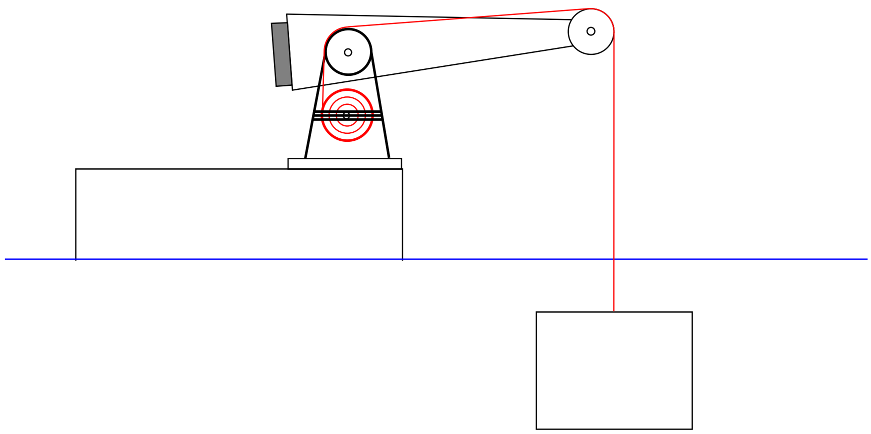

Figure 2.2: Nodding boom crane. This crane pitches its base, making the boom "nod".

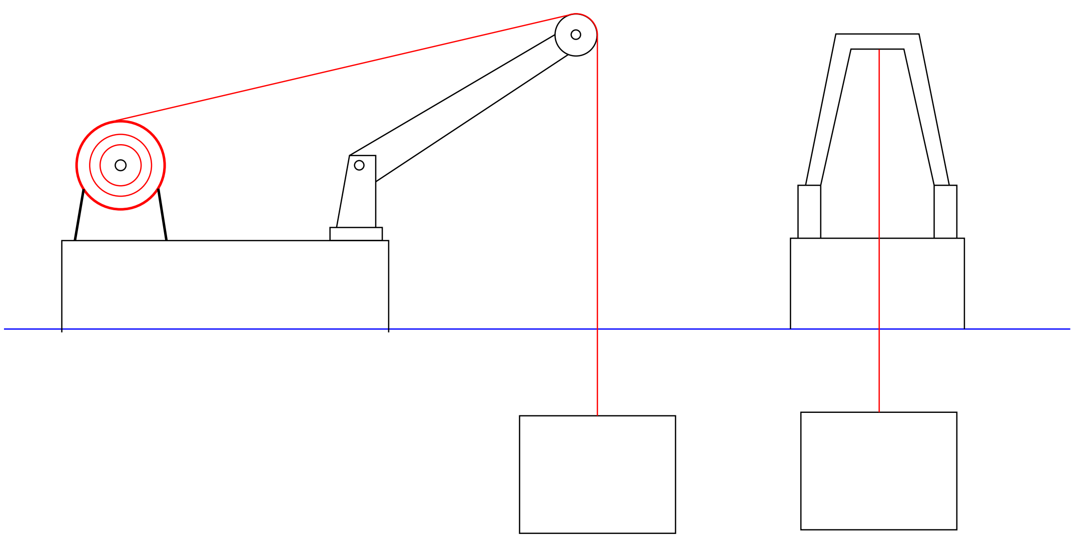

Figure 2.3: A-frame crane. This crane consists of a frame formed as an A, making a very solid construction. The illustration shows a crane mounted at the stern.

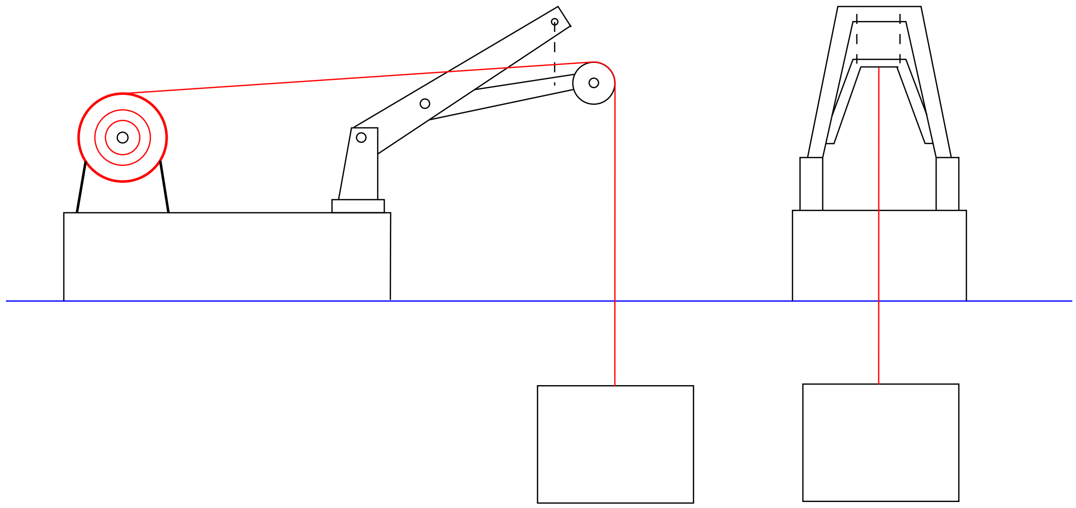

Figure 2.4: Sub-A-frame crane. This crane consists of an A-frame inside an A-frame. The illustration shows a crane mounted at the stern.

to the main frame by a winch and wire, adding one DOF.

2.1 Compensation of Heave Motion

The method used to compensate the vessel motion should be based on the crane design chosen. The most intuitive method for compensating the vessel motion is to use the winch. Since all crane systems already has a winch lifting and lowering the payload, it is easy to enhance it to compensate for the vessel motion. If the crane is a nodding boom, the boom may be driven up and down to compensate for the motion. In the case of the sub-A-frame, it is possible to place a winch on the main frame with wires connecting it to the sub-frame. This "sub"-winch may then be used to lift the sub-frame up and down, compensating the vessel motion, and hence the primary winch may be used only for lowering and lifting the payload, as it does best. It should also be noted that a combination of actuators could be used for compensation.

Knuckle-Boom Crane

In this crane design the winch is the main compensating actuator, giving out wire as the crane tip rises, and taking in wire as the crane tip lowers. This causes an extra stress on the winch compared to normal operation, and more frequent service is needed. Since this system has at least three DOF, it is also possible to include the actuators controlling the crane in the motion compensation. In doing so, it should be possible to reduce the wear and tear on the winch and at the same time accomplish better performance.

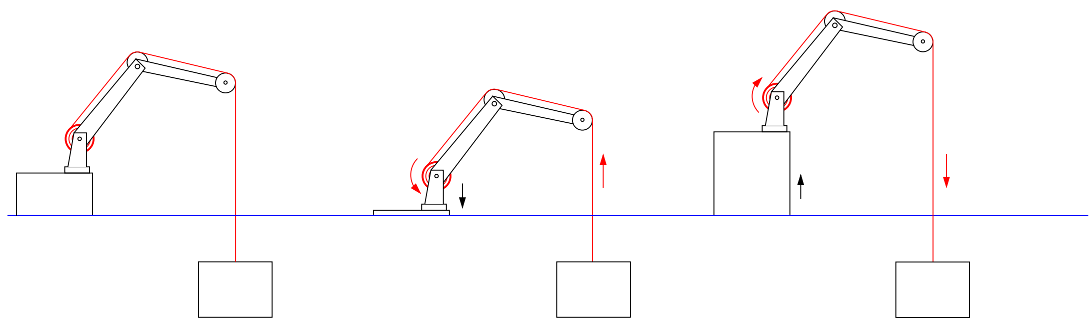

Figure 2.5: Knuckle-boom crane. The winch is shortening the wire when the vessel lowers, and making the wire longer when the vessel rises, keeping the payload stationary relative to the seabed.

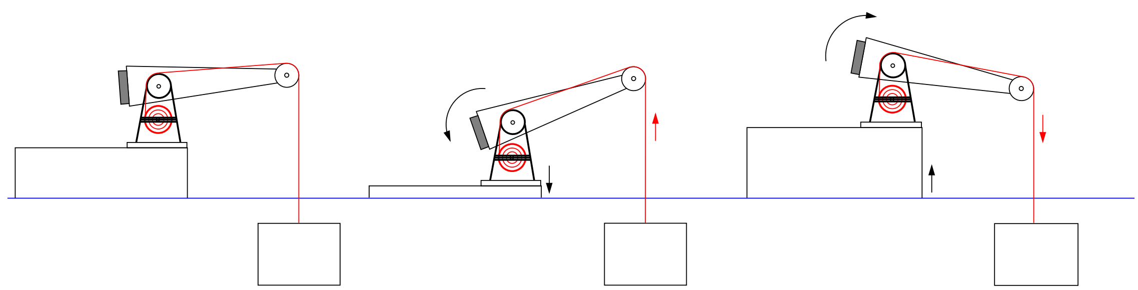

Figure 2.6: Nodding boom crane. The boom rotates counter-clockwise when the vessel is lowered, and clockwise when rising, keeping the payload stationary relative to the seabed.

Nodding-Boom Crane

This design is very kind to the wire, as there is no winch operation. The boom in itself is doing the compensation, by nodding up and down, following the vessel motion. As with the knuckle-boom crane, it might be desirable to use the winch as an extra actuator, increasing the operating range of the system.

A-frame

In this design, as with the knuckle-boom crane, the winch is compensating the vessel motion. Here it is difficult to use extra actuators in the motion compensation, as there is only one link from crane base to crane tip. Still, this is the most commonly used crane design because of its sturdiness and ability to do heavy lifts.

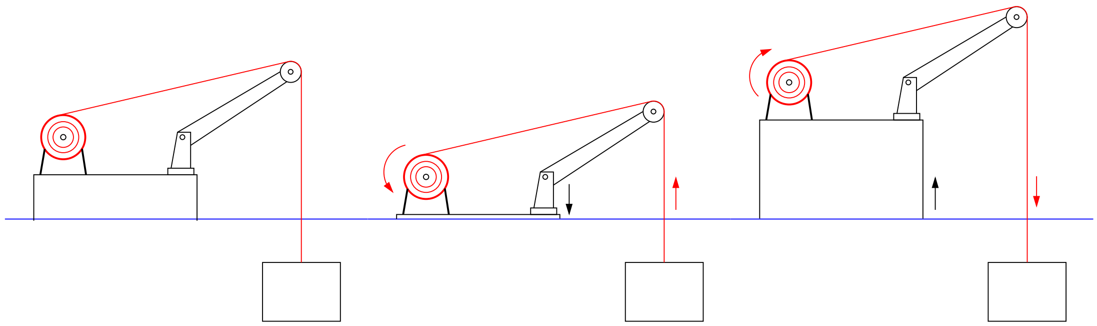

Figure 2.7: A-frame crane. The winch is shortening the wire when the vessel lowers, and making the wire longer when the vessel rises, keeping the payload stationary relative to the seabed.

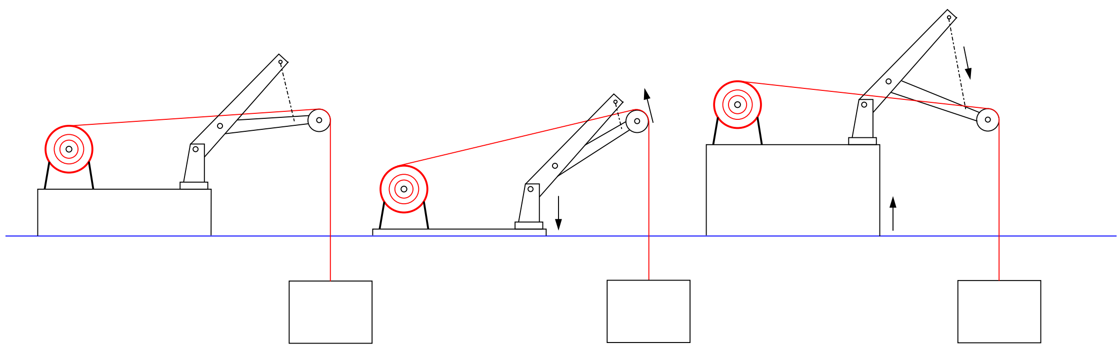

Figure 2.8: Sub-A-frame crane. The sub-frame is raised and lowered by a winch mounted on the main-frame, keeping the payload stationary relative to the seabed.

Sub-A-frame

In this design, a winch placed on the main frame raises and lowers the sub-frame, compensating the vessel motion. As with the knuckle- and nodding-boom cranes, this design lends itself well to multiple actuators performing the compensation, since the winch may contribute. This may be the most promising construction, as the best is gained from two worlds.

For the A- and sub-A-frame, where the winch is placed outside the base of the crane, there is one more way to achieve motion compensation, as described by [5]. A cylinder is placed between the winch and the crane, as illustrated in Figure 2.9. This cylinder has a tackle on each end, with the wire running over. When the cylinder is expanded, the payload raises, and when it is drawn in, the payload is lowered. If, in addition, the winch is used in the compensation and the crane is of the sub-A-frame design, there may be 3 actuators

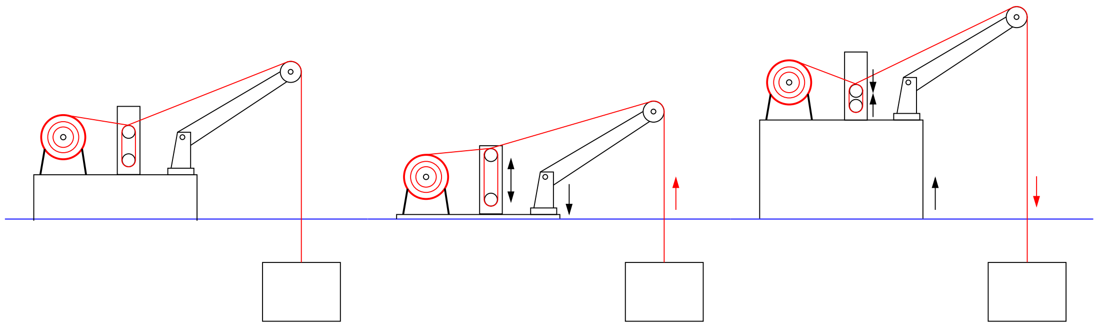

Figure 2.9: A-frame crane with compensating cylinder installed. The cylinder gives and takes wire as the vessel is affected by waves, compensating the motion.

playing a role in the compensation. This of course increases the complexity of the system, but very good results should be expected if done successfully.

By using the kind of control suggested above it is possible to compensate heave and pitch motions for a longitudinally mounted crane, and heave and roll motion for a laterally mounted crane. This is because the pitch motion of a vessel with a longitudinally mounted crane is mainly moving the crane tip up and down, while the same is the case for roll motion in a laterally mounted crane.

2.2 Compensation of Roll and Pitch Motion

Another aspect of the motion compensation is that the wire and payload system will behave as a pendulum when the vessel is rolling or pitching. This is only a problem while the payload is in the air, as the damping becomes significantly larger when the payload is submerged. Both longitudinally and laterally mounted cranes suffer from this effect. If the crane is mounted longitudinally, see Figure 2.10, the pendulum motion will be most affected by the roll motion of the vessel, which will cause the payload to swing along the lateral axis. The pitch motion of the vessel will also have an effect, but it is much smaller than the effect of roll motion, and will mostly affect the heave position of the payload. If the crane is mounted laterally, see Figure 2.11, the pendulum motion will be most affected by the pitch motion of the vessel, which will cause a pendulum motion along the longitudinal axis. The roll motion will in this case have the same effect as pitch motion does for longitudinally mounted cranes.

The surge, sway, and even yaw motion of the vessel is also affecting the pendulum motion of the payload, making this a very complex system. Even though the vessel is positioned with a DP system, the vessel will always have some motion in all 6 DOFs. If the vessel is moving in surge, the payload will swing in the longitudinal direction. If the vessel is moving in

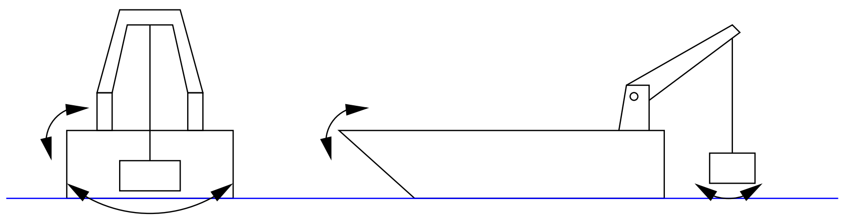

Figure 2.10: Illustration of how the payload will be affected by roll and pitch motion for a longitudinally mounted crane.

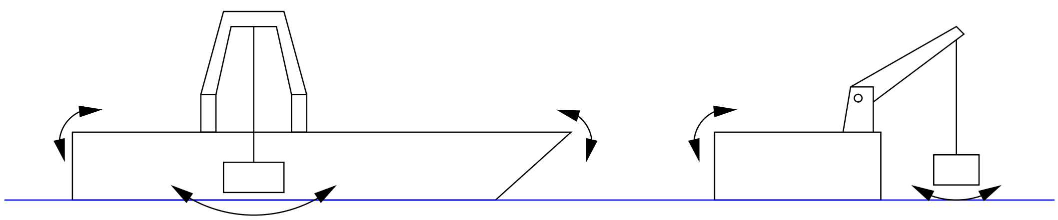

Figure 2.11: Illustration of how the payload will be affected by roll and pitch motion for a laterally mounted crane.

sway, the payload will swing in the lateral direction. In the following sections, some ideas on how the "extra" DOF in the crane designs may be used to reduce the pendulum motion of the payload are presented. The descriptions are given for a laterally mounted crane, but is applicable for a longitudinally mounted crane as well. Position or rate-of-change measurement of the angle between the hanging wire and a thought vertical line, can be a good way to receive feedback on how large the pendulum motion is. These measurements can also be used as control variables.

Knuckle-Boom Crane

As mentioned earlier, this crane has 3 or more DOFs. One way to reduce the longitudinal pendulum motion of the payload is to use actuation of the base actively. If this is done, the tip of the crane may swing back and forth, stealing energy from the system. It is also possible that this system may be passive. By use of an elastic element in the base, the base will be pivoting around a set reference position as the payload is pulling the crane back and forth. This should steal energy from the system, and damp the pendulum motion. The lateral motion of the payload caused by roll and sway may be reduced by applying a telescopic boom.

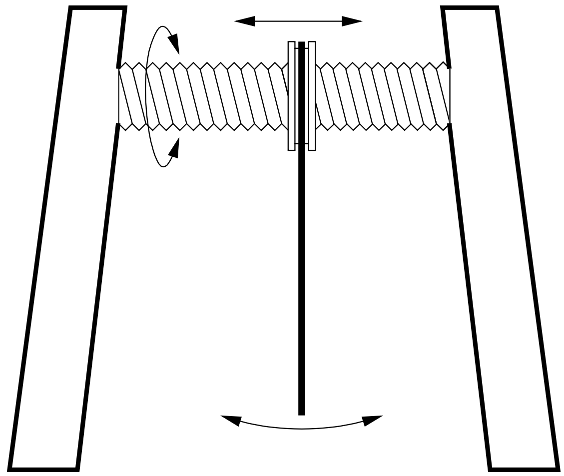

Figure 2.12: A-frame with longitudinal pendulum motion compensation. As the rod is turned, the running wheel travels along it, stealing energy from the pendulum motion.

Nodding-Boom Crane

This crane has 2 DOFs. One at the base and one at the pitching boom. As with the knuckle boom crane, the swinging base can compensate the longitudinal motion. The lateral motion on the other hand, is not controllable for this crane. This means that this design is under actuated. One solution could be to mount two rails under the base, making it possible to drive the whole crane in the lateral direction. This could lead to some additional problems, as moving an entire crane on the deck of the vessel could prove difficult. One would also have to constraint the movement on the railing system to prevent collisions between the payload and the side of the vessel.

A-frame/Sub-A-frame

The A-frame only has 1 DOF, while the sub-A-frame has 2 DOFs. None of these DOFs may be used to compensate longitudinal or lateral motion, as heave motion is what they are designed to compensate. This means that it is difficult to compensate the pendulum motion of the payload by using one of these crane designs. One way to achieve longitudinal compensation is to use a variant of the cranes. This would include mounting a device on the A-frame/sub-A-frame that moves the running wheel in the longitudinal direction. This could be done either by a threaded rod, guiding the running wheel mounted on the A-frame/sub-A-frame, see Figure 2.12, or by wires pulling the running wheel back and forth. It should also be possible to make this system passive by using a smooth rod, and attaching a spring-damper system to the running wheel, see Figure 2.13.

In addition to the methods described here, another way of compensating lateral motion

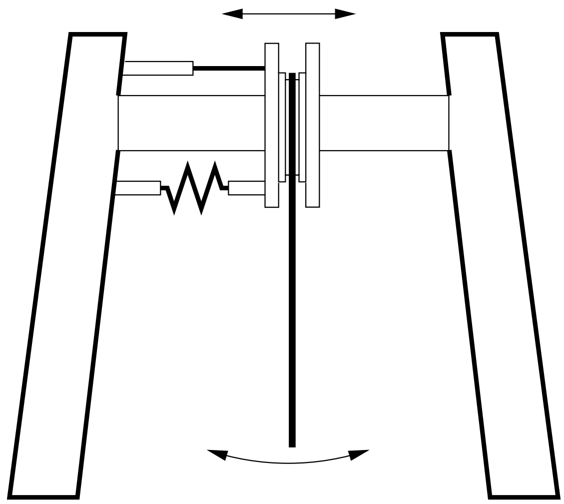

Figure 2.13: A-frame with passive longitudinal pendulum motion compensation. As the payload is dragging the wire to one side, the running wheel is moved a little, and in the process stealing energy from the pendulum motion.

due to roll is presented in [6]. Here, the pendulum motion consists of luffing the crane. This means that the crane tip is rised and lowered, creating movement in the vertical and lateral direction.

3

Theoretical Background

3.1 Operational Concerns

To minimize the forces that waves exert on the payload during transition into the sea, it is possible to position the vessel in such a way that the wave-induced motion of the vessel is damped. This is done by calculating the response of the vessel when affected by waves from different directions. A typical way of doing this is by response amplitude operator (RAO) analysis. The RAOs may be obtained by numerical analysis of the hull or by physical testing of the response of the vessel. When considering heave compensated cranes, a good rule would be to minimize roll and pitch motion. This is a crude motion control, and getting completely rid of all the roll and pitch motions is very difficult, especially as the waves get higher. However, it is still a good idea to consider the roll and pitch motion when designing motion compensation systems.

Heavy objects on the deck will lift the centre of gravity of the vessel, reducing GM. As mentioned in Section 3.7, this might easily lead to the vessel acquiring a constant roll angle. A consequence of this is that when lifting heavy objects over the side of the vessel, it might be necessary to introduce ballast control [1] in order to keep the vessel upright.

3.2 Wave Model

The waves used is generated by a block called "Waves" supplied by the MATLAB toolbox Marine Systems Simulator (MSS) [7]. To find the waves affecting the vessel, the wave elevation for any x and y position as a function of time is computed as

$$\zeta(x, y, t) = \sum_{i=1}^{n} \zeta_a(i) cos[\omega(i)t + \varphi(i) - k(i)(xcos(\psi(i)) + ysin(\psi(i))], \tag{3.1}$$

where

ζ_a : Vector of harmonic wave amplitudes [m]
ω : Vector of harmonic wave frequencies [rad/s]
φ : Vector of harmonic wave phases (random) [rad]
k : Vector of harmonic wave numbers [1/m]
ψ : Vector of harmonic wave directions [rad].

This wave model is using superposition of different sine frequencies and amplitudes to generate a wave pattern.

3.3 Vessel Model

The vessel motion is computed by the use of motion RAOs. A RAO is, according to [8], a transfer function which takes wave elevation, and gives the response of a vessel to this elevation. In the MSS toolbox there is a "Motion RAO"-block. This block takes the output from the "Waves"-block as input, and calculate the resulting vessel motion based on the vessel motion RAOs. The RAOs are generated from the vessel CAD/CAM drawings [1]. A provider of software to do these calculations is the Norwegian Marine Technology Research Institute (MARINTEK), which has developed a program called Vessel Response program (VERES) [9].

3.4 Kinematics

An inertial frame is used to describe the position, velocity and acceleration of the vessel. It is assumed that the vessel is kept positioned by a DP system with heading according to the Nort-East-Down (NED) frame, which is assumed inertial for flat earth navigation [1]. The vessel motion is described by NED position and Euler angles, as well as body-fixed linear and angular velocities and accelerations. Rotation matrices are used to transform the motion vectors from the BODY frame to the NED frame. It is also necessary to know the motion of the crane tip. The crane base is fixed in the BODY frame, with a position vector describing its position. The position of the crane tip on the other hand, is not fixed in the body frame. It is described by a position vector from the crane base to the crane tip, and may be time varying. The notation used is adopted from SNAME [2] from 1950.

3.4.1 Vessel/Crane Kinematics

It is assumed that the position of the vessel in the NED frame, and velocity and acceleration in the BODY frame are known. It is convenient to introduce some new notation to easily describe the relations between the different frames. This notation is adapted from [1]:

$$\mathbf{v}_o^n \;:=\; \text{linear velocity of point O decomposed in frame } n$$
$$\boldsymbol{\omega}_{nb}^n \;:=\; \text{angular velocity of frame } b \text{ with respect to frame } n \text{ decomposed in frame } n$$

It is also practical to introduce SNAME-notation in vector form, with NED position and attitude denoted as

$$\mathbf{p}^n = [x, y, z]^T \quad \boldsymbol{\Theta} = [\phi, \theta, \psi]^T, \tag{3.2}$$

and BODY linear and angular velocities denoted as

$$\mathbf{v}_o^b = [u, v, w]^T \quad \boldsymbol{\omega}_{nb}^b = [p, q, r]^T. \tag{3.3}$$

Here, o is the point that the vessel rotates about. If the crane base is defined as a point $base$ and the crane tip is defined as a point tip, a vector from the vessel origin to the crane base may be defined in the BODY frame as $\mathbf{r}_{base}^b = [x_{base}^b, y_{base}^b, z_{base}^b]^T$. The vector from the crane base to the crane tip may be defined as $\mathbf{r}_{basetip}^b = [x_{basetip}^b, y_{basetip}^b, z_{basetip}^b]^T$, see Figure 3.1. The vector from the vessel origin to the crane tip $\mathbf{r}_{tip}^b$ is then given in the BODY frame as

$$\mathbf{r}_{tip}^b = \mathbf{r}_{base}^b + \mathbf{r}_{basetip}^b. \tag{3.4}$$

Since the crane base is fixed in the BODY frame, and the crane tip may be moved relative to the base, $\mathbf{r}_{base}^b$ is constant while $\mathbf{r}_{basetip}^b$ is time varying.

Rotation matrices are necessary to use in order to get a description of the position, velocity and acceleration of the crane tip in NED. With $\boldsymbol{\Theta} = [\phi, \theta, \psi]^T$ representing the Euler angles, the rotation matrix from BODY to NED is given by $\mathbf{R}_b^n(\Theta) = \mathbf{R}_{z,\psi}\mathbf{R}_{y,\theta}\mathbf{R}_{x,\phi}$, with the different components given as

$$\mathbf{R}_{z,\phi} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos(\phi) & -\sin(\phi) \\ 0 & \sin(\phi) & \cos(\phi) \end{bmatrix}, \quad \mathbf{R}_{z,\theta} = \begin{bmatrix} \cos(\theta) & 0 & \sin(\theta) \\ 0 & 1 & 0 \\ -\sin(\theta) & 0 & \cos(\theta) \end{bmatrix}, \quad \mathbf{R}_{z,\psi} = \begin{bmatrix} \cos(\psi) & -\sin(\psi) & 0 \\ \sin(\phi) & \cos(\psi) & 0 \\ 0 & 0 & 1 \end{bmatrix}. \tag{3.5}$$

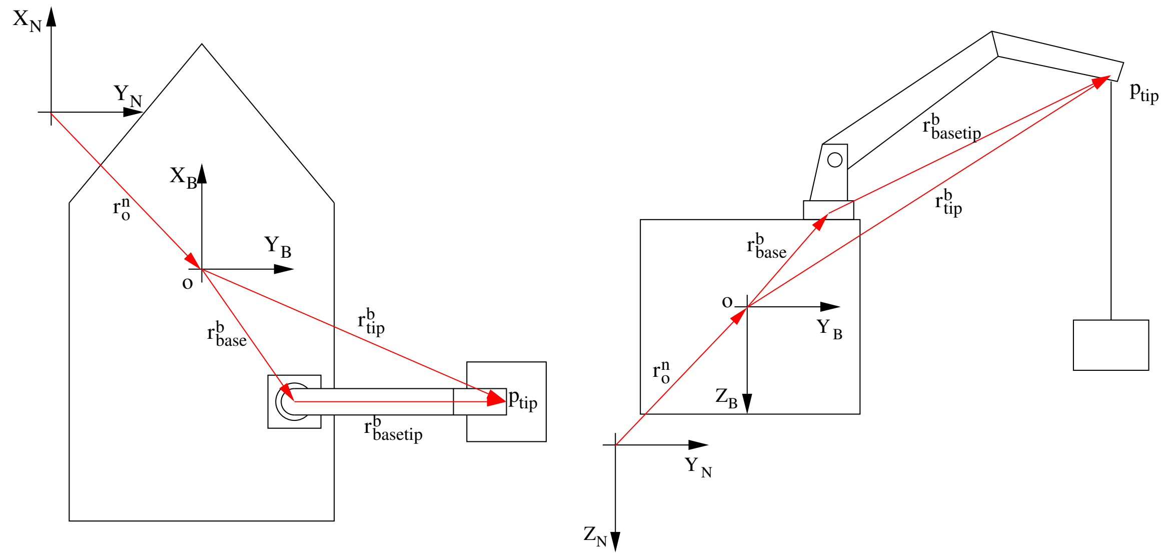

Figure 3.1: Illustration of the crane geometry. At the left, the vessel is viewed from above, while at the right the vessel is viewed from the stern.

The resulting rotation matrix then becomes

$$\mathbf{R}^n_b(\Theta) = \begin{bmatrix} \cos(\psi)\cos(\theta) & -\sin(\psi)\cos(\phi)+\cos(\psi)\sin(\theta)\sin(\phi) & \sin(\psi)\sin(\phi)+\cos(\psi)\cos(\phi)\sin(\theta) \\ \sin(\psi)\cos(\theta) & \cos(\psi)\cos(\phi)+\sin(\phi)\sin(\theta)\sin(\psi) & -\cos(\psi)\sin(\phi)+\sin(\theta)\sin(\psi)\cos(\phi) \\ -\sin(\theta) & \cos(\theta)\sin(\phi) & \cos(\theta)\cos(\phi) \end{bmatrix}. \tag{3.6}$$

It is custom to just write $\mathbf{R}^n_b$ simplisity. This matrix may be used to get the vector r^b_{tip} described in the NED frame as

$$\mathbf{r}^n_{tip} = \mathbf{R}^n_b \mathbf{r}^b_{tip}. \tag{3.7}$$

The position of the crane tip may now be described in the NED frame as.

$$\mathbf{p}^n_{tip} = \mathbf{r}^n_o + \mathbf{r}^n_{tip} \tag{3.8}$$

To find the velocities of the crane tip in the NED frame, it is practical to introduce the vector cross product operator $\mathbf{S}$. The cross product $\mathbf{a}\times\mathbf{b}$ may be written as $\mathbf{S}(\mathbf{a})\mathbf{b}$, with $\mathbf{S}(\mathbf{a})$ defined as

$$\mathbf{S}(\mathbf{a}) := \begin{bmatrix} 0 & -a_3 & a_2 \\ a_3 & 0 & -a_1 \\ -a_2 & a_1 & 0 \end{bmatrix}, \quad \mathbf{a} = [a_1, a_2, a_3]^T. \tag{3.9}$$

Assuming the velocities of the vessel in the BODY frame are known, the same procedure may be used to get the velocity of the crane tip in the NED frame. Denoting the crane tip velocity in the NED and BODY frame as $\dot{\mathbf{p}}^n_{tip}$ and $\mathbf{v}^b_{tip}$, the resulting velocity is given as

$$\dot{\mathbf{p}}^n_{tip} = \mathbf{R}^n_b \mathbf{v}^b_{tip}, \quad \mathbf{v}^b_{tip} = \mathbf{v}^b_o + \boldsymbol{\omega}^b_{nb} \times \mathbf{r}^b_{tip} \Rightarrow \mathbf{v}^b_{tip} = \mathbf{v}^b_o + \mathbf{S}(\boldsymbol{\omega}^b_{nb})\mathbf{r}^b_{tip}. \tag{3.10}$$

Using this notation, the differential equation for the rotation matrix from the BODY frame to the NED frame is given by

$$\dot{\mathbf{R}}^n_b = \mathbf{R}^n_b \mathbf{S}(\boldsymbol{\omega}^b_{nb}). \tag{3.11}$$

This result is used when deriving the acceleration of the crane tip. Assuming the linear and angular accelerations of the vessel in the BODY frame are known, the accelerations of the vessel in the NED frame is found by differentiation of (3.10). Then $\ddot{\mathbf{p}}^n_{tip}$ is derived to be

$$\begin{aligned} \ddot{\mathbf{p}}^n_{tip} &= \mathbf{R}^n_b(\dot{\mathbf{v}}^b_o + \dot{\boldsymbol{\omega}}^b_{nb} \times \mathbf{r}^b_{tip} + \boldsymbol{\omega}^b_{nb} \times \dot{\mathbf{r}}^b_{tip}) + \dot{\mathbf{R}}^n_b(\mathbf{v}^b_o + \boldsymbol{\omega}^b_{nb} \times \mathbf{r}^b_{tip}) \\ &= \mathbf{R}^n_b[\dot{\mathbf{v}}^b_o + \mathbf{S}(\dot{\boldsymbol{\omega}}^b_{nb})\mathbf{r}^b_{tip} + \mathbf{S}(\boldsymbol{\omega}^b_{nb})\dot{\mathbf{r}}^b_{tip} + \mathbf{S}(\boldsymbol{\omega}^b_{nb})\mathbf{v}^b_o + \mathbf{S}^2(\boldsymbol{\omega}^b_{nb})\mathbf{r}^b_{tip}] \qquad (3.12) \\ &= \mathbf{R}^n_b\{\dot{\mathbf{v}}^b_o + [\mathbf{S}(\dot{\boldsymbol{\omega}}^b_{nb}) + \mathbf{S}^2(\boldsymbol{\omega}^b_{nb})]\mathbf{r}^b_{tip} + \mathbf{S}(\boldsymbol{\omega}^b_{nb})[\dot{\mathbf{r}}^b_{tip} + \mathbf{v}^b_o]\}. \qquad (3.13) \end{aligned}$$

All these results are derived with the help of [1].

3.5 Wire Behaviour

The wire behaviour is in reality described by advanced partial differential equations [10]. These equations are difficult to implement in control laws, and simpler descriptions are very often sufficient. By modeling the wire and payload as a second order mass-spring-damper system in its linear range, a good model usable for control purposes can be acquired.

3.5.1 Wire Deformation

The wire is considered to be elastic, and will be deformed both under its own weight and by the payload attached at the wire end. This deformation may be found by considering the incremental deformation $d\delta$ of a slice dy a distance y from the wire bottom [11], as displayed in Figure 3.2. The total deformation δ is found by integrating over the length of the wire

$$\begin{aligned} d\delta &= \frac{\gamma A y + \gamma_l V_l}{AE} dy \qquad (3.14) \\ \Rightarrow \quad \delta &= \int_0^L d\delta = \frac{\gamma}{E}\frac{y^2}{2}\bigg|_0^L + \frac{\gamma_l V_l y}{AE}\bigg|_0^L = \frac{L}{E}\left(\frac{\gamma L}{2} + \frac{\gamma_l V_l}{A}\right). \qquad (3.15) \end{aligned}$$

Here, the wire length is L, the cross section area is A, the weight density is $\gamma = \rho g$, and the elastic modulus of the wire is E. The payload attached has weight density γ_l and volume V_l.

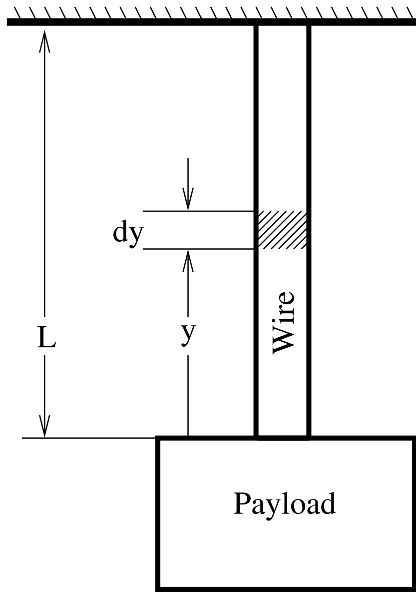

Figure 3.2: Wire deformation. The wire is deformed both by its own as well as the payload weight.

If the payload is considerably heavier than the wire, the weight of the wire may be neglected when calculating the wire elongation. As long as the wire is not overloaded, the wire may be modeled as a massless spring in its linear range, while the wire and payload together may be modeled as mass-spring-damper system. The spring stiffness k is given by

$$k(l) = \frac{EA}{l}, \tag{3.16}$$

which is a function of modulus of elasticity E, wire area A, and wire length l. This implies that the k-value of the wire will vary as the length of the wire changes. It is more problematic to find the damping coefficients, but the damping is considered to be very low. The model of the wire with a mass attached is shown in Figure 3.3. The elongation of the wire δ caused by static forces F_s (forces such as gravity, that does not vary), may now be found as

$$\delta = \frac{F_s}{k(l)}, \tag{3.17}$$

by using the k-value of the wire. This is more practical than using (3.14), as the deformation is expressed as a function of forces acting on the payload. The pre-stressed wire will now be of total length $l_{tot} = l + \delta$. The equation of motion for the system illustrated in Figure 3.3 is given by

$$m\ddot{z} + k(z - z_0) + d(\dot{z} - \dot{z}_0) = F. \tag{3.18}$$

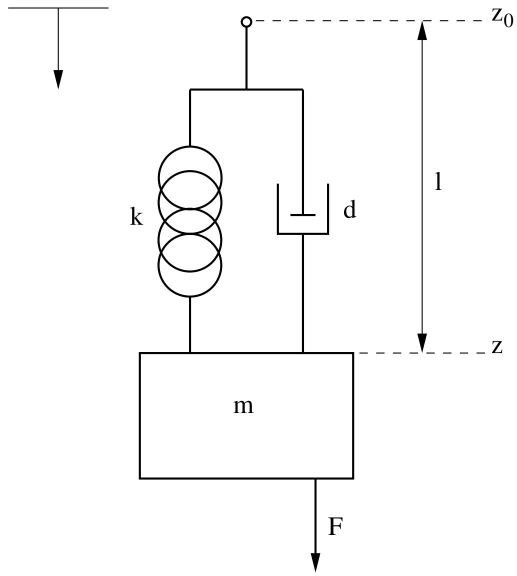

Figure 3.3: Wire and payload modelled as a mass-spring-damper system.

Here, the point z_0 may be moved by a motor, or it may be the crane tip moving up and down, z is the position of the payload, and F the force acting on it. All values given are according to the pre-stressed wire length. This system may easily be implemented as a two-port [12], where one port has input $\dot{z}_0$, and output F_0 given by

$$F_0 = k(z - z_0) + d(\dot{z} - \dot{z}_0). \tag{3.19}$$

The other port has input F and output $\dot{z}$.

There are actually two mass-damper-spring systems in play here. One is the system mentioned above, where the vessel motion is moving the crane tip. In this system it is assumed that the vessel motion is not affected by the forces acting on the payload or wire. The other system describes hoe the winch is handling the wire, and is illustrated in Figure 3.4. The equations of motion for this system is derived by the help of [12]. The motor equation of motion becomes

$$\frac{J_m}{r^2}\ddot{z}_m + d(\dot{z}_m - \dot{z}) + k(z_m - z) = F_m, \tag{3.20}$$

while the payload equation of motion becomes

$$m\ddot{z} + d(\dot{z} - \dot{z}_m) + k(z - z_m) = F_h, \tag{3.21}$$

which may be rewritten to

$$m\ddot{z} - d(\dot{z}_m - \dot{z}) - k(z_m - z) = F_h. \tag{3.22}$$

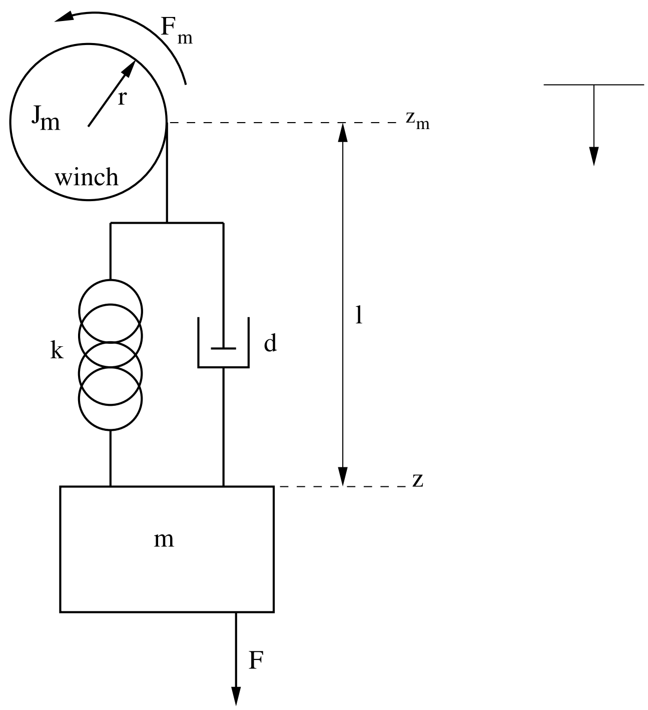

Figure 3.4: Motor, wire and payload modelled as a mass-spring-damper system.

This system is also practical to implement as a two port. One port with F_m as input and $\dot{z}_m$ as output, and one port with F_h as input and $\dot{z}$ as output.

By the use of superposition, the effect of these two systems may be added, and the total behaviour is given. There will be two different spring stiffness values k, as the wire is shorter from the crane tip to the payload, than from the winch to the payload, and the wire will have a static elongation due to gravity, and the payload oscillation point is given by this elongation added to the wire length.

3.5.2 Pendulum Motion of the Wire

In this section a model of the passive pendulum compensation of an A-frame crane, as described in Section 2.2, is derived. The Lagrangian method is used. The system is modelled as illustrated in Figure 3.5. It is assumed the mass of the running wheel is negligible compared to that of the payload. This is why it is modeled as a moving point, instead of a moving mass. First the Lagrangian L of the system has to be found. This is the total energy of the system, consisting of kinetic an potential energy, denoted T and V respectively. In the following x_0 is the position of the crane tip, while x is the position of the running wheel. The kinetic energy

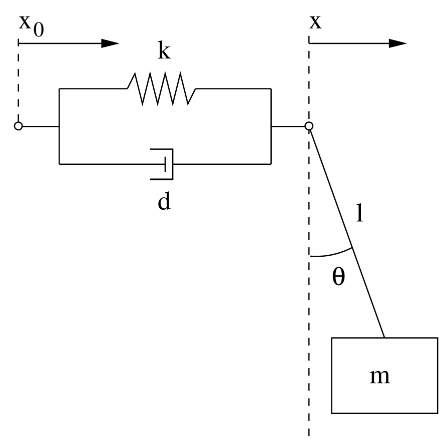

Figure 3.5: Illustration used to model the pendulum motion of the payload.

of the system is given by

$$T = \frac{1}{2}mv^2, \quad (3.23)$$

where v denotes the velocity of the payload. The velocity of the payload is then given by

$$\begin{aligned} v^2 &= \left(\frac{d}{dt}(lcos\theta)\right)^2 + \left(\frac{d}{dt}(x + lsin\theta)\right)^2 && (3.24)\\ &= \left(-l\dot{\theta}sin\theta)\right)^2 + \left(\dot{x} + l\dot{\theta}cos\theta\right)^2 && (3.25)\\ &= l^2\dot{\theta}^2sin^2\theta + \dot{x}^2 + \dot{x}^2 + \dot{x}^2\dot{\theta}lcos\theta + l^2\dot{\theta}^2cos^2\theta && (3.26)\\ &= \dot{x}^2 + \dot{x}\dot{\theta}lcos\theta + \dot{\theta}^2l^2. && (3.27)\end{aligned}$$

The kinetic energy may then be written

$$T = \frac{1}{2}m\left(\dot{x}^2 + \dot{x}\dot{\theta}lcos\theta + \dot{\theta}^2l^2\right). \quad (3.28)$$

The potential energy consists of the energy stored in the spring, and the energy conserved by gravity. This gives

$$V = V_{spring} + V_{grav}, \quad (3.29)$$

with

$$\begin{aligned} V_{spring} &= \frac{1}{2}k(x - x_0)^2 && (3.30)\\ V_{grav} &= mgl(1 - cos\theta). && (3.31)\end{aligned}$$

The Lagrangian $L = T - V$ is now found to be

$$L = \frac{1}{2}m\left(\dot{x}^2 + \dot{x}\dot{\theta}lcos\theta + \dot{\theta}^2 l^2\right) - \frac{1}{2}k(x - x_0)^2 - mgl(1 - cos\theta) \tag{3.32}$$

The equations of motion of the running wheel and payload may now be calculated by using Lagrange's Equation:

$$-d(\dot{x} - \dot{x}_0) = \frac{d}{dt}\left(\frac{\partial L}{\partial \dot{x}}\right) - \frac{\partial L}{\partial (x - x_0)}. \tag{3.33}$$

for the running wheel, and

$$-b\dot{\theta} = \frac{d}{dt}\left(\frac{\partial L}{\partial \dot{\theta}}\right) - \frac{\partial L}{\partial \theta}. \tag{3.34}$$

for the payload. In these equations $-d(\dot{x} - \dot{x}_0)$ and $-b\dot{\theta}$ denotes the damping and friction of the running wheel and pendulum motion respectively. These forces can not be included in the Lagrangian since they are non-conservative, unlike kinetic and potential energy. The partial derivatives becomes

$$\begin{aligned}
\frac{\partial L}{\partial \dot{x}} &= m\dot{x} + ml\dot{\theta}cos\theta && (3.35)\\
\frac{\partial L}{\partial (x - x_0)} &= -k(x - x_0) && (3.36)\\
\frac{\partial L}{\partial \dot{\theta}} &= ml\dot{x}cos\theta + m\dot{\theta}l^2 && (3.37)\\
\frac{\partial L}{\partial \theta} &= -ml\dot{x}\dot{\theta}sin\theta - mglsin\theta && (3.38)\\
\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{x}}\right) &= m\ddot{x} + ml\ddot{\theta}cos\theta - ml\dot{\theta}^2 sin\theta && (3.39)\\
\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{\theta}}\right) &= -ml\dot{x}\dot{\theta}sin\theta + ml\ddot{x}cos\theta + m\ddot{\theta}l^2. && (3.40)
\end{aligned}$$

The equations of motion for the system is then written as

$$\begin{aligned}
-d(\dot{x} - \dot{x}_0) &= m\ddot{x} + ml\ddot{\theta}cos\theta - ml\dot{\theta}^2 sin\theta + k(x - x_0) && (3.41)\\
-b\dot{\theta} &= ml\ddot{x}cos\theta + m\ddot{\theta}l^2 + mglsin\theta. && (3.42)
\end{aligned}$$

The equations of motion given above describe the motion of the payload, based on x_0, x and θ. The position,velocity and acceleration of the payload relative to the running wheel, is given by

$$\begin{aligned}
\text{position} \quad (x_p, y_p) &= (x + lsin\theta, lcos\theta) && (3.43)\\
\text{velocity} \quad (\dot{x}_p, \dot{y}_p) &= \dot{x} + l\dot{\theta}cos\theta, -l\dot{\theta}sin\theta) && (3.44)\\
\text{acceleration} \quad (\ddot{x}_p, \ddot{y}_p) &= \ddot{x} + l\ddot{\theta}cos\theta - l\ddot{\theta}^2 sin\theta, -l\ddot{\theta}sin\theta - l\dot{\theta}^2 cos\theta). && (3.45)
\end{aligned}$$

Once the position of the payload is known, it is possible to use the "Waves" block in the MSS-toolbox to compute the wave height in the area where the payload is.

3.6 Forces Acting on the Payload

As long as the payload is in the air, the only force acting on it is gravity. This is a static force, and how it is handled is described in Section 3.5. As the payload is lowered through the splash zone, which is the zone of transition between air and water, a number of effects must be considered.

It is important that the snap loads are small enough so that the winch, wire, crane and payload is not put under more stress than it is designed to handle. Snap loads are the forces that occur when the wire tension is lessened and suddenly increased again. This can be caused by waves slamming into the payload, making it push and pull the wire as waves pass by. This effect may result in damage to the winch, wire, crane or payload and is a problem especially if the waves pass by at a frequency close to the resonance frequency of the system.

Another effect coming into play when lowering the payload through the splash zone is buoyancy. This means that the weight of the payload will be less than in air and will decrease according to the volume of the submerged part of the payload. The added mass of the payload must also be considered, as well as linear and non-linear drag forces. All forces put together in one equation, and denoted F_h becomes [13]

$$F_h = F_B + F_A + F_D, \tag{3.46}$$

where F_B, F_A and F_D is the forces caused by buoyancy, added mass and drag effects respectively. Written out this equation becomes

$$F_h = -\rho g \nabla(z_r) - \rho \nabla(z_r)\ddot{z}_r - Z_{\ddot{z}_r}(z_r)\ddot{z}_r - \frac{\partial Z_{\ddot{z}_r}(z_r)}{\partial z_r}\dot{z}_r^2 - \frac{1}{2}\rho C_D A_{pz}\dot{z}_r|\dot{z}_r| - d_l \dot{z}r. \tag{3.47}$$

Here, ∇ represent buoyancy, $Z_{\ddot{z}_r}$ is position dependent added mass, A_{pz} is the projected effective drag area, C_D is the drag coefficient, and d_l represent the linear drag. The position of the payload relative to the water surface is z_r. The slamming force acting when the waves hit the payload is given by the fourth element in the equation. It should be noted that this force is directed upwards [13], since the partial derivative of $Z_{\ddot{z}_r} > 0$.

3.7 Metacentric Stability

When considering which vessel to use for lifting operations, the roll and pitch motions of the vessel should be minimized. One of the factors affecting these motions is the metacentric height (GM). GM is the distance between the metacentre and the centre of gravity of a vessel, and is used to calculate the stability of vessels. There is one metacentre for roll (lateral metacentre) and one for pitch (longitudinal metacentre] denoted $\overline{GM}_T$ and $\overline{GM}_L$, respectively [1]. The metacentre is defined as [14]:

The theoretical point at which an imaginary vertical line through the centre of buoyancy intersects another imaginary vertical line through a new centre of buoyancy created when the body is displaced, or tipped, in the water, however little.

The notation used in naval architecture is as follows; M denotes the metacentre, G denotes the centre of gravity, and B denotes the centre of buoyancy [1]. The centre of buoyancy is the centre of gravity of the water displaced by the hull of the vessel. In Figure 3.6, an illustration of the metacentre position is shown. It should be noted that in reality the metacentre and centre of buoyancy is moving as the vessel rolls or pitches. There are different metacentric heights for any combined roll and pitch motion, but they are usually stated as specific values for pure roll and pitch motion. The righting force on the vessel is caused by gravity acting through the centre of gravity of the vessel, and the buoyancy pushing the vessel upwards through the vertical line passing through the centre of buoyancy and metacentre. This creates a torque which rotates the hull upright. The righting force is proportional to the metacentric height times the moment arm. This makes the metacentric height important when determining the roll and pitch properties of the vessel, which in turn is important when considering the motion compensation system. The effect of a small GM is long roll and pitch periods, while increasing the GM leads to shorter roll and pitch periods. If the GM is very small or even negative, the risk of capsizing increases, and the possibility of a large constant roll angle increases. If the GM is too large on the other hand, the vessel will have a quick response and will try to follow the slope of the waves. The vessel will roll and pitch with a short period and large amplitude, resulting in high angular acceleration. Vessels with a low GM tends to lag behind the motion of the waves, leading to reduced angular acceleration, and smaller roll and pitch amplitude.

When discussing motion compensation of cranes considering the vessel properties, it would be preferable with a small GM. This would lead to less strain on the winch and other actuators included in the compensation, since smaller angular acceleration of the vessel is bound to reduce the speed in which the payload needs to be winded up or down. The problem with constant roll and pitch angles may be reduced by including ballast control [1], shifting the ballast position as the payload is lifted across the deck and over the side of the

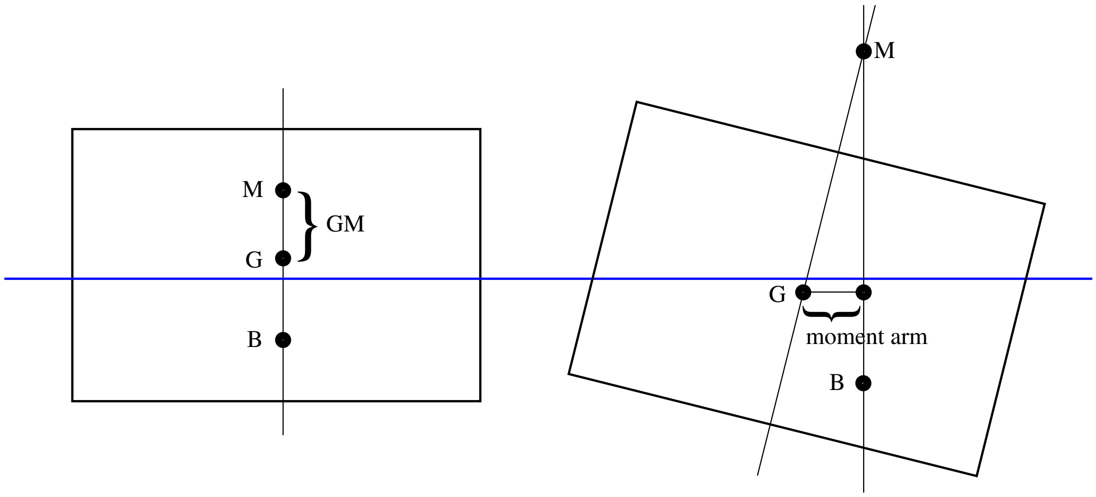

Figure 3.6: Metacentre of a vessel. The vertical line drawn in the two illustrations intersects at a given point, revealing the metacentre. The moment/righting arm may be seen to the right, as the distance between G and the vertical line. The illustration is inspired by [15].

vessel. This is often referred to as trim and heel control. A good example of a system using ballast control is the Sea Launch platform [16].

4
Motion Compensation

In the following, some ideas on how motion compensation may be achieved are presented. In the first section, position, velocity and tension control are discussed, which applies when the payload is below the sea surface. The second section treats wave synchronisation, which basically is a method for synchronising the payload with the waves, making it rise and fall with the sea surface. The wave synchronisation is also usable when lifting cargo from one vessel to another. In this case, it is possible to make the payload follow the heave motion of the deck on the receiving vessel, and thus perform a controlled descent of the cargo, which greatly reduce the risk of damaged goods or personnel injuries. A good way to validate the compensator performance is to measure the wire tension and the hydrodynamic forces acting on the payload [13]. The wire tension must never be negative, and if this should occur a snap load will be the result. The peak values and variance of both wire tension and hydrodynamic forces should be minimized to reduce wire and payload damages.

4.1 Heave Compensation

The objective of heave compensation is to make the payload follow a given reference in an inertial reference frame, and hence keep the vessel motion from influencing the payload motion. This is the classic way of compensating for the wave induced vessel motion, where the only motion compensated is the heave motion of the vessel. This is mainly applied when an object is to be placed at the seabed, and precision is essential. As an example, several oil fields could not have been realised if the possibility of performing advanced lifting operations in heavy sea was not there. There are many manufacturers producing this kind of system available, and the systems are showing good performance in keeping the payload at rest in the desired position.

4.1.1 Position/Velocity Control

The simplest form of motion compensation is to measure the position or the velocity of the crane tip, and compensate directly using the winch. This is used in [13], where the vertical velocity of the crane tip is added to the motor speed reference signal. The equation for this simple controller becomes

$$\dot{z}_d = \dot{z}_m^* + \dot{z}_0, \tag{4.1}$$

where $\dot{z}_m^*$ is the motor speed reference signal, commanded by an operator or a higher level control system, $\dot{z}_0$ is the crane tip vertical velocity, and $\dot{z}_d$ is the resulting desired motor speed. It should also be possible to use this method in position control. Then the vertical position of the crane tip is added to the motor position reference signal. The resulting equation is similar to (4.1). The only difference is that velocity is replaced with position. The equation then becomes

$$z_d = z_m^* + z_0. \tag{4.2}$$

This could be advantageous in applications where a given distance from the payload to an object is desired, rather than approaching with a given speed.

4.1.2 Wire Tension Control

With this method, the vessel motion is compensated by keeping a constant wire tension. If the tension is high, the wire is let out, if the tension is low, the wire is reeled in. This method is only suitable when the payload is in the water, since when wave synchronisation is active, it is intended that the wire tension will vary significantly. Another aspect is the pendulum motion that occurs above the surface. The wire tension will be a function of both θ and payload velocity, making it hard to compensate the vessel motion with tension control.

4.2 Wave Synchronisation

With wave synchronization it is possible to let the payload track a moving reference, like wave height or the deck of another vessel. The main aspect is the possibility to lower the payload through the surface with greatly reduced risk of snap load and damage of equipment. This is beneficial in many ways, since it leads to a safer and more robust way of conducting lifting operations, including a higher percentage of time operative, and a lessened risk of personnel injuries.

4.2.1 Position/Velocity Control

The method used in heave compensation may also be used for wave synchronisation [13]. The difference is that in this case wave elevation has to be considered as well. Looking at Figure 1.2, the distance from crane tip to sea surface is ζ_0, and the distance from sea surface to payload is z_r, the distance from the payload to the sea surface is then given by

$$z_r = z - \zeta_0. \tag{4.3}$$

To find the velocity of the surface relative to the payload, it is not sufficient to find the time derivative of this equation. This is because there is an orbital movement of water as the waves pass by. The vertical velocity of this water has to be included in the computation of $\dot{z}_r$. The resulting velocity of the surface relative to the payload then becomes

$$\dot{z}_r = \dot{z} - \dot{\zeta}_0 \kappa(z), \tag{4.4}$$

where $\kappa(z)$ is the function describing how the vertical wave velocity decreases with water depth. Since the vertical water velocity decreases rapidly with increasing water depth, a good approximation may be to let the function be exponentially decreasing depending on water depth. The function then becomes

$$\kappa(z) = \begin{cases} 1 & , z \leq h_v \\ exp(-k(z - h_v)) & , z > h_v. \end{cases} \tag{4.5}$$

Here, h_v may be the depth of the vessel keel, or it might be the height of the trough of the waves relative to still water. It might be necessary to tune h_v to get acceptable performance. Looking at the hydrodynamic forces acting on the payload (3.47), it is seen that a reduction in $\dot{z}_r$ would lead to a reduction of slamming loads and linear and viscous drag forces. Minimizing high-frequency motion of $\dot{z}_r$ will have a beneficial effect on added mass, since this will lead to a reduction in $\ddot{z}_r$. The wave synchronization is achieved by use of the feed-forward compensator

$$\dot{z}_d = \dot{z}_m^* + \dot{\zeta}_0 \kappa(z). \tag{4.6}$$

This control scheme is also applicable for position control, as done in Chapter 5, where the feed-forward compensator is implemented as

$$z_d = z_m^* + \zeta_0. \tag{4.7}$$

There will also have to be some kind of blending between heave compensation and wave synchronization [13, 17, 18], since wave synchronization should only be used during the water entry phase. By introducing a factor $\alpha(z)$ that goes from zero to one as the payload is lowered through the surface, the heave compensation may be blended with wave synchronization by writing $\dot{z}_d$ as

$$\dot{z}_d = \dot{z}_m^* + \dot{\zeta}_0 \alpha(z)\kappa(z) + \dot{z}_0(1 - \alpha(z)\kappa(z)). \tag{4.8}$$

There are many ways of defining $\alpha(z)$, and this function is a tuning factor, like h_v. An example may be [13]

$$\alpha(z) = \begin{cases} 0 & , z < 0 \\ z/h_p & , 0 \leq z \leq h_p \\ 1 & , z > h_p, \end{cases} \tag{4.9}$$

where h_p is the height of the payload, meaning that $\alpha(z)$ is zero when the payload is fully submerged.

5
Simulation

Simulating systems is a very good tool for examining their performance. This approach is cost efficient and in many cases easier to use than building models and testing the system since there is no need for special equipment. In this chapter simulations of the A-frame system described in Chapter 2 and Chapter 3 are performed. For simplicity in calculating added mass, drag force and other variables, it is assumed that the payload is a sphere. All simulations are performed by use of MATLAB/Simulink.

5.1 Waves

The waves used to simulate the vessel motion is generated by a MATLAB toolbox called MSS (Marine Systems Simulator) [7]. This toolbox contains a block called "Waves", and the output from this block is a bus containing a vector of harmonic wave amplitudes, frequencies, phases, wave numbers, and directions. All these variables are extracted from a given frequency and direction spectrum. The time series for surface elevation may be calculated as

$$\zeta(x, y, t) = \sum_{1}^{i} \zeta_a(i) cos[\omega(i)t + \varphi(i) - k(i)(xcos(\psi(i)) + ysin(\psi(i))], \tag{5.1}$$

where

ζ_a : Vector of harmonic wave amplitudes [m]

ω : Vector of harmonic wave frequencies [rad/s]
φ : Vector of harmonic wave phases (random) [rad]
k : Vector of harmonic wave numbers [1/m]
ψ : Vector of harmonic wave directions [rad].

This is also useful when simulating the distance from the payload to the ocean surface, as the wave elevation below/above the payload may be computed from (5.1), given the x and y position of the payload.

Simulations are performed with different sea states to test the motion compensation system. A Pierson - Moskowitz wave spectrum is used, where the sea states are defined as in Figure 5.1.

5.2 Hydrodynamic Forces

In [19] the hydrodynamic force on a payload shaped as a sphere is written as

$$Fh = \begin{cases} m_a(\ddot{z} - \ddot{z}_0)\left(\frac{d_p}{d}\right) + \frac{1}{2}\rho C_d A_p(\dot{z} - \dot{z}_0)|\dot{z} - \dot{z}_0|\left(\frac{d_p}{d}\right) + \rho g V_p(d_p) & , d_p > 0 \\ 0 & , \text{otherwise}, \end{cases} \tag{5.2}$$

where d_p is the payload amount which is below the surface, and the term (d_p/d) see to that F_h increases with d_p. The other terms are

m_a : added mass of payload (sphere) [kg]
C_d : drag coefficient of payload [-]
ρ : density of water [kg/m^3]
$V_p(d_p)$: Volume of submerged part of payload, depending on depth [m^3]
Ap : projected area of payload [m^2].

The water depth $d_p \geq 0$ is defined as

$$d_p := \begin{cases} -z_w & , -d \leq z_w < 0 \\ d & , z_w \leq -d \\ 0 & , z_w \geq 0, \end{cases} \tag{5.3}$$

Wind Speed (Kts)	Sea State	Significant Wave (Ft)	Significant Range of Periods (Sec)	Average Period (Sec)	Average Length of Waves (FT)
3	0	<.5	<.5 - 1	0.5	1.5
4	0	<.5	.5 - 1	1	2
5	1	0.5	1 - 2.5	1.5	9.5
7	1	1	1 - 3.5	2	13
8	1	1	1 - 4	2	16
9	2	1.5	1.5 - 4	2.5	20
10	2	2	1.5 - 5	3	26
11	2.5	2.5	1.5 - 5.5	3	33
13	2.5	3	2 - 6	3.5	39.5
14	3	3.5	2 - 6.5	3.5	46
15	3	4	2 - 7	4	52.5
16	3.5	4.5	2.5 - 7	4	59
17	3.5	5	2.5 - 7.5	4.5	65.5
18	4	6	2.5 - 8.5	5	79
19	4	7	3 - 9	5	92
20	4	7.5	3 - 9.5	5.5	99
21	5	8	3 - 10	5.5	105
22	5	9	3.5 - 10.5	6	118
23	5	10	3.5 - 11	6	131.5
25	5	12	4 - 12	7	157.5
27	6	14	4 - 13	7.5	184
29	6	16	4.5 - 13.5	8	210
31	6	18	4.5 - 14.5	8.5	236.5
33	6	20	5 - 15.5	9	262.5
37	7	25	5.5 - 17	10	328.5
40	7	30	6 - 19	11	394
43	7	35	6.5 - 21	12	460
46	7	40	7 - 22	12.5	525.5
49	8	45	7.5 - 23	13	591
52	8	50	7.5 - 24	14	566
54	8	55	8 - 25.5	14.5	722.5
57	8	60	8.5 - 26.5	15	788
61	9	70	9 - 28.5	16.5	920
65	9	80	10 - 30.5	17.5	1099
69	9	90	10.5 - 32.5	18.5	1182

Figure 5.1: Pierson-Moskowitz sea states.

where z_w is the distance between payload position and water surface. That is, $z_w = 0$ when the payload touches the water. For a sphere the terms in (5.2) is given as [19]

$$\begin{aligned} m_a &= 0.5\rho\left(\frac{4}{3}\pi\left(\frac{d}{2}\right)^2\right) \text{ [kg]} && (5.4)\\ C_d &\approx 1.0 \text{ [-]} && (5.5)\\ \rho &= 1000 \text{ [kg/m}^3\text{]} && (5.6)\\ V_p(d_p) &= \frac{\pi}{3}d_p^2\left(\frac{3}{2}d - d_p\right) \text{ [m}^3\text{]} && (5.7)\\ A_p &\approx \pi\left(\frac{d}{2}\right)^2 \text{ [m}^2\text{]} && (5.8) \end{aligned}$$

5.3 Numerical Values

The sphere used in the simulation has the specifications

$$d = 2.5 \text{ [m]} \quad \text{and} \quad \rho = 5000 \text{ [kg/m}^3\text{]}$$

The mass of the sphere is then given as

$$\begin{aligned} m &= \rho V && (5.9)\\ &= \rho\left(\frac{4}{3}\pi\left(\frac{d}{2}\right)^2\right) && (5.10)\\ &\approx 21000 \text{ [kg]}. && (5.11) \end{aligned}$$

This is the value of m used in the simulations. It is assumed that the crane is place at the starboard side of the vessel, with crane tip coordinates given in BODY frame as

$$x_b = -5 \quad y_b = 20 \quad z_b = -25. \tag{5.12}$$

It is assumed that the wire has an elasticity $E_w = 1.4 \times 10^{11}$, which is two thirds of that of a solid steel rod, a diameter $d_w = 0.075$ [m], and a length $l_w = 75$ [m] when the payload touches the surface. With these variables given, the spring coefficient k is found to be

$$k = \frac{E_w A_w}{l_w} \quad => k \approx 8.2 \times 10^6. \tag{5.13}$$

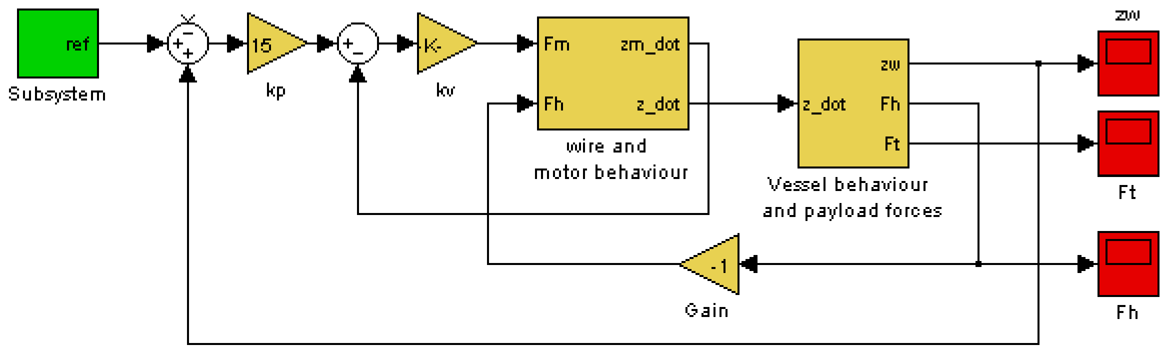

Figure 5.2: Top level block diagram of the total system with wave synchronization.

The winch is assumed to be a cylinder with a radius of $R = 1$ [m], and a mass density of $\rho = 5000$ [kg/m^3] (equal to that of the payload). The payload inertia is then given by

$$I = \frac{1}{2}\rho V r^2 \quad => I \approx 15700. \tag{5.14}$$

In the motor control loop, the gain k_v and k_p is set to

$$k_v = 25000, \quad k_p = 15.$$

5.4 Vessel and Crane Tip

The kinematics described in Subsection 3.4.1 are implemented in MATLAB/Simulink. RAOs of a vessel, along with a modified Pierson-Moskowitz wave-spectrum, are used to generate the vessel motion. The block diagram for the vessel motion simulation in Simulink is shown in Figure 5.4. The "Waves" block in Figure 5.4 and "RAO motion" block in Figure 5.5 are included in [7], while the "crane motion" block in Figure 5.5 is custom made. The vector r_tip gives the position of the crane tip in BODY coordinates. Then, by using the vessel kinematics described in Subsection 3.4.1, the crane tip motion may be calculated. This is implemented as a part of the subsystem called "Vessel behaviour and payload forces" in Figure 5.2. The subsystem is shown in Figure 5.4. The top level block diagram including wave synchronization is shown in Figure 5.2, while the top level block diagram without motion compensation is shown in Figure 5.3.

5.5 Motor, Wire and Payload

The behaviour of the motor, wire and payload is calculated by the subsystem called "Wire and motor behaviour" in Figure 5.2. This subsystem is shown in Figure 5.6. This block

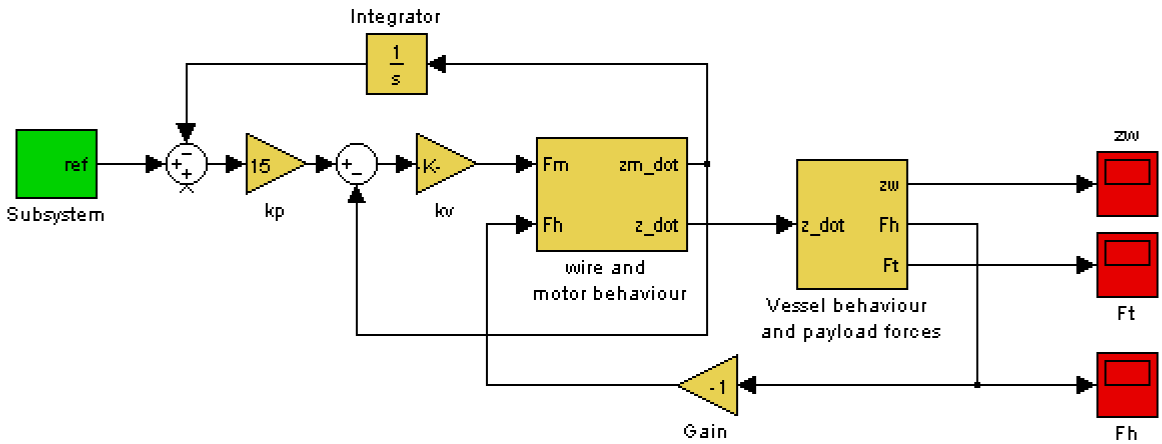

Figure 5.3: Top level block diagram of the total system without motion compensation.

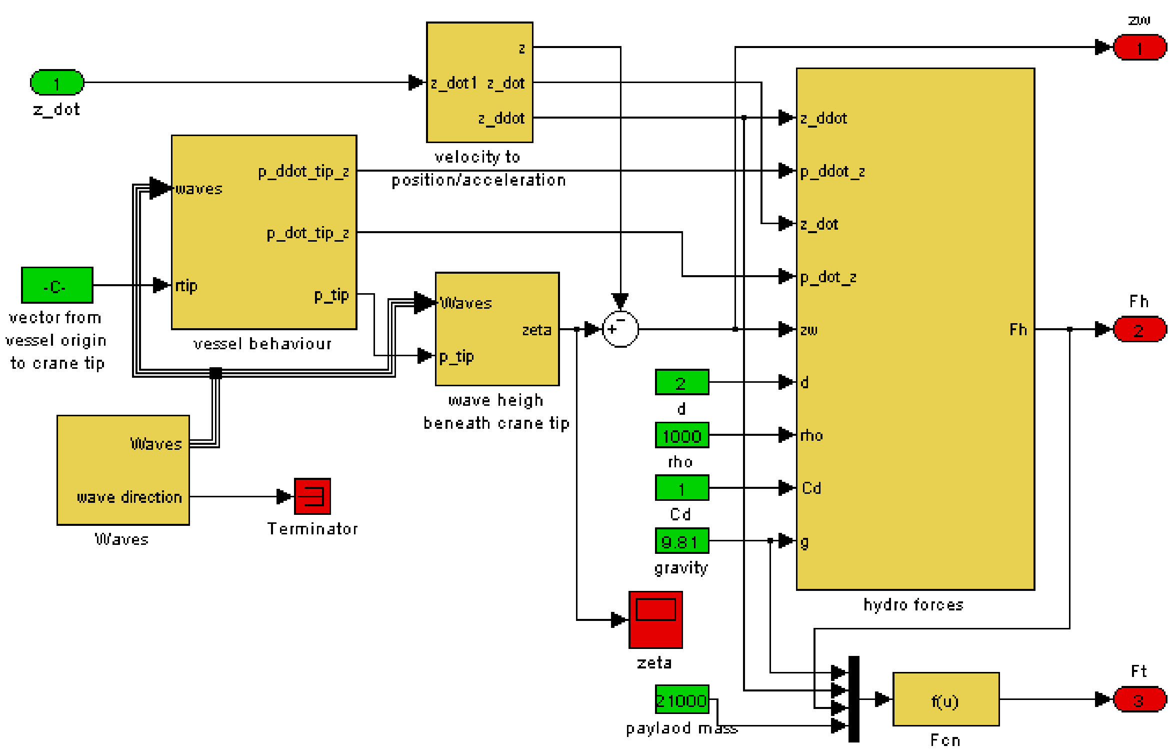

Figure 5.4: Illustration of the subsystem called "Vessel behaviour and payload forces" in Figure 5.2.

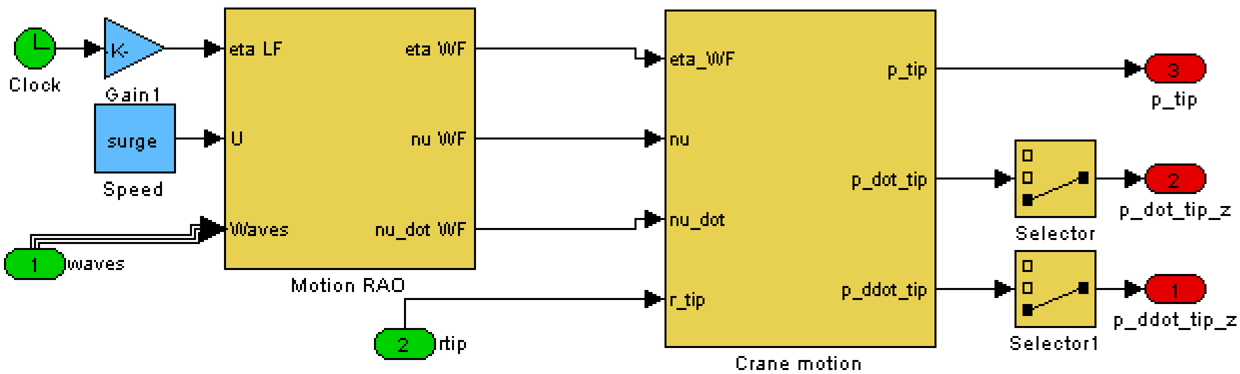

Figure 5.5: Illustration of the subsystem called "Vessel behaviour" in Figure 5.4..

takes the motor force and the hydrodynamic force as input, and gives the speed of the motor zm_dot and payload z_dot as output. The variable zm_dot is used for speed control of the motor, while z_dot is used for finding the position of the payload, and calculating the hydrodynamic forces acting on the payload.

5.6 Compensation

In this section simulations with and without motion compensation are performed. The control schemes used are wave synchronization and motor position control, where only wave synchronization is compensating wave-induced motion. The motor position control is only concerned with keeping the winch in a set position, and will try to follow the reference signal regardless of waves.

The simulations are performed with sea states according to the Pierson - Moskowitz spectrum, as described in table Table 5.1. The interesting parameters are; hydrodynamic force acting on the payload F_h, wire force F_t, and distance between the payload and sea surface z_w. Plots of these variables are shown for every sea state simulated. The reference signal used is the desired distance between the payload and the ocean surface z_w^*. This variable is positive upwards, and is inverted in the plots in order to be comparable to the actual distance. It should be noted that the blending between wave synchronization and heave compensation is not implemented, which means that only wave synchronization is simulated. The use of several DOFs for control are not simulated either.

Only two variables are changed between the simulations. These are *Significant wave height*, and *Peak frequency* in the "Waves" block, which are changed according to the sea states given in Table 5.1.

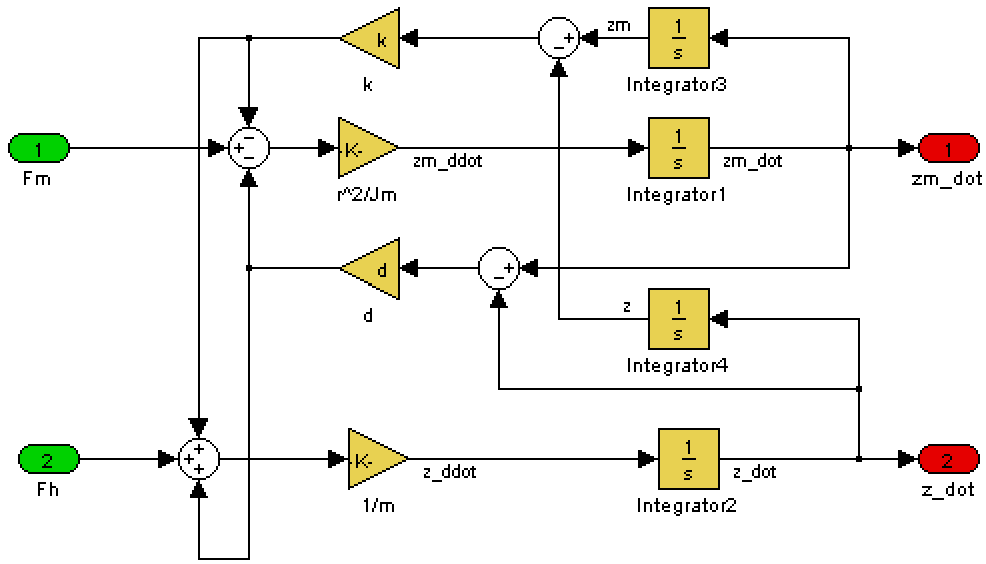

Figure 5.6: Illustration of the subsystem called "Wire and motor behaviour" in Figure 5.2.

Simulation Number	Sea State	Significant Wave Hight (H_s) [m]	Average Period (ω_0) [rad/s]
1	3	1.2	1.57
2	4	2.1	1.25
3	5	3.0	1.05
4	6	5.4	0.74

Table 5.1: Sea states used when simulating the total system. These simulations are used to measure the performance of the motion compensation system.

5.6.1 Simulation with Sea State 3

Here the parameters described as Simulation 1 in Table 5.1 are used. This means that $Hs = 1.2$ and $\omega_0 = 1.57$. The results for simulation without motion compensation are given in Figure 5.7, while the results for simulation with wave synchronization is shown in Figure 5.8.

5.6.2 Simulation with Sea State 4

Here the parameters described as Simulation 2 in Table 5.1 are used. This means that $Hs = 2.1$ and $\omega_0 = 1.25$. The results for simulation without motion compensation are given in Figure 5.9, while the results for simulation with wave synchronization is shown in Figure 5.10.

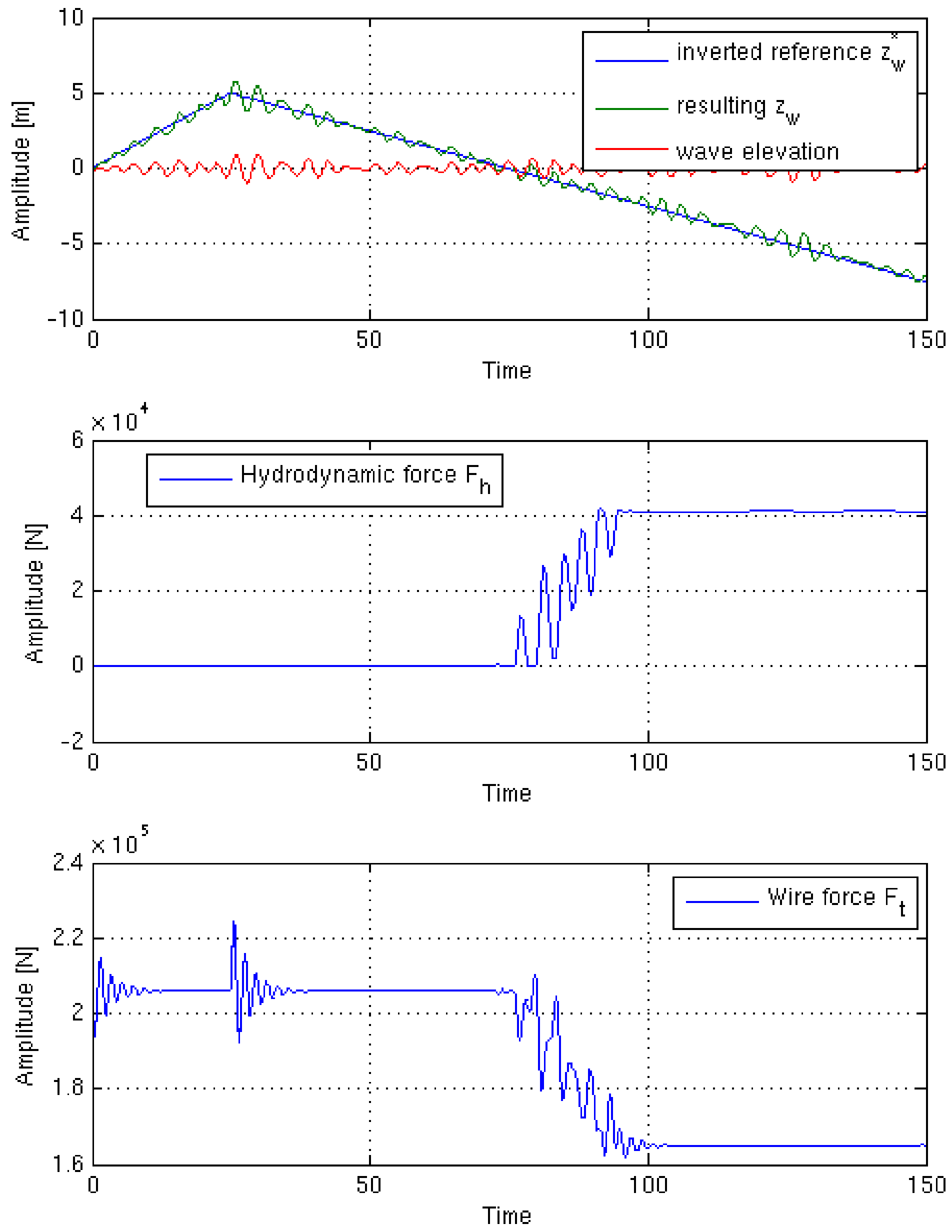

Figure 5.7: Sea state 3 without control. The reference signal is inverted in order for it to be comparable with the resulting distance.

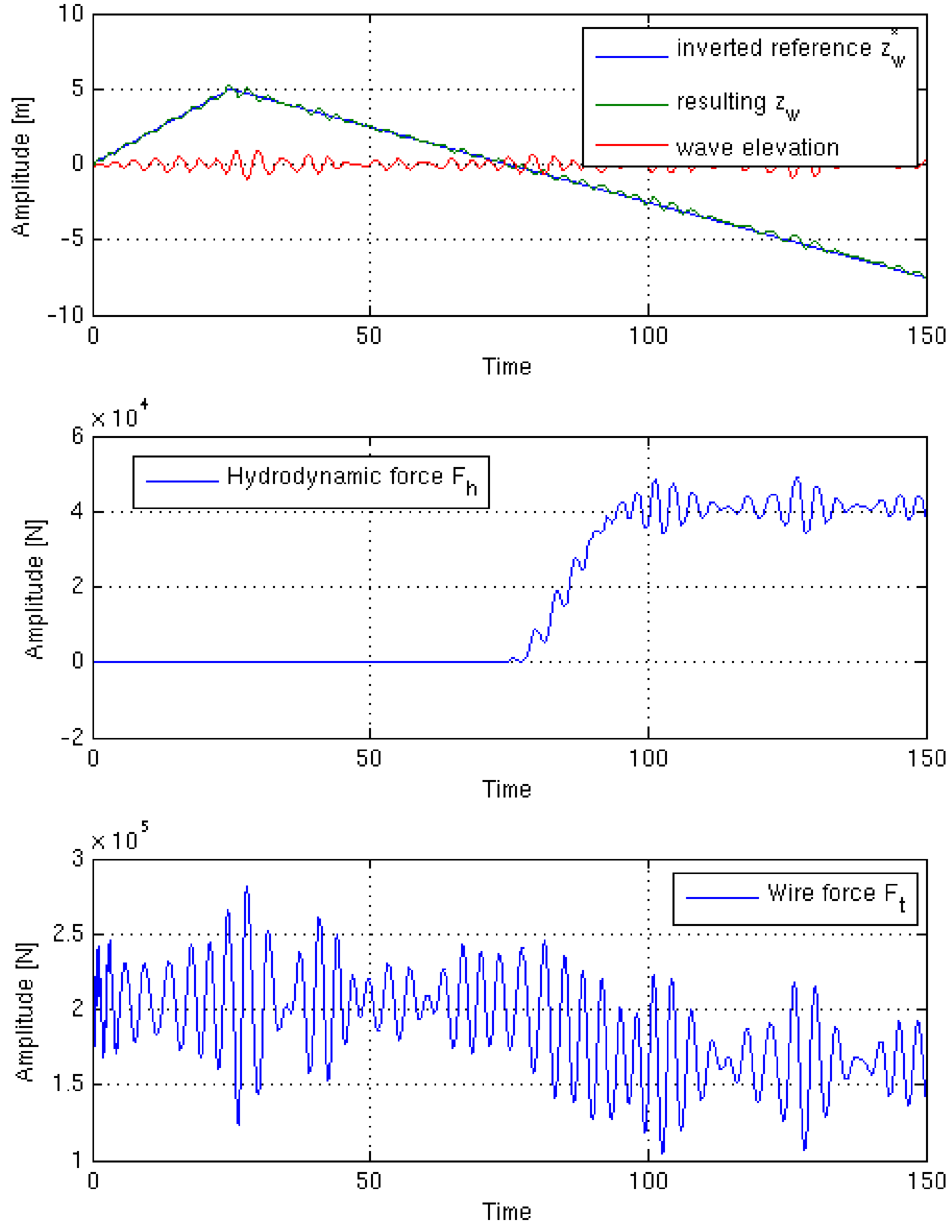

Figure 5.8: Sea state 3 with control. The reference signal is inverted in order for it to be comparable with the resulting distance.

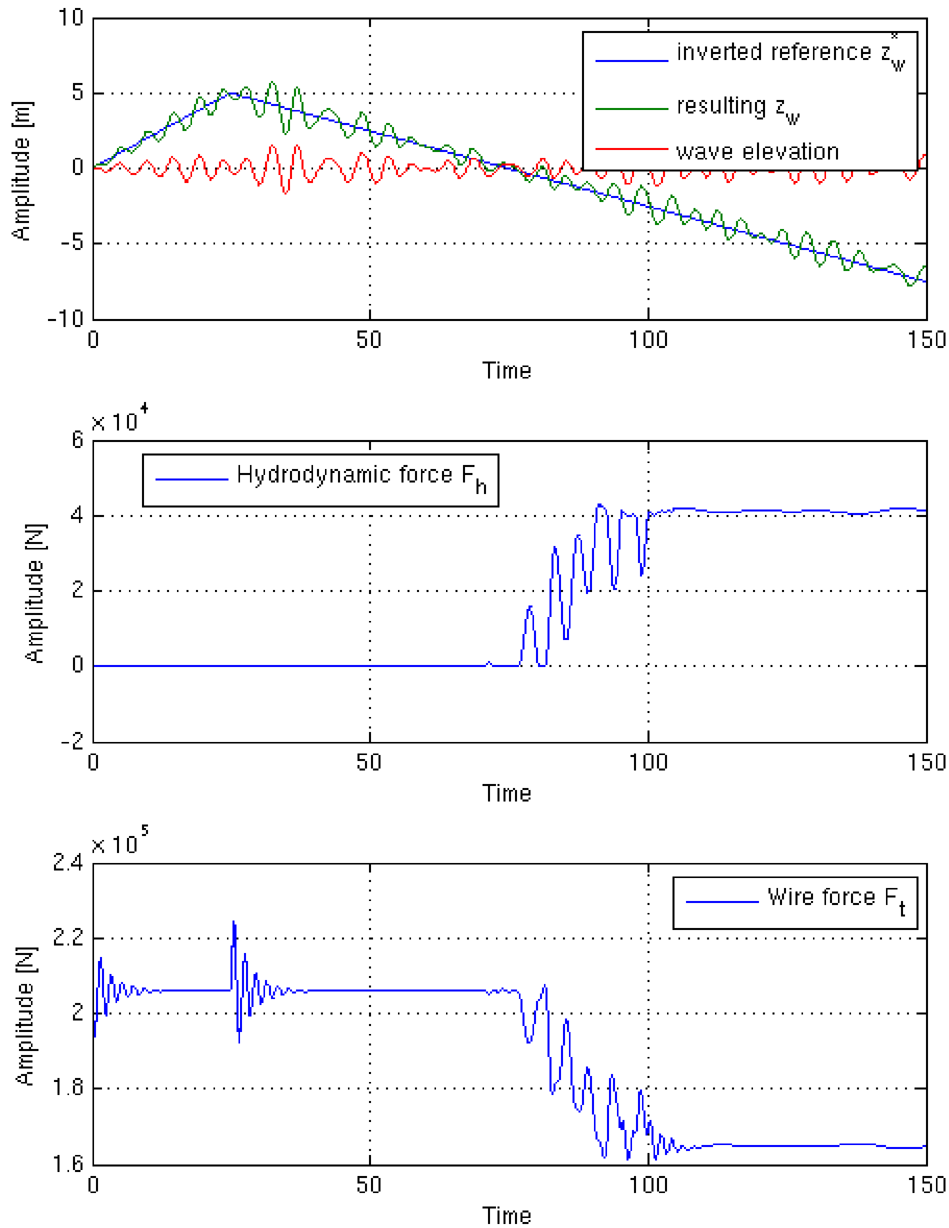

Figure 5.9: Sea state 4 without control. The reference signal is inverted in order for it to be comparable with the resulting distance.

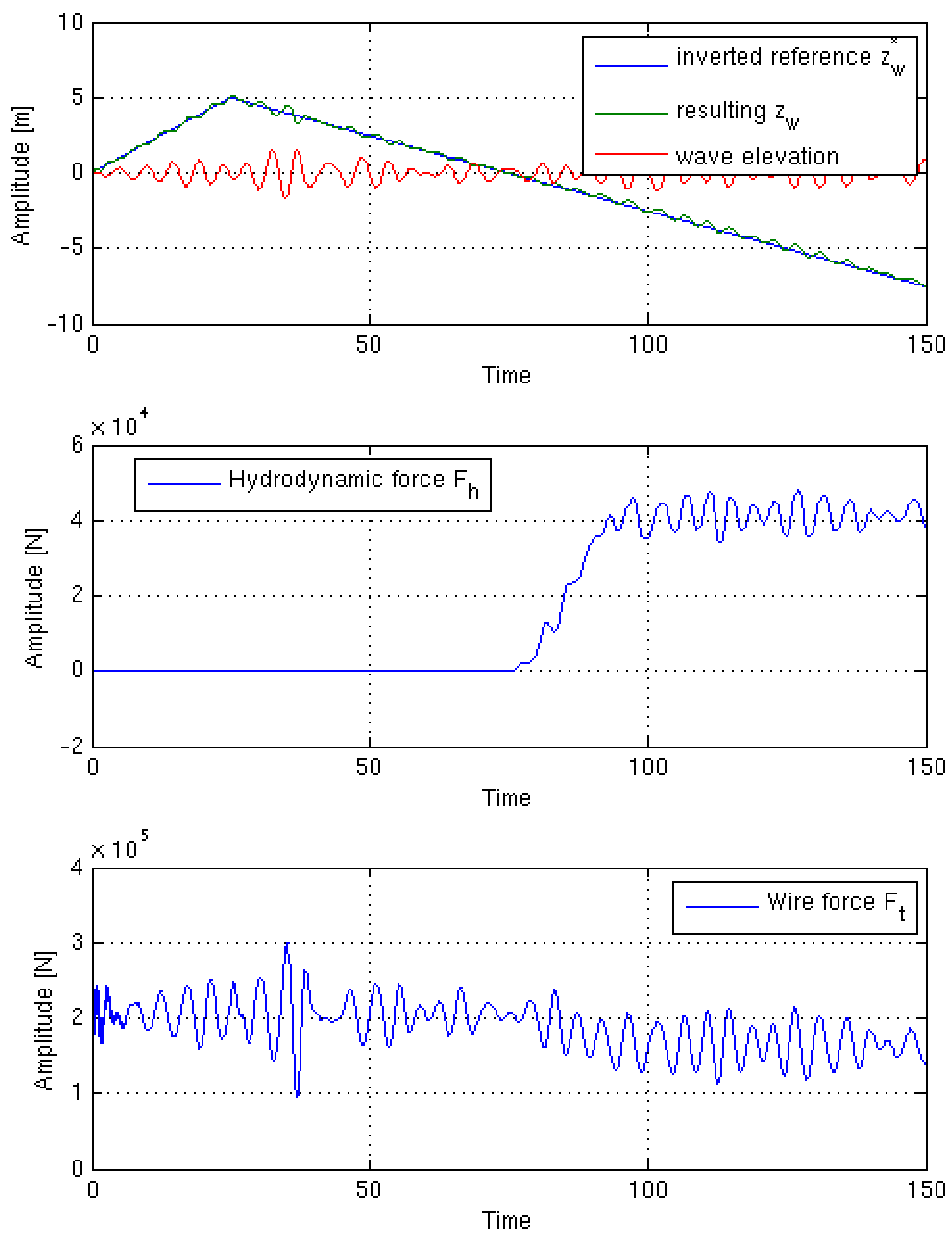

Figure 5.10: Sea state 4 with control. The reference signal is inverted in order for it to be comparable with the resulting distance.

5.6.3 Simulation with Sea State 5

Here the parameters described as Simulation 3 in Table 5.1 are used. This means that $Hs = 3.0$ and $\omega_0 = 1.05$. The results for simulation without motion compensation are given in Figure 5.11, while the results for simulation with wave synchronization is shown in Figure 5.12.

5.6.4 Simulation with Sea State 6

Here the parameters described as Simulation 4 in Table 5.1 are used. This means that $Hs = 5.4$ and $\omega_0 = 0.74$. The results for simulation without motion compensation are given in Figure 5.13, while the results for simulation with wave synchronization is shown in Figure 5.14.

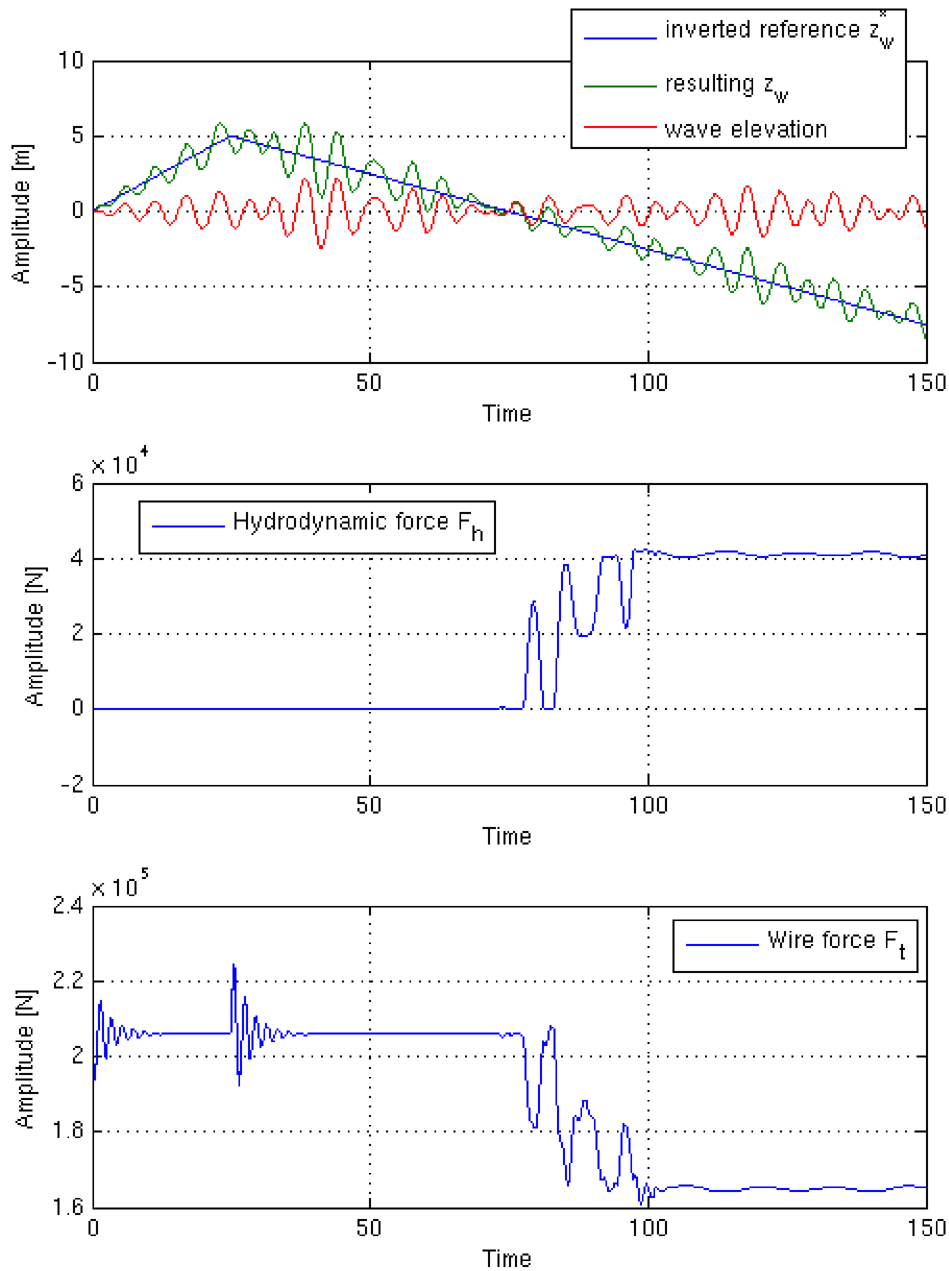

Figure 5.11: Sea state 5 without control. The reference signal is inverted in order for it to be comparable with the resulting distance.

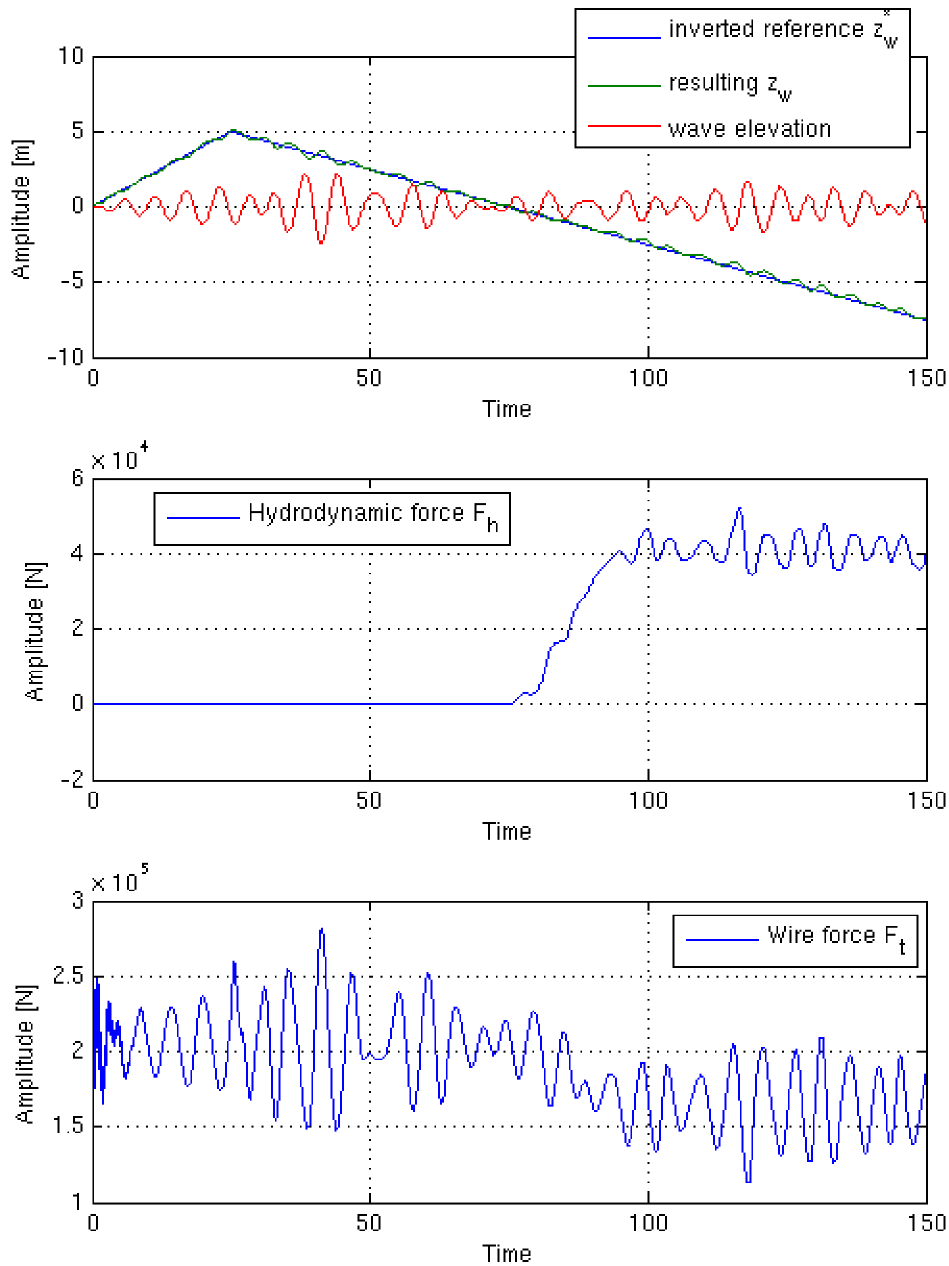

Figure 5.12: Sea state 5 with control. The reference signal is inverted in order for it to be comparable with the resulting distance.

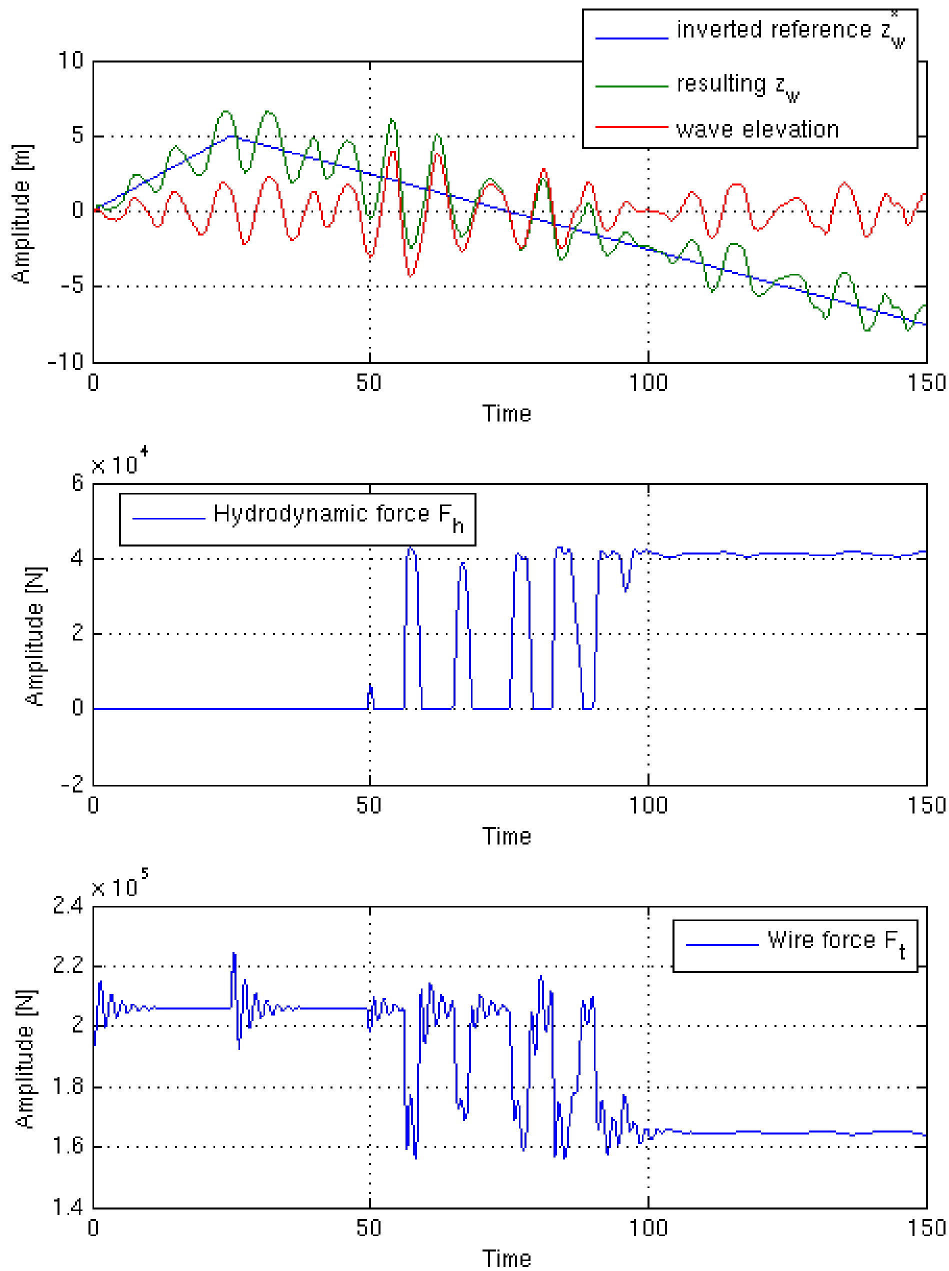

Figure 5.13: Sea state 6 without control. The reference signal is inverted in order for it to be comparable with the resulting distance.

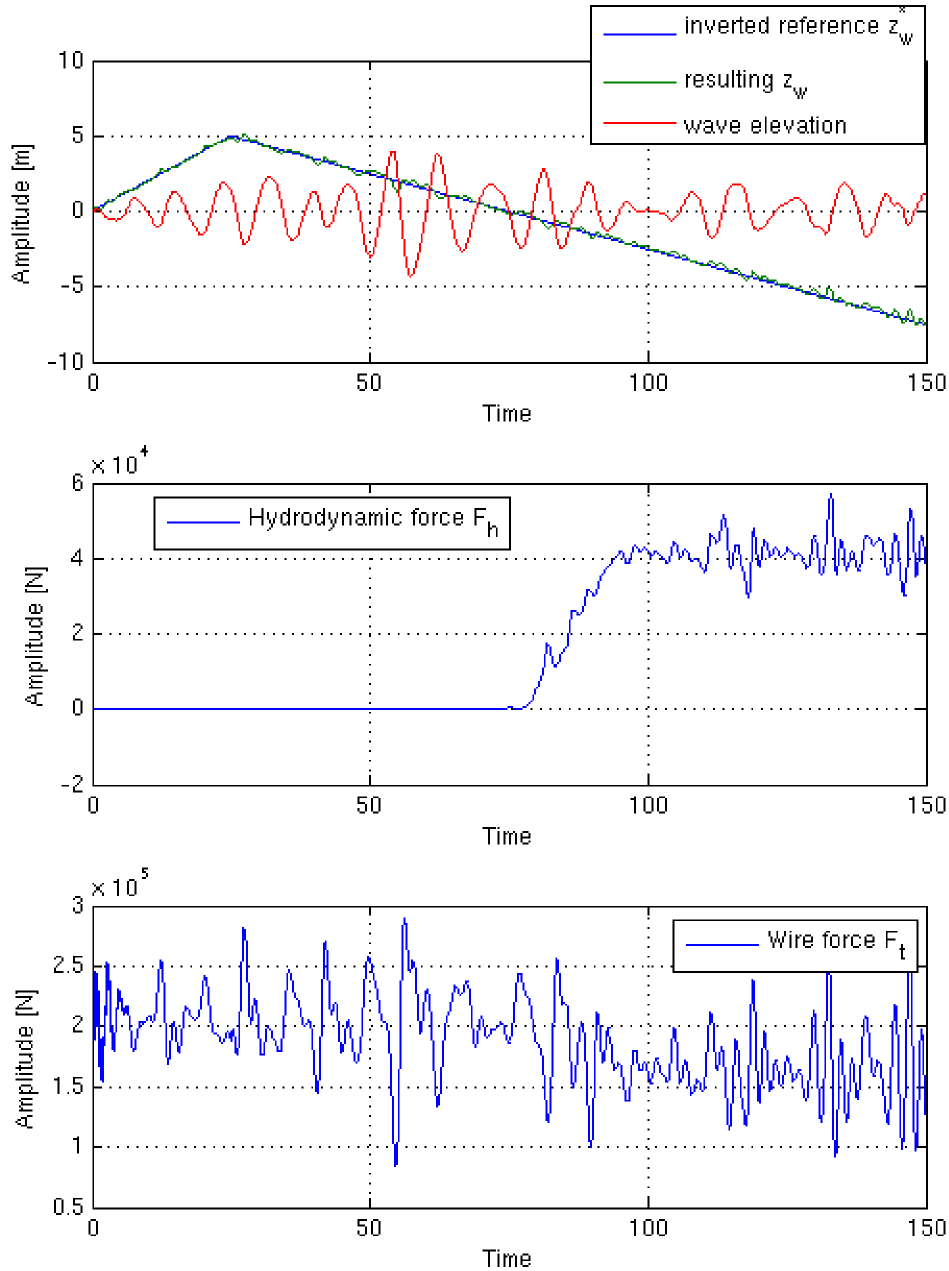

Figure 5.14: Sea state 6 with control. The reference signal is inverted in order for it to be comparable with the resulting distance.

5.7 Simulation Results and Comments

In this section the simulation results are discussed and some comments are made on the validity of the results.

The simulations are somewhat inconclusive as to what is the best of wave synchronization and no motion control. If the object is to reduce hydrodynamic loads F_h on the payload, it is advantageous to use wave synchronization. However, it is seen from the simulation that the wire force F_t increases when wave synchronization is turned on, especially for high sea states. In the splash zone, the simulations including wave synchronization perform better and is smooth compared to the simulations without motion control. This suggests that the use of a blending between wave synchronization and heave compensation is a good idea, as done in [13, 17, 18] with good results. The idea of blending different compensation modes should be further examined for cranes working over the side of the vessel.

In the simulations without wave synchronization a ringing is seen in F_t at about time 25, this is probably caused by the discontinuity in the reference, when a sudden shift from negative to positive slope in the reference is performed. The ringing also occurs when the sea surface slams into the payload, causing a sudden drop or rise in wire tension. This is not desired, as the wear and tear on the wire is increased.

The simulations performed may be inaccurate, since it is not known which gain is the optimal one for getting realistic behaviour of the winch. No power calculations has been performed, and it may be that the power-consumption or the rate-of-change in the motor position and speed is unrealistic.

Another aspect not included in the simulation is the pendulum motion of the payload caused by surge, sway, roll and pitch motion of the vessel, as well as cranes containing multiple DOFs for motion compensation. These systems are very complex, and contain coupled dynamics. It is an interesting problem which should be examined further.

6 Conclusions

It is confirmed that using wave synchronization as a means to reduce hydrodynamic forces on the payload in the splash zone is a good idea. It is seen that the wave synchronization systems used for motion compensation of moon pool cranes are also applicable in cranes operating over the side of the vessel. However, a transition phase between wave synchronization and heave compensation is needed, as the wire force is higher outside the splash zone with wave synchronization control than without. The hydrodynamic force is on the other hand positively affected.

When it comes to the winch speed and power required, more work has to be done to find realistic conditions. The most plausible way of improving the response of motion compensation systems is probably to improve the control laws and include actuation of more DOFs in the motion compensation. There are many different crane designs to consider, and more research should be performed in this area.

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