A Cutter Suction Dredger in Operational Condition
Implementation of a Cutter Force Function in a dynamic multi-body model

V.B. Bakhuizen
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Implementation of a Cutter Force Function in a dynamic multi-body model

Master of Science Thesis

For the degree of Master of Science in Offshore- and Dredging Engineering at Delft University of Technology

V.B. Bakhuizen

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Faculty of Mechanical, Maritime and Materials Engineering (3mE)
Delft University of Technology
Abstract

Modern large Cutter Suction Dredgers are often suitable for large-scale dredging projects since they are able to directly pump an almost continuous flow of dredging material over large distances (over 10 km) to disposal sites. Moreover, the excavation process, which utilizes anchors, swing wires, a cutter head and a spud, enables the CSD to excavate sand and tough rock masses with a relatively high accuracy.

A drawback of the relatively stiff mooring of the CSD is that the ship often encounters problems in certain wave conditions. Depending on wave direction and period, even significant wave heights between 1m and 1.5m may cause inoperable conditions. Several models exist which study the dynamic effects of the mooring system, when the cutter is not being operated. These models have led to innovations in the development of new spud mooring systems which are more reliable and allow the operators to safely work up to the operability boundaries.

From the desire to study innovative concepts for the cutter ladder as well to improve the range of operability, IHC has offered a thesis project to examine a dynamic simulation program of a CSD in operational condition. This project is based on the program Dynamic Operations in Dredging and Offshore (DODO) which couples the hydrodynamic solutions of a diffraction solving algorithm (DIFFRAC) to multi-body dynamics. The RK4-method is used for the numerical integration of the equations of motion. Although IHC already has several highly detailed cutting models, none of them were used since early on in the project it became clear that they were unsuitable in combination with the multi-body models of CSDs in DODO. Especially the level of detail and the scale size of these programs eventually led to the idea of developing a new and computationally cheaper model. This new Cutter Force Function is based on a case study.

The model for the case study is constructed according to a conventional multi-body method in which the cutter ladder body and the floater body are connected to each other via constraints. The hydrodynamic coefficients and the Froude-Krylov wave forces of the external DIFFRAC solution are applied to the floater body. The Morison forces are applied to the cutter ladder body where it is assumed that the inundated ladder oscillates in a static medium. The linearized stiffness matrices of the system consist of swing wire and spud mooring stiffness. The transfer functions of the Linear Time Invariant-system are studied to verify the system and to act as a stepping stone for future simulations that include the Cutter Force Function.

In the Cutter Force Function an orthogonal spacial discretization is used for a cutter and a seabed matrix which store a template of force unit vectors and the instantaneous surface
height of the seabed, respectively. With an elementwise overlay method the cut volumes and their according force vectors are determined for each time step. Two-dimensional rock cutting theory is used. The Cutter Force Function provides a framework to which clay- and sand cutting models can be added as well as other effects such as bulldozing forces, snow-plough effects, etcetera.

Several time domain simulations have been run to verify the functionality of the Cutter Force Function. The primary results indicated that the function gave adequate signal forms although their amplitudes were not limited to a maximum power. Due to the fact that this could generate energy in the system at moments at which it should not be physically possible, a feedback control loop based on the cutter drive torque should be implemented. The LTI-model of the CSD and the transfer functions are ready to be used for the study of implementing PID-controls on swing wires.
Preface and Acknowledgements

This report describes the graduation project as part of my thesis to obtain the Master of Science degree in Offshore- and Dredging Engineering.

As a young sailor of eight years old I discovered my passion for water from my Optimist. I always wanted to sail better and faster and wanted to understand the forces which made it happen. The first time I saw dredging vessels work on the Dutch coastline it sparked my interest. I had barely begun with my Bachelor of Applied Science in Mechanical Engineering and I already knew that I wanted to continue my studies with the Master Offshore and Dredging Engineering at the Delft University of Technology.

During my MSc program I worked an average of 12 hours per week at Royal IHC on the standardization and the development of the Beaver Series (standard dredging equipment). I have had a very informative, and useful time at the Engineering Department. Here I discovered that I had the same yearning to understand the forces acting on dredgers as I did with sailing boats. The R&D group MTI of Royal IHC gave me the chance to research the dynamics of a dredger for my thesis.

I would like to thank Tonie van der Vlies from Royal IHC for helping me find an interesting project and getting me in contact with the right people at MTI. I also would like to thank Peter de Graaff from Royal IHC and Amir Blanken from MTI for their effort to help me understand the countless remarkable aspects of dredging and modeling. Cheers to my co-students at MTI, who have become good friends of mine. Your help was much appreciated and we have had great times together.

Soon I found out that Sape Miedema was cut-out to be the chairman of my graduation committee. I would like to thank Sape for sharing his advice and knowledge in cutting theory and numerical modeling. I also enjoyed the discussions on many other topics that we had.

A special thank you is reserved for my family and friends. I am grateful for their support and encouragement throughout my study.

I wish you an interesting read.

Delft, November 2018

Vincent
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Dredging engineering comprises land reclamation, port construction, flood control, mining, and many other activities in coastal- and inland waters around the world. Cutter Suction Dredgers (CSDs) are commonly used for dredging projects because of their efficiency, precision, and ability to dredge relatively strong cohesive material. The environmental conditions to which they are exposed can be very challenging. Ship building companies and dredging contractors are urged to mitigate the risks that are caused by these conditions. With this in mind, technical specifications are documented. One of these is the ‘workability’.

The workability gives an uptime percentage with respect to the total project duration under prevailing environmental conditions. This is tested for several arbitrary values, such as safety of personnel and design limits of the structure. In order to quantify this, several operability limits are determined. Some limits are notoriously related to wave induced motions, forces on the mooring system, and forces on the excavating structure.

A workability assessment for a CSD therefore entails a breakdown of a wave climate (belonging to the project area in question) into a number of sea states and their respective probabilities of occurrence. On the basis of a numerical model of the CSD its operability limits are checked on exceedance to finally retrieve the uptime percentage. To be able to simulate the behavior of a CSD in operational condition, a model for the soil-structure interface of the excavator is essential.

This chapter provides some background on the numerical modeling of CSDs in waves. This is followed by a problem definition, a set of research goals, and eventually a description of the research approach and the project outline.

1-1 Background of Numerical Modeling

The motions of a CSD in sea waves have been studied for a couple of decades. In the early 1980s J.E.W Wichers wrote a paper ‘On the Forces on a Cutter Suction Dredger in Waves’ [7], in which two separate working conditions were distinguished: the mooring condition and
the operational condition. To obtain insight in the operability limits of these conditions the equations of motion were numerically solved. Wichers stated that the mooring condition could be accurately linearized, therefore a Frequency Domain analysis was sufficient.

The operational condition of a CSD however, was dominated by the cutting force. According to Wichers, the restoring force due to the cutter impression in the seabed in longitudinal, transverse and vertical direction had non-linear and time dependent load-compression characteristics, while coming free from the seabed the restoring force had to be zero [7]. Because of this non-linear effect the equations of motion had to be solved in the time domain for which the well known Cummins equation was used (see Section 2-1). Wichers’s research led to the program DREDSIM, which was developed in consultation with Maritime Research Institute Netherlands (MARIN).

Meanwhile, during the 1980s at The Delft University of Technology the DREDMO program was developed [6][8]. DREDMO solved the equations of motion in the time domain and obtained the hydrodynamic coefficients, retardation functions and frequency domain wave load series from the SEAWAY program which was based on strip theory. Extensive research was done to be able to incorporate the cutter force in DREDMO. Instead of using non-linear springs, as was done by Wichers, a cutting theory was developed and the cutting forces were integrated over the excavating part of the cutter head. A short overview of rock cutting theory can be found in Section 2-3.

Late 1990s the program CUDAS was developed, which was run in the Matlab environment [9]. This software package was used by ship building company IHC to perform dynamic simulations for the newly designed flexible spud guard system [10]. This flexible spud guard system comprises hydraulic cylinders and valves to release pressure when the load on the spud is too close to the design limits. Nowadays most large CSDs are equipped with such mechanisms to prevent spud overload. Again, the cutter force was modeled as non-linear springs so these models did not involve the cutting theory as described in Section 2-3.

In the first decade of this century MARIN started the aNySIM-pro project. The program aNySIM offered a simulation environment for combined DP-systems (Dynamic Positioning), multi-body modeled lifting operations, and multiple floater hydrodynamic problems. One of the commercial aspects of the aNySIM package was that partial licenses were offered. This way the industry only had to buy licences for their own specific use cases. This was utilized by IHC later.

In the early 2010s the Matlab-based program Dynamic Operations in Dredging and Offshore (DODO) was developed by IHC Merwede [11]. This tool was created to evaluate dredging-, lifting-, and many sorts of other dynamic equipment in their early design phase as well as more refined design evaluation later on in the product life cycle. The philosophy of the creators of DODO was to make a versatile and modular, multi-body solving program as an expansion of aNySIM. In the simulation environment of DODO the operability limits of a CSD can also be explored.

1-2 Problem Definition

Large CSDs with water displacements around 10.000 ton and more, already exceed their operability limits in moderate sea states. The paper of Overhagen et al. describes limits for
such CSDs at significant wave heights ($H_s$) of about $0.5m < H_s < 1.5m$ with peak periods ($T_p$) of $7s < T_p < 9s$ at 15m water depth [12]. To obtain insight in this problem an adequate numerical model is required.

This limited workability is partly because of the spud mooring system which connects the CSD to the seabed with a certain stiffness (explained in Appendix A and Section 4-4). Numerical models in CUDAS and DODO have led to an improvement of the spud mooring system. These models represented the CSD in mooring condition in which the cutter-to-soil interaction was left out. A hydraulic spud-guard system has been developed which prevents spud bending overload due to wave induced motions. This safety mechanism gives confidence to push the operability limits of the spud, yielding a higher workability of the CSD in general.

The other cause of this modest workability lies with the cutter ladder (explained in Appendix A) which is also a stiff component that makes contact with the seabed. The ability of the cutter ladder to compensate vessel motions in operational condition is very limited. Appendix B shows a concept which could improve the operability of the cutter ladder [13]. A dynamic feasibility study for this concept is desired but at the start of this project DODO missed a suitable force module that adds cutting theory to the CSD model. To investigate the dynamics of a CSD in operational condition this cutter-to-soil interface must be modeled.

Royal IHC has several stand-alone models to calculate cutting forces. However, these have been set up to optimize the geometry of the cutter head and its respective production. This is not quite the same goal as obtaining a force signal to implement in the considered multi-body models of relatively large scale CSDs. The level of detail and thus the amount of computational work of these simulations cause them to be inefficient for this purpose. A new function for the cutter-to-soil interaction which is both adequate and computationally feasible needs to be developed.

### 1-3 Research Goal

The goal of this master thesis is to incorporate the cutter-to-soil interface in a multi-body, dynamic model of a CSD. The main research question to be answered is:

“How can the cutter-to-soil interface be adequately modeled and verified to make a reliable statement about workability of a CSD in operational condition?”

This encompasses the following research questions:

- How can the cutter-soil intersection be adequately determined?
- What features does the Cutting Force Function require?
- What features does the CSD model require?
1-4 Approach and Project Outline

The project outline is summarized below and visualized by a flowchart in Figure 1-1. A case study will be done for a single dredging depth and three wave directions. All relevant model details will be described in Chapter 4. For a series of sea states the moored condition will be investigated. Results of the FD-analysis can be found in Chapter 5. Eventually some Time Domain simulations are run for the operational condition. Results can be found in Chapter 6.

- A **Frequency Domain** analysis is performed for Model 0, representing the mooring condition, in order to get a feel for the motions of the system. Wichers stated that the motions of a cutter head in operational condition are generally smaller than the motions of a cutter head in mooring condition [14]. However, one can argue if Wichers’s statement is justified. Discussion on this matter can be found in Chapter 7. Model 0 can be sufficiently linearized so it is suitable for Frequency Domain analysis.

- The **Cutter Force Function** is implemented in the model. Chapter 3 shows how this is done. Although the CSD in reality swings around its spud from port to starboard and back, it is modeled with a constant heading. The swing effect is modeled by moving a seabed matrix through the model space. This is illustrated in Figure 1-1.

- Eventually, **Time Domain** simulations are done including cutting force. Model 1 represents a dredger with conventional ladder hoisting system while it is in operational condition. Because of the non-linear nature of the cutter force the operational condition is subjected to a Time Domain simulation.

The goal of this project is to prepare DODO for dynamic studies of CSDs in operational condition. This provides the opportunity to study different concepts that may improve the operability limits of the cutter ladder.

![Figure 1-1: Left: Illustration of the model of a CSD. The heading of the CSD is constant while a seabed matrix passes through the model space with haulage speed $v_h$. Right: Project outline](image-url)
Chapter 2

Engineering Disciplines

For this project the dynamic behavior of the CSD is simulated in DODO. Three engineering disciplines need to be combined into one model. This chapter provides an overview of the disciplines that are involved (see Figure 2-1).

1. The largest component in this system is obviously the floater. The floater is modeled as a rigid body which is subject to hydrodynamic wave forces. These hydrodynamic forces strongly depend on the wave climate of the CSD project location. The **Hydrodynamic Theory** is discussed in Section 2-1.

2. DODO itself is based on **Multi-Body Theory**. This makes it possible to connect a system of rigid bodies, constraints, and force modules to the floater. Hence, the cutter ladder can also be modeled as a rigid body which is connected to the hull by a revolute joint. To understand how the equations of motion are solved in DODO, one must know the working principles of multi-body theory. This is discussed in Section 2-2.

3. As was briefly addressed in Chapter 1, the forces on the cutter ladder in operational condition are non-linear. The cutter-to-soil interface may thus play an essential role in the model. The newly developed force module in DODO involves **Rock Cutting Theory**, this is discussed in the Section 2-3.

![Figure 2-1: The combined disciplines for a simulation of a CSD in DODO](image-url)
2-1 Hydrodynamic Theory

This section addresses some basics of the hydrodynamic theory that are used for the external wave forces on the rigid floater. The following aspects are discussed:

1. To perform a workability study for a project, the prevailing wave conditions can be derived from **Long Term Wave Statistics**, see Section 2-1-1. One area is chosen for this case study.

2. All waves in the model are assumed uni-directional, though in reality the angle of incoming waves can be quite diffused. The wave signal is based on a spectrum. More information on the chosen **Wave Spectrum** can be found in Section 2-1-2.

3. A well known formula of the wave forces on a floating object in the time domain was developed by W.E. Cummins [15]. The **Cummins Equation** is a second order Ordinary Differential Equation (ODE), described in Section 2-1-3.

4. This Cummins equation uses **Hydrodynamic Coefficients** which are plugged into the mass and damping terms of a linear ODE. This is done for all 6 degrees of freedom of the floater, resulting in the set of equations of motion. Section 2-1-4 explains how these coefficients are determined for DODO.
2-1-1 Long Term Wave Statistics

The Global Wave Statistics (GWS) database [1] provides probabilities of meeting certain combinations of wave height, period and direction. This database covers relatively large areas, as shown in Figure 2-2.

![Figure 2-2: Part of the world map, indicating GWS areas [1]](image)

Some of this data is useful for the case study of the CSD model. Please note the following:

- The data are retrieved from visual observations. It might be adequate to use other resources, such as hindcast data and instrument measurements. Especially local effects may cause the wave climate to differ from the GWS data. Sea states might be more forgiving because of local shelter effects. On the other hand, sea states could also be worse than indicated by GWS due to local effects such as shoaling and refraction of waves. For the fictitious workability study of this thesis the data give a good impression and local circumstances are not considered.

- Both annual and seasonal data can be found in the GWS database. The seasonal data are assumed most appropriate for dredging projects since the CSD does not have year-round exposure.

- The mooring lay-out and dredging method is adjusted to the dominant wave direction. The observations in all directions are therefore used. The waves are assumed uni-directional.

- The data for areas 11 and 85 are shown in Figure 2-3. Note that the scatter plots show quite different wave statistics. Area 85 (along the coast of Namibia and South Africa) is not sheltered from oceanic swell whilst Area 11 (North Sea) is. Because of this, Area 85 shows more pronounced low frequency waves than Area 11. Each area around the world has a unique wave spectrum due to meteorological and topographical circumstances. A well known spectrum for the North Sea (Area 11) is the JONSWAP spectrum. For Area 85 a Pierson-Moskowitz spectrum should be more appropriate.
As was already mentioned in Section 1-2, the limiting significant wave heights of the CSD in this case study are in the range of $0.5m < H_s < 1.5m$ with peak periods of $7s < T_p < 9s$ [12]. Any sea state with $H_s \geq 2m$ will definitely cause trouble. From Figure 2-3 can be readily seen that for Area 11 this is the case about 30% of the time during the period June to August. For Area 85 this is much worse, with a downtime over 75%.

For significant wave heights $H_s < 2m$ the operability of the CSD also depends on the peak period $T_p$ of the sea state. For these sea states simulations should point out which combinations of $H_s$ and $T_p$ represent acceptable wave conditions. Note that the GWS scatter data for $H_s < 2m$ are relatively coarse for such analysis, since it only provides two bins for $H_s$. However, the operability scatter could look like Table 2-1. For a full blown workability assessment of a CSD it would be neat to have $H_s$ divided into more bins.

*Table 2-1*: Example of operability scatter diagram (not based on results!), indicating which combinations of $H_s$ and $T_p$ could be acceptable for the CSD

<table>
<thead>
<tr>
<th>$H_s$ [m]</th>
<th>1.50</th>
<th>0.50</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_p$ [s]</td>
<td>4.5</td>
<td>5.5</td>
</tr>
<tr>
<td>3.5</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>4.5</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>5.5</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>6.5</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>7.5</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>8.5</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>9.5</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>10.5</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>11.5</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>12.5</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>13.5</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Multiplying the binary values of the operability scatter diagram with the corresponding scatter diagrams of Figure 2-3 will result in the workability of the CSD. Suppose the operability scatter for both areas are the same as Table 2-1 (although this is not likely since the sea states have different wave spectra). The workability for Area 11 would be about 65% and the workability of Area 85 would be about 10%.

Please note: The case study for this CSD model will only be done for Area 11. Results are shown in Chapter 5.

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2-1-2 Wave Spectrum

The sea can be modeled as a stochastic sum of waves using the superposition principle [2], this is illustrated in Figure 2-4. This is founded on the Linear Wave Theory of individual harmonic components [2][16]. The following spectra are commonly used in the industry:

- **The Pierson-Moskowitz spectrum** (a specific type of Bretschneider spectrum) is a good representation of open seas, having pronounced swell waves of relatively low frequency. This type of wave spectrum would be quite suitable for Area 85 of Figure 2-2. For the case study in this thesis the Pierson-Moskowitz spectrum **is not used**.

- **The Joint North Sea Wave Project (JONSWAP) spectrum** is a good statistical representation of wind generated (young) sea states. It is adequate when the waves are not yet fully developed, which is when the wind speed is larger than the wave speed of the longest waves. It is characteristic for the North Sea because it represents a mix of wind generated sea waves and oceanic swell waves. The **JONSWAP spectrum is used** for this case study since it is appropriate for Area 11 of Figure 2-2.

\[
S_\zeta(\omega) = 320 \frac{H_s^2}{T_p^4} \omega^{-5} \exp\left\{ -1950 \frac{\omega^4}{T_p^4} \right\} \gamma^A \tag{2-1}
\]

The peakedness factor has exponent

\[
A = \exp\left\{ - \left( \frac{\omega_p}{\sigma \sqrt{2}} - 1 \right)^2 \right\} \tag{2-2}
\]

The circular frequency \( \omega_p \) at the spectral peak is

\[
\omega_p = \frac{2\pi}{T_p} \tag{2-3}
\]

A step function is used to change the shape on the left and the right side of the spectral peak

\[
\sigma = \begin{cases} 
0.07, & \text{for } \omega < \omega_p, \\
0.09, & \text{for } \omega > \omega_p.
\end{cases} \tag{2-4}
\]

With this spectrum a complete description of the surface elevation is given, in a statistical sense. The signals for the wave elevation and linear hydrodynamic forces on the floater can be obtained from this spectrum.
2-1-3 Cummins Equation

The equations of motion of the CSD can be represented with the Cummins equation as the following set of equations [8] \((k = 1...6)\):

\[
\sum_{j=1}^{6} \left\{ (M_{kj} + m_{kj}) \cdot \ddot{x}_j(t) + \int_{-\infty}^{t} K_{kj}(t - \tau) \cdot \dot{x}_j(\tau) \cdot d\tau + C_{kj} \cdot x_j(t) \right\} = F_k(t) \tag{2-5}
\]

With retardation function

\[
K_{kj}(t) = \frac{2}{\pi} \int_{0}^{\infty} b_{kj}(\omega) \cos(\omega t) d\omega \tag{2-6}
\]

and added mass coefficients

\[
m_{kj} = a_{kj}(\omega) + \frac{1}{\omega} \int_{0}^{\infty} K_{kj}(\tau) \sin(\omega \tau) d\tau \tag{2-7}
\]

The retardation function requires the potential flow field to be calculated for all 6 degrees of freedom of the floater at desired wave angles.

2-1-4 Hydrodynamic Coefficients

The added mass and hydrodynamic damping are derived from an impulse response potential flow field around the floating object. The damping terms are calculated with a convolution integral. The hydrodynamic coefficients and first order wave forces can be calculated with aNySIM, which uses the DIFFRAC solver of MARIN.

The hydrodynamic coefficients are calculated without speed or current in water. Low waves and small motions are assumed to be able to linearize the free surface condition of the water and the boundary condition on the floater. The mean wetted floater surface is divided into panels to compute pressures and water velocities of the potential flow field. These are integrated over the surface to obtain hydrodynamic coefficients and first-order wave forces for a range of wave frequencies with unit wave amplitude [17].

2-1-5 Hydrodynamic Theory - Summary

A background on long term wave statistics is provided and the choice of a wave spectrum is justified. The basic form of the equations of motion of a floater were briefly described. Also is explained how the mass- spring and damping terms of the ordinary differential equations can be obtained to complete the Cummins equation.
2-2 Multi-Body Theory

The following aspects of the CSD as multi-body model are discussed in this section:

1. To be able to describe the equations of motion of the system, a body fixed coordinate system is embedded in the center of gravity of each body. The positions and rotations of the System Bodies are stored in a vector according to classical multi-body convention, briefly noted in Section 2-2-1.

2. The equations of motion of a multi-body system are commonly written in the Newton-Euler Notation. This is elaborated in 2-2-2. In this notation the floater can be assigned as first rigid body of the system. Constraint Equations provide a kinematic relation between the bodies of the system. The application of these constraint equations is explained.

3. A linearized equation of motion can be used for an analysis in Frequency Domain, this is shown in Section 2-2-3.

4. For a simulation in Time Domain the set of Cummins equations is substituted into the Newton-Euler notation. This is discussed in Section 2-2-4. In the Time Domain the equations of motion are numerically integrated. The phenomenon of bodies drifting away from their constraints is also discussed, as well as a way to mitigate that during a simulation. Two stabilization methods are mentioned which can mitigate the numerical drift phenomenon that is inherent to multi-body dynamic simulations.

2-2-1 System Bodies

Each body can be located by specifying translational and rotational coordinates of the origin of the body fixed reference system with respect to the global reference system. This vector of coordinates in three dimensional space for body $i$ is denoted as follows:

$$ q_i \equiv [x, y, z, \phi, \theta, \psi]^T_i \quad (2-8) $$

A mechanism with $b$ bodies in a spatial system needs $n = 6 \times b$ coordinates. The overall vector of coordinates for a system can be written as

$$ q = [q_1^T, q_2^T, \ldots, q_b^T]^T \quad (2-9) $$

For the CSD model in DODO this will be:

$$ q = [q_1^T, q_2^T]^T \quad (2-10) $$

2-2-2 Newton-Euler Notation

The motions of a system can be described by Newton-Euler equations, combined with Lagrange multipliers as can be seen in Equation 2-11 [3]. The Equations of Motion of the interconnected system of bodies can be written in the form:

$$ \begin{bmatrix} M & \Phi_q^T \\ \Phi_q & 0 \end{bmatrix} \begin{bmatrix} \dot{q} \\ \lambda \end{bmatrix} = \begin{bmatrix} F_{\text{ext}} \\ \gamma \end{bmatrix} \quad (2-11) $$
Here $M$ is the mass- and inertia matrix of all bodies. The symbol $\Phi_q$ is the Jacobian matrix of the constraint equations. The symbol $\lambda$ represents the vector of Lagrange multipliers and $\gamma$ is the vector of convective acceleration terms of constraints. $F_{ext}$ is the vector of external forces ($F_w$ on floater and $F_{mb}$ on the rest of multi-body system) and $\dot{q}$ is the vector which describes the accelerations of the bodies.

The bodies of the system have kinematic relations. These relations are described by the equations of constraints. For a system having $m$ independent constraint equations and $n$ coordinates the number of DoF’s is: $k = n - m$. This only yields a correct answer when all $m$ constraint equations are independent [3]. Constraint equations can be denoted in the following form:

$$\Phi \equiv \Phi(q, t)$$  \hspace{1cm} (2-12)

The Jacobian matrices are derived as follows:

$$\Phi_q \equiv \begin{bmatrix} \frac{\partial \Phi}{\partial q} \end{bmatrix} ; \quad \Phi_t \equiv \begin{bmatrix} \frac{\partial \Phi}{\partial t} \end{bmatrix}$$  \hspace{1cm} (2-13)

In this project all constraint equations are independent of time, in other words: the kinematic relation between bodies are constant and $\Phi_t = 0$. When the equations of motion are solved the second derivative of the Jacobian matrix with respect to time (Equation 2-15) is kept zero, this results in Equation 2-16.

$$\dot{\Phi} = \Phi_q \dot{q} + \Phi_t = 0$$  \hspace{1cm} (2-14)

$$\ddot{\Phi} = \Phi_q \ddot{q} + \dot{\Phi}_q \dot{q} + \Phi_t = 0$$  \hspace{1cm} (2-15)

$$\Phi_q \ddot{q} = -\dot{\Phi}_q \dot{q}$$  \hspace{1cm} (2-16)

The vector of Langrange multipliers is

$$\lambda = \Phi_q \dot{q}$$  \hspace{1cm} (2-17)

The vector of convective acceleration terms is

$$\gamma = -\dot{\Phi}_q \dot{q}$$  \hspace{1cm} (2-18)

### 2-2-3 Frequency Domain

A multi-body system can be linearized by a finite difference method. The finite difference $\Delta q$ of Equation 2-19 is used to calculate the total linearized stiffness and damping matrices of the system.

$$q = q_{eq} + \Delta q$$  \hspace{1cm} (2-19)

The vector $F_{ext}$ can be divided into a couple of terms, as shown in Equation 2-20. The first term indicates the force at equilibrium, the second term is related to restoring spring stiffness, the third term is related to damping, and the last term is the external force parameter that defines the force.

$$F_{ext} = F_{eq} + \frac{\partial F}{\partial q} \Delta q + \frac{\partial F}{\partial q} \Delta \dot{q} + \frac{\partial F}{\partial p} \Delta p$$  \hspace{1cm} (2-20)

In the case of a moored floater, reacting to waves, this parameter $p$ is the Froude-Krylov force $F_{FK}(\zeta)$ that depends on surface elevation.

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After linearization the equations of motion of the system can be written as Equation 2-21. The hydrodynamic correction terms for the Froude-Krylov force (added mass, and hydrodynamic damping) are on the left hand side of the equation, just as all other restoring spring forces and damping forces on the system.

$$\begin{bmatrix} M_{t,eq} & \Phi_{q,eq}^T \\ \Phi_{q,eq} & 0 \end{bmatrix} \begin{bmatrix} \Delta \dot{q} \\ \Delta \lambda \end{bmatrix} + \begin{bmatrix} B_{t,eq} \Delta \dot{q} + K_{t,eq} \Delta q \\ 0 \end{bmatrix} = \begin{bmatrix} \partial F_{FK}/\partial \zeta \end{bmatrix} \Delta \zeta$$ \hspace{1cm} (2-21)

The Froude Krylov force on the right hand side contains the external loads in all six degrees of freedom for two bodies:

$$F_{FK}(t) = \begin{bmatrix} F_{FK,1} \\ F_{FK,2} \end{bmatrix}; \quad F_{FK,1} = \begin{bmatrix} F_x \cos(\omega t + \epsilon_x) \\ F_y \cos(\omega t + \epsilon_y) \\ F_z \cos(\omega t + \epsilon_z) \\ M_{xx} \cos(\omega t + \epsilon_{xx}) \\ M_{yy} \cos(\omega t + \epsilon_{yy}) \\ M_{zz} \cos(\omega t + \epsilon_{zz}) \end{bmatrix}; \quad F_{FK,2} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix}$$ \hspace{1cm} (2-22)

In the frequency domain the equations of motion become:

$$\begin{bmatrix} -M_{t,eq}\omega^2 + B_{t,eq}\omega + K_{t,eq} & \Phi_{q,eq}^T \\ \Phi_{q,eq} & 0 \end{bmatrix} \begin{bmatrix} Q(\omega) \\ \Lambda(\omega) \end{bmatrix} = \begin{bmatrix} F_{FK}(\omega) \\ 0 \end{bmatrix}$$ \hspace{1cm} (2-23)

The vectors $Q(\omega)$ and $\Lambda(\omega)$ are both size 1x12. The matrices $M_t$, $B_t$, $K_t$, and $\Phi_{q,eq}$ have size 12x12, so 12 transfer functions to harmonic waves can be calculated. With Equation 2-24 the transfer function can be calculated. This contains both real and complex numbers.

$$\begin{bmatrix} Q(\omega) \\ \Lambda(\omega) \end{bmatrix} = \begin{bmatrix} -M_{t,eq}\omega^2 + B_{t,eq}\omega + K_{t,eq} & \Phi_{q,eq}^T \\ \Phi_{q,eq} & 0 \end{bmatrix}^{-1} \begin{bmatrix} F_{FK}(\omega) \\ 0 \end{bmatrix}$$ \hspace{1cm} (2-24)

The vector $Q(\omega)$ contains the transfer functions for all 12 responses of the two bodies. Each of these response transfer functions is from now on in this document referred to as $H_{R,\zeta}(\omega)$ (subscript $R$ indicates the type of response, e.g. surge).

### 2-2-4 Time Domain

The Cummins equation can be substituted into Equation 2-11 by assuming that the floater is the first rigid body of the system of interconnected bodies. [11]. In Equation 2-5 the state of the floater is denoted as $x_j$, wit $j = 1, 2, ..., 6$. In the multi-body notation of Equation 2-11 it is $q$.

$$\begin{bmatrix} (M + m) & 0 \\ 0 & M_{mb}(q) \end{bmatrix} \Phi_{q}^T(q, \dot{q}) \begin{bmatrix} \dot{q} \\ \lambda \end{bmatrix} = \begin{bmatrix} F_w(t) \xi - \int_{-\infty}^{t} K(t - \tau) \cdot \dot{q}(\tau) \cdot d\tau - Cq + F_s(q, \dot{q}) \\ F_{mb}(q, \dot{q}) \end{bmatrix}$$ \hspace{1cm} (2-25)

The added mass is placed on the left hand side of the equation. The external wave force, the hydrodynamic damping term and the hydrostatic spring term are placed on the right hand side. The force $F_s(q, \dot{q})$ represents all forces that connect the floater to the rest of the multi-body system.
Numerical Integration

For the numerical integration of the equations of motion the Runge-Kutta 4 method is used. [18].

Stabilization

The system of differential equations from Equations 2-11 and thus also 2-25 suffer from constraint stabilization errors. As only the acceleration constraint equations have been imposed, the positions and velocities provided by the integrator suffer from a ‘drift’ phenomenon [19].

There are two methods in DODO to cope with this problem: the Baumgarte stabilization method and the stabilization by correcting with the least-squares distance. Note that constraint stabilization is equivalent to a numerical transfer of energy in the model. This may cause the numerical solution to differ from the exact solution.

- **Baumgarte’s stabilization method** is described in [20] and briefly addressed below. Instead of solving $\Phi = 0$ from Equation 2-15 the following condition is upheld:

$$\dot{\Phi} + 2\alpha \Phi + \beta^2 \Phi = 0$$

(2-26)

In which coefficients $\alpha$ and $\beta$ are chosen positive constants. The convective acceleration terms $\gamma$ are iterated

$$\gamma(q, \dot{q}) = \gamma(q, \dot{q}) - 2\alpha \dot{\Phi} - \beta^2 \Phi$$

(2-27)

- Alternatively, **Stabilization by least-square correction** can be used. For this the Gauss-Newton method is used. A detailed explanation of this stabilization technique can be found in lecture notes of A. Schwab. [21]

The effect of stabilization feedback is illustrated in the figure below. For nonzero $\alpha$ and $\beta$ the solution oscillates about the exact solution. The feedback is critically damped if $\alpha = \beta$, which usually results in the quickest stabilization. The higher $\alpha$ and $\beta$ are chosen, the stronger the feedback is.

![Figure 2-5: Schematic representation of the exact and numerical solutions with Baumgarte stabilization][1]

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2-2-5 Multi-Body Theory - Summary

The Newton-Euler notation and the implementation of the Cummins equation in DODO have been described. The linearized equations of motion have been discussed, this is used to calculate the transfer functions of the system to perform a Frequency Domain analysis. The resulting equations of motion in the Time Domain can be numerically solved by a Runge-Kutta algorithm. The concept of drift (numerical error) and two stabilization techniques have been addressed.
2-3 Rock Cutting Theory

Many factors play a role in the cutting process of a CSD. This section embraces a simplification of the rock cutting process, though it is still considered adequate enough for implementation in the equations of motion of the CSD. The following topics are covered:

1. Several types of rock can be encountered in the seabed: igneous rock, sedimentary rock and metamorphic rock. These types of rocks have very different properties, can occur at different places and each have their own history. Regarding the excavation process, the most important characteristic of this rock is the Failure Envelope. This is discussed in more detail in Section 2-3-1.

2. For the modeling of cutting forces it is very important to describe The Cutting Process on cutter head scale. Section 2-3-2 describes how the cutter head rotates, what a typical haulage speed is, what the geometry of the face of the seabed is and what effects the joint spacing and -orientation of the rock mass may have.

3. Zooming in, the Individual Tooth Cutting Theory can be used to calculate time averaged cutting forces on teeth of the cutter head. Several cutting theories from literature are addressed. In the end a specific energy is determined for the cutting of a small volume of soil. Zooming out again, this specific energy can be used to estimate the required energy for the CSD to excavate.

2-3-1 Failure Envelope

A cutting tool induces stress tensors in the rock mass. These stresses build up until the boundaries of the failure envelope are reached. Some key characteristics of the failure envelope are indicated in Figure 2-6 and pointed out below:

- The BTS indicates the minimum principal stress of the failure envelope. Region I in Figure 2-6 indicates the stress states of the rock where both tensile and shear failure may occur.
- The UCS can be used to find the failure envelope of the unconfined rock mass.
- Region II in Figure 2-6 indicates the stress states of the rock where failure is brittle shear. Around the Mogi criterion lies the brittle-ductile transition point, indicated III in Figure 2-6. Mogi [22] found that for silicate rocks this ratio of the maximum principle stress over the minimum principle stress was \( \sigma_1/\sigma_3 = 3.4 \). For carbonate rocks he found \( \sigma_1/\sigma_3 = 4.2 \). Up to the Mogi criterion the microscopic structure of the rock is mostly intact.
- For increasing normal stresses on the boundary of the failure envelope above the Mogi criterion more and more damage is done at microscopic level. This ductile cataclastic failure, region IV in Figure 2-6, causes the failure envelope to drop.
- Hydrostatic triaxial compression tests on intact rock have indicated that Hydrostatic Compressive Strength (HCS) can reach few times the UCS. Miedema [5] gives an example of \( HCS + UTS = 3.5 \cdot UCS \). The sharp decline of the failure envelope towards the
HCS is called the volumetric cap. Note that with this decline of the failure envelope also the internal friction angle decreases. Although it does become very small, the internal friction angle of the crushed rock does not necessarily go to zero. A failure envelope beyond the HCS can be defined by the remaining internal friction angle of the crushed material.

A large part of the brittle region in the failure envelope can be approached by the Mohr-Coulomb criterion, where $\phi$ is the angle of internal friction and cohesion coefficient $c$ is the shear strength at zero normal stress

$$\tau = c + \sigma \cdot \tan \phi$$  \hspace{1cm} (2-28)

in which

$$c = \frac{UCS}{2} \cdot \frac{1 - \sin \phi}{\cos \phi}$$  \hspace{1cm} (2-29)

The failure of rock can have a ductile or brittle mechanism. The ductility number of the rock mass (strength ratio) $m = UCS/BTS$ indicates which failure mechanism is dominant. An overview of ductility numbers can be found in Appendix H. A rough assumption, without specifying a tool geometry, can be made that for $UCS/BTS > 15$ the failure is brittle and for $UCS/BTS < 9$ the failure is ductile. This was stated by Gehring in 1987, cited by Helmons [23].
2-3-2 The Cutting Process

Cutting of rock includes brittle failure, this causes highly frequent and sometimes intermittent forces on the cutter teeth. Chip formation of the cut rock and imperfections in the rock mass imply that cutting teeth make erratic changes to the rock geometry. Because of this the cutter force is noisy, as illustrated in Figure 2-7.

![Figure 2-7: Brittle Force Signal [4]](image)

However, it should not be forgotten that the cutter force is just a part of a system in a bigger picture. A cutter suction dredger is two to three orders of magnitude larger than a single pickpoint. The floater and the cutter ladder are insensitive to the frequency of chip formation.

![Figure 2-8: Scale comparison of cutter teeth, cutter head and CSD. [4]](image)

The rotational speed of the cutter head is for most dredgers around 30 RPM. Common cutter designs have 6 arms. On these arms a number of teeth is mounted, which are staggered for optimal cutting. The cutter in Figure 2-11 therefore has $n_R = 3$ teeth in the same path per revolution. If a power spectral density would be plotted from a cutter force signal, it would probably show distinctive peaks at the arm passing frequencies and even at higher frequencies where the tooth penetration frequencies are. Once again, on the relatively large scale of the CSD model the floater and cutter ladder will filter out such high frequencies so these high frequency effects are neglected.

The rotational speed, the swing speed (haulage velocity) and the number of staggered teeth determine the maximum penetration (cutting depth) of the teeth in the rock. This can be seen in Figure 2-9. The swing speed varies between 5m/min for strong rocks and 20 m/min for weak rocks. In rock, the face is normally 0.5 to 1 times the diameter of the cutter. In very hard rock the face may become less than 0.5 times the diameter [4].
The face stays mostly intact but in case of very small cohesion the face can collapse. Chunks of fractured rock may also fall down in the process. The face (or breach) is visualized in Figure 2-10. The production in cubic meters per hour can be approached by the step size, face height and haulage speed

\[ Q = H_{\text{face}} \cdot L_{\text{step}} \cdot v_h \]  

(2-30)
At the scale of one cutter head the rock mass often has multiple discontinuities, such as a fracture, joint, fault, shear, weak bedding plane or contact that has relatively low tensile strength [4]. Different mechanisms can be characterized in the excavation process of a cutter head in rock mass: ripping, cutting or both ripping and cutting. To what extent these mechanisms occur, depend very strongly on the discontinuities in the rock mass. Often these discontinuities exist in patterns, as illustrated in Figure 2-12. Although not all discontinuities are strictly joints, the term ‘joint spacing’ is loosely used to indicate the frequency of occurrence of discontinuities in rock mass.

Figure 2-12: Discontinuities of rock mass as illustrated by Verhoef [4]

Verhoef addressed that the discontinuity spacing of the rock mass is very important for the resulting forces on a cutter head [4].

- If the joint spacing is larger than the size of the cutter, the cutter is essentially cutting intact rock.
- If the joint spacing is smaller than the effective size of the cutter, but wider than the maximum penetration of a pickpoint, the cutting force depends on the orientation of discontinuity surfaces. This is illustrated in Figure 2-13.
- If the joint spacing is even smaller than the tooth penetration, the cutter is probably excavating fractured rock.

Figure 2-13: Discontinuity orientation and its effects on the overcutting and undercutting process schematized by Verhoef [4]
2-3-3 Individual Tooth Cutting Theory

If the rock is assumed a homogeneous isotropic material, the analytical cutting models for individual teeth can provide a good estimation for the cutting forces. The main cutting mechanism for rock cutting under atmospheric conditions (normal dredging), as described by Miedema [5], is the Chip Type. This process is dominated by the UCS, UTS, BTS, internal friction angle and external friction angle. The Chip Type cutting mechanism is a mix of the shear failure and tensile failure with a crushed zone near the tip of the blade. A 2D illustration of this cutting mechanism is shown in Figure 2-14.

![Figure 2-14: Chip Type cutting mechanism as illustrated by Miedema [5]](image)

Helmons and Miedema explained the phenomena of the Chip Type, which are briefly addressed below [23] [5]:

- **Crushing.** First penetration leads to a crushed zone at the tool tip. Inside the crushed zone the stress state is on or beyond the theoretical HCS of Figure 2-6. At the boundary of the crushed zone the stress state is pseudo-ductile and close to the brittle-ductile transition point on the failure envelope. As the tool moves further into the intact rock the crushed zone will grow proportionally to the penetration depth of the tool.

- **Shear failure.** With the pickpoint moving further, shear failure will occur just outside the crushed zone. This shear failure is governed by Region II of the failure envelope in Figure 2-6.

- **Tensile failure.** Further away on the shear plane, the isotropic stress is smaller. However, one of the principal stresses can become negative such that the shear crack bifurcates into the start of a tensile crack. This bifurcation leads to chip formation.

- Depending on tool geometry and the failure envelope, the brittle-tensile and brittle-shear transitions can be analytically derived. The shape of the failure envelope and the geometry of the tool can be used to find the theoretical planes for shear- and tensile failure. One failure plane, or a combination of failure planes, can be found at which
minimum horizontal force is required for the pickpoint to proceed. At small blade angles \( \alpha \) the (brittle) tensile failure mechanism can be dominant whereas at large blade angles the shear failure mechanism prevails. For cutter heads of CSDs most blade angles are about 60 degrees.

- Based on these findings, the different failure mechanisms can be combined into a **Resulting Cutting Force** on the tooth.

**Crushing failure**

The Chip Type cutting mechanism shows crushing failure near the tip of the pickpoint [5]. It is a pseudo ductile cataclastic mechanism. This means that the compressed rock shows stress-strain behaviour like plastic deformation, region IV in Figure 2-6. It is called pseudo-ductile since crushing is irreversible, destroying the rock structure at microscopic level.

![Figure 2-15: The crushed zone at the tip of a pickpoint](image)

Inside the crushed zone the principal stresses can be beyond the Hydrostatic Compressive Strength. The remaining internal friction angle at this region of the failure envelope is very small. A very simple approximation of the crushing forces on the pickpoint tip would be to integrate the HCS stress over the area as indicated in Figure 2-15. The crushed zone can cover the complete wear flat of the pickpoint.

The crushing stress at the tip of the pickpoint is spread out to the shear and tensile stress surfaces where the chip eventually breaks off. The peak forces (Figure 2-7) of the Chip Type mechanism are caused by brittle failure, not by the pseudo-ductile crushing failure. These peak forces are dominated by the brittle shear and brittle tensile failure mechanisms. The average cutting force can be assumed from these peak forces without going into the details of the crushing type mechanism. The Crushing failure mechanism is therefore not required to model average cutting forces.
Shear failure

The Chip Type cutting mechanism can be partly consisting of shear failure, as indicated in Figure 2-14. The shear failure can be described by Nishimatsu theory [5]. Nishimatsu neglects the crushed zone. A 2D-representation of a shear failure plane is illustrated in Figure 2-16. The pickpoint in Figure 2-16 is moving horizontally. This is an analytical way to determine peak forces. Time averaged forces are 30% to 60% of the peak forces.

![Figure 2-16: Nishimatsu cutting mechanism as illustrated by Miedema (edited) [5]](image)

The horizontal cutting force, in which $\lambda_H$ is the brittle horizontal shear force coefficient and $n$ represents a stress distribution factor. If $n = 0$ the Nishimatsu model becomes the same as the Merchant theory.

$$F_h = \frac{1}{n + 1} \cdot \lambda_H \cdot c \cdot h_i \cdot w \quad ; \quad \lambda_H = \frac{2 \cdot \cos(\phi) \sin(\alpha + \delta)}{1 + \cos(\alpha + \delta + \phi)}$$

(2-31)

The vertical cutting force, in which $\lambda_V$ is the brittle vertical shear force coefficient:

$$F_v = \frac{1}{n + 1} \cdot \lambda_V \cdot c \cdot h_i \cdot w \quad ; \quad \lambda_V = \frac{2 \cdot \cos(\phi) \cos(\alpha + \delta)}{1 + \cos(\alpha + \delta + \phi)}$$

(2-32)

The coefficient $c$ is the cohesion

$$c = \frac{UCS}{2} \cdot \left(1 - \frac{\sin(\phi)}{\cos(\phi)}\right)$$

(2-33)

The angle of the shear plane is found by minimization of the cutting force with respect to $\beta$. This leads to:

$$\beta = \frac{\pi}{2} - \frac{\alpha + \delta + \phi}{2}$$

(2-34)

The specific energy $E_{sp}$ for this mechanism is

$$E_{sp} = \frac{1}{n + 1} \cdot \lambda_H \cdot c$$

(2-35)

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**Tensile failure**

The Chip Type cutting mechanism can be partly consisting of tensile failure, as indicated in Figure 2-14. The tensile failure is a brittle phenomenon which can be described by Evans theory [5]. A 2D-representation of a chip that breaks off due to tensile failure is illustrated in Figure 2-17. The pickpoint in Figure 2-17 is moving horizontally. This is an analytical way to determine peak forces. Time averaged forces are 30% to 60% of the peak forces.

![Figure 2-17: Evans cutting mechanism as illustrated by Miedema (edited)[5]](image)

Evans theory is based on a failure mechanism that depends on the tensile stress in which a sharp wedge or chisel is used. Miedema [5] extended this theory for a pickpoint with wearflat under an angle of $\alpha = \varepsilon$. Note that $\alpha$ is half the blade angle. The angle $\beta$ is determined by using the principle of minimum energy:

$$\frac{dF_c}{d\beta} = 0 \quad (2-36)$$

This leads to the total cutting force

$$F_c = \sigma_t \cdot h \cdot w \cdot \frac{2 \cdot \sin(\alpha + \delta)}{1 - \sin(\alpha + \delta)} \quad (2-37)$$

in which $\sigma_t$ is the tensile strength BTS, $w$ is the width of the pickpoint and $h$ the cutting depth. In this equation, $\delta$ is the external friction coefficient and is assumed to be $\delta = 2/3 \cdot \phi$, where $\phi$ is the internal friction angle. The horizontal and vertical force components of the blade are

$$F_{ch} = F_c \cdot \cos(\alpha) \quad ; \quad F_{cv} = F_c \cdot \sin(\alpha) \quad (2-38)$$

The specific energy $E_{sp}$ for this cutting mechanism is

$$E_{sp} = \frac{F_{ch} \cdot v_{ch}}{h \cdot w \cdot v_c} = \sigma_t \cdot \frac{2 \cdot \sin(\alpha + \delta)}{1 - \sin(\alpha + \delta)} \cdot \cos(\alpha) \quad (2-39)$$

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Brittle tensile - Brittle shear transition

By means of the Flow Type, Shear Type and Tear Type cutting mechanisms, Miedema was able to derive analytical boundaries of brittle tensile and brittle shear failure for a given blade angle [5]:

- The shear failure plane and the $\tau/\sigma$ ratio on this plane are geometrically determined based on the Shear Type mechanism. A linear Mohr-Coulomb failure criterion is assumed. Since this stress state must be on the failure envelope, $\tau_{S1}$ and $\sigma_{N1}$ are known. The point $(\sigma_{N1}, \tau_{S1})$ is exactly the tangent point of the Mohr circle for shear failure, hence the center of this Mohr circle is also known, as well as its minimum principal stress $\sigma_{\text{min},S}$. This is illustrated in Figure 2-18. Please note that only a part of the failure envelope is displayed. Shear failure will occur if the minimum principal stress $\sigma_{\text{min},S}$ is larger than the tensile strength of the rock $\sigma_T$.

$$\sigma_{\text{min},S} > \sigma_T \quad (2-40)$$

Or in absolute values, if:

$$|\sigma_T| > |\sigma_{\text{min},S}| \quad (2-41)$$

- The tensile failure plane and the $\tau/\sigma$ ratio on this plane are also geometrically determined, in this case with the Tear Type mechanism. The Mohr circle for tensile failure can be determined with $(\sigma_{N2}, \tau_{S2})$ in the same manner as above. The same linear Mohr-Coulomb failure criterion is assumed. Tensile failure will occur if:

$$|\sigma_T| < |\sigma_{\text{min},T}| \quad (2-42)$$

![Figure 2-18: Mohr circles of shear failure and tensile failure](image-url)
Resulting Cutting Force

For rocks with tensile strengths smaller than the lower analytical boundary $|\sigma_T| < |\sigma_{\text{min}},S|$, the failure mechanism must be tensile. For rocks with tensile strengths larger than the analytical upper boundary, pure shear failure must occur $|\sigma_T| > |\sigma_{\text{min}},T|$. In between, a mix of shear and tensile failure can exist. The analytical boundaries which Miedema derived and the effect on horizontal cutting forces are visualized in Figure 2-19. The graph for vertical cutting forces shows a similar trend.

![Horizontal Cutting Force vs. Tensile Strength BTS-UTS](image)

**Figure 2-19:** Horizontal cutting forces as calculated by Miedema. Blade angle $\alpha = 60^\circ$, $h_i = 0.1m$, $w = 0.1m$, UCS = 100 MPa, $\phi = 20^\circ$[5]

2-3-4 Rock Cutting Theory - Summary

The important characteristics of rock mass are captured in the failure envelope. Different regions of a typical rock failure envelope have been addressed. On the scale of a cutter head, some important aspects of the excavation process have been described. All in all the excavation process depends very much on effects at cutter scale. On the other hand, small scale effects (at tooth level) are also very important. For a generalized tool geometry a good estimation of cutting forces can be made. The calculated cutting forces, which are dominated by brittle tensile and/or brittle shear failure, are peak forces. Brittle cutting leads to erratic geometry changes. This causes a highly frequent and sometimes intermittent cutting force. The time averaged force on a brittle cutting tooth may be only 30% to 60% of the peak forces. These time averaged cutting forces can be used in the model of the CSD since the bodies in this model have relatively large inertia.

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2-4 Engineering Disciplines - Summary

Three disciplines have to be combined for a dynamic model of a CSD in operational condition. The discussed hydrodynamic theory is based on linear wave theory and provides first order wave forces. Because of the relatively stiff connection of spud and ladder with the seabed, these first order wave forces are sufficiently accurate. Furthermore, there is no need to model hydrodynamic interaction with another floating body and the wetted surface of the floater is not changing too much. This allows for an autonomous simulation in DODO, which means that the diffraction problem is solved once by aNySIM before the multi-body model is built. During the simulation of the multi-body system, all relevant hydrodynamic parameters are used from that one single set of aNySIM solutions.

The basics of multi-body theory have been discussed. For DODO, the Cummins equation has been implemented in the Newton-Euler notation. This Newton-Euler notation is a common notation for multi-body theory. It formulates the equations of motion and the kinematic relations all together and is a good way to approach the CSD dynamics. However, the drift effect in multi-body dynamics is inevitable. There are ways to mitigate this but this yields numerical transfer of energy. High sudden loads can also cause bodies to accelerate and drift too far away from their constraints. As a result, the model can become numerically instable if sudden forces with high magnitude occur.

The third important engineering discipline of this model encompasses the theory of rock cutting. On the scale of a cutter head, the cutting process has been explained. This places the relatively small-scale brittle chip formation caused by the many cutter teeth in the perspective of a relatively large scale CSD. Since multiple teeth around the cutter head are in action at the same time, and the body masses of the CSD model imply large inertia, these average cutting forces are a good representation of the cutter forces in a large scale CSD model.
The Cutter Force Function

A function for the cutter forces is essential if the cutter-to-soil interface is incorporated in the system. This chapter describes how the cutter forces are established.

1. A local Coordinate System for the cutter is defined. Based on a constant cutter orientation all relevant unit vectors are described that are used later. This can be found in Section 3-1.

2. The Discretization of the cutter-to-soil interface is treated in Section 3-2. Each time step a couple of samples is taken from the seabed matrix and eventually updated.

3. All discrete volumes that are cut away are stored in an array. Their position vectors with respect to the local cutter coordinate system are known. This information is used to introduce the Cutter-to-Soil Interface Forces, as can be found in Section 3-3.

4. Verification of the Cutter Force Function is based on four benchmark CSDs. This can be found in Section 3-4.

Figure 3-1: Left: CAD-model of a cutter head. Right: simplified representation of cutter head by hemisphere
3-1 Coordinate System

The cutter is represented by a hemisphere as illustrated in Figure 3-1. The backring of the cutter coincides with the so-called ‘great circle’ of the hemisphere, this is the flat plane which divides a sphere into two equal volumes. The origin of the cutter coordinate system is in the centroid of the great circle.

In this model the cutter has a constant orientation because the motions of the cutter ladder do not change the cutter orientation too much. Note that in Section 1-4 was explained that the heading of the CSD stays more or less constant while the seabed passes through the model space (see also Figure 1-1). The cutter axis \( a = z’ \) is pointing from the origin to the pole of the hemisphere, as seen in Figure 3-2.

![Figure 3-2: A hemisphere with axis of rotation \( a \). Left: the unit vectors \( u_r, u_t \) and \( u_r \) for an arbitrary point with position vector \( r \). Right: the polar angle \( \phi_p \) and azimuthing angle \( \theta_a \)](image)

To map the general Cartesian coordinate system \((x, y, z)\) onto the rotated Cartesian coordinate system \((x’, y’, z’)\) a rotation matrix \( R \) is used. The cutter orientation is such that \( \theta \neq 0 \), \( \psi = 0 \) and \( \phi = 0 \), so \( R \) becomes:

\[
R = \begin{bmatrix}
\cos \theta & 0 & \sin \theta \\
0 & 1 & 0 \\
-\sin \theta & 0 & \cos \theta 
\end{bmatrix}
\]  \hspace{1cm} (3-1)

The unit axis \( u_a \) in the model is orientated such that \( \theta = \pi/2 + \theta_{\text{ladder}} \) in the general coordinate system, in which \( \theta_{\text{ladder}} \) is the pitching angle of the ladder body with respect to the x-axis. In reality the cutter shaft usually has a small \( \theta \) angle with respect to the ladder body but the model is simplified such that the cutter axis is parallel to the longitudinal axis of the ladder.

\[
u_a = \begin{bmatrix}
\cos \theta & 0 & \sin \theta \\
0 & 1 & 0 \\
-\sin \theta & 0 & \cos \theta
\end{bmatrix}\begin{bmatrix}0 \\ 0 \\ 1\end{bmatrix} = \begin{bmatrix} \sin \theta \\ 0 \\ \cos \theta \end{bmatrix} = \begin{bmatrix} \cos(\theta_{\text{ladder}}) \\ 0 \\ -\sin(\theta_{\text{ladder}}) \end{bmatrix}
\]  \hspace{1cm} (3-2)
The rotation matrix $R$ maps any point of Cartesian system $(x', y', z')$ onto the general Cartesian coordinate system $(x, y, z)$. The inverse of $R$ can map any general coordinate onto the rotated system from which the spherical coordinates and unit vectors are calculated.

$$
\begin{bmatrix}
x \\
y \\
z
\end{bmatrix} = R
\begin{bmatrix}
x' \\
y' \\
z'
\end{bmatrix} = R^{-1}
\begin{bmatrix}
x \\
y \\
z
\end{bmatrix}
$$ (3-3)

Any point $P$ on the surface or inside this hemisphere, as indicated in Figure 3-2, has its own set of unit vectors that are based on a spherical coordinate system. The pole of this spherical coordinate system lies on the axis of revolution $z'$. The spherical coordinates are the radial distance, azimuthing angle and polar angle, respectively $(r, \theta_a, \phi_p)$, indicated on the right side of Figure 3-2. These are related to the Cartesian $(x', y', z')$ coordinates by

$$
r = \sqrt{x'^2 + y'^2 + z'^2}; \quad \theta_a = \tan^{-1}\left(\frac{y'}{x'}\right); \quad \phi_p = \cos^{-1}\left(\frac{z'}{r}\right)
$$ (3-4)

**Local unit vectors**

The radial component $u'_r$ points from the origin to point $P$. The tangent component $u'_t$ is parallel to the 'great circle'. The longitudinal component $u'_l$ is pointing along the meridian of point $P$ towards the pole of the hemisphere. The unit vectors are

$$
\begin{bmatrix}
\cos\theta_a\sin\phi_p \\
\sin\theta_a\sin\phi_p \\
\cos\phi_p
\end{bmatrix}; \quad
\begin{bmatrix}
-sin\theta_a \\
\cos\theta_a \\
0
\end{bmatrix}; \quad
\begin{bmatrix}
\cos\theta_a\cos\phi_p \\
\sin\theta_a\cos\phi_p \\
-sin\phi_p
\end{bmatrix}
$$ (3-5)

The unique unit vectors for point $P$ are mapped back to the general coordinate system with:

$$
\begin{align*}
\mathbf{u}_r &= Ru'_r; \\
\mathbf{u}_t &= Ru'_t; \\
\mathbf{u}_l &= Ru'_l;
\end{align*}
$$ (3-6)

Any vector associated to point $P$ can now be expressed in terms of $\mathbf{u}_r$, $\mathbf{u}_t$ and $\mathbf{u}_l$ as visualized in Figure 3-2. These unit vectors in terms of the general coordinate system can be used for the force components of the cutter-to-soil interface, since the force components of the 2D-cutting theory of Section 2-3-3 are aligned with these unit vectors.
3-2 Discretization

3-2-1 Temporal Discretization

The complete multi-body model of the cutter suction dredger leads to a coupled system of ordinary differential equations. To approximate the solution of these equations of motion a fourth order Runge-Kutta (RK4) solution scheme is used. The simulated time is divided into equal time steps $\Delta t$. Within each time step the cutter force function is called four times by the RK4-scheme.

In this model the cutter force is constant during one timestep, therefore the cutter force is only calculated in the first instance of the RK4-loop. The rest of the RK4-loop the force is constant.

3-2-2 Spacial Discretization

In the first version of the cutter force function the cutter hemisphere is discretized by means of an $m \times n \times o$ matrix $C$, as can be seen in 3-3. The elements in this matrix correspond with position vectors $r_{i,j,k}$ in $\mathbb{R}^3$ which are located inside an imaginary box around the cutter. The size of this matrix is odd-numbered, which means that there is one element exactly on the origin of the cutter coordinate system. For the discretization constant grid sizes are used, though $\Delta x$, $\Delta y$, $\Delta z$ are not necessary equal. Each grid cell $c_{i,j,k}$ contains a binary state which indicates whether it is inside the cutter hemisphere or not. The discretized cutter approaches the shape of the hemisphere.

Please note: The cutter is attached to the ladder body in the multi-body system. As bodies change states during simulations the orientation of the cutter contour should strictly change along with the cutter ladder. Since the cutter ladder rotations will be very small during the simulations the orientation of the cutter is assumed constant for the calculation of the force. Therefore the cutter can be discretized prior to the simulation and stored in one matrix $C$.

![Figure 3-3: Representation of the cutter in an $m \times n \times o$ matrix.](image)
In the first version of this cutter force function the seabed profile is discretized by means of a larger $p \times q \times r$ matrix $S$, which has the exact same constant grid spacings $\Delta x$, $\Delta y$ and $\Delta z$ as the cutter matrix. Each grid cell $s_{i,j,k}$ contains a fraction which indicates a percentage of soil that is represented in the control volume. The seabed is assumed to be homogeneous. A representation of the seabed matrix is shown in Figure 3-4. Although the displayed matrix elements in the figure are either 0 or 1, they can also contain any fraction between 0 and 1.

![Figure 3-4: Representation of the seabed in an $p \times q \times r$ matrix](image)

3-2-3 Box Samples of Seabed Matrix

The equations of motion of the cutter dredger are solved in the time domain, so the momentary position of the cutter is numerically approximated for each time step. To obtain the interface forces of the cutter at $r_{i,j,k}$, the cut volume $(V_{cut})_{i,j,k}$ is required. The following steps are taken:

- It is near certain that the position of the cutter head origin is not located exactly on one of the grid cells of the seabed. The center of the cutter head matrix snaps to the closest grid cell of the seabed.

- Around this grid cell a box matrix $B_0$ is taken from the seabed matrix. It is the size of the $m \times n \times o$ cutter matrix and the grid cell is exactly in its center. This box sample is updated (see Section 3-2-4) and placed back into the seabed. Since the cutter matrix and the seabed matrix have equal grid spacing, the intersection of all elements with $(c_{i,j,k} = 1 \cap b_{i,j,k} > 0)$ can be used to find $V_{i,j,k}$. 

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• One box at a time, the cutter matrix is shifted in six orthogonal directions and the seabed is conditionally updated. This is elaborated in Section 3-2-4 and illustrated in 2D in Figure 3-5.

### Table 3-1: Shifts corresponding to the box samples of the seabed

<table>
<thead>
<tr>
<th>Box</th>
<th>Shift</th>
</tr>
</thead>
<tbody>
<tr>
<td>B</td>
<td>i + 1</td>
</tr>
<tr>
<td>B</td>
<td>i - 1</td>
</tr>
<tr>
<td>B</td>
<td>j + 1</td>
</tr>
<tr>
<td>B</td>
<td>j - 1</td>
</tr>
<tr>
<td>B</td>
<td>k + 1</td>
</tr>
<tr>
<td>B</td>
<td>k - 1</td>
</tr>
</tbody>
</table>

The volume that is cut away, which not only depends on intersection \((c_{i,j,k} = 1 \cap b_{i,j,k} > 0)\) but also on a ‘snapping offset’ \(e\), also eventually contributes to \(V_{\text{cut}}\) of Equation 3-10.

**Figure 3-5:** Representation of the box matrices in 2D

### 3-2-4 Updating Sequence

Figure 3-6 shows a 1D example of one element \(c_{i,j,k}\) of the cutter matrix. First, the cutter matrix snaps to the closest grid of the seabed matrix (indicated in the first out of the three scenarios in the figure).

**Figure 3-6:** Spacial discretization illustrated in 1D
The central box $B_0$ is updated and placed back into matrix $S$. The discrete volume elements that are cut depend on $(c_{i,j,k} = 1 \cap (b_0)_{i,j,k} > 0)$

$$V_{i,j,k} = \Delta x \cdot \Delta y \cdot \Delta z \cdot (b_0)_{i,j,k} \cdot c_{i,j,k}$$

(3-7)

However, $c_{i,j,k}$ was most likely not exactly on the grid of matrix $S$, as can be seen in the second and third scenario of Figure 3-6. If only the central box would be updated, the snap-to-grid behavior of the sample taking method would reduce to a binary signal. Therefore a ghost of $c_{i,j,k}$ is placed on six sides, shifted one grid spacing.

A shifted box $B_n$ is taken from matrix $S$, conditionally updated and placed back into matrix $S$. This is consequently done for $n = 1, 2, \ldots, 6$. If $B_n$ is updated, a fraction $p$ is removed from $(c_{i,j,k} = 1 \cap (b_n)_{i,j,k} > 0)$. The cutter head passes through the seabed matrix with a speed close to $v_h = 1/3 m/s$. The fraction $p$ depends on the gridsize in $y$-direction $\Delta y$ and the average swing speed.

$$p = \frac{|v_h|}{\Delta y}$$

(3-8)

In short, the updating conditions (for the boxes of Table 3-1) are:

- If $(e_x)_t > 0$, $B_1$ has to be updated. $p_1 = \frac{|v_h|}{\Delta y}$ ; $p_2 = 0$
  - If $(e_x)_t < 0$, $B_2$ has to be updated. $p_2 = \frac{|v_h|}{\Delta y}$ ; $p_1 = 0$
- If $(e_y)_t > 0$, $B_3$ has to be updated. $p_3 = \frac{|v_h|}{\Delta y}$ ; $p_4 = 0$
  - If $(e_y)_t < 0$, $B_4$ has to be updated. $p_4 = \frac{|v_h|}{\Delta y}$ ; $p_3 = 0$
- If $(e_z)_t > 0$, $B_5$ has to be updated. $p_5 = \frac{|v_h|}{\Delta y}$ ; $p_6 = 0$
  - If $(e_z)_t < 0$, $B_6$ has to be updated. $p_6 = \frac{|v_h|}{\Delta y}$ ; $p_5 = 0$

The discrete volume elements that are cut away by the shifted cutter matrices $C$ are

$$(V_n)_{i,j,k} = \Delta x \cdot \Delta y \cdot \Delta z \cdot (b_n)_{i,j,k} \cdot c_{i,j,k} \cdot p_n ; \quad n = 1, 2, \ldots, 6$$

(3-9)

During this sequence it is made sure that the updated seabed matrix will never contain any negative element. If the fraction $p$ turns out to be larger than box element $b_{i,j,k}$, it will just remove the remainder of whatever positive number is left. The total volume that is cut away is:

$$(V_{cut})_{i,j,k} = V_{i,j,k} + \sum_{n=1}^{6} (V_n)_{i,j,k}$$

(3-10)

These volumes correspond to $r_{i,j,k}$ from the discretized cutter and are used for the calculation of cutter forces.

### 3-2-5 Revisions

The computational time and the cost of writing all this information to the workspace turned out to be far beyond acceptable levels. One of the main culprits of this excessive demand was the fact that a box matrix, filled with binary elements, was used to determine the discretized body of the cutter inside the seabed matrix. A lot of information had to be processed for the intersection finding algorithm just to find out that a large part of the matrix $C$ was empty.
An inclined cutter axis, starting from the cutter origin, was used to define the orientation of the cutter and the plane of the ‘great circle’ of this hemisphere, where the back ring of the cutter is located. This can be seen in Figure 3-7. On one side of this plane none of the discretized volumes of the box were identified as cutter. Hence all these elements had a value of $c_{i,j,k} = 0$. Divide the volume of a hemisphere by the volume of a box that fits seamlessly around a sphere of equal radius and one finds a ratio of

$$\frac{V_{\text{hemisphere}}}{V_{\text{box}}} = \frac{\frac{1}{2} \frac{4}{3} \pi r^3}{2 \cdot r \cdot 2 \cdot r \cdot 2 \cdot r} = \frac{\pi}{12} \approx 0.26 \quad (3-11)$$

About 74% of all elements in the box matrix is outside the hemisphere volume, having a value of $c_{i,j,k} = 0$. They will never remove anything in the first place. To make matters worse, the elements on the inside of the hemisphere will do nothing either. Any element of the seabed will long be emptied before it reaches the core of the hemisphere.

![Figure 3-7: Cutter Discretized, 2D representation](image)

**Revision one**

All this waste had to be eliminated in order to speed up the simulations. The simulation of the CSD should at least be capable of running real-time. All rudimentary box elements were therefore removed from the sampling and updating sequence. This means that the first revised function performs a targeted search based on a ‘Reduced Box’ array. Only the elements on the outer shell of the hemisphere can be active so only those elements are checked on the presence of seabed. The computational demand was reduced by over 70%. With this revised function the simulation could perform realtime with time steps of $\Delta t = 0.1$ s and element volumes $\Delta x \cdot \Delta y \cdot \Delta z \approx 10^{-3} m^3$.

**Revision two**

Despite the smoothing techniques of the production signal, as described in Section 3-2-3 and Section 3-2-4, a much finer grid spacing was desired. For the second revision of the cutter force function the seabed-cutter interface is discretized by means of a 2D-matrix. An illustration is provided in Figure ???. The differences from the original function and revision one are:

- **The cutter hemisphere** is now discretized by means of an $m \times n$ matrix $C$. In these matrix entries the points on the surface of the cutter are stored, as calculated in Equation 3-12. The unit vectors (on which the cutting forces, and normal forces
are applied) for all relevant points \( P \) (see Figure 3-2) on the surface of the cutter are determined in the same way as was described in Section 3-1. This is stored in matrix \( C \).

\[
c_{i,j} = \sqrt{\frac{D_c^2}{4} - x_{i,j}^2 - y_{i,j}^2 - \frac{D_c}{2}}
\] (3-12)

- **The seabed matrix** \( S \) is reduced to a \( p \times q \) matrix, storing the height of the seabed surface \( b_{i,j} \) for each time step.

- The algorithm is fairly similar to the original version, so the methods explained in Section 3-2-3 and Section 3-2-4 also apply. The difference is that the box samples \( B_5 \) and \( B_6 \) are not taken. For each sample that is taken, the height of that box sample \( b_{i,j} \) is evaluated. This is the surface height of the seabed.

- For each time step the height of the ladder tip is calculated by DODO. This is done to determine the height of the discretized points on the surface of the cutter. Equation 3-13 is used to find the volume \( V_{i,j} \):

\[
V_{i,j} = \begin{cases} 
\Delta x \cdot \Delta y \cdot (b_{i,j} - c_{i,j} + z_{tip}) & \text{if } c_{i,j} + z_{tip} < b_{i,j} \\
0 & \text{if } c_{i,j} + z_{tip} > b_{i,j}
\end{cases}
\] (3-13)

The volumes \((V_n)_{i,j}\) that are found for box samples \( B_n \) are multiplied by fraction \( p \) (same as in Section 3-2-3), but now for \( n = 1, 2, ..., 4 \).

The total volume that is cut away (during time step \( \Delta t \)) is:

\[
(V_{\text{cut}})_{i,j} = V_{i,j} + \sum_{n=1}^{4} (V_n)_{i,j}
\] (3-14)

**Figure 3-8:** Left: Illustration of revised volume finding method. Right: recommendation for improvement

In the end, this method is capable of detecting the soil volumes at the interface of cutter and seabed. In fact it makes sure that all volumes crossing the boundary of the cutter will be removed. It will result in the right force vectors provided that the face height \( H_{\text{face}} < D/2 \). This can be improved if the base plane is rotated such that it is perpendicular to the cutter axis when the system is in equilibrium.
3-3 Cutter-to-Soil Interface Forces

Now that the interface volumes are identified, the forces on the cutter can be calculated. This is done as follows;

1. The Cutting Mechanism is modeled, using a generalized tooth with blade angle $\alpha = 60^\circ$, width $w = 0.1m$ and cutting depth $h = 0.1m$. Any soil volume intersecting the cutter will be removed as if it would be cut by this generalized tooth. This leads to a specific energy. The modeling of the cutting mechanism can be found in Section 3-3-1.

2. This specific energy is also the ratio of Cutting Power over production. This means that the removed volumes (described in previous section) can be related to a torque on the cutter shaft. This is described in Section 3-3-2.

3. From this torque the Cutting Forces are derived in Section 3-3-3. The tangent cutting forces are exclusively caused by the cutter torque. The radial cutting forces are a ratio of the tangent cutting forces. The ratio of radial/tangent forces can depend on the type of cutting mechanism, blade angle, external friction coefficient, and changed tooth geometry due to wear.

This cutter force function does not make a distinction between the tool geometry of different regions on the cutter head. The function also does not notice the difference between a soil volume being part of a thick layer or a thin layer. Hence, all five cut volumes from the example in Figure 3-9 are removed with the same cutting mechanism and thus the same energy corresponding to one generalized tooth.

![Figure 3-9: Left: schematic side view of cutter removing 2 discretized volumes. Right: schematic axial view of cutter removing 3 discretized volumes.](image-url)
3-3-1 Cutting Mechanism

The cutting mechanism is modeled, using the theory for tensile failure of Section 2-3-3 and the theory for shear failure of Section 2-3-3. For simplicity of the model, the transition between brittle tensile and brittle shear failure occurs at the intersection around $m \approx 5.7$. The assumption is compared to the Chip Type zone in Figure 3-10. As can be seen the cutting force is a bit underestimated. This is because pure tensile failure is assumed while analytically a mix of tensile and shear failure may exist. The maximum underestimation is about 15% at $m = 8$ ($BTS = 12.5 MPa$ in Figure 3-10).

![Figure 3-10: Comparison of the modeled Fh and calculated Fh by the Shear-, Chip- and Tear Type. Blade angle $\alpha = 60^\circ$, $h_i = 0.1 m$, $w = 0.1 m$, $UCS = 100 MPa$, $\phi = 20^\circ$[5]](image)

For the cutting of soil a specific energy can be assigned to the process.

$$E_{sp} = \frac{F_h}{h_i \cdot w}$$  \hspace{1cm} (3-15)

This parameter is related to the required cutting power.

3-3-2 Cutting Power

Power is required to move the cutter through the seabed. The specific energy relates the required cutter drive power to production.

$$E_{sp} = \frac{P_c}{Q} = \frac{P_c}{dV/dt}$$  \hspace{1cm} (3-16)
The relation between power, torque and angular velocity of the cutter head is:

\[ P = T \cdot \omega \quad ; \quad \omega = \frac{2\pi \text{RPM}}{60} \]  

(3-17)

Combining Equations 3-16 and 3-17

\[ T \cdot dt = \frac{dV \cdot E_{sp}}{\omega} \]  

(3-18)

Integrating the above gives the required torque to remove a discrete volume \((V_{cut})_{i,j,k}\) in a discrete time step

\[ T \int dt = \frac{E_{sp}}{\omega} \int dV \]  

(3-19)

\[ T_{i,j,k} = \frac{(V_{cut})_{i,j,k} \cdot E_{sp}}{\omega \cdot \Delta t} \]  

(3-20)

The sum of all torques is the total torque on the cutter drive shaft

\[ T_c = \sum_{i=1,j=1,k=1}^{m,n,o} \{T_{i,j,k}\} \]  

(3-21)

### 3-3-3 Cutting Forces

The tangent cutting forces are exclusively caused by the cutter torque. From the torque \(T_{i,j,k}\) and the corresponding position vector \(r_{i,j,k}\) the tangent force vector can be calculated. The scalar \(d_{i,j,k}\) is the shortest distance to the cutter axis. It is the lever arm of the tangent force vector. It can be calculated with polar angle \(\phi_p\) and the position vector \(r_{i,j,k}\) from Figure 3-2.

\[ d_{i,j,k} = \|r_{i,j,k}\| \cdot \sin(\phi_p) \]  

(3-22)

The tangent force is

\[ (F_t)_{i,j,k} = \frac{T_{i,j,k}}{d_{i,j,k}} \cdot (u_t)_{i,j,k} \]  

(3-23)

If the tensile failure mechanism from Section 2-3-3 is used, the radial force depends on the blade angle. In the model a generalized blade is used for all cut volumes, with blade angle \(\alpha = 60^\circ\)

\[ (F_r)_{i,j,k} = \tan(\alpha/2) \cdot (F_t)_{i,j,k} \]  

(3-24)

If the shear failure mechanism from Section 2-3-3 is used, the radial force depends on the blade angle \(\alpha\) and the external friction angle \(\delta\):

\[ (F_r)_{i,j,k} = \frac{1}{\tan(\alpha + \delta)} \cdot (F_t)_{i,j,k} \]  

(3-25)

The sum of all forces on the cutter is the total force on the cutter drive shaft

\[ F_c = \sum_{i=1,j=1,k=1}^{m,n,o} \{(F_t)_{i,j,k} + (F_r)_{i,j,k}\} \]  

(3-26)
In order to verify the cutter force function, it has been applied to 4 different CSD’s of which the theoretical production data and technical specifications have been provided by IHC. The relevant data can be found in Table 3-2.

**Table 3-2: Technical specifications of four CSD’s**

<table>
<thead>
<tr>
<th>Max. Cutter Power $P_{c,max}$ [kW]</th>
<th>Average Cutter Diameter $D_c$ [m]</th>
<th>Max. Swing Force $F_{h,max}$ [kN]</th>
<th>Swing Speed $v_h$ [m/min]</th>
<th>Rotational Speed $[RPM]$</th>
</tr>
</thead>
<tbody>
<tr>
<td>#1 1350</td>
<td>2.563</td>
<td>600</td>
<td>20</td>
<td>30</td>
</tr>
<tr>
<td>#2 2600</td>
<td>2.673</td>
<td>900</td>
<td>24</td>
<td>30</td>
</tr>
<tr>
<td>#3 3500</td>
<td>2.673</td>
<td>1000</td>
<td>20</td>
<td>30</td>
</tr>
<tr>
<td>#4 4400</td>
<td>3.080</td>
<td>1700</td>
<td>24</td>
<td>30</td>
</tr>
</tbody>
</table>

**3-4 Verification Cases**

For the four cases from Table 3-2, simulations were done in which the cutter maintained a constant face height, step size, swing speed and cutter rotation. In each simulation a short
swing was made to be able to check the production rate, torque and forces. During these simulations the seabed had an ideal shape, as illustrated in Figure 3-12 below. The rock mass was homogeneous and had a specific energy of \( E_{sp} = 1Pa \).

![Figure 3-12: Illustration of removed seabed (each removed element is a red dot). The cutting started at the left and stopped at the right.](image)

This was done for all combinations of the parameters shown in Table 3-3, which yields a total number of \( 2 \cdot 2 \cdot 5 \cdot 5 = 100 \) simulations per CSD.

**Table 3-3: Inputs for the 4 verification cases**

<table>
<thead>
<tr>
<th>Cutting mechanism</th>
<th>Shear ( L_{step} )</th>
<th>Tensile ( H_{face} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Over cutting</td>
<td>0.1 ( \cdot D_c )</td>
<td>0.2 ( \cdot D_c )</td>
</tr>
<tr>
<td>Under cutting</td>
<td>0.3 ( \cdot D_c )</td>
<td>0.4 ( \cdot D_c )</td>
</tr>
<tr>
<td></td>
<td>0.5 ( \cdot D_c )</td>
<td></td>
</tr>
</tbody>
</table>

**3-4-2 Verification of Cutter Power**

There is a direct relation between cutter power, cut volume and specific energy. Since the maximum production of a CSD is limited by the installed cutter power (other limits are not considered now), a production envelope can be drawn which shows \( E_{sp} \) versus \( Q \). Such an envelope is plotted in Figure 3-13. The following is important:

- The figure shows 25 unique data points, which originate from 25 test simulations of one CSD (5 Step Sizes x 5 Face Heights, Table 3-3).
- The two Cutting Modes and the two Cutting Mechanisms of Table 3-3 have no direct relation between cutter power, cut volume and specific energy.
- From the time averaged torque signals of each simulation the required cutter power was calculated. Remember: the simulation input was \( E_{sp} = 1Pa \).

\[
P_{c,signal} = T_{c,signal} \cdot \omega [W] ; \quad \omega = \frac{2 \cdot \pi \cdot RPM}{60} [rad/s] \quad (3-27)
\]
The installed power is much more than the required cutting power that was found for just $E_{sp} = 1 Pa$. The specific energy that the cutter should be able to handle, can thus be calculated with Equation 3-28. This was done for each of the 25 simulations.

$$E_{sp, envelope} = \frac{P_{c, max}}{P_{c, signal}} [Pa]$$  \hspace{1cm} (3-28)

The production for each of the 25 cases was calculated with Equation 3-29. The difference between the seabed matrices at the start and the end of the simulations should reflect this (this was verified).

$$Q_{envelope} = L_{step} \cdot H_{face} \cdot |v_h| [m^3/s]$$  \hspace{1cm} (3-29)

Figure 3-13 shows the production rates of CSD #1 in a specific energy range of the cut soil. The curve of # 1.A represents the cutter power envelope from the benchmark. The 25 points of # 1-Pc represent the results that are based on the 25 torque signals. From this, the cutter power envelope of the simulations is drawn. The cutting mechanism and the cutting mode of Table 3-3 have no effect in the relation between torque, specific energy and production. The benchmark graphs of the other three CSDs can be found in the Appendix C.

![Figure 3-13: Capability envelope of cutter drive](image)

The results from the cutter force function show a similar trend as the benchmark. This is the case for all four CSDs, having different technical specifications. As can be seen from Figure 3-13, the scatter points do not fit the smooth line of the envelope exactly. This has to do with the temporal and spacial discretization of the cutter force function. All in all this difference is acceptable and the benchmark shows clearly the same trend as the simulations. It can be concluded that the relation between cutter power and cut volume is well implemented in the Cutter Force Function. It is accurate enough for an implementation in a multi-body model of a CSD.
3-4-3 Verification of Swing Wire Force

There is no direct relation between swing wire force, cut volume and specific energy. The swing wire forces depend on the orientation of the tangent, longitudinal and radial forces that the teeth around the cutter may or may not induce. For the four verification cases, however, the influx of new seabed volume is known a priori. With some assumptions the swing wire force can be verified in the same way as was done for the cutter torque signal.

- One of the swing winches is braking, as illustrated in Figure 3-14. This costs approximately 5% of the maximum swing winch power.
- The hauling swing wire is not always pulling at an ideal angle but it may pull at an angle of about 45 degrees.
- The swing winches should be able to cope with current and wind forces, but these are relatively small compared to the total forces on the cutter head. This is therefore neglected.
- The lever arm of the swing wires is smaller than the lever arm of the cutter head. As can be seen from the sketch in Figure 3-14. Nonetheless, this difference is neglected.
- When the face height is small the cutter drive creates horizontal traction (as a wheel), this has to be counteracted by the swing wires.
- If the y-component (in the general coordinate system) of the cutter force is verified, all forces that are perpendicular to the cutter axis are verified.

\[ E_{sp,nom} = \frac{F_{h,max}}{F_h} \cdot 0.95 \cdot \sqrt{2} \cdot E_{sp} \]  

Figure 3-14: Top view of CSD, illustrating effects that demand swing winch pulling power

Simulations are performed for all combinations of Table 3-3. Again, the production Q is calculated with Equation 3-29. The required swing wire force is calculated with Equation 3-30 (remember: specific energy of the simulation was \( E_{sp} = 1Pa \)).
Figure 3-15 shows the same cutter drive envelope as Figure 3-13. In addition to the envelope of the cutter drive, 50 scatter points are plotted for the swing winches (some are off the chart). The plus- and minus signs represent the swing winch capability for respectively over- and under cutting. The results for the brittle shear mechanism are shown below (for brittle tensile the figure is almost the same). The rest of the graphs can be found in the Appendix.

The production is limited by the power of the cutter drive. None of the scatter points of the swing wire is below the envelope. The scatter points that are far from the envelope are not really meaningful, since the cutter drive will never be able to provide the required power for such cases. However, some points are very close to the envelope of the cutter drive. These points correspond with simulations in which the face height $H_{\text{face}}$ was smallest ($H_{\text{face}} = 0.1 \cdot D_c$). When the face height is small, the ‘cutting action’ is at the lowest region of the cutter head. This causes horizontal traction, hence larger y-components in the general coordinate system. In short: some situations exist at which the cutter drive and the swing winches are both operating close to their design limits.

All in all the plotted swing winch capabilities are considered very reasonable in comparison to the technical specifications of the benchmark CSDs. All benchmark plots show similar results. The swing winch forces at small face heights show the expected trends, which means that the lateral force at the tip of the cutter ladder of the CSD model is verified. If this lateral force on the tip of the cutter ladder is verified (which is in the plane perpendicular to the cutter shaft), then it can be assumed that any force perpendicular to the cutter shaft is verified. This can be assumed because of the way that the algorithm just sums up all found cutting force vectors. It can be concluded that the nett resultant cutting force has the right direction and magnitude.
3-4-4 Closing Remarks

The sole purpose of this verification section was to check the force and torque signals that are generated by the cutter force function. Figures 3-13 and 3-15 only illustrate the capability regarding swing winches and cutter drive, not the complete production envelope of the CSD. Note that the goal of this thesis is not to simulate the full dredging process including material transport. Despite their significance to the eventual production of a CSD the following effects are left out of the model:

- The dredge pump has limited suction power to transport the dredged material. Based on the density of mixture, flow speed and suction pipe diameter the maximum in situ production can be estimated. Although Figures 3-13 and 3-15 show a vertical asymptote at specific energy $E_{sp} = 0$, the actual production envelope will be limited. Moreover, a value of $E_{sp} = 0$ is for a cutter dredger irrelevant, since it would render the function of the cutter head obsolete.

- For low $E_{sp}$ the cutter drive may have the available power for even higher swing speeds than 20$m/min$. However, the cut material needs to fit through the opening between the arms of the cutter head.

- Clearance angle of the cutting tooth (visualized in Figure 2-16). At high swing speeds compared to the rotational speed of the cutter head, the clearance angle becomes smaller. There is a maximum swing speed at which the clearance angle is zero. For larger swing speeds there is no clearance left. In fact, the cutting tooth is pushed to the rock surface on the wrong side. This type of bulldozing leads to excessive wear of pickpoints and does not yield higher production.

3-5 The Cutter Force Function - Summary

The coordinate system and the discretization of the cutter have been discussed. In the first revision, the function processes the cutter-soil interface by means of 3D matrices and 7 box samples to quantize the cut volume.

The cutter matrix was stripped to an array of elements in order to reduce computational waste. This way the function targets only the elements of the seabed matrix that have to be updated. The second revision uses 2D matrices to speed up the quantizing process. Based on the box samples the matrix elements can be conditionally and partially emptied, effectively smoothing the binary signal.

Verification was done based on four benchmark CSDs. It was shown that the function generates reasonable production rates, swing winch power and cutter power. This force signal from the cutter-to-soil interface is simple, robust and can be easily implemented in the CSD model in DODO. The Time Domain simulations of the CSD model with this Cutter Force Function can be run faster than realtime.
Chapter 4

CSD Model

The engineering disciplines of Chapter 2 can be combined into a dynamic model of the CSD. An overview of this chapter is given below:

1. The **Multi-Body System** is discussed in Section 4-1. The Euclidian space of the model is defined with a global reference frame. Inside this space the floater and cutter ladder can move. The kinematic relations between the bodies of the system are implemented with constraint equations. Some system dimensions are provided.

2. Section 4-2 addresses the **Floater Hydrodynamics**. Hydrodynamic data from a diffraction solution are used as input. However, there are more damping effects than just the damping from potential flow theory. Therefore some viscous damping effects are added.

3. A hydrodynamic force will act on the submerged cutter ladder due to oscillations with respect to the water medium. For **Ladder Hydrodynamics**, see Section 4-3

4. The **Mooring Forces** are discussed in Section 4-4. The section starts with the characteristics of the spud mooring system. The swing wires are also considered as part of the mooring system.

5. Considering all aforementioned effects the model can be sufficiently linearized for a **Frequency Domain Analysis**. The Response Amplitude Operators in Section 4-5 show the sensitivity of the system to a frequency spectrum.

6. The **Cutter Force** (discussed in more detail in Chapter 3) is implemented. Section 4-6 provides some specifications that are used for this CSD model.

7. Finally, the **Time Domain simulations** of **Model 1** are performed. Some important aspects of the CSD model are explained in Section 4-7.

Three lay-outs are studied to obtain insight in the behavior of the CSD model in mooring condition. The analyzed lay-outs are shown in Table 4-1 and Figure 4-1 below. The results
of the Frequency Domain analysis of the moored condition can be found in Chapter 5. One lay-out is studied in Time Domain, in operational condition. Results of the Time Domain simulations are shown in Chapter 6.

**Table 4-1: Models and model lay-outs**

<table>
<thead>
<tr>
<th>Cutting</th>
<th>Model 0</th>
<th>Model 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Simulation Type</td>
<td>Frequency</td>
<td>Time</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Lay-out</th>
<th>#1</th>
<th>#2</th>
<th>#3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Swing Angle [deg]</td>
<td>-20</td>
<td>0</td>
<td>20</td>
</tr>
<tr>
<td>Wave Angle [deg]</td>
<td>160</td>
<td>180</td>
<td>200</td>
</tr>
<tr>
<td>Swing Wire1 Angle [deg]</td>
<td>110</td>
<td>90</td>
<td>70</td>
</tr>
<tr>
<td>Swing Wire2 Angle [deg]</td>
<td>70</td>
<td>90</td>
<td>110</td>
</tr>
<tr>
<td>$L_{sw1} + L_{sw2}$ [m]</td>
<td>120</td>
<td>120</td>
<td>120</td>
</tr>
</tbody>
</table>

**Figure 4-1:** Top views of model. Left: Lay-out 1. Middle: Lay-out 2. Right: Lay-out 3

### 4-1 Multi-Body System

#### 4-1-1 Model Overview

To be able to describe the model it is necessary to define a global coordinate system and some local coordinate systems. The following reference frames are right-handed Cartesian coordinate systems. A model overview is given in Figure 4-2.

- The global reference frame is earth-fixed, the z-axis points upwards and the xy-plane coincides with the still water plane.

- The Center of Gravity (CoG) of the floater is used as origin of its body-fixed reference frame. The x-axis points along the longitudinal axis of the hull, from the CoG to the bow. The y-axis points from the centerline of the hull towards port side and the z-axis points upward.

- The CoG of the ladder is used as origin of its body-fixed reference frame. The x-axis is aligned with the longitudinal axis and points to the cutter head, the y-axis points towards port side and the z-axis points upward.
The cutter head is at the tip of the cutter ladder. The cutter head is not modeled as a separate body. The forces that are found by the cutter force function are simply introduced at the tip of the cutter ladder, on the axis of the cutter shaft.

- The swing wires are connected to two anchors, which are modeled as fixed points in the global coordinate system. The anchors may have different positions than Figure 4-2 shows. Firstly they may be further away from the ladder, secondly they may be more towards the front or aft of the CSD. In reality the swing wires run through two fairleads near the tip of the cutter ladder, towards two controlled winches. In the model the swing wires are tied to the tip of the cutter ladder. Their unstretched length $L_0$ is not controlled but fixed.

### 4-1-2 Body Dimensions

The following dimensions of the model bodies and inertia terms are used:

- **Floater**: $m = 8183 \cdot 10^3 kg$, $L = 104.3 m$, $B = 21.7 m$, $T = 5m$.
  
  \[
  I_{xx} = \frac{1}{12} \cdot m \cdot (B^2 + T^2) \approx 3.4 \cdot 10^8 kg \cdot m^2
  \]
  
  \[
  I_{yy} = I_{zz} = \frac{1}{12} \cdot m \cdot L^2 \approx 6.8 \cdot 10^9 kg \cdot m^2
  \]

- **Ladder**: $m = 1000 \cdot 10^3 kg$, $L = 39.8 m$, $B = 8 m$, $H = 5m$.
  
  \[
  I_{xx} = \frac{1}{12} \cdot m \cdot (B^2 + H^2) \approx 8.3 \cdot 10^6 kg \cdot m^2
  \]
  
  \[
  I_{yy} = I_{zz} = \frac{1}{12} \cdot m \cdot L^2 \approx 1.3 \cdot 10^8 kg \cdot m^2
  \]
The mass matrix \( M \) for one body is shown below. For the two-body system the total mass matrix will thus be 12x12.

\[
M = \\
\begin{bmatrix}
    m & 0 & 0 & 0 & 0 & 0 \\
    0 & m & 0 & 0 & 0 & 0 \\
    0 & 0 & m & 0 & 0 & 0 \\
    0 & 0 & 0 & I_{xx} & 0 & 0 \\
    0 & 0 & 0 & 0 & I_{yy} & 0 \\
    0 & 0 & 0 & 0 & 0 & I_{zz}
\end{bmatrix}
\]

### 4-1-3 Kinematic Relations

The following kinematic relations are used in the CSD model:

- **A Revolute Joint** is used for the ladder hinge. The pinned connection of the cutter ladder inside the floater restricts 3 translations and 2 rotations of the cutter ladder with respect to the hull. In total \( m = 5 \) constraint equations are defined. Three times a ‘point constraint’ is used in order to block relative translations. Two times a ‘parallel vector constraint’ is used in order to cope with rotations. Upon implementation of these equations of constraints the system has \( k = 7 \) remaining DoF. The coordinates of the revolute joint with respect to the COG of the bodies is given below:

\[
x_{\text{hinge-on-floater}} = \begin{bmatrix} 22.5 \\ 0 \\ -1 \end{bmatrix} m \\
x_{\text{hinge-on-ladder}} = \begin{bmatrix} -19.9 \\ 0 \\ -2.2 \end{bmatrix} m
\]

- The CSD model is provided with a rigid ladder pitch system (Note: This is a heavily simplified model since in reality the CSD has a hoisting winch and hoisting wire strain). The modeled rigid wire connects the A-frame at the bow of the CSD to the tip of the cutter ladder. For this a **Constant Distance Constraint** is used. This constraint eliminates the 7th degree of freedom of the system model. The coordinates of the hoisting wire attachments with respect to the COG of the bodies is given below:

\[
x_{\text{wire-on-floater}} = \begin{bmatrix} 58 \\ 0 \\ 10 \end{bmatrix} m \\
x_{\text{wire-on-ladder}} = \begin{bmatrix} 15.9 \\ 0 \\ 2.8 \end{bmatrix} m
\]

A simple example of a constraint equation and its application in a set of equations of motion can be found in Appendix F-1.

### 4-2 Floater Hydrodynamics

1. The generation of a **Wave Signal** is described in Section 4-2-1. An appropriate wave spectrum should be chosen. For this case study the JONSWAP spectrum was chosen.

2. Two different **Simulation Modes** can be used in DODO. The choice of simulation mode determines how the solution of the diffraction problem is incorporated in the simulation. This can be found in Section 4-2-2.
3. To narrow the scope of this project, only one dredging depth is considered. The considered dredging depth is 20 meters. The simulations are run with relatively small wave heights, therefore Linear Wave Theory is justified. The First Order Linear Wave Forces are calculated with potential flow theory. The vessel motions do not change the wetted surface too much, so only one set of diffraction solutions at the floater equilibrium is used. This means that the DODO multi-body solver can be run in ‘autonomous mode’. All relevant hydrodynamic data are presented in Section 4-2-3.

4. Since potential flow theory does not account for viscous effects, the hydrodynamic damping from the diffraction solution is underestimating. Added damping is described in Section 4-2-4.

4-2-1 Wave Signal

This section shows how the irregular wave signal is generated from the wave spectrum. The frequency range of the wave spectrum is divided by a constant frequency interval $\Delta \omega$. For each $\omega_n$ a harmonic component is generated with amplitude $\zeta_{an}$ depending on the spectrum $S_\zeta$.

$$\zeta_{an} = \sqrt{2 \times S_\zeta(\omega_n) \times \Delta \omega}$$  \hfill (4-1)

The total surface elevation of the irregular wave signal is determined for $0 \leq t \leq t_{\text{end}}$. It is the sum of all generated harmonic components with random phase shifts $\varepsilon_n$. All phase shifts have a uniform distribution and $0 \leq \varepsilon_n \leq 2\pi$.

$$\zeta(t) = \sum_{n=1}^{N} \zeta_{an} \cos(\omega_n t + \varepsilon_n)$$ \hfill (4-2)

A constant frequency interval $\Delta \omega$ is used for the irregular wave signal. This signal will repeat itself after $2\pi/\Delta \omega$ seconds. The smallest interval that the wave building algorithm uses is 0.001 rad/s. The maximum unique wave signal length is therefore:

$$\frac{2\pi}{0.001} = 6283 \ [s]$$ \hfill (4-3)

Note that the wave signal of Figure 4-3 is repeating every 6283 seconds. The unique length of the wave signal can be much larger if the frequency interval is not constant. Smaller intervals could be used around the peak of the spectrum and larger intervals at the tail of the spectrum. For this case study however, such long unique wave signals are not needed.
Figure 4-3: Left: A wave signal of 6 hours and 15 minutes, generated from JONSWAP. Right: first 10 peak periods of same signal $H_s = 0.5m$; $T_p = 7.5s$, $\gamma = 3.3$

4-2-2 Simulation Modes

- The **Hybrid simulation mode** of DODO has a built-in interface with aNySIM-Pro. The aNySIM package can simulate multiple floating bodies including hydrodynamic interactions. This is done by solving the diffraction problem for each time step. To combine the multi-body solver from Matlab and aNySIM-Pro from MARIN a staggered approach can be used. Each time step information is exchanged between the Matlab multi-body solver and the aNySIM hydrodynamic solver.

- **Autonomous simulation mode** The hydrodynamics and multi-body dynamics can also be combined without constant use of the aNySIM interface. Some hydrodynamic cases do not require the diffraction model to be solved for each time step. If for instance the wetted surface of the floater does not change much when it oscillates close to equilibrium, or if there is no interaction between multiple floaters via radiating waves, one can choose not to re-iterate the diffraction problem for each time step but to solve the diffraction problem once in advance. This way the DODO simulation is autonomous once the diffraction problem has been solved for the first time. The **autonomous mode** of DODO is used.

4-2-3 First Order Linear Wave Forces

For the first order linear wave forces the following is assumed:

- Small displacements and rotations of the floater are assumed. The shape of the wetted surface remains close to its equilibrium and is assumed constant for the ‘autonomous mode’ of DODO.

- The DIFFRAC module of AnySIM computes the Froude-Krylov forces of the undisturbed wave for all panels around the floater, and translates their total sum to the
COG of the floater. The Froude-Krylov forces and moments are calculated for a series of wave frequencies and wave angles. DIFFRAC accounts for radiation and diffraction effects via frequency dependent matrices that contain the added mass and hydrodynamic damping. These matrices $B(\omega)$ and $A(\omega)$ have size 6x6.

- The solution set provides the following frequency range for the added mass, damping, and Froude Krylov forces: $\omega = 0 : 0.05 : 3 \ [\text{rad/s}]$.
- The solution set provides the following wave angles for the Froude-Krylov forces: $\text{WaveAngle} = 0 : 5 : 355 \ [\text{degrees}]$.
- Table 4-2 shows which types of input and output are [17]
- The wetted surface is divided into quadrilateral panels. An illustration of the quadrilateral panel model is shown in Figure 4-4 below.

Appendix E-1 shows a selection of the resulting .hyd-file. The full file is not shown since it is over 14 thousand lines. The shown data is sufficient for a simple verification of natural frequencies.

<table>
<thead>
<tr>
<th>Input</th>
<th>Output (.hyd-file)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Floater Geometry (PATRAN-file)</td>
<td>Added mass and damping coefficients</td>
</tr>
<tr>
<td>Floater COG</td>
<td></td>
</tr>
<tr>
<td>Floater Radii of inertia</td>
<td>First order wave exciting forces, moments, and phase shifts</td>
</tr>
<tr>
<td>Water Depth</td>
<td></td>
</tr>
</tbody>
</table>

### 4-2-4 Added Damping

The potential flow solver of AnySIM underestimates the damping for some degrees of freedom. Especially since the CSD is rather sensitive to beam waves, sufficient roll damping is important for the model behavior. The added roll-damping is assumed 4% of the critical damping.
4-3 Ladder Hydrodynamics

In reality the hydrodynamic force on the cutter ladder will depend on surrounding water particles. The following can be said about the ladder hydrodynamics in the model:

- When the cutter ladder makes a relative oscillation with respect to the water, the water has to move around. These water particles induce drag-, and inertia forces on the slender ladder body. The resulting forces can be calculated with the Morison equation [16]. The Morison equation acts perpendicular to the slender ladder body. The force per unit length is integrated over the submerged length of the ladder body.

\[ f = f_i + f_d = \dot{\rho} V \hat{u} + \rho C_a V (\hat{u} - \hat{v}) + \frac{1}{2} C_d \rho A \cdot |u - v| \cdot (u - v) \]  

(4-4)

In Equation 4-4 the vector \( v(t) \) is the localized velocity of a water particle, \( u(t) \) is the localized velocity of the ladder section of unit length. \( V \) is the volume per unit length, \( A \) is the area per unit length. The coefficients \( C_d \) and \( C_a \) are respectively the drag force coefficient and the added mass coefficient.

- The Morison force in the model treats the water as static medium. This means that the potential flow field does not play a role. Neither the undisturbed wave, nor the diffracted-, and radiated waves from the floater have an effect on the ladder hydrodynamics. The vector of water particle velocities is \( v(t) = 0 \). The Morison equation reduces to:

\[ f = \dot{\rho} V \hat{u} + \rho C_a V \hat{u} + \frac{1}{2} C_d \rho A \cdot |u| \cdot (u) \]  

(4-5)

The first term is the entrapped mass of water inside the ladder. The second term is the added mass of the water because it has to move around the cutter ladder. The third term is the quadratic drag force term.

- If the Morison forces would be also based on the potential flow field of the water particles, a first step could be to use the potential flow field of an undisturbed wave. The water particles in such a flow field have cyclic motions as indicated in Figure 4-5. Note: this is not modeled.

![Figure 4-5: Flow field of undisturbed wave around cutter ladder](image)

- The term \( f_i \) of Equation 4-5 is integrated over the length of the cutter ladder to obtain the force vector on the ladder body. The added mass and added inertia can be put into matrix \( M \) of the Newton-Euler set of equations of motion. Although it depends on the frequency of oscillation of the model, the added mass is set as a constant to keep the model simple.
The quadratic drag force term is not used in the linearized equation of motion. As such it does not contribute to the transfer function.

All in all the following added mass is used for the cutter ladder:

\[
A_{\text{ladder}} = 10^3 \begin{bmatrix}
230 & 0 & 0 & 0 & 0 & 0 \\
0 & 1540 & 0 & 0 & 0 & 0 \\
0 & 0 & 2170 & 0 & 0 & 0 \\
0 & 0 & 0 & 12750 & 0 & 0 \\
0 & 0 & 0 & 0 & 186330 & 0 \\
0 & 0 & 0 & 0 & 0 & 134120
\end{bmatrix} \begin{bmatrix}
\text{kg} \\
\text{kg} \\
\text{kg} \\
\text{kg} \\
\text{kg} \cdot \text{m} \\
\text{kg} \cdot \text{m}
\end{bmatrix}
\]

4-4 Mooring Forces

The CSD maintains its position with a mooring system. This comprises a spud pole at the aft and swing wires near the ladder tip.

1. The Spud Force is briefly explained in Section 4-4-1. The spud force was already modeled by IHC. The effect of the spud mooring system was checked with a basic verification, which is also discussed.

2. The Swing Wire Force is described in Section 4-4-2.

4-4-1 Spud Force

The spud is designed to resist moment- and shear lines as sketched in Figure 4-6. One extreme case is where the tip of the spud is essentially a pinned connection at the seabed. Another extreme case is where the lower part of the spud is penetrating the soil, essentially being clamped by the seabed. The intermediate case is ‘soft clamped’ in which the penetrated soil provides some rotational stiffness (but no rigid clamping) [8].

![Figure 4-6: Sketch of moment- and shear lines of spud. Left: spud pinned. Right: spud clamped.](image-url)
The spud has already been modeled in DODO for previous workability studies of CSDs in mooring condition. The following aspects of the spud mooring system are important:

- The modeled spud position with respect to the COG of the floater body is $x_{spud} = -39.55m$.

- The tubular beam segments have stepwise wall thickness. The spud is relatively thin walled at the tip, where small bending moments are expected. The wall is relatively thick around the lower spud connection with the floater, where the highest bending moments are expected. The wall thickness of the tubular beam and the length of the modeled segments are given in Table 4-3. The outside diameter of the spud is $D_o = 1.8m$. The assumed Young's modulus of steel is $E = 210 \cdot 10^9 Pa$.

- The bending stiffness in the model is determined by a FEM model. The basis of the FEM model is clearly explained in the textbook ‘Fundamentals of Mechanical Vibrations’[24] and will not be described further.

- In reality, besides bending stiffness of the tubular segments and rotational stiffness of the penetrated soil, the hydraulic system (that is designed to prevent spud bending overload) also provides stiffness. The hydraulic mechanism acts as a rotational spring between the spud-keep and the floater. When the relative spud vs. floater rotation exceeds a certain threshold, the spud safety mechanism opens a hydraulic valve which enables the spud-keep to rotate some more. This has a soothing effect on the bending stress. The spud-keep has a longitudinal and a transverse buffer system. Note that this mechanism is designed to prevent permanent damage and that this valve opening event is beyond the operability limit of the CSD. As can be seen in Figure 4-7, the end of the buffer is reached when the floater pitches over 3 degrees. The spud will be damaged if the floater pitches more than 3 degrees.

- The characteristic rotational stiffness of the spud mooring system is illustrated in Figure 4-7. Around equilibrium, up to the threshold values of the buffer, the pitch stiffness of the spud is linear. The spud characteristic is different for roll than for pitch, since the longitudinal and transverse buffer systems have different designs. For FD-analysis the spud buffer system cannot be modeled. Although it is possible for TD-simulations, **the buffer system is not used** in the model. This means that the spud mooring stiffness in the model is **linear**.

- The harder the seabed is, the smaller the spud penetration generally is. Spud penetration is neglected for the case study. The model therefore has a spud with pinned connection to the seabed, as sketched on the left of Figure 4-6.

<table>
<thead>
<tr>
<th># segment</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wall Thickness [mm]</td>
<td>35</td>
<td>40</td>
<td>50</td>
<td>60</td>
<td>70</td>
<td>75</td>
<td>60</td>
<td>50</td>
<td>30</td>
</tr>
<tr>
<td>Length segment [m]</td>
<td>11.8</td>
<td>2.9</td>
<td>4.8</td>
<td>2.9</td>
<td>2.9</td>
<td>2.85</td>
<td>2.9</td>
<td>2.9</td>
<td>11.8</td>
</tr>
</tbody>
</table>
Verification of Spud Stiffness

The spud mooring stiffness can be verified with a simple calculation of eigenvectors and eigenvalues of the equations of motion. The homogeneous equation of motion of a mass-spring-damper can be written as Equation 4-6 below.

\[ M \cdot \ddot{q}_1 + B \cdot \dot{q}_1 + C \cdot q_1 = 0 \]  \hspace{1cm} (4-6)

If extra mass, spring, and damping matrices are added to the system (only spud mooring, no ladder) the following set of equations is obtained:

\[ (M + A(\omega)) \cdot \ddot{q}_1 + B(\omega) \cdot \dot{q}_1 + (C + K) \cdot q_1 = 0 \]  \hspace{1cm} (4-7)

The floater has a 6x6 mass matrix \( M \) and a 6x6 added mass matrix \( A(\omega) \). The floater has a 6x6 hydrodynamic damping matrix \( B(\omega) \) and a 6x6 hydrostatic spring stiffness matrix \( C \). The 6x6 matrix \( K \) of Equation 4-7 represents the spud mooring stiffness. The diagonal terms and the coupling terms can be calculated with Equations 4-8 to 4-17 [7] (based on Euler-Bernoulli beam theory).

- The spud length from the keel of the floater to the assumed pinned connection at the seabed is \( L = 17m \). \( K_G = 6.8m \). The Youngs modulus of steel is 210GPa. The spud position with respect to the COG of the floater is \( x_{spud} = -39.55m \).

- Please note: in the model the spud is not prismatic, for this simple check however, it is assumed prismatic. Segment #1 of Table 4-3 is the longest segment (except for segment #9 but that is above deck level), this is assumed as prismatic cross section of the spud.

- Looking ahead at Section 5-1, a very prominent natural frequency of the system is close to \( \omega = 0.6[\text{rad/s}] \). If the added mass matrix \( A(\omega) \) and the damping matrix \( B(\omega) \) corresponding to \( \omega = 0.6[\text{rad/s}] \) are plugged into Equation 4-7, at least one of the eigenvalues should be close to this frequency. The corresponding eigenvector should indicate a mode with surge, sway, and pitch. Note that the added mass and damping depend on the wave frequency, so the found natural frequencies that are far away from \( \omega = 0.6 \text{ rad/s} \) may not be accurate.
The state-space representation of the spud-moored floater is given in Equation 4-18. The matrix $E$ has the size 12x12. Top left in matrix $E$ are 6x6 zeros. Top right in matrix $E$ is a 6x6 identity matrix. Bottom left is the 6x6 spring stiffness matrix, which is left-multiplied with the 6x6 inverse mass matrix. Bottom right is the 6x6 damping matrix, which is left-multiplied with the inverse mass matrix.

$$[\dot{q} \ddot{q}] = E \cdot [\dot{q} \ddot{q}]; \quad E = \begin{bmatrix} 0 & I \\ -(M + A)^{-1}(C + K) & -(M + A)^{-1}B \end{bmatrix}$$ (4-18)

The eigenvalues of $E$ represent the natural frequencies of the spud-moored floater. The corresponding eigenvectors tell in which mode these natural oscillations of the floater occur. The resulting matrices of this check can be found in Tables G-2 and G-1.

Looking ahead to the RAOs of the full DODO model in Figure 5-1, it can be seen that the first natural frequency ($\omega \approx 0.7 \text{ rad/s}$) of the floater is close to the distinctive surge, and pitch peak that is visible in the figure. From this it is concluded that the linearized equation of motion by DODO is close to the state space system (of one body) that was constructed for this spud stiffness verification.

### 4-4-2 Swing Wire Force

The swing wires are modeled as linear springs. The stiffness of the swing wires is simply calculated with Equation 4-19, based on the left of Figure 4-8. The factor 0.6 is assumed since the steel wire consists of many bundles of smaller steel wires, reducing the nett cross sectional area.

$$W = -k\delta L; \quad k = \frac{EA}{L_0}; \quad A = 0.6 \cdot \frac{\pi}{4} D_{wire}^2$$ (4-19)

As explained in Section 1-4, the heading of the CSD model is constant while the seabed moves through the model space. The wire length $L_0$ for both swing wires is set constant during one simulation.
Important remarks on swing wire effects:

- In the model the CSD does not swing but the seabed moves through the model space. The average swing wire direction is constant in the model, since the anchors are fixed points in the model space. Also, both swing wires are parallel in the model. In reality the swing wires are not necessarily parallel and their direction will change along with the swing of the CSD.

- In reality the swing wire lengths $L_0$ change along with the CSD swing. As Equation 4-19 shows, the effective wire stiffness depends on $L_0$. However, the swing wires of the model have constant $L_0 = 50\sqrt{2}$ m. The swing wires are tied to the cutter ladder at $x = [15.9; 0; 0.3]$ m with respect to the ladder coordinate system.

- In reality the swing wires may be sagging due to gravity, as illustrated on the right side of Figure 4-8. Close to the anchors the wire may or may not be resting on the seabed for a length $l$. The swing wire could be modeled as catenary wire, introducing higher order spring forces. However, not all required parameters to model this adequately are deterministic. Only first order linear spring forces are modeled to keep the model simple.

- In reality, due to the swinging motion of the CSD, the orientation of the wires constantly changes. The wires may (for some short length close to the anchors) sweep over the seabed, illustrated in Figure 4-9. This could introduce friction, thus some sort of damping. This is not modeled.

- A catenary wire with constantly changing tension will make constant lateral motions. To some extent the swing wires move laterally through the water and may have added mass and viscous damping. This force per unit length could be modeled by the Morison formula. Compared to other assumptions on the hydrodynamic forces of the cutter ladder, this effect should be rather small. This is not modeled.

- In reality the swing winches can be controlled with ‘constant tension’ or ‘constant speed’ programs. This is done by means of a PID controller on the winches. The PID controller keeps the tension or haulage speed as constant as possible. In the model this is not controlled. The seabed passes through model space with constant average swing speed.

- In reality the swing anchors may be dragged over the seabed. In the model the anchors are fixed nodes.
Figure 4-9: Top view of a CSD making half of a swing. Close-up of the swing wire near anchor, where the swing wire may or may not sweep on the seabed (effect is not modeled)
4-5 Model 0 for Frequency Domain Analysis

A FD-analysis can be done assuming a quasi-static mooring lay-out. Taking the Laplace transform of the linearized equations of motion, a transfer function can be found for the different degrees of freedom of the system. From this the Response Amplitude Operator can be derived. This may provide some insight how the CSD tends to move while being restrained by the mooring system. A flowchart is illustrated in Figure 4-10.

The following is important:

- All mooring stiffness and external wave forces of the model are linearized. The Laplace transform of the equations of motion, Equation 2-23 will lead to transfer functions of the model. The response spectra can be treated in two ways to estimate the most probable maxima $R_a$ of the response amplitudes. $R_a$ can be calculated directly if it is already defined in a transfer function $H_{R_a}(\omega)$. In that case it simply has to be multiplied with the wave spectrum $S_\z(\omega)$. It can also be estimated indirectly, first calculating a time trace from the body states and consequently deriving all model forces based on their respective force functions. This is explained in Section 5-4.

- This FD-analysis does not represent the dynamics of the CSD in operational condition (when cutting). The cutter force is non-linear and not suitable to incorporate in the FD-analysis.

- It will take at least tens of wave periods for the CSD to perform one full swing, see Equation 4-20. Because this swing is relatively slow it makes sense to assume a quasi-
static system lay-out. This means that the swing winches are not moving in the model and therefore the equilibrium lengths $L_0$ of the wires remain constant.

- Constant water depth, constant spud setting, and constant ladder angle are assumed. The ladder angle is the angle between the longitudinal axis of the cutter ladder body and the longitudinal axis of the floater body (also explained in Section 3-1).

- In reality the CSD heading will gradually change during the swing. In the model multiple lay-outs with different wave angles and swing wire angles should be analyzed, as can be seen in Table 4-1.

One full swing will take about 5 minutes of time. This crude assumption is based on a swing angle from $\alpha = \pm 30^\circ$, swing radius $r_{\text{swing}} = 100m$ and swing speed $v_h = 20m/min$.

$$T_{\text{swing}} = \frac{r_{\text{swing}} \cdot 2 \cdot \sin(\alpha)}{v_h} = \frac{100 \cdot 2 \cdot \sin(30)}{20} \approx 5[\text{min}]$$

### 4-6 Cutter Forces

The forces on the cutter head are added for Model 1. The Cutter Force Function is separately described in Chapter 3. The following is assumed for the CSD model:

- The main specifications for the cutter in this CSD model are given in Table 4-4.

- An average face height and average step size are assumed, as shown in Table 4-5. Note that the transport of material is not accounted for in the model. This means that the model may correspond to material transport that is not realistic.

- The specific energy of the cut material is also shown in Table 4-5.

- The cutter force is induced at the ladder tip. The position vector of the ladder tip is $x_{\text{cutter}} = [19.9; 0; 0.3]m$ in the body-fixed reference system of the ladder.

<table>
<thead>
<tr>
<th>Max. Cutter Power</th>
<th>Average Cutter Diameter</th>
<th>Swing Speed</th>
<th>Rotational Speed</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{c,\max}$ [kW]</td>
<td>$D_c$ [m]</td>
<td>$v_h$ [m/min]</td>
<td>[RPM]</td>
</tr>
<tr>
<td>3500</td>
<td>2.673</td>
<td>20</td>
<td>30</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 4-5: Modeled face geometry, specific energy and production</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average Step Size</td>
</tr>
<tr>
<td>$L_{\text{step}}$ [m]</td>
</tr>
<tr>
<td>$D_c/2$</td>
</tr>
</tbody>
</table>
4-7 Model 1 for Time Domain Simulations

Model 1 will be subjected to TD analysis. An overview of the TD process is illustrated in Figure 4-11. Since the time domain simulations use historic data from previous time steps the flowchart shows some feedback arrows. The simulations result in motion- and force signals in time domain. These can be used for an operability assessment.

![Figure 4-11: TD analysis outline](image)

This section describes how the TD-simulations are performed for the model.

1. The Time Domain simulation does not immediately produce a useful result, since the response has to build-up. This is explained in Section 4-7-1 Simulation Sequence.

2. The retardation function that is used by the Cummins equation uses a set of Retardation functions in the convolution integral. Treatment of retardation functions is briefly discussed in Section 4-7-2.
4-7-1 Simulation Sequence

The Time Domain simulation runs through a certain sequence, which is explained in this section. A schematic view of the sequence is given in Figure 4-12.

- Before the simulation starts, the Cutter Force Function is run for a short time while it has a predefined path through the seabed matrix. This is done by fixing the cutter in the model space and letting the seabed matrix move through the model space with constant haulage speed $v_h$. The seabed has an ideal and constant shape. From this short time span the average cutting force is determined.

- At $t=0$ the fixed constraint of the cutter head is removed and the average cutting force is applied to the tip of the cutter ladder. The system is in quasi static equilibrium (including average cutter force). In order to calculate the system equilibrium, a force vector $F_{eq}$ is added to the floater body on the right hand side of Equation 2-25. The average cutting force is now part of $F_{mb}$.

\[
\begin{bmatrix}
(M + m) & 0 \\
0 & M_{mb}(q)
\end{bmatrix}
\begin{bmatrix}
\Phi_q(q, \dot{q}) \\
\Phi_T(q, \dot{q})
\end{bmatrix}
\begin{bmatrix}
\ddot{q} \\
\lambda
\end{bmatrix}
= 
\begin{bmatrix}
F_w(t)\zeta - \int_{-\infty}^{t} K(t-\tau) \cdot \dot{q}(\tau) \cdot d\tau - Cq + F_s(q, \dot{q}) + F_{eq}
\end{bmatrix}
\]

Figure 4-12: Sequence of a Time Domain simulation

The initial values of the system in equilibrium are $q(0) = q_{eq}$ and $\dot{q}(0) = 0$. When the quasi-static equilibrium is calculated, the wave forces are $F_w(t) = 0$. The hydrostatic restoring spring force $Cq = 0$. The floater has no motion history, so the convolution integral is also zero. The right hand side must be zero, so:

\[
F_s(q, \dot{q}) = -F_{eq}
\]

- After the average cutting force is incorporated in the model and the quasi-static equilibrium of the model is calculated, the hydrodynamic force starts to act (from $t = 0$). The hydrodynamic wave force signal will cause the system to start oscillating.

- The system will undergo a build-up time at least as long as the cut-off time of the retardation function (see Equations 2-5 and 2-6). A cut-off time of 60 seconds is assumed. This is explained in Section 4-7-2. With this 60 seconds of cutoff-time it is assumed that the hydrodynamic damping effect of the longest waves is sufficiently accounted for.
• The system may still have to build its response after the cut-off time has passed. To shorten the build-up time of the response signal (due to the wave force signal), the force could be gradually introduced by an S-curve function. This is not used in the model.

The first stage of the simulation starts with the model in default position.

4-7-2 Retardation Functions

The convolution integral of the Cummins Equation (2-5) is solved. The retardation becomes very small after \( t=60s \). Instead of integrating the complete simulation history the convolution integral is therefore cut-off at 60s. The floater has six degrees of freedom, so it has \( 6 \cdot 6 = 36 \) retardation functions. Figure 4-13 shows some of these retardation functions for the floater.

![Figure 4-13: Retardation functions of floater](image)

The domain of integration can therefore be limited to a so-called cutoff time to reduce the computational work of the simulation, while still sufficiently representing the historic effects...
of the potential flow field. The ‘ideal’ cutoff time for low frequency waves is longer than for short frequencies. Nonetheless the cutoff time is limited since the presence of low frequency waves is also limited by the used spectrum type and the sea state definition. This comes with a downside: hydrodynamic damping of the lower frequency waves that would otherwise be there, is now discarded by the cutoff time. This can lead to a ‘numerical underestimation’ of damping.

4-8 CSD Model - Summary

The CSD model has been explained. It consists of two bodies that are kinematically related via a revolute constraint. The first body of the system is the floater, the second body is the ladder. The model is built up such that it is in equilibrium position. The equilibrium state is calculated with forward kinematics. A second constraint is used in order to fix the relative ladder pitch. The constant distance constraint is kinematically defined. It eliminates the 7th degree of freedom of the system. The constant distance constraint can be easily removed by a spring or a spring-damper system to create a more advanced model that represents reality better. This would re-enable the 7th degree of freedom again.

The mooring system (spud and swing wires) have been explained. Several quasi-static mooring lay-outs and wave angles are considered. The spud force is based on a FEM-model. With a finite difference method (see Section 2-2-3) the linearized spud stiffness matrix is calculated. A basic check of the linearized spud stiffness (from the FEM model) is done by calculating the first natural frequency of the system with a prismatic tubular beam as spud. The hydrodynamic coefficients and first order hydrodynamic force vectors are imported from a DIFFRAC-file. This file contains a solution set for a range of wave frequencies and wave angles. The hydrodynamics of the cutter ladder are modeled with the well known Morison equation. The quadratic drag force term of the Morison equation is left out in Model 0. The linearized Model 0 can be used for a FD-analysis. Although the FD-analysis of Model 0 is only valid for the quasi static mooring condition, it does provide valuable information for the simulations of the operational condition.

Finally, the cutter force function is implemented on the tip of the cutter ladder to obtain Model 1. Details on the cutter force can be found in Chapter 3. The simulation goes through separate phases in order to build up its response. In the first (initialization) stage of the simulation the average cutter force is calculated. During this stage the hydrodynamic forces are zero. This average cutting force is accounted for in the equilibrium equation of the system. When the simulation is started (with the model in equilibrium), the floater starts to oscillate due to the wave forces. This startup-phase should at least be as long as the cut-off frequency of the retardation function but the response build-up may take longer. For this reason the first 30 wave periods are considered not very reliable. After the build-up time has passed, the Time Domain simulations delivers reasonable signals for the operational condition. One mooring lay-out of the CSD is considered for the TD-simulations.
Chapter 5

Results FD - Model 0

This Chapter shows the results of the Frequency Domain analysis for Model 0 (mooring condition). An overview is given below:

1. Lay-out #2 has a Swing Angle of 0 degrees. The waves travel exactly along the x-axis of the model. The RAOs of the floater are briefly discussed in Section 5-1.

2. Lay-out 1 and lay-out 3 are assumed to have equal RAOs, since the model of the CSD in mooring condition is symmetrical. The effects of the wave angle and mooring lay-out are discussed for the model with Swing Angle of 20 degrees. This can be found in Section 5-2.

3. The transfer functions give the frequency responses of the system. From this the response signals in other local positions of the floater can be determined. These oscillations will have the same mode as the COG but different amplitudes. This is briefly described in Section 5-3.

4. The responses of the linearized model can be analyzed with Short Term Statistics. This is described in Section 5-4. A CSD is no permanent structure because the operations can be ceased and the CSD can head back to port. For this reason only short term statistics are addressed.

Please note that the results of the FD-analysis depend on a set of general assumptions, which are listed below.

- Wave Angle and Water Depth. For each wave angle and water depth the water particles interact differently with the floater. The aNySIM algorithm therefore returns different hydrodynamic coefficients. In this case study the water depth is kept the same for all lay-outs. Water Depth = 20m. Two wave angles are considered: one for layout #2 and one for lay-outs #1 and #3.
• **Spud Mooring System.** The spud mooring stiffness depends on the water depth, the amount of soil penetration and the soil stiffness. It also depends on the design of the hydraulic spud guard system. The spud introduces coupling between several degrees of freedom via its K-matrix. The spud stiffness of the model is kept the same for all lay-outs. For stiffness see Section 4-4-1.

• **Swing Wires and Ladder Angle.** The direction of swing wires and their effective stiffness depend on the CSD heading and lay-out of anchors (see Section 4-4-2). The swing wires introduce coupling terms in the K-matrix of the system. These coupling terms also depend on the ladder angle. The ladder angle is kept the same for all lay-outs. Ladder angle $\theta_{ladder} = 30^\circ$. This corresponds to a dredging depth of about 20 meters.
5-1 Swing Angle 0 Degrees - Model 0

The results from the lay-out #2 are shown in this section. Two arbitrary JONSWAP wave spectra are plotted in the same figures as the RAOs of the CSD. This is to relate the distribution of wave energy to the sensitivity of the system at certain frequencies. To obtain the RAOs, the absolute value of the transfer function is divided by the wave elevation: $|H_{R,\zeta}|/\zeta$ [16]. This is because the phase lag of the system is not relevant in this case.

Please note: the plotted spectra are arbitrarily chosen and that the sea states in this case study may vary from $3.5s < T_p < 13.5s$.

![Figure 5-1: RAOs of the CSD model for 6 DOF. Compared to typical wave spectra](image)

The following can be seen in Figure 5-1:

- For this lay-out the swing wires have a negligible effect in x-direction. The large peak for surge (at $\omega \approx 0.5rad/s$) must therefore be caused by the modeled stiffness of the spud mooring system. A simple The shown wave spectra hardly harness any power at $\omega = 0.5rad/s$. The response peak however, is quite wide so part of the wave energy of a
sea state with $T_p = 7.5\text{s}$ will cause a considerable surge excitation. For sea states with longer peak periods $T_p$ their spectral peaks are shifted towards lower frequencies. The surge excitation for these sea states will be larger, which leads to higher loads on the spud mooring system. This illustrates the poor workability of the spud mooring system very clearly. For sea states with Pierson-Moskowitz spectra the problem may be even worse, since they generally have more wave energy at lower frequencies (with respect to the peak period $T_p$) than the JONSWAP wave spectrum.

- One can clearly see that the waves in layout #2 have no effect on sway motion of the model. This is because of the symmetry with respect to the xz-plane of the model space. The same applies to roll and yaw.

- The RAO for heave shows no anomalies.

- The RAO for roll is flat for the same reason as surge and yaw.

- The pitch RAO of the model also shows a large peak at $\omega \approx 0.5\text{rad/s}$, just as surge does. The RAO for the floater itself has a maximum which is only half of the maximum for Model 0. This tells that the spud mooring system introduces a strong pitch-surge coupling. The swing wires will do nothing for pitch in this lay-out, since they act only in lateral direction.

- The RAO for yaw is flat for the same reason as roll and yaw.

Looking ahead, the surge and pitch amplitudes at low wave frequencies might cause problems in the future for the TD-simulations of Model 1. As a consequence of surge and pitch, the tip of the cutter ladder might be somewhat poking the seabed. This might introduce strongly varying cutter forces (both in magnitude and direction) that could actuate this kind of behavior even more.
5-2 Swing Angle 20 Degrees - Model 0

In this section the results from the lay-out #3 are shown. The results for lay-out #1 must be the same since this is exactly a mirror of lay-out #1 with respect to the xz-plane of the model space.

The following can be seen in Figure 5-2:

- The peak in the surge RAO has slightly shifted up to \( \omega = 0.6 \text{rad/s} \). This is because of the increase of longitudinal stiffness, resulting in a higher longitudinal natural frequency. It is expected that for larger swing wire angles this effect becomes larger.

- The large sway amplitudes of the CSD only occur at very low frequencies. This is where most of the energy of the shown wave spectra is not.

- The RAO for heave shows no anomalies.

Figure 5-2: RAOs of the CSD model for 6 DOF. Compared to typical wave spectra
• The swing wires and spud mooring system restrain roll motion. For \( \omega > 0.9 \text{rad/s} \) this is effective but at lower frequencies the response amplitudes may cause problems. The RAO which was directly imported from aNySIM (the .hyd-file) shows the behavior of only the floater. It has a very sharp peak around \( \omega = 0.75 \text{rad/s} \). This is because aNySIM does not account for viscous roll damping. The model however, does. The viscous roll damping causes the sharp peak to shift to a lower frequency and it is reduced to a relatively small bump (just above \( \omega = 0.5 \text{rad/s} \)).

• The response amplitudes for pitch are almost exactly the same as for lay-out #2. Compared to the effect of the spud stiffness, the pitch effect of the swing wires is very small. This is because the swing angle is only 20\(^\circ\) and the longitudinal spring stiffness of the swing wires is relatively small for this lay-out.

• The RAO for yaw shows a very sharp peak at \( \omega = 0.2 \text{rad/s} \). Since the model does not incorporate any yaw damping, it allows the peak at \( \omega = 0.2 \text{rad/s} \) to be very sharp. It is however not in the frequency range where most of the wave energy is, so this sharp peak is not a problem. The bump between \( 0.4 \text{rad/s} < \omega < 1 \text{rad/s} \) may cause a problem in combination with the cutter force, which can excite the CSD model at the tip of the cutter ladder in lateral direction.

5-3 Response Signals - Model 0

The Response Amplitude Operators of the CSD were shown in Section 5-1 and Section 5-2. Section 2-2-3 shows how the transfer functions for the bodies were calculated. For a local point \( p \) on these bodies, the longitudinal, lateral, and vertical displacement are given by Equations 5-1 to 5-3 [16]:

\[
\begin{align*}
x_p &= x - y_b \cdot \psi + z_b \cdot \theta \\
y_p &= y + x_b \cdot \psi - z_b \cdot \phi \\
z_p &= z - x_b \cdot \theta - y_b \cdot \phi
\end{align*}
\] (5-1)

The complex transfer functions can be combined to obtain the transfer functions of local displacements:

\[
\begin{align*}
(H_{x, \zeta})_p &= H_{x, \zeta} - y_b \cdot H_{\psi, \zeta} + z_b \cdot H_{\theta, \zeta} \\
(H_{y, \zeta})_p &= H_{y, \zeta} + x_b \cdot H_{\psi, \zeta} - z_b \cdot H_{\phi, \zeta} \\
(H_{z, \zeta})_p &= H_{z, \zeta} - x_b \cdot H_{\theta, \zeta} - y_b \cdot H_{\phi, \zeta}
\end{align*}
\] (5-4)

The local displacements of the spud keep can be related to the spud bending stress (\( \sigma_{xx} \) and \( \sigma_{yy} \)), and the spud shear stress (\( \tau_x \) and \( \tau_y \)) via the Euler-Bernoulli beam theory. The transform functions for these stresses can be a composed in a similar fashion as shown above.

After the desired transfer functions are calculated, the response spectra can be found with Equation 5-7 [16].

\[
S_{R, \zeta}(\omega) = |H_{R, \zeta}(\omega)|^2 \cdot S_{\zeta}(\omega)
\] (5-7)

Another way to calculate the spud force is to use time domain signals of the floater motions at the spud keep position. The spectra \( S_{R, \zeta} \) for the local translations and rotations can be transformed to the time domain by the superposition principle of individual harmonic waves.
The method is the same as the generation of a wave signal $\zeta$ from $S_\zeta$ as described in Section 4-2-1. Using the linearized equation of motion 2-21, and 2-20 the signal of spud forces on the floater body can be calculated by DODO. The problem of using a time domain signal as a sample for an operability assessment however, is that it has finite length (thus a finite number of peaks). There is a way to mitigate this but it still is very important to use a time record of sufficient length. This will be discussed a bit more in Section 5-4.

## 5-4 Short Term Statistics - Model 0

Figure 5-3 shows a histogram of the wave crest height and a Rayleigh probability density function (pdf) fit. Due to the linear character of Model 0 its response to the waves is also Rayleigh distributed. The short term pdf of a response amplitude $R_a$ is given in Equation 5-8, in which the $m_{0R}$ is the area under the response spectrum $S_{R\zeta}$. For this assumption the response spectrum of the system should be narrow banded and the sea state should be stationary and Gaussian as described in [2].

\[ f_{st}(R_a) = \frac{R_a}{m_{0R}} \cdot \exp \left( -\frac{R_a^2}{2 \cdot m_{0R}} \right) \quad ; \quad m_{0R} = \int_{0}^{\infty} S_{R\zeta}(\omega)d\omega \quad (5-8) \]

![Figure 5-3: Probability density of wave crest height $\eta_{crest}$ (from the wave signal example of Section 4-2-1)](image)

### 5-4-1 Probability of Exceeding

There are two ways to estimate the probability that a response exceeds a certain value. First, it can be directly calculated from the 0th-order moment of the response spectrum:

\[ P\{R_a > a\} = \exp \left( -\frac{a_{\text{max}}^2}{2 \cdot m_{0R}} \right) \quad (5-9) \]
The second option is to use a time record based on the random-phase/amplitude model. From this time record (sample) the set of response peaks can be identified by means of the zero-crossing definition. When a peak is drawn from the time record, the probability that this is not the largest possible peak of that sea state is:

$$P\{r_a \neq R_a\} = \left(1 - \frac{1}{N}\right)$$ \hspace{1cm} (5-10)

When \(N\) independent peaks are drawn from the time record, the probability that none of the peaks is the most probable maximum peak can be written as Equation 5-11. This probability converges when \(N\) becomes larger.

$$P\{r_a \neq R_a\} = \lim_{N \to \infty} \left(1 - \frac{1}{N}\right)^N \approx 0.368$$ \hspace{1cm} (5-11)

Holthuijsen states that the maximum value of the probability density function for \(r_{a,max}\) is close to its mode [2]. This is interpreted as the most probable maximum \(R_a\):

$$R_a \approx \text{mod}(r_a) \approx \sqrt{2 \ln N \sqrt{m_0}}$$ \hspace{1cm} (5-12)

Figure 5-4 shows how the probability \(P\{r_a \neq R_a\}\) converges for large \(N\).

<table>
<thead>
<tr>
<th>(N) (time record)</th>
<th>(\sqrt{2 \ln N})</th>
<th>10</th>
<th>50</th>
<th>100</th>
<th>500</th>
<th>1000</th>
<th>5000</th>
<th>10000</th>
<th>50000</th>
<th>(\sqrt{2 \ln N})</th>
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<td>1</td>
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<tr>
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<td>1,329</td>
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<td>3,717</td>
</tr>
<tr>
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<td>1,923</td>
<td>1,476</td>
<td>1,360</td>
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<td>1,110</td>
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<td>0,962</td>
<td>0,887</td>
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</tr>
<tr>
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<td>1,663</td>
<td>1,533</td>
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<td>1,127</td>
<td>1,084</td>
<td>1</td>
<td>4,652</td>
</tr>
</tbody>
</table>

Figure 5-4: Probability that one maximum peak out of a sample of \(N\) is not drawn for \(N\) times

With this relation in Equation 5-12 between \(N\), \(m_0\), \(r_a\), and \(r_{a,max}\) the significant amplitude of a signal can be determined from a relatively short time record for a long time span. This also means that the maximum peak of a time record can be extrapolated to a longer time span. Table 5-1 can be used for this.

Table 5-1: Conversion table for significant response amplitudes of samples with different \(N\)
Please note that for Equation 5-12 implicitly is assumed that the signal is stationary, Gaussian, and that the peaks are Rayleigh distributed.

### 5-4-2 Operability Assessment for Spud

The operability assessment for the spud is very important since this is one of the most sensitive structures of the CSD. If the spud bending loads exceed a certain value, the spud and the spud carriage can be severely damaged. Because the spud of the model has a pinned support on the seabed, and the spud does not take any axial loads in the model, the bending stress of the spud is solely defined by the spud force as illustrated in Figure 5-5.

![Figure 5-5: Only the spud force on the tip defines bending load](image)

The maximum response of the spud is not directly calculated from the transfer function but from the time record of the localized floater motions (at spud-floater interface). From these time records the resulting force on the spud tip could be calculated. The maximum spud force

![Figure 5-6: Operability assessment regarding only the Spud Force](image)

is the square root of the x-, and y-component. The maximum allowed force on the spud tip is 1300kN. Figure 5-6 shows the operability scatter. Referring back to the example of Table 2-1, the operability of a CSD is a bit worse than expected.

The Safe Working Load (SWL) of the steel swing wires could be a very strict operability limit. For the cases of Model 0 (mooring condition) however, it is very unlikely that the SWL is ever exceeded. The loads on the swing wires will be higher when the CSD is in operational condition.

Other operability limits that relate to the wave conditions (and could be relevant) are:

- The vertical relative motion of the structure with respect to the undisturbed wave surface. This could be important when a floating pipeline is connected.
- The accelerations of the cutter repair platform at bow, when the CSD is moored while the cutter head is serviced.
- The operability limit of the auxiliary spud. This may be important when the main spud is relocated. The auxiliary spud has no buffer system.

Needless to say, operability limits are arbitrary and the list does not consist exclusively of the above. These aspects are however beyond scope of the project so will not be discussed further.
In this Chapter some results of Model 1 with lay-out #2 (see Section 4-1) are presented to point out some modeling effects.

1. Each Time Domain simulation starts by Initiating the System. Section 6-1 shows the importance of this.

2. TD- signals show different stability for Over Cutting vs. Under Cutting in 6-2.

3. Results for Model 1 with Swing Angle of 0 degrees are shown in Section 6-3.

6-1 Initiating the System

Before the simulation starts at $t = 0$ the system must be initiated. This was already explained in Section 4-7-1. The CSD model starts in equilibrium position.

Figure 6-1: Transient response with original function. Average cutter force not initiated. Under Cutting, $v_h = 30m/min$, $L_{step} = 0.5D_c$, $H_{face} = 0.5D_c$. Grid Sizes $\Delta x = 0.05m$, $\Delta y = 0.05m$, and $\Delta z = 0.05m$
Figure 6-1 shows what happens when the average cutting force signal is not accounted for in the equilibrium at $t = 0$. The system shows an unwanted transient response since the average cutter force acts like a step function on the model. The higher the $E_{sp}$ of the soil is, the more severe the transient response is. For some under cutting cases, the effects of this cost over 100 seconds to die out.

As can be seen from Figure 6-1, the torque signal shows a poor relation to the cutter head motions. This is because of the large quantization error. The large quantization error was inevitable for the original Cutter Force Function. The cutter-to-soil interface was discretized in a 3D matrix, which made it computationally unfeasible to reduce the grid size further than $\Delta x = \Delta y = \Delta z = 0.05m$. The way that the Cutter Force Function stored information in its matrices was very inefficient, since 8 bytes per element were used by Matlab to store a value which was only intended to be a percentage. After revision 2, the Cutter Force Function finally could be used for much smaller grid sizes, while it was also capable of running faster.

The torque- and motion signal of the revised function can be seen for the under cutting case in Figure 6-2.

Figure 6-2: Transient response with revised function. Average cutter force not initiated. Under Cutting, $v_h = 30m/min$, $L_{step} = 0.5D_c$, $H_{face} = 0.5D_c$. Grid Sizes $\Delta x = 0.01m$, $\Delta y = 0.01m$.

Figure 6-3: Average cutter force initiated. Under Cutting $E_{sp} = 5MPa$, $v_h = 30m/min$, $L_{step} = 0.5D_c$, $H_{face} = 0.5D_c$. Grid Sizes $\Delta x = 0.01m$, $\Delta y = 0.01m$.

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Figure 6-3 shows that indeed the system does not have to go through a transient response from the moment that the simulation starts at \( t = 0 \). The torque signal in Figure 6-2 shows that the delivered torque of the Cutter force Function may become higher than the average torque limit that was derived from cutter power specifications of the CSD. In reality the cutter drive will stall when the required torque becomes too high. As a result the cutter head will not be able to remove the seabed. Any further movement of the cutter into the seabed will not be a cutting one, but a bulldozing one. Above the torque limit of the cutter head, the cutter force signal from the model will be very different from the real situation.

Figure 6-4 shows that the model can become unstable in over cutting mode. Please note again: the modeled system has constant swing speed and unlimited cutter drive torque. In reality this behavior is thus not seen. However, it was merely to point out that the under cutting case in itself is stable, while the over cutting process in itself is unstable.

For the case of 5 MPa, the system is stable. The linearized transfer function must have negative complex poles, yielding an exponential decay. For the case of 6 MPa and 7 MPa the system seems marginally stable. For the case of 8 MPa, the system is unstable. The linearized system must have positive complex poles, yielding an exponential increase of the response amplitude.

6-2 Over Cutting vs. Under Cutting

Figure 6-4 shows that the model can become unstable in over cutting mode. Please note again: the modeled system has constant swing speed and unlimited cutter drive torque. In reality this behavior is thus not seen. However, it was merely to point out that the under cutting case in itself is stable, while the over cutting process in itself is unstable.

For the case of 5 MPa, the system is stable. The linearized transfer function must have negative complex poles, yielding an exponential decay. For the case of 6 MPa and 7 MPa the system seems marginally stable. For the case of 8 MPa, the system is unstable. The linearized system must have positive complex poles, yielding an exponential increase of the response amplitude.
The responses were checked for the same conditions in under cutting mode and they were all stable.

6-3 Swing Angle 0 Degrees - Model 1

The following plots of Time Domain simulations show a striking effect in the resulting spud forces for the mooring condition of \( H_s > 0 \), for the operational (over cutting) condition of \( H_s = 0 \), and for the operational (over cutting) condition of \( H_s > 0 \).

![Spud Force Model 0](image1)

![Spud Force Model 1](image2)

![Torque Cutter Drive](image3)

Figure 6-5: Upper left: Waves energy only. Upper right: cutter energy only (over cutting)
Lower left and right: Both wave energy and cutter energy; \( H_s = 0.7 \text{m}; T_p = 6.5 \text{s}; Esp = 4 \text{MPa}; H_{face} = D/2; L_{step} = D/2 \)

The horizontal lines in the spud plots are a reference to the operability limit of the spud. Needless to say, the blue limit is non-negotiable (broken spud). The other levels are based on the design of the spud carriage, its hydraulic components, and the probabilities of exceedance of certain threshold values. The red line in the plot of the torque signal is based on the maximum power of the cutter drive. An interpretation of the signals is given below:

- In the upper two plots in Figure 6-5 the system either got energy from the waves or it got energy from the cutter drive. As can be seen (in the plot upper left) the wave induced
motions of the floater were such that the spud load was well below its operability limit. The mean cutter force was (in the plot upper right) accounted for in the equilibrium. The spud force stayed close to zero (the system was stable).

- In the lower two plots, the system got energy from both the waves and the cutter. The system started in equilibrium, just as was the case for the upper two plots. For the first 250 seconds the required torque stayed below the torque limit. Exactly the same wave time series was used for the simulation of the upper left figure and the lower figures. Nonetheless, the spud load is very different.

- This clearly demonstrates that the cutter force takes its toll on the total damping of the system.

- For the shown simulations, the cutter torque only had three peaks that exceeded the torque limit. It can be assumed that for most of the time the haulage speed \( v_h = 0.33m/s \) would not lead to a stalling cutter.

- On \( t = 0 \) the average cutting force will not cause a transient response when the simulation starts. The incoming waves however, will cause a transient response. The spud force signal starts with \( F = 0 \) at \( t = 0 \) but quickly develops.

- Although the waves, the floater, and the mooring system are symmetrical, the motions and loads on the system are not. The cutting force module introduces coupling between the degrees of freedom in the XZ-plane (pitch, surge, heave) and the degrees of freedom out of the XZ-plane of the model (roll, sway, yaw). The roll, sway, and yaw motions of the floater cause the spud force to have a y-component.

Some final notes about the motion signals of the CSD model and the inherent load on the spud:

- The retardation functions of the hydrodynamic equations (some were plotted in Figure 4-13) contain a convolution integral which is important for at least 60 seconds of simulation history. Especially for longer wave periods this retardation function is important to account for the changing potential flow field near the floating body. After a start-up phase of approximately 60 seconds, the effect of the hydrodynamic retardation is sufficiently introduced. However, other effects may require the start-up phase to be even longer than 60 seconds.
This Chapter provides a discussion on several model features. A myriad of assumptions and choices was made in the process of designing the model. These were necessary to make the solution feasible, however, they also limited the effectiveness of real life application of the model. Thus, their effects define the credibility and reliability of the results. With this discussion a framework is provided on which the conclusions and recommendations of Chapter 8 are based. The main topics of the discussion are listed below:

1. The build-up of the **Cutter Force Function** in Section 7-1.
2. Features of the **CSD Model**, and their effects in Section 7-2.
3. Advantages and disadvantages of **Time- and Frequency Domain** in Section 7-3.

### 7-1 Cutter Force Function

In this section the used coordinate system, the Cutter Force Function and the cutting forces themselves are discussed. The different assumptions and their respective effects for the choice of coordinate system are mentioned. These influence the Cutter Force Function and the cutting forces. For each different version of the Cutter Force Function the improvements are explained.

#### 7-1-1 Coordinate System

The coordinate system of the Cutter Force Function was defined as an orthogonal basis in line with the general coordinate system. The function performs an elementwise search from a cutter matrix, $C$, and a seabed matrix, $S$. It finds which volume has to be cut at what time and at what location inside the cutter head this happens. For this method it was necessary that both matrices $C$ and $S$ had an equal spacing between rows and columns. Due to this
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restraint the seabed mesh cannot be chosen freely. This also implies that the orientation of the cutter-axis cannot change with respect to the \( S \) matrix which, in turn, means that the CSD may not change its heading. In other words, it is not possible for the CSD to make a swing. To by-pass this problem the CSD itself does not move through the model space but the \( S \) matrix does so instead. Apart from the fact that this is a counter-intuitive way of modeling, another disadvantage is that the wave direction and mooring lay-out also need to change to simulate a swing. When haulage speed (the speed at which the seabed passes) is increased, the mass and inertia of the CSD as well as additional hydrodynamic effects are completely neglected in the case that the CSD model has to retain a constant heading in the model space.

7-1-2 Discretization

The original version of the Cutter Force Function resulted in a binary force signal. This signal had a quantization error of the cut volume, \( V_{\text{cut}} \), which was too large for a decent simulation. To reduce this noise, it seemed logical to choose a finer grid spacing since the cut volume depended strongly on grid spacing of the 3D matrix and the relative speed between matrix \( S \) and \( C \). However, a finer grid spacing would yield an exponential increase of required workspace memory as well as computational effort to process the data. Matlab sets the class type and memory allocation of variables by default to ‘8-byte double’. It took some time before this was realized. In an effort to smoothen the signal without the use of larger matrices, the method from Section 3-2-3 was developed. This method somewhat smoothed the cutter production signal. To further reduce the residual noise a moving average of the last \( n \) number of time steps was used additionally. Although this was an extremely effective method of removing the noise, it introduced a time lag in the signal. For this reason, the moving average smoothing should not span more than a couple of time steps. (In the model a history length of \( n = 10 \) time steps was used, with time steps of \( \Delta t = 0.1 \) s.) Despite these efforts, the cutter force remained too noisy because the grid spacing was not small enough. Especially in cases where the soil had a higher specific energy \( E_{\text{sp}} \). It was important that the relation between cutter impression in the seabed and the resulting force should be accurately defined, meaning that the grid spacing had to be sufficiently small.

In the first revision of the Cutter Force Function all redundant matrix elements of \( C \) were removed. Only the elements that were part of the cutter hemisphere were checked on the presence of soil volume. This reduced the amount of rudimentary data by over 70% which allowed the grid size to be smaller. Although this seemed like a big leap forward it was not enough to mitigate the quantization error. In some way the quantization error had to be reduced significantly. In Matlab, the default class ‘double’ requires 8 bytes of memory to store per element. The matrices \( S \), \( C \), and 9 matrices on background to store the three unit-vector matrices of \( F_c \) all together had over a million of data points which had to be processed for each time step of 0.1 s. This alone was already a big problem, however, the way that the data points were stored was utmost inefficient which made the problem even bigger. Another malfunction of the algorithm was that the cutter head had a relatively large bandwidth in which it could move without any resulting change of production rate. The system did not respond very well to this.

In the second revision the matrices \( C \) and \( S \) were modified to be 2D matrices. This removed a vast amount of data points. Inside the elements of the 2D matrix the height of the seabed
surface was stored in the 8-byte data points instead of a filling ratio of the volumetric element as in the previous version of the function. It was a slight disappointment that the function could not make a spatial distribution of soil properties in 3D space anymore. Nevertheless, such spatial distribution could still be optional in the remaining two dimensions.

7-1-3 Cutting Forces

The implemented cutting forces are adopted from analytical 2-dimensional cutting theory. First of all, only the normal- and tangential forces are modeled. Forces that may act in axial direction, or pointing towards the pole of the hemisphere (for example a snow-plough effect), are not implemented. The specific energy which is determined for the excavation of the rock mass, and the resulting forces, are related to a strong simplification of the reality. The model does not consider the penetration depth of the cutting teeth which means that different failure-mechanisms, which can be dominant for different thicknesses of the cut layer, are averaged. Regardless of the thickness of the cut layer at the position where the volume is being cut, the same ratio of normal to tangential forces is used everywhere. This ratio is determined by the rock mass properties and the geometry of the cutting teeth. The model does not take into account the fact that these teeth wear fast and that this causes the ratio between the normal and tangential forces to become more unfavorable for the swing winches until, at a certain moment, the process could be considered more as a bulldozing than a cutting one. The average forces which the Cutter Force Function calculates for each generalized tooth can be justified on the relatively large scale of the CSD model. However, the fact that the time averaged cutting forces are between 30% and 60% of the peak power is worrisome. Even though the simulations give a clear understanding of the behavior of the total system, the accuracy of the results ought to be interpreted with a grain of salt. If only for the fact that the modeling of the excavation process brings large uncertainties with it, the quantification of the workability of a CSD in operational condition is barely feasible.

7-2 CSD Model

This section briefly discusses the most important differences between the model and reality. The choices which were made in modeling the components had several effects on the results. These are described as well.

7-2-1 Available Cutter Drive Power

There is not yet a built-in functionality to limit the available cutter drive power to a maximum level. This means that the cutter drive can deliver much more torque in the model than in reality. As a consequence, the resulting forces of the cutter head are also larger than possible in reality. This excessive influx of energy into the system worsens the already instable character of the excavation process. The Cutter Force Function is a type of complex negative damping which couples the different degrees of freedom of the model. An oscillation in the XZ-plane of the model can lead to amplified oscillations in the direction of the lateral mooring wires and vice versa. If one decomposes the movements of the cutter in vectors perpendicular and parallel to the cutter axis it is clear to see that the cutter oscillates mostly perpendicular to
the cutter axis. This movement is reinforced by the influx of energy of the unlimited available cutter drive power and the continuously moving seabed matrix. By any means a solution to limit the available cutter drive power, and to thus stop the uncontrolled oscillations which do not occur in real life, should be implemented.

7-2-2 Stiffness of the wires

The stiffness of the wires depends on the equilibrium length of the model in starting position. The effective stiffness of the wires heavily depends on the mooring lay-out. The mooring stiffness of the swing wires seems to have a large influence in the general behavior of the mooring condition. If the cutting forces are added to this, the importance of the swing wires may even become larger because of lateral cutting forces. The non-linear characteristics of a swing wire, in reality, would be very different for long and short sagging wires. The hauling swing wire would also have more pretension on it in comparison to a wire which is slacked in a controlled manner. In terms of the wires the model is a very simple representation of reality.

In this model the wires also do not have any damping. This leads to the fact that the model can oscillate in a lateral direction which limited damping on the ladder and the hull only suppress to some extent. Energy is accumulated in the wires originated from the cutter head and the waves in the model. The wires are tied on both ends: to the anchors and to the cutter ladder. In the model they can deliver larger forces than the swing winches can pull in reality. This enables the wires to accumulate more energy than should be physically possible. The oscillations of the model therefore are more severe than in reality.

7-2-3 Stiffness of Spud System

It was shown from the RAOs of the mooring condition that the system tends to oscillate in combined pitch and surge. This is a coupled movement which can be explained by the stiffness of the spud mooring system on the back of the CSD. The spud accumulates surge energy which later manifests itself in the pitch movements and vice versa. These oscillations can be clearly seen in the signals.

7-2-4 Axial Force on Cutter Shaft

In the model, the cutter ladder is directed at a downward angle. This angle in combination with the uncontrolled pitch and surge movements could lead to piercing movements of the cutter head. In reality, the cutter head is not able to pierce the ground in an axial direction at speeds higher than the haulage/swing speed.

7-2-5 Ladder Hoisting Wire

The ladder hoisting wire is, in reality, reeved multiple times in the block and has the same diameter as the swing wires. It therefore functions as a stiff spring. While dredging, the hoisting winch is not always actively regulated. The hoisting wire is implemented as a rigid bar without any weight in the model. This is done by implementing it as a constant distance...
constraint which means that during the simulations the ladder is completely coupled to the pontoon and that the model thus only had six degrees of freedom. For the results of the equation of motion in this case study it would have made no (or a minimal numeric) difference if the cutter ladder would be literally merged to the floater body. In the interest of versatility, the cutter ladder was modelled as a separate body. Due to the modular structure of DODO just one small step is necessary to replace the relative pitch constraint between the floater body and the ladder body by a different element. By modeling it this way, a seventh degree of freedom can be easily added to the system. The model is prepared to possibly be expanded to include a hydraulic passive heave compensated system or any comparable alternative.

7-3 Time- versus Frequency Domain

During this discussion the uncertainty introduced by generation of time records and the advantages and disadvantages of the usage of the Frequency Domain and Time Domain simulations are addressed. For a number of reasons the Time Domain simulations of the operational conditions are merely suitable to gain insight in the cutting process and to do qualitative studies, such as comparing the effects of different concepts and mooring lay-outs. They are not very efficient to quantify a workability for extended periods and a large set of sea states.

Spectral Representation of Sea-states

In theory, an endless series of wave signals, all having identical statistical properties, can be generated for any sea state. This spectral representation of the sea states provided a means to create the random wave signals which were used in the model. The stochastic nature in combination with the finite duration of the wave signals in the model introduces an uncertainty. The statistical results ought to be interpreted carefully, especially when an operability assessment is performed. One should determine whether the short-term statistics (of finite length) will have led to the right probability of exceedance for the response in question. To lessen this uncertainty, the simulations should cover at least hundreds of wave periods and multiple time traces per sea state should be generated.

Although Table 5-1 does provide a relation between the the number of waves and relative multiplication factors for peak responses, it is important to realize that this is a statistical trick. The estimated maximum response amplitudes become less and less reliable the more the duration of the time record is stretched.

7-3-1 Model 0 in Frequency Domain

A linearized model that accurately represents reality, is very suitable for a full-blown workability analysis. Since the CSD in mooring condition can be accurately represented by a linear model, it is straightforward to use its frequency response spectrum. By using the frequency response spectrum, the most probable maximum peak responses can be found directly. Alternatively the superposition principle can be applied. The computation of ‘linear time domain’ time traces is computationally cheap. Long time traces are thus easily obtained which efficiently lessens the statistical uncertainty. Moreover, if a Rayleigh distribution can be used for
the extreme wave height together with the linearity of the model, the same distribution can be applied to the extreme responses. This can be exploited to fit the short-term statistics to longer time frames.

7-3-2 Model 1 in Time Domain

A full-blown workability assessment for Model 1 is cumbersome and utterly inefficient. This is in view of the fact that the system could not be accurately represented by a linear model which meant that the equation of motion had to be solved in the time domain. The problem of Time Domain simulations is that they are computationally expensive. Furthermore, it is possible that the response spectrum could contain energy at entirely different frequencies than the input spectrum. Since the relation between inputs and their respective responses is not known ‘a priori’, one could definitely not assume that the extreme responses fit the same distribution as the extreme wave heights. After completion of a simulation separate distributions have to be fitted to individual response signals to extrapolate the statistics to long-term time frames. To neatly fit the tail of these distributions, where the important extreme values are located, very long signals are required. Simulations have to be run for hours or even days to obtain statistical significance. The time domain simulations of the CSD used in the model are nearly real-time. In view of this discussion, the most important point is that the simulations of the CSD in operational condition were merely suitable for short-term time frames. They could not be successfully used for a full-blown workability assessment.

7-4 Closing remarks

All in all, the interaction of model features still yields an unstable model. The model is a body which is restrained in several springs, namely: the spud, swing wires and hydrostatic springs. There are two sources of energy and the system is underdamped which is why it is quite logical that it can become unstable. In Chapter 8 several recommendations will be made to improve the model.
Conclusions and Recommendations

This project started with the goal to add a cutter force function to the DODO simulation environment. The main question was:

“How can the cutter-to-soil interface be adequately modeled and verified to make a reliable statement about workability of a CSD in operational condition?”

1. The Conclusions section refers back to the research question that marked the beginning of this project.

2. The Recommendations section offers an outline of future steps that can be taken to get the most out of this project.

Each of the conclusions is written in view of the three stated research questions that were formulated to answer the main question.

8-1 Conclusions

The main research question can be decomposed into:

1. a part that only focuses on the developed Cutter Force Function,

2. a part that focuses on the Application of this function in the CSD model,

3. and a part that answers the question about Useability of the CSD model.
### 8-1-1 The Cutter Force Function

The cutter head can be modeled as a contour of a hemisphere. During a Time Domain integration step, a discretized soil volume that moves from outside the cutter to the inside, can be removed by a cutter matrix representing this hemisphere. This will thus impose a force that depends on specific energy.

It also depends on a predefined matrix of the cutter head that contains all localized unit vectors for the cutting forces. So, when the difference of the seabed matrix between the start and the end of one time step is known, the matrix of resulting force vectors for that time step can be readily calculated with only one matrix operation. All in all this method provides a simple deterministic way to quantize the matrices for the cut volumes, and the matrices for the resulting forces. It adds sufficient detail to the CSD model to investigate important characteristics of a cutting process.

The most important take-aways are about: the **Quantization of the Cut Volume**, the **Matrix Size**, and which effects are captured in the **Interface Forces**.

#### Matrix Size and Quantization of cut Volume

The original version of the Cutter Force Function, and also its first revision, used 3D matrices of equal grid spacing for the cutter and for the seabed. Excessive memory demand turned out to be a frustration of this project. The grid size could not be chosen sufficiently small to reduce the quantization error that was inherited from the discretization.

Based on the grid spacing and the seabed matrix size, there is a bandwidth in which the cutter ideally oscillates. If the oscillations are about the same size as one grid spacing, the resulting signal makes irregular jumps, which causes very much noise. For this reason it is important to have sufficiently small grid spacing. On the other hand, the cutter matrix must stay inside the seabed matrix at all times. The seabed matrix must therefore be larger than the bandwidth in which the cutter might come during a simulation. These two requirements on the size of the seabed matrix, together with the poor memory allocation, yielded a very high memory demand. When the second revision of the discretization was made, the matrices were reduced to 2D, essentially utilizing the 8 bytes allocation per matrix element to store a part of the information (seabed surface height) in it. In the end, the Cutter Force Function works just fine the way it does but it has thrown out one dimension to store information in.

To reduce the quantization error (originating from the grid spacing limits) smoothing techniques have been used. The sampling method is the first example of that. The smaller the grid sizes become, the more it will cost to compute this while the need for smoothing diminishes. The moving average of the signal is the second example. The history length of this moving average may not be too large because it causes a lag between the production signal and the cutter force signal.

It can be concluded that the quantization method to find $(V_{cut})_{i,j}$ in the revised function is adequate. The signals that originate from this are smooth enough to capture the cutter-to-soil interface and to develop extra modeling features from it.
8-1 Conclusions

Interface Forces

The cutting forces are adopted from 2D-cutting theory, based on a generalized tooth geometry. The time averaged forces of the brittle failing mechanism are used for this. The relatively high frequent force oscillations from the cutting teeth are filtered out in the equations of motion of the CSD, this was discussed in Section 2-3. The body masses of the CSD and the ladder act as low pass filters in the equations of motion.

Verification was done based on four benchmark CSDs. It was shown that the function generates reasonable production rates, swing winch power and cutter power for an stationary cutting process. Since the second revision of the Cutter Force Function the TD-simulations of the CSD model can be run faster than realtime.

The function does not account for bulldozing effects, snow plough effects, or other effects such as wear flat. Therefore the simulated result may differ very much from reality. Moreover, the mean cutting forces can be between 30% to 60% of the cutting forces. This uncertainty is another reason why the simulation may give different forces than the real situation.

8-1-2 Application in CSD Model

This research clearly showed that the cutter drive introduces energy into the system due to which oscillations (from the cutting process only) can be stimulated. When the CSD is in under cutting mode, the system looks stable for all investigated cases while the wave forces are zero. When the CSD is in over cutting mode, the modeled system becomes unstable for tougher rocks (although the wave forces are zero).

An explanation for this is the lack of damping of the Linear Time Invariant system that represents the mooring condition. The transfer functions in the LTI- model were given in Section 5. The external cutter force may become too strong for the system to remain stable. For the unstable cases, the solution of the linearized differential equation has real valued eigenvalues. This can cause the amplitudes to exponentially grow over time. Since the cutter force provides a coupling in several degrees of freedom, the oscillations in one degree of freedom may cause oscillations in other degrees of freedom and vice versa.

A source from Wichers [14] was cited at the beginning of this report. Several publications can be found in which this cutter-ground interaction is simply applied as a non-linear spring. With the results of the Time Domain simulations in Chapter 6-3 that there is much more behind the cutter-seabed interface than a simple non-linear spring.

Obviously, the model does not fairly represent reality because the cutter power that is fed to the model is just as high as the required cutter power. In reality the cutter drive and the power limit will cause the cutter to stall if the required power is too much to ask. Also, in reality the system has torque feedback control on the swing winches. One might think that the proven instability of the model (at zero waves) therefore is no real issue, because in reality the power input on the cutter head is limited by at least two things. However, one must not forget that the cutter forces are not the only source of power in the model. In other words: the combination of wave power and cutting power may cause larger system oscillations than wave power alone (even if the cutter power never exceeds its maximum).
8-1-3 Useability of the CSD Model

The transfer functions of the LTI-system of the CSD in mooring condition (Model0) show plausible results. The system is linearized by a function in DODO, and the linearized spud stiffness, damping, and added mass for the system are verified in Section 4-4-1. It provides a lot of insight to study the transfer functions for different wave angles, and swing angles. It shows how sensitive the CSD dynamics are to the mooring lay-out. Based on this Linear Time Invariant model, a workability assessment can be performed.

Because of stochastic uncertainties that have been mentioned in the Discussion (Chapter 7), Model 1 cannot be reasonably used for a full-blown workability assessment. It can, however, be used for qualitative studies in which two systems are compared.

Model 1 can, however, be used as a stepping stone to evaluate design concepts, such as the example in Appendix B. The results in Section 6-1 show that the cutter force of a CSD in operational condition may take its toll on the damping of the total system. This negative effect on the system damping depends on the seabed geometry that is encountered, and the specific energy from which the forces are calculated. The model can be used to investigate alternative ladder pitching concepts, and the interaction between wave-, soil-, and geometrical parameters. This may support the design process.

8-2 Recommendations

In view of the whole process a number of things can be advised to take the results of this project further.

8-2-1 The Cutter Force Function

The following improvements of the model may be considered:

- Revision 2 of the Cutter force function uses a 2D-matrix to store seabed heights. The seabed matrix projects the height of the seabed onto the cutter surface, where the cutting action takes place. It should be neat to rotate the seabed matrix such that the height of the seabed is projected at an angle equal to the ladder angle.

- Add different mechanisms to the framework, such as clay cutting and sand cutting.

- For all discretized elements inside the cutter contour the same tool geometry is assumed. Several mask matrices (equal size as the cutter matrix) could be used to add location dependent effects. The snowplough effect is an example of an effect that is location dependent and can be implemented by means of such a mask. Each mask can be simply added as a different property of the cutter object.

- The modeled seabed matrix is homogeneous. In reality the rock mass is most likely inhomogeneous. A mask matrix the size of the seabed matrix could be added to the model for a distribution of rock material properties. This could be added as an extra property of the seabed object.
• The modeled seabed matrix is isotropic. In reality the rock mass may have anisotropic strength. This could also be added with a mask, enhancing the cutting forces in specific directions.

• A bulldozing effect can be added. This depends on the clearance angle of cutting teeth, the rotational- and translational speed of the cutter head. This type of bulldozing is in fact a sum of normal forces on the wear flat of the cutting teeth. Also, at the pole of the hemisphere (on the cutter axis), a kind of normal force can be implemented in order to account for the cutter pounding (punching) the face of the seabed.

• Since the cutting teeth wear a lot, there must be friction. These friction forces can be expressed in terms of the local unit vectors of Section 3-1. This is related to the bulldozing force.

8-2-2 The CSD Model

The following recommendations can be considered for further application of Model1:

• The most important recommendation is to limit the amount of energy that can flow into the system via the uncontrolled swing speed, and the unlimited cutter power.

• Move away from the assumption that the CSD retains its heading, and that the seabed passes through model space. If the seabed grid would be in a cylindrical coordinate system, with the spud at its origin, the CSD would be able to swing through the model space. Note that the volume of a seabed element in that case depends on the radial distance from the spud. This will make the model a lot more intuitive.

• A thorough analysis on the linearized effects of over-, and under-cutting on the system stability at zero waves. A sort of negative damping coefficient can be formulated that captures the effect of the cutting process around the system equilibrium. This depends on the cutter-seabed interface geometry at equilibrium. The negative damping coefficient acts in multiple degrees of freedom.

The analytical formulas of rock cutting theory are used to find Since the mass of the CSD bodies is relatively very large compared to the brittle peak forces of the excavation process, the much higher frequencies
A Cutter Suction Dredger (CSD) has a cutter ladder which hinges down to the seabed to remove sediment with a rotating cutter head. This cutter head is driven by a hydraulic or electric motor. The ladder can be rotated with a ‘ladder pitch system’, this generally comprises a hoisting wire which is spooled on a hoisting winch and reeved through a block-tackle system. An alternative ladder pitch system is one with a hydraulic cylinder instead of hoisting wires.

![Diagram of CSD while dredging](image)

**Figure A-1:** Overview of CSD while dredging
The Cutter Suction Dredger

CSDs are typically used in water depths up to 30m. The installed cutter powers reach from 50 kW (the smallest CSDs) to 5 MW and even 8.5 MW (Jumbo CSDs). CSDs mostly deal with rocks with a strength up to 20 MPa, where rock cutting is economically the most preferable method. They are also used on tougher rocks with UCS values up to 60-80 MPa [23].

A spud mooring system is used to position the aft of the vessel. The vessel can make a sweeping motion (between $+35^\circ$ and $-35^\circ$) around the spud pole. This is done by means of two anchored swing wires which run through sheeves at the lower part of the cutter ladder and are connected to two swing winches. One swing wire points to starboard and one to portside, this can be seen in figure A-1. The hauling swing winch is pulling the cutter ladder in the desired swing direction. The opposing winch is letting the wire go while retaining some tension. The spud mooring system, ladder pitch system and the swing wires together provide the necessary force and positioning for the cutter head to excavate the soil.

Once the soil is cut loose the sediment-water mixture at the cutter head is transported through a suction pipe in the lower part of the cutter ladder. A dredge pump provides pressure to discharge the mixture. The mixture is transported via a pipeline or discharged into a barge which is moored at the side of the vessel.

One can distinguish roughly two working conditions: the Operational condition (i.e. actually excavating) and the Mooring condition [7]. Since both working conditions are relevant for the scope of this project, they will be discussed with more detail in Section A-3 and Section A-4.

**A-2 Environmental conditions**

Important working conditions of a cutter dredger are wind, current and waves. These effects depend on local topography and meteorology. Although the drag force from wind and water current can be quite significant they are neglected for this thesis work. This is because the solution set from DIFFRAC is only valid for water without current.

Most water waves that the CSD has to cope with are wind generated waves. These can be classified into two basic categories: sea waves and swell waves. Sea waves are generated by local wind areas and depend on local topography. They are generally short-crested, have irregular shapes and the direction in which they propagate can deviate several tens of degrees from the mean direction [2]. Swell waves have traveled from a place far away. These waves have relatively regular shapes, move almost uni-directional and they have long crests compared to sea waves. A deterministic definition could be fairly sufficient to model swell waves but for young wind generated sea states a stochastic approach would be more adequate.

**A-3 Mooring condition**

In support of the whole excavation process, the CSD must perform many side activities: the connection and disconnection procedure of a discharge pipeline (optional), the positioning of the swing anchors and the (re)positioning of the mooring spuds. These processes take time in which the cutter head is idle. Also, the cutter teeth on the rotating cutter head must be replaced frequently since they wear down. In the scope of this thesis the CSD is said to be
in ‘mooring condition’ when the cutter head is lifted from the breach, hence the excavating forces on the cutter head are non-existent.

In the mooring condition the vessel is mostly restrained by the spud mooring system. In addition to this the cutter ladder can be restrained by the two (anchored) swing wires. The ship response depends on its hull shape, the positions of the anchors, the stiffness of the spud mooring system and the direction of environmental forces.

A-4 Operational Condition

When the CSD is operational, the cutter head interacts with the seabed while excavating the sediment. The basics of the excavation process will be discussed below. Also the shape of the breach and the phenomena of overcutting and undercutting will be described.

A-4-1 Handling of swing anchors and spud mooring system

As the Cutter Suction Dredger swings from starboard to port side and back, the cutter head creates a breach which follows the shape of an arc in a horizontal plane. The swing radius of this arc is the longitudinal distance of the cutter head measured from the centerline of the spud. The radial distance from the breach of the previous swing towards the breach of the current swing, measured from spud position, is referred to as ‘forward step size’. The lateral speed of the cutter head is provided by the hauling swing wire, this will be referred to as ‘swing speed’.

There are two spud mooring systems. The main spud mooring system has an integrated spud carriage. The spud carriage can change the swing radius, thus increase the (forward) reach of the cutter head. This is done by means of a hydraulic cylinder which keeps the longitudinal position of the spud carriage within the vessel. The spud carriage cylinder can make a number of ‘steps’ before the cylinder is at the end of its stroke.

After the spud carriage reached the end of its stroke the main spud is put at a new position in the seabed. During the repositioning the auxiliary spud mooring system is used, the main spud is hoisted, the spud carriage cylinder is moved from the end of its stroke to the beginning and the main spud is dropped into the seabed again.

Each forward step that the CSD makes it essentially causes a small degradation of the anchor positions until at some moment the swing wires have a larger backward component than lateral component. At extreme swing wire angles a very high load on the spud mooring system is required to provide the necessary lateral force on the cutter ladder. The dredging operator is thus urged to reposition the anchors on a regular basis to maintain proper swing wire angles.

A-4-2 Seabed Geometry and Mass Properties

Both the seabed geometry and rock mass properties are very important for the cutting process. The seabed mainly owes its shape to passive soil failure, i.e. caused by excavating element, but active soil failure can also play a significant role. Regarding passive soil failure the seabed
geometry is defined by the historic trajectory of the cutter head. Regarding active soil failure the seabed geometry depends on factors such as internal friction, adhesion, water depth and pore pressure. Within the scope of this thesis the geometry is only changed by passive soil failure. In other words: it solely depends on the cutter trajectory.

**Soil**

Obviously soil properties are important for the stability of the breach. Active soil failure can cause the breach to collapse and debris to fall down, resulting in a sloped surface. However, it is assumed that adhesive capabilities of rocks can keep the face intact and that it will only change shape by passive soil failure (i.e. caused by the excavating element).

Nonetheless, the cutter head cannot create a perfect breach shape since its trajectory is distorted by wave induced motions of the vessel. Moreover, each swing the CSD makes it is affected by the trajectory of the previous swing.

### A-4-3 Overcutting and Undercutting

The cutter head is always designed to rotate in just one direction, though the swing direction can be either left or right. The cutting process is therefore asymmetric, which is why a distinction between overcutting and undercutting is made.

Consider a two dimensional cutter head as illustrated in Figure A-2. Suppose, as indicated in the figure, that the cutting force $F_s$ is mostly tangent to the cutter head circumference. When the dredger swings to the left the cutter head is said to be undercutting. The cutting edge is in this case at the left side and the resulting force on the cutter head generally has a downward component. Suppose the cutter head had a lower position such that the cutting edge would have been at the bottom. The resulting force vector $F_s$ would be close to horizontal, thus the vertical component would be close to zero. The cutting process is vertically stable. The horizontal force component acts in the opposite direction of the swing direction, which also makes the undercutting process horizontally stable.
When the dredger swings to the right, the cutter head is overcutting. In this case the cutting teeth attack the uncut soil from above. This can result in an uplifting force on the cutter ladder. The horizontal component of the cutting force is in the same direction as the swing movement, hence it can become unstable. The cutter head can get horizontal traction like a wheel. When this happens the opposing swing wire has to withstand this horizontal force. The swing winch, however, often has a lot less power than the cutter motor.

Due to the aforementioned effects the process of undercutting is often preferred. Sometimes the operator exclusively uses undercutting and hoists the cutter ladder on the way back, dismissing any excavation opportunity on the return swing.
Appendix B

Patent
A method for operating a cutting dredger in open water, in particular to control a ground reaction force during dredging operations, the dredger including: a floating body, an elongate cutter ladder having a cutting head at its lowerable free end for dredging, which cutter ladder is hingedly coupled to the floating body and hingeable between an upwards position and a cutting position wherein the cutting head engages matter to be cut away at the floor of a body of water.
Fig. 4
CUTTING DREDGER

BACKGROUND

[0001] The present invention relates to a method for operating a cutting dredger in open water, in particular to control of a ground reaction force during dredging operations, the dredger comprising:

[0002] a floating body,

[0003] an elongate cutter ladder having a cutting head at its lowerable free end for dredging, which cutter ladder is hingedly coupled to the floating body and hingebale between an upwards position and a cutting position wherein the cutting head engages matter to be cut away at the floor of a body of water.

[0004] Such a method is known from US2005268499 which relates to a method and apparatus for pumping with a dredge. Herein it is disclosed to apply a predetermined amount of load force by the head against the bottom. Herein, a winch and cable and the controller are operated to lift some of the head weight until the desired predetermined head force is applied to the bottom. It is a disadvantage that head weight can only be lifted.

[0005] In general, it is known to raise and lower a ladder connected to the hull of a dredger by a hydraulic cylinder like for instance in U.S. Pat. No. 3,470,633, U.S. Pat. No. 4,095,545A, U.S. Pat. No. 4,628,623A and U.S. Pat. No. 2,850,814. Typically, these dredgers have a relative short ladder and are not suitable to operate in open water and instead intend to operate in areas where room is restricted and/or areas with no or very small waves, like in a port or harbour. These hydraulic cylinders are not operated while dredging. It is however known to operate a ram cylinder to provide crowding action for tough digging. This ram cylinder is coupled with a relative short ladder not suitable for open water. Open water here means water with possibly significant wave height.

[0006] It is also known, like in U.S. Pat. No. 2,963,801A, to hydraulically drive a cutter ladder during swing motion, which is a lateral motion with respect to vertical motion.

[0007] US 2005/0268499 A relates to an environmental dredging method having a suction head for removing a contaminated layer from a bottom of a body of water. A predetermined amount of load force may be applied by the suction head against the bottom. Herein, a winch and cable and the controller are operated to lift some of the suction head weight until the desired predetermined head force is applied to the bottom. Such a suction head suspended from a cable does not suffice for controlling a ground reaction force of a cutting dredger.

[0008] U.S. Pat. No. 3,777,376 (A) discloses a dredging apparatus for use in heavy seas having a barge and an articulated ladder. U.S. Pat. No. 3,777,376 relates to positioning of ladder parts in that a lower ladder part is angularly movable relative to the upper ladder part, and means are provided for adjustably positioning said first ladder part at a selected angle relative to said barge. The ladder is moved between its working and non-working positions by a winch carrying a cable 87. Such a cable does not suffice for controlling a ground reaction force of a cutting dredger.

SUMMARY OF THE INVENTION

[0009] The invention aims to provide an improved control of the ground reaction force during dredging operations.

[0010] Another object of the invention is to improve known methods to control the ground reaction force during dredging operations in that a problem associated with that method is at least partly solved.

[0011] Yet another object of the invention is to provide an alternative method to control the ground reaction force during dredging operations and/or to keep a cutting head at a constant dredging depth independent of wave and soil conditions.

[0012] According to a first aspect of the invention this is realized with a method for operating a cutting dredger, the dredger comprising:

[0013] a floating body,

[0014] an elongate cutter ladder having a cutting head at its lowerable free end for dredging, which cutter ladder is hingebale coupled to the floating body and hingebale between an upwards position and a cutting position wherein the cutting head engages matter to be cut away at the floor of a body of water.

[0015] a driving device comprising an onboard hydraulic cylinder (8) coupled with the cutter ladder for hinging the cutter ladder between the upwards position and the cutting position, the method comprising the steps:

[0016] operating the driving device, when the cutter ladder is in the cutting position, in order to urge the cutter ladder down in a controlled manner for controlling a ground reaction force F between the cutting ladder and the bottom of the body of water. For controlling the dredging process, in an embodiment operating the driving device, when the cutter ladder is in the cutting position, for controlling a ground reaction force between the cutting ladder and the bottom of the body of water, or alternatively.

[0017] The operating of the driving device, when the cutter ladder is in the cutting position, enables to provide control of the ground reaction force. In addition, the driving device enables not only to lift the cutter ladder but also to urge the cutter ladder down in a controlled manner. In addition, cumbersome winch system can be omitted. This enhances safety on board because the amount of breakable cables at deck level is reduced. In addition, the omission of the winch system saves space at deck level and provides a better survey ability.

[0018] The ground reaction force is the force between the cutting head of the cutter ladder and the bottom of a body of water. The ground reaction force supports at least a part of the weight of the cutter ladder. The ground reaction force is not to be confused with cutting forces between the cutting head and the matter to be cut away.

[0019] In an embodiment, the driving device comprises a number of hydraulic cylinders like for example a pair of hydraulic cylinders.

[0020] In an embodiment, the driving device comprises hydraulic cylinders set in parallel.

[0021] In an embodiment, the method comprises operating the cutting dredger under swell conditions. Since the ground reaction force is much more controllable, the cutting dredger is much more suitable to operate under swell conditions, specifically relative heavy swell conditions up to a significant wave height of e.g. 3 m. The method preferably comprises operating the dredger in hard soil conditions.

[0022] In an embodiment of the method according to the invention, the ground reaction force is between about zero and about twice the weight of the cutter ladder. This is a beneficial result of the driving device enabling not only to lift the cutter ladder but also to urge the cutter ladder down in a controlled manner. A known winch system only provides a
ground reaction force between zero and one times the weight of the cutter ladder. Such cable systems which suspend a ladder are known from e.g. US 2005/0268499 A and U.S. Pat. No. 3773736 (A).

0023] In an embodiment, the length of the cutter ladder is about tens of meters, specifically between 10 and 70 meters. The length of the cutter ladder is such that the cutting dredger may operate at sea and thus under swell conditions.

0024] In an embodiment, the method according to the invention comprises operating the driving device, when the cutter ladder is in the cutting position, for maintaining the cutting head at a constant dredging depth.

0025] The invention further relates to a cutting dredger for use in the method according to the invention. The cutting dredger comprises:

- a floating body,
- an elongate cutter ladder having a cutting head at its lowerable free end for dredging, which cutting ladder is hingeably coupled to the floating body and hingeable between an upwards position and a cutting position wherein the cutting head engages matter to be cut away at the floor of a body of water, and
- a driving device comprising an onboard hydraulic cylinder (8) coupled with the cutter ladder for hinging the cutter ladder between the upwards position and the cutting position, and for urging the cutter ladder down in a controlled manner for controlling a ground reaction force F between the cutting ladder and the bottom of the body of water.

0029] In an embodiment of the cutting dredger according to the invention, the driving device comprises means for pressurizing the hydraulic cylinder in order to operate the hydraulic cylinder.

0030] In an embodiment, the means for pressurizing the hydraulic cylinder comprise an accumulator coupled with the hydraulic cylinder for providing the ground reaction force. The accumulator controls in a passive way the ground reaction force in that it operates the hydraulic cylinder at a substantially constant pressure which pressure determines the ground reaction force.

0031] In an embodiment, the driving device comprises a control valve operationally coupled with the means for pressurizing the hydraulic cylinder for controlling pressurizing of the hydraulic cylinder. This results in an active control of the ground reaction force or alternatively or additionally the dredging depth independent of soil and wave conditions. In an embodiment, the cutting dredger comprising a measuring device for measuring the ground reaction force.

0032] In an embodiment, the cutting dredger comprises a display device operationally coupled with the measuring device for displaying the ground reaction force.

0033] In an embodiment, the cutting dredger comprises a control unit operationally coupled with the measuring device and the hydraulic cylinder for controlling the ground reaction force.

0034] In an embodiment, the cutting dredger comprises a control unit operationally coupled with the measuring device and the hydraulic cylinder for controlling a dredging depth. The dredging depth is measured in a usual manner based on the angle α of the cutter ladder with respect to the floating body and the submersion of the floating body at the hinge point of the cutter ladder.

0035] In an embodiment, the cutting dredger comprises an input device operationally coupled with the control unit for setting a desired ground reaction force.

0036] In an embodiment, the measuring device comprises a pressure sensor operationally coupled with the hydraulic cylinder.

0037] In an embodiment of the cutting dredger, the driving device is directly coupled with the cutter ladder. This provides a more direct control of the ground reaction force.

0038] In an embodiment of the cutting dredger, the hydraulic cylinder piston rod is directly coupled with the cutter ladder and a hydraulic cylinder barrel is coupled with the floating body. This ensures that a maximum driving force is available when moving the cutter ladder from the cutting position to the upwards position. This is all the more of importance when, as preferred, the cutter ladder in its upwards position is at least partly above the water surface.

0039] In an embodiment of the cutting dredger, a line of action of the hydraulic cylinder extends at an angle of about 10° to about 50° with respect to the horizontal during moving of the cutter ladder between the upwards position and the cutting position. This ensures an optimal arm between the line of action and the hinge point of the cutter ladder during the entire movement of the cutter ladder between the upwards position and the cutting position and vice versa.

0040] In an embodiment, the cutting dredger comprises a spud device for providing support to the cutting dredger during dredging operations. Such a spud device is well known per se.

0041] In an embodiment of the cutting dredger, the length of the cutter ladder is about tens of meters, specifically between 10 and 70 meters.

0042] The hydraulic cylinder is an onboard cylinder. This is beneficial in terms of environmental effects because the hydraulics is not in direct contact with the outboard water.

0043] The invention further relates to a method comprising one or more of the characterising features described in the description and/or shown in the attached drawings.

0044] The invention further relates to a method comprising one or more of the characterising features described in the description and/or shown in the attached drawings.

0045] The various aspects discussed in this patent can be combined in order to provide additional advantageous advantages.

DESCRIPTION OF THE DRAWINGS

0046] The invention will be further elucidated referring to an preferred embodiment shown in the schematic drawings wherein shown in:

- FIG. 1 in side view a cutting dredger with the cutter ladder in cutting position;
- FIG. 2 the cutting dredger according to FIG. 1 with the cutter ladder in an intermediate position;
- FIG. 3 the cutting dredger according to FIG. 1 with the cutter ladder in the upward position; and
- FIG. 4 a functional scheme of the driving device.

DETAILED DESCRIPTION OF EMBODIMENTS

0051] The invention will now be described in more detail referring to FIGS. 1-4. The cutting dredger 1 for use in the method according to the invention, comprises a floating body 2, like a hull etc. The cutting dredger 1 comprises an elongate cutter ladder 3 for reaching the floor 6 of a body of water 7. The length of the cutter ladder is about tens of meters, specifically between 10 and 70 meters for reaching the floor 6 of a body of water 7 like the sea. The cutter ladder 3 is hingeably
coupled to the floating body 2. The cutter ladder 3 is hinge-
able coupled to the floating body 2 at a hinge point 13. The
cutter ladder 3 is hingebly between an upwards position
shown in FIG. 3 and a cutting position shown in FIG. 1. The
elongate cutter ladder 3 has a cutting head 4 at its lowerable
free end for dredging. When the cutter ladder 3 is in the
cutting position, the cutting head 4 engages matter to be cut
away at the floor 6 of a body of water 7. When the cutter ladder
3 is in the cutting position, the cutter ladder 3 is supported
both by the floor 6 and the pivot point 13. The cutter ladder 3
support at the floor 6 is referred to a ground reaction force F.
This ground reaction force plays an important role to be able
to penetrate hard soils and as a result of this to be able to cut
away this soil at the floor 6.

[0052] The cutting dredger 1 comprises a driving device for
hinging the cutter ladder 3 between the upwards position and
the cutting position. Here, the driving device comprises an
hydraulic cylinder 8. Here, the hydraulic cylinder 8 is an
onboard cylinder. It is conceivable to use a number of hydrau-
lic cylinders, like for example a pair of parallel hydraulic
hydraulic cylinders on both sides of the cutter ladder 3. The hydraulic
cylinder 8 is coupled with the cutter ladder 3 for hinging the
cutter ladder 3 between the upwards position and the cutting
position. Here, the hydraulic cylinder 8 is directly coupled
with the cutter ladder 3 which improves controllability of the
ground reaction force F. The hydraulic cylinder piston rod 11
is directly coupled with the cutter ladder 3. The hydraulic
cylinder piston rod 11 is hingebly coupled with the cutter
ladder 3 by a piston rod hinge point 12. The hydraulic cylinder
barrel 9 is coupled with the floating body 2 to provide a
maximum driving force when the cutter ladder moves
towards its upward position. The hydraulic cylinder barrel 9 is
hingebly coupled with the floating body 2 by a barrel hinge
point 14. Here, the line of action 10 of the hydraulic cylinder
8 extends at about 45° with respect to the horizontal during
moving of the cutter ladder 3 between the upwards position
and the cutting position. This ensures even more an optimal
arm between the line of action and the hinge point of the cutter
ladder during the entire movement of the cutter ladder
between the upwards position and the cutting position
vice versa.

[0053] As shown in FIG. 4, the driving device comprises a
hydraulic cylinder 8 as well as means 15, 16 for pressurizing
the hydraulic cylinder. The means for pressurizing the
hydraulic cylinder 8. Here, the hydraulic cylinder 8 comprises
an accumulator 15 coupled with the hydraulic cylinder 8 for
providing a controlled ground reaction force F. In addition, here a hydraulic pump 16 is
provided for driving the cutter ladder 3. The driving device
comprises a control valve 18 operationally coupled with the
means 15, 16 for pressuring the hydraulic cylinder and the
hydraulic cylinder 8 for controlling pressurizing of the hydraulic
cylinder 8.

[0054] The driving device comprises a measuring device 17
for measuring the ground reaction force. Here, the measuring
device is a pressure sensor 17 operationally coupled with the
hydraulic cylinder 8 for measuring the ground reaction force
F.

[0055] The driving device comprises a control unit 19
operationally coupled with the measuring device 17 and the
hydraulic cylinder 8 for controlling the ground reaction force.
Here, the control unit 19 is operationally coupled with the
hydraulic cylinder 8 via the control valve 18. The ground
reaction force is continuously controlled. The control unit 19
may be operationally coupled with the measuring device 17
and the hydraulic cylinder 8 such the dredging depth or
thickness of a cutaway layer may be controlled as well wherein
the dredging depth is measured in a known manner.

[0056] The driving device here comprises a display device
21 operationally coupled with the measuring device 17 for
displaying the actual ground reaction force to an operator.
The display device 21 is operationally coupled with the measu-
ring device 17 via the control unit 19.

[0057] The driving device here comprises an input device
20 operationally coupled with the control unit 19 for setting a
desired ground reaction force by an operator.

[0058] In use of the schematically shown cutting dredger 1
the cutter ladder 3 is lowered towards its cutting position.
Then the hydraulic cylinder 8 is operated such that the ground
reaction force F between the cutting ladder 3 and the bottom
6 of the body of water 7 is controlled in a continuous manner.
Here, the ground reaction force F is controlled between about
zero and about twice the weight of the cutter ladder 3. The
dredging may be performed under relative heavy swell
conditions.

[0059] It will also be obvious after the above description
drawings are included to illustrate some embodiments of
the invention, and not to limit the scope of protection. Starting
from this disclosure, many more embodiments will be evident
to a skilled person which are within the scope of protection
and the essence of this invention and which are obvious
combinations of prior art techniques and the disclosure of this
patent.

1-22. (canceled)
23. Method for operating a cutting dredger (1) in open
water, the dredger comprising;
a floating body (2),
an elongate cutter ladder (3) having a cutting head (4) at its
lowerable free end for dredging, which cutter ladder is
hingebly coupled to the floating body and hingebly
between an upwards position and a cutting position
wherein the cutting head engages matter (5) to be cut
away at the floor (6) of a body of water (7),
a driving device (8) comprising an onboard hydraulic cy-
linder (8) coupled with the cutter ladder for hinging the
cutter ladder between the upwards position and the cut-
ting position, the method comprising the step:
operating the driving device, when the cutter ladder is in
the cutting position, in order to urge the cutter ladder
down in a controlled manner for controlling a ground
reaction force F' between the cutting ladder and the bot-
tom of the body of water.
24. Method according to claim 23, wherein the driving
device comprises a number of hydraulic cylinders, like for
example a pair of hydraulic cylinders.
25. Method according to claim 24, wherein the driving
device comprises hydraulic cylinders set in parallel.
26. Method according to claim 23, comprising operating
the cutting dredger under swell conditions.
27. Method according to claim 23, wherein the ground
reaction force is between about zero and about twice the
weight of the cutter ladder.
28. Method according to claim 23, wherein the length of
the cutter ladder is about tens of meters, specifically between
10 and 70 meters.
29. Method according to claim 23, comprising operating the driving device, when the cutter ladder is in the cutting position, for maintaining the cutting head at a constant dredging depth.

30. Cutting dredger (1) for use in the method according to claim 23, the cutting dredger comprising:

a floating body (2),

an elongate cutter ladder (3) having a cutting head (4) at its lowerable free end for dredging, which cutter ladder is hingeably coupled to the floating body and hingeable between an upwards position and a cutting position wherein the cutting head engages matter to be cut away at the floor of a body of water, and

a driving device (8) comprising an onboard hydraulic cylinder (8) coupled with the cutter ladder for hinging the cutter ladder between the upwards position and the cutting position, and for urging the cutter ladder down in a controlled manner for controlling a ground reaction force F between the cutting ladder and the bottom of the body of water.

31. Cutting dredger according to claim 30, wherein the driving device means for pressurizing the hydraulic cylinder.

32. Cutting dredger according to claim 31, wherein the means for pressurizing the hydraulic cylinder comprise an accumulator coupled with the hydraulic cylinder for providing the ground reaction force.

33. Cutting dredger according to claim 31, wherein the driving device comprises a control valve operationally coupled with the means for pressurizing the hydraulic cylinder and the hydraulic cylinder for controlled pressurizing of the hydraulic cylinder.

34. Cutting dredger according to claim 30, comprising a measuring device for measuring the ground reaction force.

35. Cutting dredger according to claim 34, comprising a display device operationally coupled with the measuring device for displaying the ground reaction force.

36. Cutting dredger according to claim 34, comprising a control unit operationally coupled with the measuring device and the hydraulic cylinder for controlling the ground reaction force.

37. Cutting dredger according to claim 36, wherein the control unit is operationally coupled with the measuring device and the hydraulic cylinder for controlling the dredging depth.

38. Cutting dredger according to claim 36, comprising an input device operationally coupled with the control unit for setting a desired ground reaction force.

39. Cutting dredger according to claim 34, wherein the measuring device comprises a pressure sensor operationally coupled with the hydraulic cylinder.

40. Cutting dredger according to claim 30, wherein the driving device is directly coupled with the cutter ladder.

41. Cutting dredger according to claim 40, wherein a hydraulic cylinder piston rod (11) is directly coupled with the cutter ladder and a hydraulic cylinder barrel (9) is coupled with the floating body.

42. Cutting dredger according to claim 30, wherein a line of action (10) of the hydraulic cylinder extends at between about 10° to about 50° with respect to the horizontal during moving of the cutter ladder between the upwards position and the cutting position.

43. Cutting dredger according to claim 30, comprising a spud device for providing support to the cutting dredger during dredging operations.

44. Cutting dredger according to claim 30, wherein the length of the cutter ladder is about tens of meters, specifically between 10 and 70 meters.

* * * * *
Figure C-1: Simulation set for Shear
Figure C-2: Simulation set for Tensile
Appendix D

Transfer Functions

- **Surge | Wave angle = 140 degrees**
- **Sway | Wave angle = 140 degrees**
- **Heave | Wave angle = 140 degrees**
- **Roll | Wave angle = 140 degrees**
- **Pitch | Wave angle = 140 degrees**
- **Yaw | Wave angle = 140 degrees**

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Surge | Wave angle = 145 degrees

- Response Amplitude Model
- Response Amplitude Floater only

Spectral Density

$H_s = 1$ [m]; $T_p = 7.5$ [s]
Spectral Density

$H_s = 1$ [m]; $T_p = 3.5$ [s]

Sway | Wave angle = 145 degrees

Roll | Wave angle = 145 degrees

Pitch | Wave angle = 145 degrees

Heave | Wave angle = 145 degrees

Yaw | Wave angle = 145 degrees
Surge | Wave angle = 150 degrees

Roll | Wave angle = 150 degrees

Sway | Wave angle = 150 degrees

Pitch | Wave angle = 150 degrees

Heave | Wave angle = 150 degrees

Yaw | Wave angle = 150 degrees

\[ H_s = 1 \text{ [m]}; T_p = 7.5 \text{ [s]} \]

\[ H_s = 1 \text{ [m]}; T_p = 3.5 \text{ [s]} \]
Surge | Wave angle = 155 degrees

- Response Amplitude Model
- Response Amplitude Floater only
- Spectral Density

- $H_s = 1\ [m]$, $T_p = 7.5\ [s]$
- Spectral Density
- $H_s = 1\ [m]$, $T_p = 3.5\ [s]$

Sway | Wave angle = 155 degrees

- Response Amplitude Model
- Response Amplitude Floater only
- Spectral Density

Roll | Wave angle = 155 degrees

- Response Amplitude Model
- Response Amplitude Floater only
- Spectral Density

Pitch | Wave angle = 155 degrees

- Response Amplitude Model
- Response Amplitude Floater only
- Spectral Density

Heave | Wave angle = 155 degrees

- Response Amplitude Model
- Response Amplitude Floater only
- Spectral Density

Yaw | Wave angle = 155 degrees

- Response Amplitude Model
- Response Amplitude Floater only
- Spectral Density
Surge | Wave angle = 160 degrees

- Response Amplitude Model
- Response Amplitude Floater only
- Spectral Density

- $H_s = 1 \text{ (m)}$, $T_p = 7.5 \text{ (s)}$
- Spectral Density

- $H_s = 1 \text{ (m)}$, $T_p = 3.5 \text{ (s)}$

Roll | Wave angle = 160 degrees

- Response Amplitude Model
- Response Amplitude Floater only
- Spectral Density

- $H_s = 1 \text{ (m)}$, $T_p = 7.5 \text{ (s)}$
- Spectral Density

- $H_s = 1 \text{ (m)}$, $T_p = 3.5 \text{ (s)}$

Sway | Wave angle = 160 degrees

- Response Amplitude Model
- Response Amplitude Floater only
- Spectral Density

- $H_s = 1 \text{ (m)}$, $T_p = 7.5 \text{ (s)}$
- Spectral Density

- $H_s = 1 \text{ (m)}$, $T_p = 3.5 \text{ (s)}$

Pitch | Wave angle = 160 degrees

- Response Amplitude Model
- Response Amplitude Floater only
- Spectral Density

- $H_s = 1 \text{ (m)}$, $T_p = 7.5 \text{ (s)}$
- Spectral Density

- $H_s = 1 \text{ (m)}$, $T_p = 3.5 \text{ (s)}$

Heave | Wave angle = 160 degrees

- Response Amplitude Model
- Response Amplitude Floater only
- Spectral Density

- $H_s = 1 \text{ (m)}$, $T_p = 7.5 \text{ (s)}$
- Spectral Density

- $H_s = 1 \text{ (m)}$, $T_p = 3.5 \text{ (s)}$

Yaw | Wave angle = 160 degrees

- Response Amplitude Model
- Response Amplitude Floater only
- Spectral Density

- $H_s = 1 \text{ (m)}$, $T_p = 7.5 \text{ (s)}$
- Spectral Density

- $H_s = 1 \text{ (m)}$, $T_p = 3.5 \text{ (s)}$
Transfer Functions

\[ H_s = 1 \text{[m]}; T_p = 7.5 \text{[s]} \]

\[ H_s = 1 \text{[m]}; T_p = 3.5 \text{[s]} \]

Surge | Wave angle = 165 degrees

Sway | Wave angle = 165 degrees

Heave | Wave angle = 165 degrees

Roll | Wave angle = 165 degrees

Pitch | Wave angle = 165 degrees

Yaw | Wave angle = 165 degrees
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Transfer Functions

Surge | Wave angle = 175 degrees

- Response Amplitude Model
- Response Amplitude Floater only

Spectral Density

\( H_s = 1 \text{ [m]} \); \( T_p = 7.5 \text{ [s]} \)

Spectral Density

\( H_s = 1 \text{ [m]} \); \( T_p = 3.5 \text{ [s]} \)

Roll | Wave angle = 175 degrees

Spectral Density

Sway | Wave angle = 175 degrees

Pitch | Wave angle = 175 degrees

Heave | Wave angle = 175 degrees

Yaw | Wave angle = 175 degrees

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Appendix E

Hydrodynamic force

E-1 Hyd-file

The following pages show the format of the DIFFRAC solution set, that is used as input for DODO. Only a small selection is shown since the full document has over 13 thousand lines of data (WD = [0:5:255] degrees ; ω = [0.05:0.05:3.00] rad/s).

1 IDENT CASE STUDY
2 IDENT lpp(m):104.30 B(m) : 21.70 T(m) : 5.00
3 IDENT kxx(m): 6.700 kyy(m) :28.700 kzz(m) : 28.800
4 IDENT mass(ton) : 8183.00
5 IDENT displacement(m3) : 7939.00
6 IDENT waterdepth(m) : 20.00
7 IDENT Additional B44 in hydfile: .00
8 IDENT Xg(st10)(m) : -3.500
9 IDENT Yg(cl)(m) : .000
10 IDENT Zg(keel)(m) : 6.820
11 IDENT Xf (wrt G)(m) : .256
12 IDENT Aw1 (m2) : 2068.571
13 IDENT GMT (m) : 4.052
14 IDENT GM1 (m) : 176.500
15 IDENT
16 REFS 20.00 5.000 -1.820
17 SPRING 8959. 2069. 0.2560 .000 10.87 183.3
18 PARA 60 73 0
19 OMEGA 0.05
20 ADMAS A(1,1) A(1,2) A(1,3) A(1,4) A(1,5) A(1,5)
21 ADMAS A(2,1) A(2,2) A(2,3) A(2,4) A(2,5) A(2,5)
22 ADMAS A(3,1) A(3,2) A(3,3) A(3,4) A(3,5) A(3,5)
23 ADMAS A(4,1) A(4,2) A(4,3) A(4,4) A(4,5) A(4,5)
24 ADMAS A(5,1) A(5,2) A(5,3) A(5,4) A(5,5) A(5,5)
25 ADMAS A(6,1) A(6,2) A(6,3) A(6,4) A(6,5) A(6,5)
26 BDAMP B(1,1) B(1,2) B(1,3) B(1,4) B(1,5) B(1,5)
Hydrodynamic force

27 BDAMP B(2,1) B(2,2) B(2,3) B(2,4) B(2,5) B(2,5)
28 BDAMP B(3,1) B(3,2) B(3,3) B(3,4) B(3,5) B(3,5)
29 BDAMP B(4,1) B(4,2) B(4,3) B(4,4) B(4,5) B(4,5)
30 BDAMP B(5,1) B(5,2) B(5,3) B(5,4) B(5,5) B(5,5)
31 BDAMP B(6,1) B(6,2) B(6,3) B(6,4) B(6,5) B(6,5)
32 WDIR 0.000
33 FAMP Fx Fy Fz Mxx Myy Mzz
34 FEPS Ex Ey Ez Exx Eyy Ezz
35 WDIR 5.000
36 . . .
37 . . .
38 WDIR 355.00
39 FAMP Fx Fy Fz Mxx Myy Mzz
40 FEPS Ex Ey Ez Exx Eyy Ezz
41 OMEGA 0.100
42 . . .
43 . . .
44 . . .
45 . . .
46 . . .
47 OMEGA 1.000
48 ADMAS 702.4 -0.5001E-02 154.6 -1.466 0.2397E+05 0.1970
49 ADMAS -0.4936E-01 4774. -0.2735E-01 0.1000E+05 18.42 0.3888E+05
50 ADMAS 189.9 -0.4977E-01 0.1214E+05 1.349 0.4740E+05 23.27
51 ADMAS 0.5406 9958. 0.2378 0.2840E+06 7.115 0.1722E+06
52 ADMAS 0.2387E+05 -0.1394 0.4747E+05 -0.2053 0.6091E+07 265.4
53 ADMAS 0.6170 0.3887E+05 0.1573 0.1787E+06 61.95 0.4911E+07
54 BDAMP 573.9 -0.7752E-02 -0.5324 -1.276 0.5528E+05 -0.2789E-01
55 BDAMP -0.1317 0.5217 -0.1507 -0.2750 -0.1616 0.2033E+05
56 BDAMP -481.5 0.2109E-01 9393. 4.109 -0.4523E+05 26.71
57 BDAMP 0.2723E-01 -0.2687 0.1672E-01 5995. -0.1436 0.2412E+05
58 BDAMP 0.5422E+05 -6.640 -0.4426E+05 -100.4 0.6549E+07 59.72
59 BDAMP -4.837 0.2027E+05 3.147 0.2422E+05 -0.8085 0.3284E+07
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62 FEPS 288.1 0.6018 251.2 0.3404 0.2738 177.1
63 . . .
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68 WDIR 355.0
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70 FEPS Ex Ey Ez Exx Eyy Ezz
71 . . .
72 . . .
73 . . .
74 . . .
75 . . .
76 OMEGA 3.000
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78 ADMAS 0.6490E-01 1345. 0.5705 -0.1946 0.2901 0.1420E+05
79 ADMAS 197.3 -0.9458E-01 0.1775E+05 0.1108 0.4627E+05 32.82

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Appendix F

MultiBody Theory

F-1 Constraint equations and Lagrange multipliers

The Lagrange Multiplier method is illustrated with the simple 1-DoF model from figure F-1 below.

Two carriages of 0.25kg and 0.50 kg are connected with a towing hook (the constraint). On carriage 2, a force of 2N is applied. By Newton’s 2nd law of motion it can be easily calculated that the acceleration of both carriages will be $2N/0.75kg = 2.67m/s^2$. The force in the towing hook remains unknown.

Now the same problem will be solved by making use of the equations of motion combined with the Langrange Multiplier method.

The constraint equation in this example is:

$$\phi(q) = x_1 - x_2 = 0 \quad (F-1)$$

The Jacobian of the constraint equation, $\phi_q$, becomes:
\[
\phi_q = \begin{bmatrix}
\frac{\partial \phi(q)}{\partial x_1} & \frac{\partial \phi(q)}{\partial x_2}
\end{bmatrix} = \begin{bmatrix} 1 & -1 \end{bmatrix} \tag{F-2}
\]

Since the constraint is constant, from the perspective of both bodies, the convective acceleration terms of the constraints are zero:

\[
\gamma = -\dot{q}^T \phi_{qq} \ddot{q} - \phi_{tt}
\tag{F-3}
\]

The equation of motion from 2-11 will in this example become:

\[
\begin{bmatrix}
0.25 & 0 & 1 \\
0 & 0.5 & -1 \\
1 & -1 & 0
\end{bmatrix}
\begin{bmatrix}
\dot{x}_1 \\
\dot{x}_2 \\
\lambda
\end{bmatrix} =
\begin{bmatrix}
0 \\
2 \\
0
\end{bmatrix} \tag{F-4}
\]

Solving this set of equations we obtain the accelerations of the bodies \(2.67\, m/s^2\) and the Lagrange Multiplier:

\[
\begin{bmatrix}
\dot{x}_1 \\
\dot{x}_2 \\
\lambda
\end{bmatrix} =
\begin{bmatrix}
2.67 \\
2.67 \\
-0.67
\end{bmatrix} \tag{F-5}
\]

Now, the force in the constraints can be found with the equation:

\[
-\dot{\phi}_q^T \lambda =
\begin{bmatrix}
0.67 \\
-0.67
\end{bmatrix} \tag{F-6}
\]
Appendix G

Some Matrices
Table G-1: Mass-, Added mass, stiffness- and damping matrices of floater-spud combination for $\omega = 0.6$

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Table G-2: Eigenvectors and Eigenvalues of floater-spud combination for $\omega = 0.6$

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Master of Science Thesis

V.B. Bakhuizen
Appendix H

Rock properties

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Figure H-1: Some ductility numbers of rocks from Hoek and Brown[5]
Bibliography


Glossary

List of Acronyms

CSD  Cutter Suction Dredger
UCS  Unconfined Compressive Strength
LWT  Linear Wave Theory
TD   Time Domain
FD   Frequency Domain
EoM  Equations of Motion
JONSWAP Joint North Sea Wave Project
CoG  Center of Gravity
DoF  Degree(s) of Freedom
BTS  Brazilian Tensile Strength
DODO Dynamic Operations in Dredging and Offshore
DP   Dynamic Positioning
RK4  fourth order Runge-Kutta
HCS  Hydrostatic Compressive Strength
ODE  Ordinary Differential Equation
MARIN Maritime Research Institute Netherlands
pdf  probability density function
TD   Time Domain
FD   Frequency Domain
List of Symbols

\( H_s \) Significant wave height
\( T_p \) Peak period
\( S \) Spectrum
\( \zeta \) Wave surface elevation
\( \omega \) radial frequency
\( \gamma \) peakedness factor JONSWAP
\( \omega_p \) Peak frequency
\( M_{kj} \) Mass matrix
\( m_{kj} \) Added mass matrix
\( x \) position vector
\( t \) time
\( K_{kj} \) Spring stiffness matrix
\( \tau \) time
\( C_{kj} \) Hydrostatic spring coefficient
\( F_k \) External wave force
\( b_{kj} \) damping
\( a_{kj} \) added mass
\( q \) six dimensional state vector of body i
\( \Phi_q \) Jacobian matrix of constraint equations
\( \lambda \) vector of lagrange multipliers
\( \gamma \) vector of convective acceleration terms
\( \mathbf{F}_w \)  
external force vector on floater

\( \mathbf{F}_{mb} \)  
external force vector on multi-body system

\( \mathbf{F}_{FK} \)  
Froude-Krylov force vector

\( B_{t,eq} \)  
total damping matrix of system, evaluated at equilibrium

\( K_{t,eq} \)  
total spring matrix of system, evaluated at equilibrium

\( Q(\omega) \)  
state vector in frequency domain

\( \Lambda \)  
vector of lagrange multipliers in frequency domain

\( \sigma_1 \)  
first principal stress

\( \phi \)  
internal friction coefficient

\( \tau \)  
shear stress

\( c \)  
cohesion

\( m \)  
ductility number

\( v_h \)  
haulage velocity, or swing speed

\( n_r \)  
number of staggered teeth

\( Q \)  
production

\( H_{face} \)  
face height

\( L_{step} \)  
step size

\( v_c \)  
cutting speed

\( \alpha \)  
blade angle

\( \lambda_H \)  
brittle horizontal force coefficient

\( \delta \)  
external friction coefficient

\( h_i \)  
thickness of cut layer

\( w \)  
width of generalize cutting tooth
$F_h$  horizontal cutting force

$F_v$  vertical cutting force

$c$  cohesion factor

$\beta$  angle of shear plane

$E_{sp}$  specific energy

$\sigma_t$  tensile stress

$\tau$  shear stress

$\theta$  pitch angle

$\psi$  yaw angle

$\phi$  roll angle

$u_r$  radial unit vector

$u_t$  tangential unit vector

$u_l$  longitudinal unit vector

$C$  cutter matrix $C$

$S$  seabed matrix $S$

$c_{i,j}$  element $i,j$ in cutter matrix

$s_{i,j}$  in seabed matrix element $i,j$

$r_{i,j}$  position vector in cutter coordinate system of element $i,j$

$V_{cut}$  cut volume

$B_n$  box sample matrix

$b_0$  element in box sample matrix

$V_n$  quantized cut volume for sample $n$

$D_c$  Cutter diameter
$z_{tip}$  \text{z-coordinate of tip of cutter ladder in general coordinate system}

$P$  \text{power}

$T$  \text{torque}

$T_c$  \text{torque on cutter drive}

$d_{i,j}$  \text{shortest distance of element $c_{i,j}$ to cutter axis}

$T_{i,j}$  \text{torque that element $i,j$ imposes on cutter axis}

$P_c$  \text{power on cutter}

$L$  \text{length}

$m$  \text{mass}

$B$  \text{breadth of hull}

$T$  \text{draft of hull}

$I$  \text{mass moment of inertia}

$S_\epsilon$  \text{phase shift}

$A(\omega)$  \text{frequency dependent added mass matrix}

$f_i$  \text{inertia force per unit length}

$f_d$  \text{drag force per unit length}

$\rho$  \text{density}

$C_d$  \text{drag force coefficient}

$C_a$  \text{added mass force coefficient}

$V$  \text{volume}

$A$  \text{area}

$E$  \text{Youngs modulus}
$D_o$ outside diameter

$D_i$ inside diameter

$k$ spring stiffness coefficient

$\overline{KG}$ height of center of gravity above keel

$f_{st}$ short term probability density function

$R_a$ most probable response amplitude

$m_0$ 0-th order spectral moment

$N$ number of waves