Development of relative horizontal motion reduction systems

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Challenge the future

Development of relative horizontal motion reduction systems

By

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This thesis is confidential and cannot be made public until September 28, 2021.

Preface

This report is written in partial fulfilment of the requirements for the degree of 'Master of Science' in 'Offshore and Dredging Engineering' at the Delft University of Technology. The research study is performed in cooperation between Dockwise/Boskalis and the Delft University of Technology and it focuses on the development of a relative horizontal motion reduction system, which can be used during offshore loading and discharge operations.

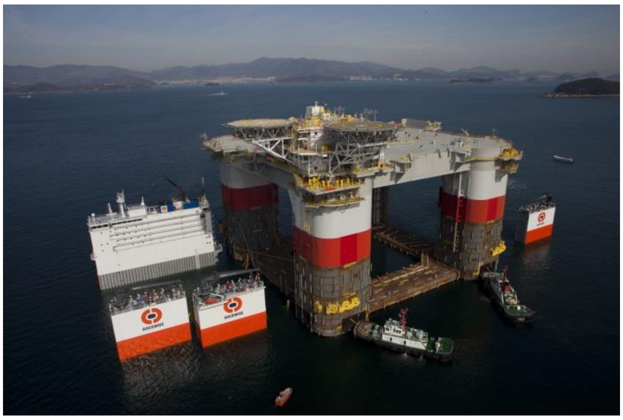
I would like to thank ir. Onno Peters for his direct guidance and help. During the past year we had many discussions about this research study, and I always ended up with new ideas and insights. I would also like to express my gratitude to ir. A. Jarquin Laguna and prof. dr. A. Metrikine for their supervision and support as part of the graduation committee.

Furthermore I would like to thank all my friends for their support and for offering the sufficient study avoiding activities. At last, I would like to thank my parents for their support and providing me the opportunity to go to the university.

Jelle van Beelen
Voorschoten, 18 september 2016

Abstract

In the recent years, the demand for energy resources kept rising worldwide and it is expected that it keeps rising. This caused a shift from the traditional exploration areas to more remote areas. The large offshore structures (FPSOs, Semi-subs) used at these locations need to be transported from the fabrication yard to the exploration area. The transportation of these structures (cargo) is done by self-propelled semi-submersible heavy transport vessels (HTVs).



At this moment the locations of the loading and discharge operations are limited to areas with benign environmental conditions i.e. sheltered locations. This provides a severe limitation for projects which are located far away from a shore. Therefore Dockwise is investigating the possibility to perform the loading and discharge operations offshore in harsh environments, close to the intended production site of the cargo. However challenges need to be solved before this is possible. One of the main challenges involves the amplitude of the relative motions which occur when the HTV and the cargo are floating on top of each other, with only a small gap of water in between them. The amplitude of these motions should be reduced in order to ensure a safe and reliable operation.

The aim of this thesis is to develop a system which is capable of reducing the relative horizontal motions of the cargo with respect to the HTV. A multibody hydrodynamic model is used to gain insight in the behaviour of the HTV-cargo system when the standard cargo handling system is used. The standard cargo handling system is the system which is presently used to control the motions of the cargo. The performed simulations showed that this system is not capable of controlling the amplitude of the motions sufficiently, during an offshore loading and discharge operation. Therefore a relative horizontal motion reduction system needs to be developed. For this purpose a range of concepts is generated. Two concepts are selected of which a preliminary design is obtained; a *Clamping system* and a *Line tension actuator*. The principle of the *Clamping system* is to increase the stiffness of the connection between the HTV and the cargo over time. Due to the increase in stiffness, the amplitude of the relative horizontal motions is reduced. The principle of the *Line tension actuator* is to control the tensions in the lines connecting the HTV and the cargo.

Different MATLAB-Simulink simulations are conducted to investigate the effect of the design parameters on the performance of the systems. The simulations show that both systems are able to sufficiently reduce the relative horizontal motions. The presented preliminary design forms a good basis for detailed design, during which also operational aspects need to be covered.

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Nomenclature

List of symbols

a	[kg]	Added mass
$a_{operation}$	[m]	Wave amplitude at which the operation can be performed
A_f	[m^2]	Accumulator face plate area
A_p	[m^2]	Piston head area
A_r	[m^2]	Piston rod area
A_v	[m^2]	Valve area
A_{v_cv}	[m^2]	Valve area of control valve
A_{v_owv}	[m^2]	Valve area of one way valve
c	[Ns/m]	Damping coefficient
c_{crit}	[Ns/m]	Critical damping coefficient
C_p	[J/kg K]	Specific heat at constant pressure of the liquid/gas
C_v	[J/kg K]	Specific heat at constant volume of the liquid/gas
D	[m]	Diameter cylinder
D_t	[m]	Diameter tube
D_r	[m]	Diameter piston rod
D_{cv}	[m]	Diameter control valve
D_{prv}	[m]	Diameter pressure relief valve
D_{owv}	[m]	Diameter one way valve
E	[N/m^2]	Modulus of elasticity
f	[-]	Friction coefficient
F	[N]	Force
F_c	[N]	Critical force
F_e	[N]	Force error
F_{ext}	[N]	External force
F_d	[N]	Damping force
F_{line}	[N]	Line force
F_{min}	[N]	Minimum force
F_p	[N]	Piston force
F_r	[N]	Required force
F_x	[N]	Force in surge direction
F_y	[N]	Force in sway direction
F_0	[N]	Force amplitude
h_f	[m]	Fender height
h_r	[m]	Roughness height
H_s	[m]	Significant wave height
$H_{s,design}$	[m]	Significant design wave height
I	[m^4]	Area moment of inertia
k	[N/m]	Stiffness
K	[-]	Effective length of the column factor
k_{line}	[N/m]	Line stiffness
K_a	[N/m]	Accumulator spring stiffness
k_d	[-]	Discharge coefficient valve
k_{d_cv}	[-]	Discharge coefficient of control valve
k_{d_prv}	[-]	Discharge coefficient pressure relief valve
k_{d_owv}	[-]	Discharge coefficient one way valve
K_p	[-]	Proportional gain value
K_s	[-]	A constant representing isentropic conditions
K_t	[-]	A constant representing isothermal conditions
L	[m]	Length
m	[kg]	Mass
$\dot{m}_{in,out}$	[kg/s]	Mass flow rate

N	[-]	Number of waves
P	[Pa]	Pressure
P_{cr}	[Pa]	Critical pressure
P_{prv_set}	[Pa]	Pre-set pressure level at which the pressure relief valve opens
P_{prv_reg}	[Pa]	Pressure increase over the pre-set level to fully open the valve
Q	[m^3/s]	Flow rate
$q_{in,out}$	[W/m^2]	Heat transfer terms
R	[J/kg K]	Specific gas constant
Re	[-]	Reynolds number
r_v	[-]	Volume of entrained air
S	[m]	Piston stroke
T	[K]	Temperature
t	[s]	Time
T_p	[s]	Wave period
$\dot{U}$	[J]	Change in internal energy
V	[m^3]	Volume (within cylinder chamber)
v_v	[m/s]	Fluid velocity through valve
$\dot{W}$	[Nm]	Rate of change in performed work
W_d	[J]	Work performed by damper
X	[m]	Response amplitude
x	[m]	Displacement
x_c	[m]	Cargo position
x_p	[m]	Piston displacement
x_s	[m]	Stretch of the line
$\dot{x}$, v	[m/s]	Velocity
$\ddot{x}$	[m/s^2]	Acceleration
x_a	[m]	Accumulator faceplate displacement
β_s	[Pa]	Isentropic bulk modulus
β_t	[Pa]	Isothermal bulk modulus
γ	[-]	Heat capacity ratio of liquid/gas
ζ	[-]	Damping ratio
ν	[m^2/s]	Kinematic viscosity
ρ	[kg/m^3]	Density
φ	[^{o}]	Phase lag
ω	[rad/s]	Excitation frequency
ω_n	[rad/s]	Natural frequency

Abbreviations

AV	Across Variable
CoG	Centre of Gravity
DLL	Dynamic Link Library
DOF	Degree of freedom
DP	Dynamic Positioning
FPSO	Floating Production Offloading and Storage unit
FPU	Floating Production Unit
HMT	Heavy Marine Transport
HTV	Heavy Transport Vessel
LMU	Leg Mating Unit
LNG	Liquefied Natural Gas
MCA	Multi criteria analysis
PID	Proportional-Integral-Derivative
TLP	Tension-Leg Platform
TV	Through Variable

1 Introduction

In the offshore industry, large and heavy structures are used. These structures need to be transported around the world between the fabrication yard and the final destination or between different operating areas; this is known as Heavy Marine Transport (HMT). One of the methods which can be used to transport these structures is dry tow.

Dry tow means that the structure is transported on a barge or self-propelled semi-submersible vessel, which transports the structure from one location to another. Therefore loading and discharge operations are needed. The vessels are known as heavy transport vessels (HTV) and the transported structures are referred to as cargo. This thesis will focus on the equipment needed for the loading and discharge operations at open sea.

1.1 Background information

In the recent years, the demand for energy resources has been rising worldwide and it is expected that it continues rising. This has caused a shift from the traditional exploration areas to deeper and more remote areas. As a result, the offshore structures have been growing in size due to the increasing water depths; there is a link between the size of a structure and the water depth in which it is operating.

Most of the times, the large and heavy structures (see Appendix A.1) are transported from the fabrication yards, often located in Asia, to the exploration areas, often in Africa, Brazil and the Arctic region (see Figure 1-1) . HMT is a widely accepted method to perform these transportations. Dockwise (since 2013 part of Boskalis) is a Dutch shipping company and the global market leader in dry HMT operations. Dockwise operates a fleet of 23 HTVs of different sizes and designs. The flagship is the Dockwise Vanguard.

Figure 1-1 Major shipping routes [1]

However HMT has its limitations. The most important one is that HMT is limited by the capacity of the HTV. Another limitation is provided by the occurring motions between the HTV and the cargo during the loading and discharge operations. These motions are caused by wind, waves and currents. The operations can become extremely risky when the motions of the cargo and the ship exceed a certain limit. For this reason the present loading and discharge operations are executed in sheltered areas; such as harbours and other inland locations. At these locations the environmental conditions are benign, making it possible to execute the operations safely and without the occurrence of excessive motions between the cargo and the vessel. Due to the increase of the cargo dimensions also the draft of the cargo increases. This will limit the number of inshore locations available for loading and discharge operations, which implies that there is a growing need for offshore loading and discharge operations.

Another motivation is that Dockwise wants to reduce the voyage time from the yard to the exploration area. At this moment, the transportation of an offshore structure is done in two parts. The first part is done by a dry tow transport from the fabrication yard to a sheltered location near the exploration area where the HTV is discharged. The second part is done by wet tow from the discharge location to the final destination. Dockwise wants to eliminate the wet tow part, because the final destinations become more distant from the sheltered discharge locations. This causes the wet tow part to form a significant part of the journey. Therefore the competiveness of the HMT can be strengthened with the addition of the offshore loading and discharge capability [2].

On top of this, Dockwise discovered that the offshore dry-docking of Floating Production Offloading and Storage units (FPSOs) could be a new market in the HMT industry [3]. At this moment, the FPSOs are disconnected from their mooring lines and risers when they need to be transported to a repair yard. In the proposed concept, the FPSO is loaded onto an HTV and stays connected to the mooring and riser lines. The FPSO is dry docked onto an HTV with opportunities to keep producing while it is repaired and maintained offshore. However this is only possible if the HTV can perform offshore loading and discharge operations.

The loading and discharge operations follow a specific sequence, which is employed in both sheltered and offshore locations. The sequence of the discharge operation is almost the same as the loading operation, but then in reverse. Therefore only a short overview of the sequence of the loading operation will be given (for a more elaborated description see Appendix A.2). The loading operation starts with the preparation of the HTV and the cargo. The cribbing beams and guideposts (see Appendix A.3) are positioned on the HTV and the HTV and cargo are submerged to their loading draft. The cargo is towed to the HTV by tugs and connected to the standard cargo handling system which positions it correctly above the cribbing beams. The standard cargo handling system consists of Tugger winches and lines (see Appendix A.4). The HTV is de-ballasted and the cargo is placed on the cribbing beams. If the cargo is positioned correctly, the tugs are disconnected and the HTV will prepare for its voyage.

1.2 Problem statement

The prospects for offshore loading and discharge operations seem promising; however there are challenges which need to be solved before these can be applied. The environmental conditions in which the HTV is operating are more harsh than at the sheltered locations. The operating limits need to be higher than the standard limits which are considered at sheltered locations; to ensure that the loading/discharge operation can be executed with limited waiting-on-weather time.

It is necessary to be able to predict the hydrodynamic behaviour of the cargo above the HTV during the loading and discharge operation. The HTV and the cargo represent a multi-body dynamic problem, due to the wind, current and waves [1]. The vessels experience different motions (see Figure 1-2 and Table 1-1); these motions are dominant in the horizontal or vertical plane. The horizontal motions are the surge, sway and yaw motion, while the vertical motions are the heave, pitch and roll motion [4]. The occurring motions in one plane have a limited influence on the motions in the other plane, i.e. the influence of the vertical motions on the horizontal motions is limited. Therefore the problem can be splitted in vertical and horizontal relative motions between the bodies.

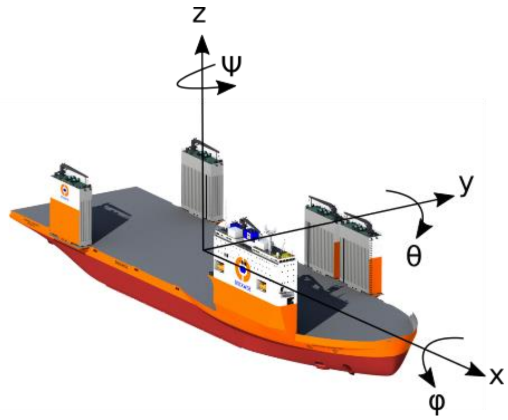

Figure 1-2 Coordinate system ship motions

Table 1-1 Ship motions

Translational			Rotational		
Surge	x	[m]	**Roll**	φ	[rad]
Sway	y	[m]	**Pitch**	θ	[rad]
Heave	z	[m]	**Yaw**	ψ	[rad]

Research performed by Dockwise showed that the prediction of the relative vertical motions is inaccurate. It is shown that these inaccuracies are caused by the narrow gap of water between the HTV and the cargo [1]. Research to solve this hydrodynamic problem is in progress and is not part of this thesis.

It is necessary to control the relative horizontal motions between the HTV and the cargo in order to ensure reliable and safe operations. Excessive motions can lead to collisions between the HTV and the cargo or wrong positioning of the cargo on the cribbing beams; both can lead to significant damages. Besides this, large motions will create slack in the mooring lines, which increases the chance on snapping when the lines are rapidly loaded. Furthermore the loads caused by the wind, wave and current are larger compared to sheltered locations. This causes the bodies to have larger motions which means that the relative horizontal motions also become larger. To control the relative horizontal motions between the cargo and the HTV, the cargo handling system needs to be optimized or extended.

In the recent years several studies were conducted to develop the optimal cargo handling system, different systems were considered; an extension of the standard cargo handling system, an active tension control system and the use of ShoreTension. The first system consisted of the standard cargo handling system with the addition of fenders. The idea was that the fenders would absorb the relative motions; however fenders are too stiff. Therefore the fender acted as a spring, feeding back energy to the motions [5]. Therefore this system only changes the stiffness of the system, increasing the risk on resonance. The active tension control system reduced the relative horizontal motions by increasing the tension in the lines, which connect the HTV and the cargo. The required tension is calculated by a PID controller which allocates the required tension to the lines through the Tugger winches. Therefore this system is actively controlled. However the required power and speed, the winches need to deliver is too high to reduce the high frequency relative horizontal motions (for a more elaborate description see Appendix A.5) [6]. ShoreTension is a mooring device which keeps a constant tension on the lines connecting the vessels. The system consists of a hydraulic cylinder and a piston. The piston moves out if the tension in the line is larger than a certain limit and moves in if the tension is lower than a certain limit. This causes the ShoreTension system to be a passive system. Simulations showed that the system reduces the relative motions if they are excited in the natural frequency or low frequency area of the system, through the addition of damping and stiffness. It is also noticed that the system operates more as a safety system to prevent the line tensions from becoming too high (for a more elaborate description see

Appendix A.5) [7]. The mentioned systems are only capable of reducing the low frequency relative horizontal motions; therefore these systems cannot be applied on offshore loading and discharge operations. However the studies showed that it is likely that, if a system can manage the high frequency motions, it is also able to manage the low frequency motions. In hydrodynamics the low frequency motions are caused by small low frequency forces which occur at very soft systems.

1.3 Objectives

The objective of this thesis is:

To develop a technical feasible system, capable of reducing the relative horizontal motions of the cargo with respect to the HTV during offshore loading or discharge operations.

This thesis will focus on the most critical state of the loading or discharge operation. The most critical state is when the HTV is ballasting or de-ballasting and the cargo is floating above the HTV with only a small clearance between its keel and the deck of the HTV. In this phase the relative horizontal motion reduction system is used, which makes this the interesting phase of the loading or discharge operation. Consequently, only the ballasting or de-ballasting phase is taken into account when a reference is made to offshore loading or discharge during this study.

The proposed relative horizontal motion reduction system will have an effect on the other phases of the loading and discharge operations. However, the other phases of the operations are only of interest when it is proven that the proposed system minimizes the relative horizontal motions sufficiently.

1.4 Outline and methodology

In order to obtain the objective of this thesis, the design trajectory will be followed. At first the problem will be analysed in the analysing phase. A multibody hydrodynamic model is used to gain insight in the behaviour of the HTV-cargo system when the standard cargo handling system is used. At the end of the analysing phase the design requirements will be set for the to be developed relative horizontal motion reduction system (Chapter 2). In the next phase, conceptual designs are created of the possible solutions to solve the offshore loading and discharge problem. The concept of diverging and converging will be used. At first all the possibilities will be listed (diverging part). Based on these possibilities, feasible concepts will be created (start of the converging part). At the end of the conceptual design phase the two best concepts will be selected for further development (Chapter 3). These concepts will be taken to the preliminary design phase. In this phase a more comprehensive design of the concepts will be developed. Furthermore the performances and characteristics of the concepts are investigated. These analyses will be performed with different MATLAB-Simulink models (Chapters 4 and 5). This thesis will conclude with the conclusions about the feasibility of the preliminary designs and will provide recommendations for future research projects (Chapter 6). The design trajectory is graphically presented in Figure 1-3. Unfortunately it is not possible to perform the complete design trajectory during this master thesis; the scope of this thesis is limited to the preliminary design phase. The complete design trajectory is presented to provide a clear overview how a final design of a usable system should be obtained. For a more elaborate description of the design trajectory see Appendix A.6.

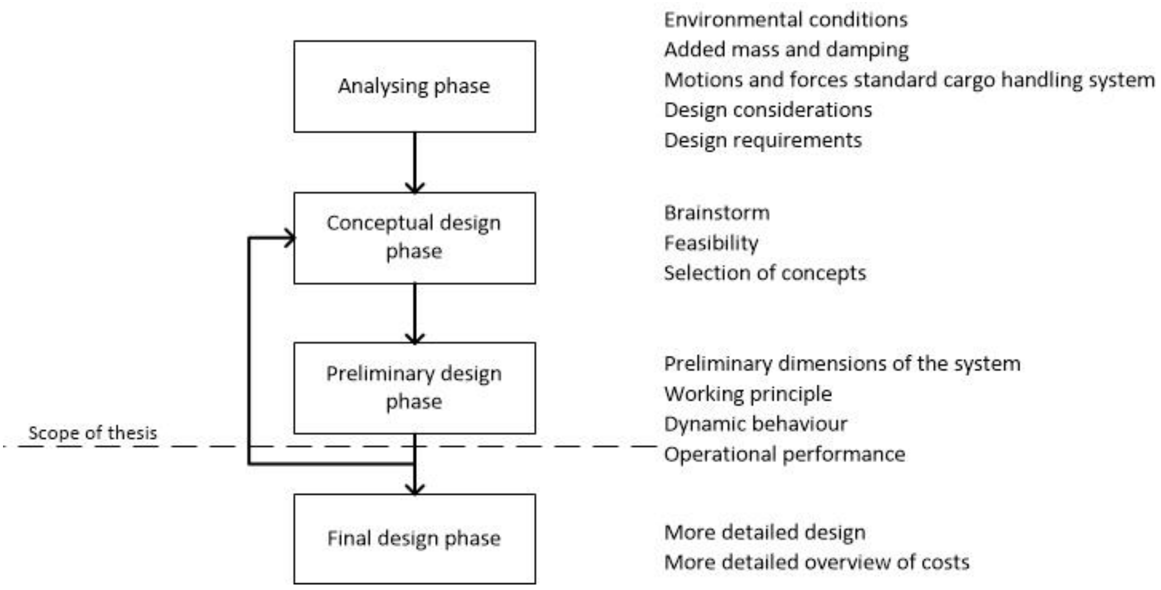

Figure 1-3 Design trajectory

2 Analysing phase

This chapter will describe the behaviour of the Dockwise Vanguard and the cargo in its initial state, in which the standard cargo handling system is used. It is also explained which type of simulation models are used. The chapter also focuses on the design values which are used for the new relative horizontal motion reduction system, as well as the design considerations for this new system. The chapter is concluded with the design requirements.

2.1 System description

This section will describe the starting points which are used to simulate the relative horizontal motions of the standard cargo handling system. The simulations will be conducted for a base case and will be performed with a multibody hydrodynamic model.

Multibody hydrodynamic model

The behaviour of the base case will be analysed with a multibody hydrodynamic model in AQWA. AQWA is an engineering toolbox to investigate the effects of environmental loads, like waves, wind and currents, on floating offshore and marine structures [8] (for a more elaborated description of AQWA see Appendix F.1).

In order to analyse the problem, a hydrodynamic diffraction model of the coupled system, the HTV with the cargo, is set up in AQWA. The diffraction analysis predicts the hydrodynamic diffraction characteristics of the system given a specific configuration. The hydrodynamic diffraction characteristics can be used to perform a time domain simulation in AQWA. The AQWA model produces the following outputs: all the forces working in the system, the positions of the HTV and the cargo, the frequency dependent added mass and damping.

Base case

The Dockwise Vanguard (see Figure 2-2) and the FPSO MOHO Nord (see Figure 2-3) are selected as the system's HTV and cargo respectively, for this thesis. However the system which will be presented at the end of this thesis will not be limited to this specific HTV or cargo. The dimensions of the vessels are given in Table 2-1 and Table 2-2. More specifications of the Dockwise Vanguard are given in Appendix B.1.

Figure 2-1 Dockwise Vanguard

Figure 2-2 FPSO MOHO Nord

Table 2-1 Dimensions of the Dockwise Vanguard

Length overall	275	[m]
Length between casings	155.5	[m]
Width moulded	70	[m]
Width max	78.75	[m]
Depth	15.5	[m]
Draft submerged	31.5	[m]
Maximum water depth above deck	16	[m]
Deadweight	117,000	[t]

Table 2-2 FPSO dimensions

Length	245	[m]
Width	44	[m]
Height	18	[m]
Draft	7	[m]
Weight	78,500	[t]

Environmental conditions
A JONSWAP swell wave spectrum with a significant wave height (H_s) of 2 [m] and a peak period (T_p) of 10 [s] is employed in the simulations. This H_s-T_p combination is chosen based on preliminary studies which show a relation between the wave period and the wave height (see Appendix B.1). It is considered to be a good representation for a conceptual design.

Global mooring system
In the simulations, the Dockwise Vanguard is kept on position by a global mooring system; four anchored mooring lines. This global mooring system represents an actual mooring system or temporary tug assistance. The stiffness of this mooring system is chosen such that the natural frequencies in surge, sway and yaw direction do not coincide with the excitation frequencies of the wave spectrum. Therefore, drifting of the Dockwise Vanguard is not a critical issue, similar to a real offshore loading or discharge operation. The global mooring system is visualized in Figure 2-3.

Configuration cargo handling system
The FPSO is connected to the Dockwise Vanguard with 8 lines. The stiffness of the lines varies significantly for every line, as the lines differ in length. The lines will approach the Dockwise Vanguard almost horizontally. The clearance between the keel of the cargo and the deck of the Dockwise Vanguard is kept constant during the simulations.

Three types of mooring configurations of the cargo handling system are tested: a soft configuration, a stiff configuration and a joint configuration. The soft configuration has the stiffness of the standard cargo handling system. The stiff configuration is three times as stiff as the soft configuration. The joint represents an infinitely stiff connection between the cargo and the Dockwise Vanguard.

The layout of the cargo handling system is shown in Figure 2-3, this is the layout in case the cargo is positioned above the deck of the Dockwise Vanguard and the (de-) ballasting operation is started. During the positioning operation the cargo will enter the area above the deck from the aft of the Dockwise Vanguard. Therefore the lines will have different lengths and angles between the Dockwise Vanguard and the cargo during the start of the operation, compared to what is shown in Figure 2-3. The focus of this thesis will be on the motions during the critical state of the loading and discharge operation; the ballasting and de-ballasting phase of the HTV when the cargo is floating above the HTV (see Chapter 1.3). Therefore the layout of Figure 2-3 is used during these simulations.

Motion measurements
During the AQWA simulations the relative motions are measured between the centre point of the cargo at the keel and the corresponding point on the deck of the HTV. If these two points are exactly on top of each other, the relative motions are zero and the cargo is located at its correct final location. In this thesis the right hand coordinate system will be used, for example the surge motion is positive in the direction of the bow. The origin of the coordinate system is in the Centre of Gravity (CoG) of the vessel in the submerged condition.

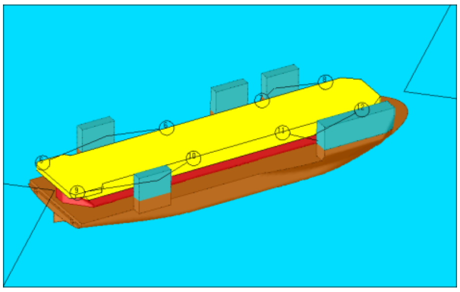

Figure 2-3 Mooring configuration of present cargo handling system

2.2 AQWA results

This section describes the results of the AQWA simulations for the HTV-cargo system, in an irregular wave environment. It is also discussed what the effect of a different system stiffness' is on the resulting relative horizontal motions and the relative excitation force working on the system. The basic theory behind the calculation of the motions is explained in Appendix B.2.

Motions

Figure 2-4 and Figure 2-5 show the relative horizontal motions as measured from the centre of the FPSO. The absolute maximum and minimum values encountered are plotted for all wave directions. Also the maximum allowed relative motion amplitude is plotted. It can be concluded from the figures that the amplitude of the relative motions is larger than the maximum allowed motion. Therefore a new relative horizontal motion reduction system needs to be developed.

The figures show that the relative motions are larger if a stiff system is used than if a softer system is used. The increase in stiffness causes the natural frequency of the system to shift to the excitation frequency of the waves, therefore larger motions occur. The amplitude of the relative motions in the sway direction stays more or less the same. This is due to the large angle of the lines between the cargo and the Dockwise Vanguard. Therefore the stiffness of the mooring lines in the sway direction is very small. If a very small value is multiplied with a small value the product is still a small value. For this reason the relative horizontal motions in the sway direction are approximately the same for the different system stiffness's. The relative motions for the joint connection are not plotted in the figures because the joint is infinite stiff and therefore the relative motions associated with the joint connection will be zero.

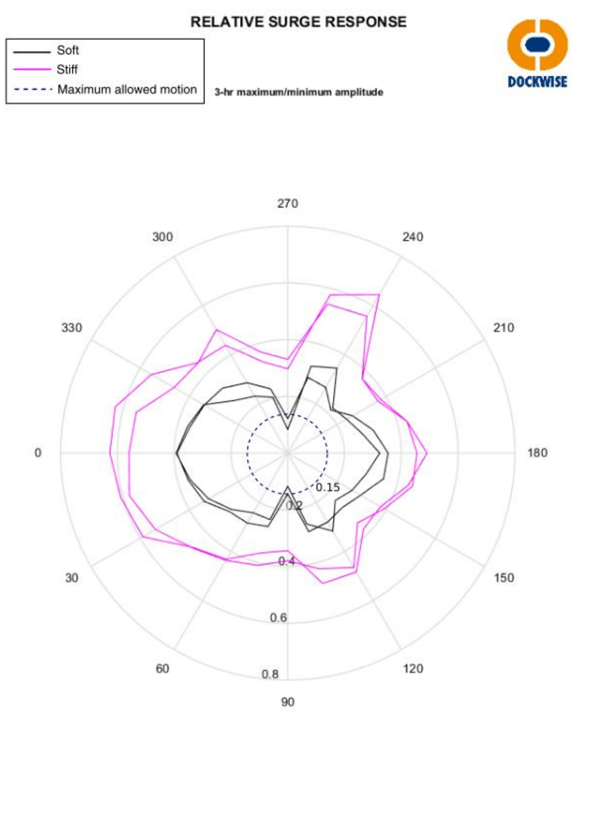

Figure 2-4 Relative surge response in [m]

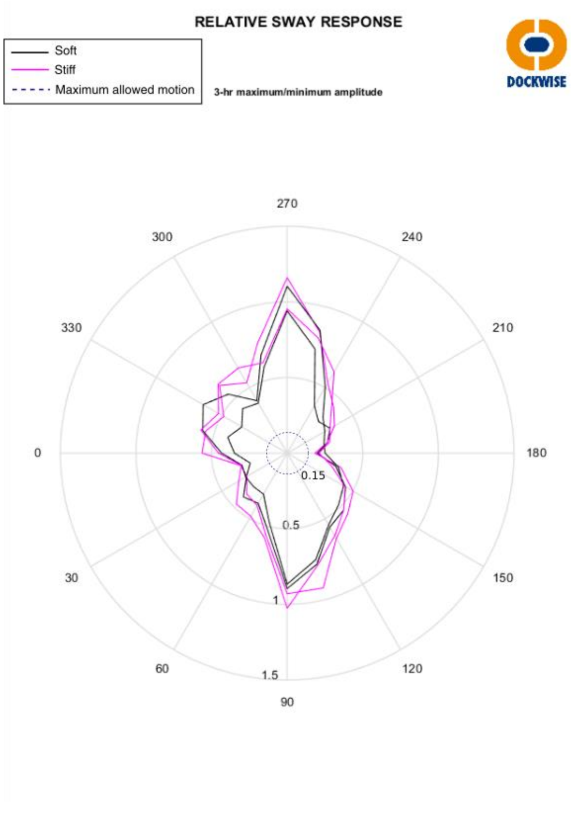

Figure 2-5 Relative sway response in [m]

Frequency

Figure 2-6 shows the relative surge motions (cargo's keel, HTV's deck) over the frequency when the Dockwise Vanguard and the cargo are positioned in head waves (180 degrees wave direction). Figure 2-7 shows the relative sway motions over the frequency when the Dockwise Vanguard and the cargo are positioned in beam waves (90 degrees wave direction). The blue dotted line shows the wave frequency for which the motions and forces are plotted in the previous figures (T_p = 10 [s]).

In Figure 2-6 two resonance peaks are visible for the two different systems. The peaks at 0.06 [rad/s] are the resonance motions caused by the global mooring system. The stiffness of the cargo handling system does not influence these motions; therefore the responses are the same. The peaks at 0.25 [rad/s] for the soft system and 0.42 [rad/s] for the stiff systems are the natural frequencies of the HTV-cargo system in surge direction. The figure clearly shows that the natural frequency of the stiff system is larger than that of the soft system. The natural frequency of a system is dependent on the stiffness and the mass of the system (see equation (B-5)). The mass of the system will be approximately the same for the two systems; therefore the natural frequency will be dependent on the stiffness of the system. Consequently the natural frequency of a stiffer system will be larger than that of a less stiff system.

In Figure 2-7 multiple resonance peaks are visible, in addition to the sway response also the yaw response is included in the figure. It is noticed that the natural frequency of the global mooring system and the cargo handling system are very close to each other. The shift in natural frequency, due to the increase in line stiffness, is less clear because the different natural frequencies are less distinct.

The amplitude of the relative motions is calculated according to equation (B-3). The amount of damping within this system is very small. Therefore the amplitude of the motions is heavily dependent on the stiffness of the system; i.e. the natural frequency of the system. If the wave frequency equals to the natural frequency the motions will become very large, because little damping is applied to the system to prevent these motions from occurring. For this reason it is possible that the motions of the soft system are smaller than those of the stiff system. If the stiff system is acting near its resonance region the motions are automatically larger.

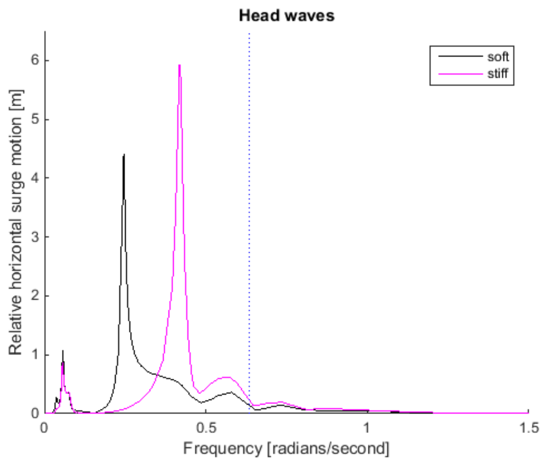

Figure 2-6 Relative surge motion over the frequency

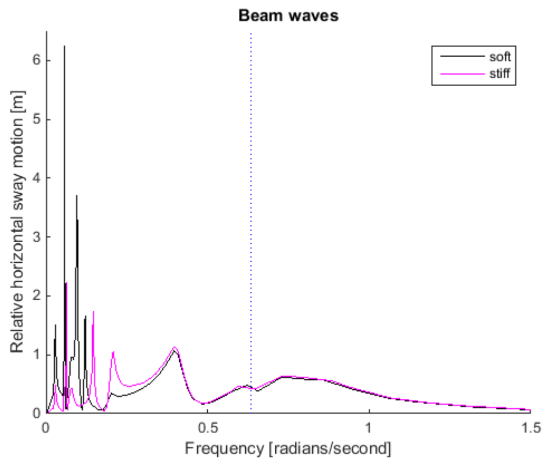

Figure 2-7 Relative sway motion over the frequency

Loads

It is also important to know which loads are working on the standard cargo handling system. Figure 2-8 and Figure 2-9 show the loads on the connection between the Dockwise Vanguard and the cargo. The absolute maximum and minimum values are plotted for all wave directions. The wave force on the cargo is also plotted; the blue line. It is noticed that the loads on the cargo handling system are smaller than the wave force. This is caused by the fact that the Dockwise Vanguard is allowed to move with the cargo, reducing the load on the cargo handling system.

It can be concluded from the plots that the loads working on the cargo handling system increase if the stiffness of the cargo handling system is increased. However the plots are produced for a certain wave frequency (T_p = 10 [s]). On top of this, the natural frequencies of the three systems are different. The natural frequency is dependent on the stiffness of the system (see equation (B-5)). If the wave frequency is near the natural frequency of the system, the motions of the system will become large, increasing the loads in the cargo handling system. Therefore it is still possible that the soft system induces a larger load on the system than the stiff system in a certain frequency range.

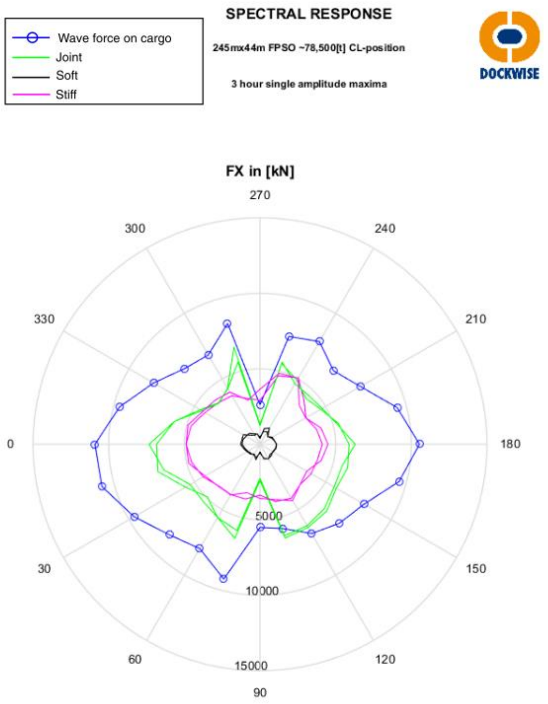

Figure 2-8 Force in surge direction

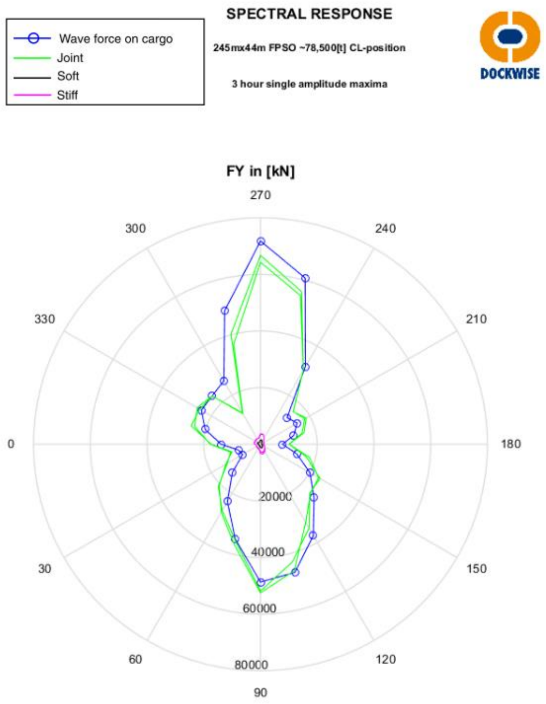

Figure 2-9 Force in sway direction

2.3 Design approach

In order to develop additional equipment for the standard cargo handling system or a new relative horizontal motion reduction system, simulations need to be performed to analyse these systems. These simulations will be performed with different MATLAB models. For a more elaborated description of MATLAB see Appendix F.2. The obtained results from the AQWA model can be used as input values for the MATLAB models.

The focus for the conceptual phase and the preliminary phase lies on defining a suitable solution for the offshore loading and discharge problem. The performed simulations in these phases serve to gain insight in the working principles of the system and to find out if the system is capable of reducing the amplitude of the relative horizontal motions. This can be done with a simplified model. The advantage of such a model is that it is better controllable and it is also easier to adjust features and parameters and to investigate the response of the system to these adjustments. The HTV-cargo system represents a multi-body system, each body has 6 degrees of freedom (DOFs). Therefore the HTV-cargo system has 12 degrees of freedom. As only the horizontal motions are considered, the system can be reduced to 6 DOFs. Another reduction is obtained by taking only the relative horizontal motions into account, the system has 3 DOFs. The last reduction is obtained by allowing the cargo to move only in the surge or sway direction (1 DOF). The system is reduced from having 12 degrees of freedom to 1 degree of freedom.

The advantage of using a simplified MATLAB model instead of an AQWA model is that it is easier to program the models in the MATLAB environment. It is also not possible to implement hydraulic or pneumatic actuator models or control systems directly into AQWA. This needs to be done through a Dynamic Link Library (DLL) file. Therefore only MATLAB models will be used in the preliminary design phase. Although the results of the MATLAB models will be less accurate than for the AQWA models, due to the simplifications which are made. However the focus of these phases is on the global performance of the systems. Therefore MATLAB provides sufficient accuracy for the current research.

The HTV-cargo system can be modelled as a mass-spring-dashpot system. The following simplifications are made for this model, compared to the AQWA model:

- The HTV is simulated as a fixed reference point,
- The system is modelled as a 1 DOF system. The cargo is allowed to move in one direction only. During this study the focus will be on either the surge or the sway motion,
- The cargo is simulated as a point mass. The position of the Dyneema lines, cargo handling system, or excitation force are neglected. The resulting moments are not taken into account,
- The Dyneema lines are simulated as springs with a constant linear stiffness. Contrary to AQWA where the direction of the springs and thus the stiffness can change in time, depending on the position of the cargo relative to the HTV. The Dyneema lines are presented as one spring,
- The added mass and damping are taken as constants, in contrary to AQWA where a frequency dependent added mass and damping is used. In reality the values of the cargo are influenced by the nearby HTV and vice versa.
- The system will be excited by a regular wave sea state instead of an irregular sea state, as in the AQWA simulations.

Of course the simplified model is not as accurate as a 12 DOF model excited by an irregular sea state. However the inaccuracies in the model caused by the simplified added mass, damping and excitation force do not change the principles of the system. The performance of the system will be investigated for low and high frequencies. The performance of the system will also be investigated near the natural frequency of the system. In this case this will be the natural frequency which corresponds to the simplified model and not to the real system. However if the system performs well at these frequencies, it is likely that the system also performs well at the corresponding frequencies for the real system.

The next step is to create a hydrodynamic diffraction model of the cargo only. This diffraction model provides valuable information like the frequency dependent added mass, damping and the motions of the cargo. This information can be used to create a 6 DOF model to analyse the motions of the cargo in different directions under the influence of different excitation forces. However the system will still be excited by a regular sea state. Therefore the frequency dependent added mass and damping still need to be converted to frequency independent values. In this case the added mass needs to be chosen such that the model has the same natural frequency as the real system. If the system performs well for 6 DOF, the model can be extended to 12 DOF for a regular sea state. Finally the system can be made such that the system has 12 degrees of freedom and is excited in an irregular sea state. Furthermore the added mass and damping are frequency dependent in this case. However this is outside the scope of this research, because this research focuses on the preliminary design of a relative horizontal motion reduction system.

2.4 Design values

The output values, like the forces, of the multibody hydrodynamic model can be used as input values for the simulations of the motion reduction system. They can also be used to compare the performance of the standard cargo handling system and the new motion reduction system. However in order to compare the performance of the two systems, the obtained relative motions in the MATLAB models and the AQWA model for the standard cargo handling system should be the same. This section explains how the excitation force, added mass and damping values for the MATLAB model are determined. With these values known, it is possible to determine the corresponding wave excitation force for a certain frequency with a simple MATLAB mass-spring-dashpot model (see Appendix B.3). The total response of the cargo in the transverse direction is composed of sway and yaw motions of the cargo. While in the longitudinal direction the response is mainly composed of surge motions and the influence of yaw is little. For the 1 DOF model only the total response and its corresponding excitation force is taken into account.

Regular wave

An important difference between the AQWA model and the MATLAB model is that AQWA employs an irregular wave field and MATLAB employs a regular sea state. Therefore the regular design wave approach [9] is used to convert the irregular sea state into a regular sea state. The following steps are performed in order to determine the regular design wave. At first the governing irregular wave characteristics are determined (see Chapter 2.1). The next step is to select an appropriate regular wave frequency that causes the maximum response of the system. Finally the corresponding wave height, which together with the wave frequency produces the same motion amplitudes as the irregular wave, needs to be determined.

According to this approach, the regular wave has the following characteristics:

- H_s = 2 [m]
- T_p = 10 [s]

Added mass and damping

In reality the added mass and damping are frequency dependent. However in order to be able to understand how the to be designed systems operate and perform, relative simple models will be used. These models will be frequency independent; constant coefficients will be used, making them simple and easy to apply [10]. The systems can be modelled according to the low frequency model, because this model is accurate for manoeuvring, station-keeping and control in a seaway [10]. According to this model the added mass and damping values are determined (see Appendix B.3). The constant added mass and damping values are shown in Table 2-3.

Table 2-3 Added mass and damping values

	Added mass	Damping
Surge direction	7500 [t]	1 [t/s]
Sway direction	80,000 [t]	131.4 [t/s]

Wave excitation force

Until this point all the important input parameters for the MATLAB model are known, except for one; the wave excitation force. However the AQWA simulations also provided the maximum amplitude of the relative horizontal motions between the HTV and the cargo. With the motions known, it is possible to calculate the corresponding wave excitation force (see Appendix B.3).

In this way the following excitation forces in the surge and sway direction are determined:

- For surge direction: F_{ext} = 8000 [kN]
- For sway direction: F_{ext} = 25,000 [kN]

2.5 Design considerations

The previous sections provided a clear overview of the factors which influence the motion behaviour of the HTV-cargo system. It is concluded that a connection between the HTV and the cargo is needed, which has a higher stiffness than the standard cargo handling system, in order to reduce the relative horizontal motions. It is also important to minimize the chance that resonance occurs, or to find a way to reduce the amplitude of the motions if the excitation frequency is close to the natural frequency of the system.

The motional behaviour of the HTV-cargo system is dependent on the natural frequency of the system and the excitation frequency to which the system is subjected. Figure 2-10 gives an overview of the three frequency areas which can be distinguished with respect to the motional behaviour of the system:

- The low frequency area, $\omega << \omega_n$. In this area, the motions are dominated by the spring terms in the equation of motion (see equation (B-1)) [4]. The spring terms represent the stiffness of the system.
- The natural frequency area, $\omega \approx \omega_n$. In this area, the motions are dominated by the damping terms in the equation of motion [4].
- The high frequency area, $\omega >> \omega_n$. In this area, the motions are dominated by the mass terms in the equation of motion [4].

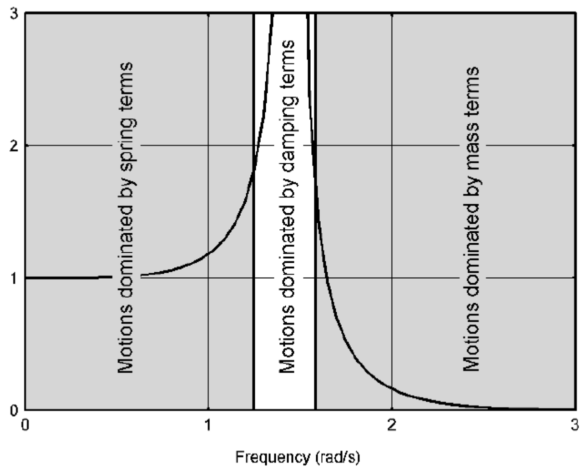

Figure 2-10 Frequency areas with respect to motional behaviour [4]

This means that it is possible to reduce the motions of a dynamic system by changing the stiffness of the system or the amount of damping which the system experiences.

It is possible to reduce the relative horizontal motions by connecting the HTV and the cargo to each other by a stiff connection. However the natural frequency of this system should be outside the excitation frequency. The natural frequency of the system shifts to the right (see Figure 2-10), if a stiffer connection is established. Therefore the minimum required stiffness of the system would be (see Appendix B.4):

- k_{surge} = 344,000 [kN/m]
- k_{sway} = 634,000 [kN/m]

However it is practically impossible to make such a stiff connection between two moving objects in a few seconds. The forces associated with the ceasing of the motions are too large for all the involved structures to accommodate, if this is done within a few seconds.

Another possibility to reduce the relative horizontal motions is to add damping to the system. The damping will absorb the kinetic energy of the system; the motions will be reduced because less energy is available to generate large motions (see Appendix B.2). However in order to reduce the relative motions of the system a lot of damping is required, if the stiffness of the system stays the same (see Appendix B.4). This is caused by the fact that the natural frequency of the standard cargo handling system and the excitation frequency are not that close, therefore the effect of adding damping to the system is small. The amount of damping needed is larger than the critical amount of damping; the system will be overdamped.

It can be concluded that it is very unlikely to reduce the relative horizontal motions by only adding damping to the system or by only increasing the stiffness of the system. Therefore it is recommended to develop a system in which both stiffness and damping are utilized.

The working principle of such a system can be as follows. The system will have a low stiffness at the start of the positioning operation and a high stiffness at the end of the positioning operation. The natural frequency consequently shifts from a low frequency at the start to a high frequency at the end. At the start of the operation, the motions of the system are dominated by the mass terms. As the stiffness becomes higher the changing natural frequency will coincide with the wave frequency of the system and the system will start to resonate. As the stiffness is still increasing (i.e. the natural frequency is increasing), the system will enter the phase in which it is dominated by the spring terms. This is the desired situation in which the HTV will start the de-ballast or ballast phase of the offshore loading or discharge operation.

The resonance motions can be reduced by adding damping to the system, because the motions in the natural frequency area are dominated by damping. If a small amount of damping is added, a large resonance peak is expected (see equation (B-3)). Thus if a large amount of damping is added, a small resonance peak is expected. On top of this, time is needed for the system to reach the large resonance motions [11]. Therefore if enough damping is added and if the stiffness is increased fast enough it is unlikely that large resonance motions will occur. Therefore a solution which uses both stiffness and damping seems promising.

2.6 Design requirements

Before the design process can truly commence, the design requirements, starting points and assumptions have to be defined. The design requirements specify which tasks the cargo handling system should accomplish. The starting points are limitations and requirements which are necessary to design a system. In every design process certain assumptions will be made to be able to create the design of the system.

Requirements

- The system should be able to position the cargo within certain limits,
- The system should be fail safe,
- The system should be able to cope with the protrusions on the cargo,
- The large motion amplitude if resonance occurs should be minimised,
- The system should be safe for the cargo; the system is not allowed to damage the cargo.

Starting points

- The system should be able to handle different types of cargo,
- The maximum relative horizontal motions are 15 [cm] to each side (half of the width of one cribbing beam) (see Appendix A.3),
- The system should fit onto different HTVs,
- The system is operating in the following JONSWAP maximum wave spectrum:
  - H_s = 2 [m]
  - T_p = 10 [s]
- The design case consists of the Dockwise Vanguard and an FPSO,
- The stiffness of the standard cargo handling system is 4000 [kN/m] in surge direction and 1150 [kN/m] in sway direction.
- It should be able to place the system at certain spots on the HTV which can be different per operation.
- The total installed power on the Dockwise Vanguard is 28,500 [kW] (see Appendix B.1).

Assumptions

- The wind and current loadings are not taken into account. As a result, the actual mean loads on the HTV and the cargo will be different in reality. However, influence of wind and current load on the dynamic behaviour and loads are assumed negligible; in fact wind and current may contribute to damping in an oscillating system.
- The height of the vertical supporting structure ranges between 0.3 and 1 [m].
- The clearance between the keel of the cargo and the top of the supporting structure is 2 [m], at the start of the operation.
- The bilge radius of the cargo is approximately 2 [m].
- The current sea fastenings have a holding capacity of 200 tonnes.

3 Conceptual design phase

In this chapter a range of concepts, to solve the offshore loading and discharge problem, is developed. The requirements and restrictions of the previous chapters are taken into account. The development of the concepts starts with a brainstorm session with engineers from different disciplines. During this brainstorm session, a large amount of concepts is developed. Some of these concepts already appear to be unrealistic during the conceptual phase. The concepts are reviewed on their strong and weak points. The focus in this phase of the design trajectory is on finding a suitable solution to the offshore loading and discharge problem. Therefore the analysis of the concepts is confined to qualitative considerations. After the concept development, the concepts are compared to each other by a Multi Criteria Analysis (MCA). With the help of this analysis, two favourable concepts are selected for further development.

3.1 Concepts

This section describes the concepts which appear to be realistic at the end of the conceptual design phase. For an overview of the developed concepts which appeared to be unrealistic or which can be used to widen the operational limitations see Appendix C.1.

Cargo bottom lines

The underlying idea of this concept is to restrict the motions of the cargo at its keel level, by using Tugger winches and lines. The keel of the cargo needs to be positioned correctly on its supporting structure; therefore the motions of the keel are of greater importance than the motions at the top level of the cargo. To accomplish this, the winches are located on the deck of the HTV. The lines are attached to the side of the cargo opposite of the winch; the line of the portside winch is connected to the starboard side of the cargo (see Figure 3-1). In this way the motions are restricted at the keel level of the cargo. For a more elaborated description of this concept see Appendix C.1.

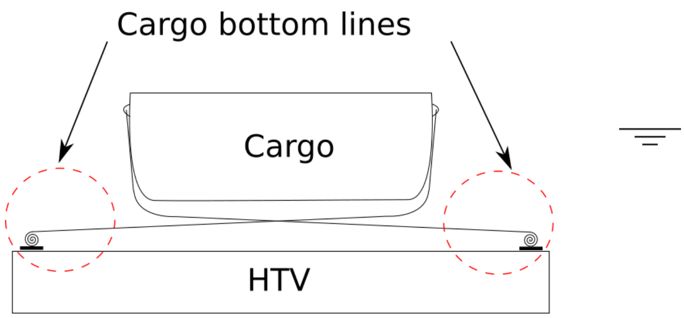

Figure 3-1 Cargo bottom lines, schematic cross section

Advantages and disadvantages

The advantages and disadvantages for this concept are discussed below.

Advantages

- The system is easy to construct and to install,
- The used technology is known,
- It is a relatively cheap system.

Disadvantages

- The system is heavily dependent on the stiffness of the used lines,
- The system can control the motions only in the transverse direction, another system is needed to control the motions in the longitudinal direction,
- Heavy equipment needs to be placed on the cargo.

Stretch compensation system

The principle of a stretch compensation system is to use the stretch in the mooring lines to transfer the wave energy working on the cargo into an energy absorber. The tension in the mooring lines is regulated by a hydraulic system. The idea is derived from the heave compensation systems used to place the subsea structures in the offshore industry.

The hydraulic cylinder is connected to the mooring arrangement. The motions of the cylinder compensate the stretch in the mooring line and in this way the cargo can be kept at the same relative horizontal position, because the horizontal length of the line stays the same. For a more elaborated description of this concept see Appendix C.1.

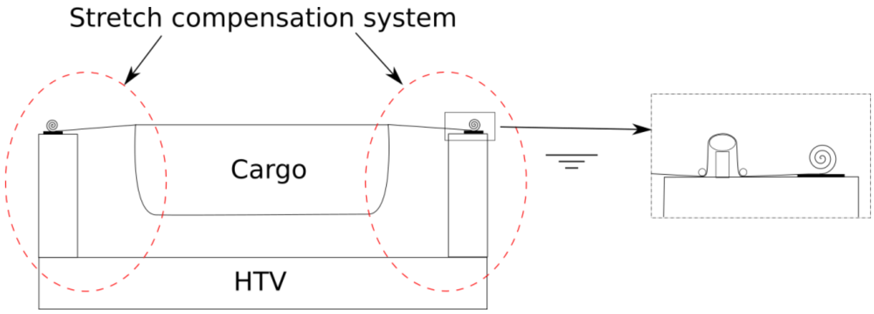

Figure 3-2 Stretch compensation system, schematic cross section

Advantages and disadvantages

The advantages and disadvantages for this concept are discussed below.

Advantages

- The system reduces the motions in both directions (longitudinal and transverse),
- The system is easy to install,
- Minor adjustments to the cargo are needed (pad eyes),
- The system is easy to connect to the cargo.

Disadvantages

- The system is difficult to construct, a lot of different elements are included,
- Due to all the moving parts, a lot of maintenance is required,
- It is an active system.

Teflon pads

The principle of this solution is to absorb the induced wave energy onto the cargo in order to reduce the relative horizontal motions. The idea is to use friction plates to overcome the horizontal excitation loads, reducing the relative horizontal motions. The friction plates will be mounted onto a jacking system which is placed on Teflon pads. Examples of jacking systems are the Unideck system and the Smart-Leg system [12].

The cargo will be placed on the jacking system (see Figure 3-3), during the (de-)ballasting process. As soon as the cargo stops moving on the jacking system, the (de-)ballasting process is paused. This is the moment to correctly position the cargo above the supporting structure. The jacking structure will be pushed over the Teflon pads by hydraulic cylinders. The Teflon pads are used to minimise the amount of friction, between the jacking structure and the deck of the HTV, the cylinders need to overcome. When the cargo is positioned correctly the (de-)ballasting process can be recommenced and the cargo is positioned on the supporting structures. For a more elaborated description of this concept see Appendix C.1.

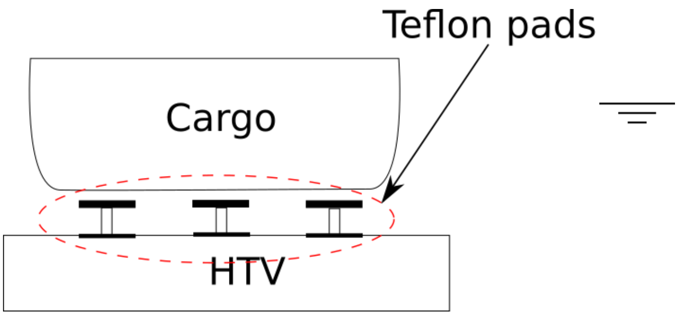

Figure 3-3 Teflon pads, schematic cross section

Advantages and disadvantages

The advantages and disadvantages for this concept are discussed below.

Advantages

- The system can position the cargo correctly in all the horizontal directions,
- The system is relatively easy to manage,
- The system is relatively simple to position on the deck of the HTV.

Disadvantages

- It is an active system, it is not certain if it is possible to control the motions if the system fails,
- A larger number of systems is needed, because the free area under the keel of the cargo is small due to the protrusions,
- The system is maintenance sensitive because it is located in a very corrosive environment,
- The system is operating just above the supporting structure.

Clamping

The underlying idea of this concept is to reduce the relative horizontal motions by connecting the cargo to a stiff system. A stiff connection can be utilised by a moveable structure like the system in Figure 3-4. The HTV and the cargo are connected with each other by the system which is placed on the deck of the HTV and will clamp the cargo in between the structures, accomplishing a stiff connection between the HTV and the cargo.

The proposed system consists of a stiff part and a moveable part (see Figure 3-4), and is located at opposite sides of the cargo. At the start of the operation the cargo is free to move in between the structures. However in time the hydraulic cylinder (the blue line) will start to move out, reducing the free space between the cargo and the structures. In this way the cargo will be clamped in between the structures and the motions of the cargo will be restricted. For a more elaborated description of this concept see Appendix C.1.

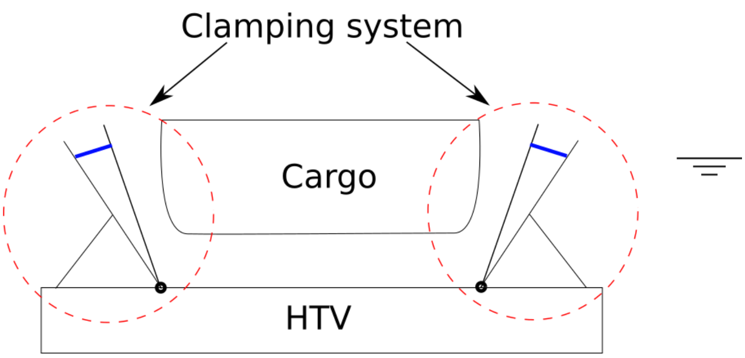

Figure 3-4 Clamping system, schematic cross section

Advantages and disadvantages

The advantages and disadvantages for this concept are discussed below.

Advantages

- The forces working on the system will slowly increase,
- It is possible to design the system such that if the control system fails the system will react as a stiff structure,
- The system is relative easy to connect to the cargo.

Disadvantages

- The system can control the motions only in one direction, another system is needed to control the motions in the other direction,
- It is (partly) an active system, if the control system does not work correctly the motions can become out of control.

Converging vertical system

In this solution a rigid structure is used to place the cargo correctly onto the supporting structure. Rigid structures will be placed at opposite sides of the cargo on the deck of the HTV; the cargo will be located in between the structures (see Figure 3-5). The basic idea of this solution is that the available space, wherein the cargo is allowed to move, is slowly reduced over the clearance height between the keel of the cargo and the top of the supporting structure. Therefore the possible relative horizontal motions between the cargo and the HTV will converge until they are within the design requirements (see Chapter 2.6). For a more elaborated description of this concept see Appendix C.1.

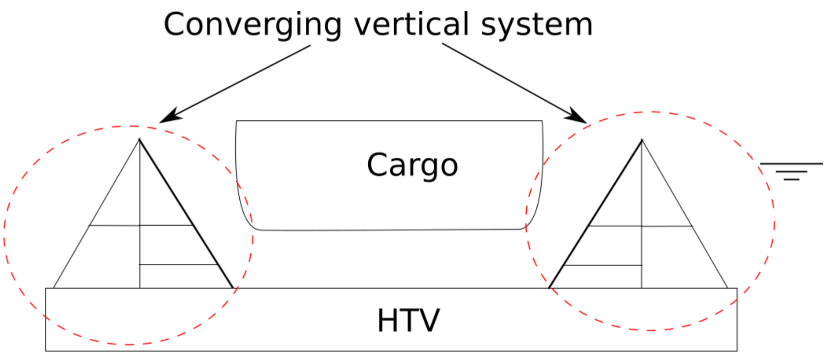

Figure 3-5 Converging vertical system, schematic cross section

Advantages and disadvantages

The advantages and disadvantages for this concept are discussed below.

Advantages

- The system is passive,
- Simple steel structures are used,
- The used technology is known,
- It is a relatively cheap system,
- The system is easy to construct and install,
- No extra attachments to the cargo are needed.

Disadvantages

- The system can position the cargo correctly in only one horizontal direction,
- It is difficult to locate the impact loads at the strong points of the cargo,
- The system is hard to adjust to different cargo hull shapes,
- The available height over which the motions can be reduced is limited.

3.2 Multi criteria analysis

The multi criteria analysis (MCA) is a quantitative method to determine the most viable concept. In this analysis the concepts are scored relative to each other on certain criteria. These criteria are different per project, but are considered important aspects of the project (for the explanation of the criteria see Appendix C.2). However some criteria are more important than other criteria. Therefore the criteria have different weight factors (for the determination of the weight factors see Appendix C.2). The concepts are scored on a scale from 1 to 10. In this scale is 1 lowest score, thus the worst rating and 10 is the best score, thus the best possible rating.

Table 3-1 MCA

Criteria	Weight factors	Concept 1: Cargo bottom lines	Concept 2: Stretch compensation system	Concept 3: Teflon pads	Concept 4: Clamping	Concept 5: Convergin g vertical system
Construction	0.04	8	3	7	5	9
Installation	0.05	6	9	3	6	7
Maintenance	0.07	4	3	5	6	8
Reuse/adjustability	0.09	8	9	5	5	2
Attachments to cargo	0.07	3.	8	9	9	9
Complexity	0.12	8	8	6	3	2
Efficiency	0.09	3	9	6	7	9
Energy consumption	0.02	3	3	4	5	9
Controllability	0.14	7	7	8	7	2
Safety to cargo	0.14	3	7	4	7	5
Fail safe/Redundancy	0.18	2	4	4	8	9
Score		**4.77**	**6.49**	**5.47**	**6.14**	**5.49**

3.3 Conclusion

The concept selection process started with a brainstorm session which provided a long list of possible solutions. Using a feasibility analysis this list was reduced to a shorter list with 5 concepts, of which the strong and weak points have been identified. An MCA is conducted to determine the best concepts. Based on the MCA and taking the challenges into account, the following 2 concepts are selected for further development:

- Concept 2: Stretch compensation system,
- Concept 5: Clamping.

4 Preliminary design Clamping system

This chapter will cover the outlook and performance of the clamping system. First the general outlook and working principle of the clamping system are discussed. The next section focuses on the description of the damping device. The third section describes the models which are used to analyse the behaviour of the damping device. A parametric study is conducted to see how the important parameters effect the behaviour of the damping device. The next study focused on the performance of the damping device. Finally an overall conclusion about the usability of the system is presented.

4.1 General outlook and working principle

The clamping system consists of two main parts:

- A stiff construction which is moveable and is used to clamp the cargo such that the motions of the cargo are restricted,
- A damping device which is a hydraulic system. This system is designed to absorb the motions of the cargo due to the harmonic loads, such as waves. It is decided to use a hydraulic system because hydraulic fluids are almost incompressible instead of gasses which are used in a pneumatic system (see Appendix D.1). Therefore the controllability of the system is better if liquids are used.

The clamping systems are placed at two sides of the cargo; portside and starboard side or front and aft of the cargo. This is depended on the type of cargo which is loaded or discharged onto the HTV. In this way, the motions of the cargo are restricted in the sway direction if the operation is performed in head waves or in surge direction if the operation is performed in beam waves.

The purpose of the clamping system is to increase the stiffness of the connection between the HTV and the cargo over time, to reduce the relative horizontal motions. Therefore the natural frequency of the complete system shifts from a low frequency to a high frequency. In this way the system changes from a mass driven system to a stiffness driven system (see Chapter 2.5). The working principle of the clamping system, to accomplish this, is divided into four phases:

1. In the first phase, the cargo is positioned in between the clamping systems (see Figure 4-1). The clamping systems are positioned around the deck of the HTV such, that the motions of the cargo are restricted only by the lines which are used to position the cargo in between the systems. In this phase the system stiffness is determined by the line stiffness and the amount of damping is zero.
2. In the second phase, the clamping systems are positioned against the cargo (see Figure 4-2). During this phase the HTV and cargo are still moving relative to each other. Therefore impact loads will occur when the cargo makes contact with the clamping system. To keep this impact loads as low as possible, the stiffness of the system should be low during this phase.
3. In the third phase, the motions of the cargo are restricted in time. The stiffness of the system depends on the line stiffness and the stiffness associated with the damping devices. During this phase the amount of damping and stiffness the damping devices generate is increased. This causes the relative horizontal motions to reduce.
4. In the fourth phase, the cargo is completely clamped between the clamping structures (see Figure 4-3). The damping and stiffness properties of the system are very large in this phase. Therefore the amplitude of the relative horizontal motions is small and within the design requirements. It is possible to guide the external loads working on the cargo to the stiff construction through the damping devices. It is also possible to guide the loads directly to the stiff construction by completely compressing the damping devices. In this case, the compressed damping device should fit inside the stiff construction allowing the stiff construction and cargo to be in direct contact with each other.

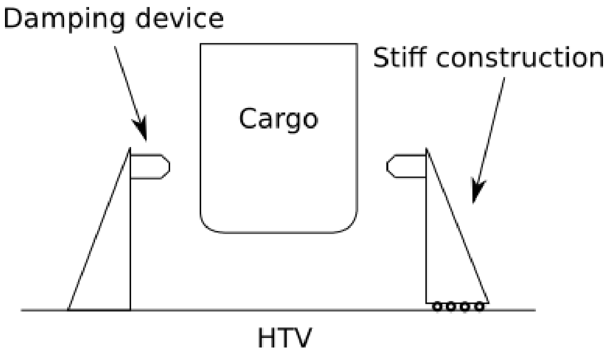

Figure 4-1 Clamping system phase 1

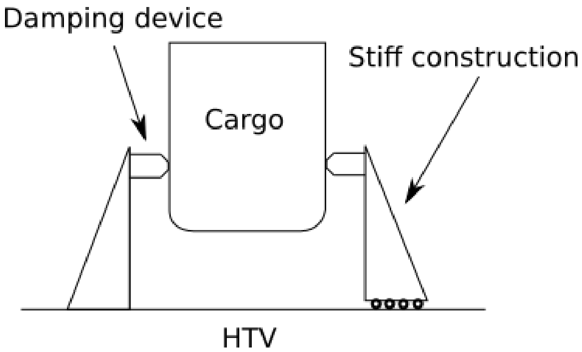

Figure 4-2 Clamping system phase 2+3

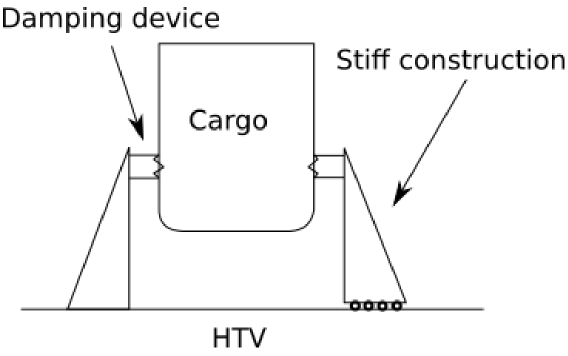

Figure 4-3 Clamping system phase 4

The clamping system is partly an active and partly a semi-active system. The clamping constructions, on which the damping devices are mounted, are active systems because external energy during the operation is needed. This energy is needed to push the structures against the cargo. The damping device is a semi-active system. The term semi-active means that the device requires a very small amount of external power [13]. This small amount of power is used to control the characteristics of the system. Of course the damping device needs to be activated once before use by a hydraulic system in order to set the right settings.

4.2 Description damping device

It is important to understand how the damping device works, before it is implemented in the clamping system. Therefore this section focuses on the principles of the damping device.

In Figure 4-4 a simplified overview of the used damping device is shown. The semi-active damper consists of a passive fluid damper in combination with an external control valve [13]. The device also possesses a fender and a pressure relief valve.

The numbers in Figure 4-4 relate to:

1. Chamber 1,
2. Chamber 2,
3. Piston head,
4. Piston rod,
5. Seal,
6. Control valve,
7. One-way valve,
8. Accumulator,
9. Fender,
10. Pressure relief valve.

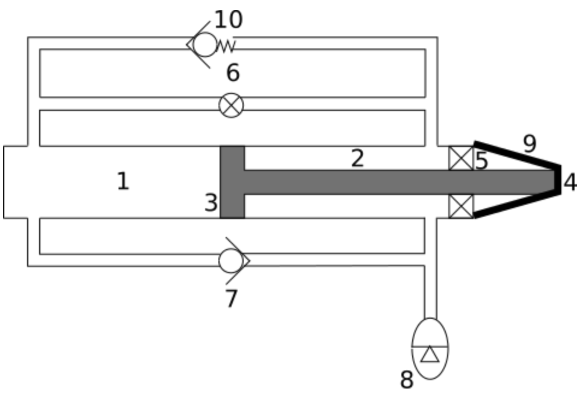

Figure 4-4 Simplified overview damping device

One side of the hydraulic cylinder is coupled to an accumulator; a closed reservoir for hydraulic fluid and gas (8). The accumulator is used to store the overflow of hydraulic fluid which occurs when the piston (4) heaves in. If the piston heaves in, the piston rod will take up some amount of space inside the cylinder. Normally this space is filled with the hydraulic fluid. Therefore less space is available for the hydraulic fluid. As the hydraulic fluid is nearly incompressible, the reduction in fluid volume will result in a restoring force. This is prevented by the use of an accumulator [14].

Different type of accumulators can be used: spring charged and gas charged. In a spring charged accumulator a spring is used to provide the restoring force of the accumulator. In a gas charged accumulator gas at a certain pre-load pressure used to provide this force [15]. If the pressure in chamber 2 (2) is higher than the pre-load pressure, fluid will flow into the accumulator while compressing the gas or spring. If the pressure in the accumulator is higher than the pressure in chamber 2, fluid will flow back into the system. Therefore the pressure of the fluid in the system and the accumulator will be equal during the operation.

The force generated by the damper is the result of a pressure differential across the piston head (3). A control valve (6) and an one way valve (7) are used to control this pressure differential. The generated force should be different for compression and tension forces. The cargo will only introduce compression forces in the damping device and no tension forces, because the cargo is not rigidly connected with the damping device; the cargo can only push against the device. If the cargo pushes against the device a damping force is required to reduce the motions of the cargo. The control valve is used to control the amount of damping the device will produce. However to get the piston back at its initial position, when the cargo moves away of the device, a pulling force needs to be introduced to the system. To keep this force as small as possible, the one way valve is used. In this way the hydraulic fluid in the device is allowed to flow freely from chamber 2 to chamber 1. The pressure difference in this direction is small thus a small force is required to move the piston back to its initial position. The force output of the device consists of a damping component and a stiffness component and is therefore not purely a damping force. However the largest part of the force consists of damping and the purpose of the device is to reduce (damp) the relative horizontal motions. It is therefore decided to call the system a damping device which produces a damping force, although it is not purely a damping force.

The fender (9) is connected to the end of the piston rod on one side and to the cylinder housing on the other side. A fender can be modelled as a spring, because a fender will always return to its original shape. The fender has two functions in the system. During the second phase of the clamp operation, in which the HTV and cargo are still moving, the fender is used to absorb the impact loads. To keep these loads small, the stiffness of the fender should be low. During the third phase of the operation, in which the control valve is closed, the fender is used to pull the piston back to its outer position.

The pressure relief valve (10) is used to limit the occurring pressure in the cylinder. The maximum pressure within a hydraulic cylinder is typically 350 bar. The majority of the hydraulic cylinders have a working pressure below 350 bar. Especially during the first phase of the operation it is possible that the occurring pressure in the cylinder is larger than 350 bar (1 bar is 100,000 [Pa]). During this phase the cargo has space to move in and can build up a certain velocity at which it hits the damping device. The output force of the device and thus the pressures inside the cylinder are dependent on the velocity of the piston rod. This can cause the pressure inside the device to become too high. The pressure relief valve will prevent this by letting the fluid flow from chamber 1 to chamber 2.

It is possible to equip the system with an electronic transmitter, such that the information about the system behaviour can be passed to a remote location where it can be monitored. Detectors connected to the transmitter can measure the pressure in the cylinder and the velocity and/or the position of the piston.

4.3 Description how basic damping device works

It is important to understand the basic physics of the damping device, before more complex performance simulations are performed. Therefore this section will describe the physics of a basic damping device. The basic damping device consists of the same elements as the damping device of the previous section, however without the fender and the pressure relief valve.

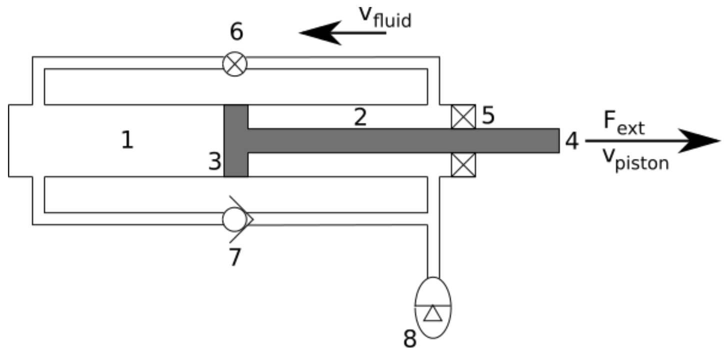

Figure 4-5 Simplified overview basic damping device

The characteristics of the basic damping device can be modelled in two ways: mathematically by creating a MATLAB model according to the fluid mechanics formulas or by creating a physical model in Simulink. The MATLAB model will be explained in the following section, for the Simulink model see Appendix D.3.

Fluid mechanics based model

The principles of fluid mechanics can be used to develop a simplified model to describe the dynamic behaviour of the basic damping device. A schematic overview of this device is shown in Figure 4-5. Taking the boundary deformations and the fluid compressibility into account and assuming a constant fluid density, the following mass flow rate continuity equation is found for the chambers [13]:

$$\frac{dV}{dt} + \frac{V}{\beta}\frac{dP}{dt} = Q \tag{4-1}$$

The used variables are:

P	=	Pressure within fluid	[Pa]
Q	=	Flow rate into the chamber	[m^3/s]
t	=	Time	[s]
V	=	Volume	[m^3]
β	=	Bulk modulus	[Pa]
ρ	=	Density fluid	[kg/m^3]

The damping device displayed in Figure 4-5 performs different for a positive and negative velocity of the piston. If the piston moves with a positive velocity, forcing the fluid to flow from chamber 2 to chamber 1, the fluid is allowed to travel through the control valve and the one way valve. However if the piston has a negative velocity, forcing the fluid to flow from chamber 1 to chamber 2, the fluid is only allowed to travel through the control valve. Therefore the fluid flow in the cylinder and thus the system characteristics will be different for the two velocity directions.

From Bernoulli's equation follows that the flow rate of the fluid through the control valve is equal to equation (4-2) [15]. This equation utilizes the conservation of energy, mass flow rate continuity, and an incompressible, inviscid fluid is assumed. In an inviscid fluid the pressure forces on the particles dominate over the viscous forces [14].

$$Q = k_d A_v \sqrt{\frac{2\Delta P}{\rho}} \tag{4-2}$$

The introduced variables are:

A_v	=	Valve area	[m^2]
k_d	=	Discharge coefficient of valve	[-]

With the current system configuration (see Figure 4-5), a negative piston velocity and only considering the effect of the control valve, the mass conservation equations become:

$$\frac{dV_1}{dt} + \frac{V_1}{\beta_1}\frac{dP_1}{dt} = k_d A_v \sqrt{\frac{2|P_1 - P_2|}{\rho}} sign(P_2 - P_1) \tag{4-3}$$

$$\frac{dV_2}{dt} + \frac{V_2}{\beta_2}\frac{dP_2}{dt} = -k_d A_v \sqrt{\frac{2|P_1 - P_2|}{\rho}} sign(P_2 - P_1) \tag{4-4}$$

The sign function indicates the sign of the real number which is enclosed in the function, in the above equations. The function is used because it is impossible to extract the square of a negative value, in this way it is possible to get the correct sign of the velocity dependent parts in the equations.

For sake of simplicity, the accumulator is modelled as a spring which is connected to an accumulator face plate. The spring stiffness and the face plate area can be adjusted to properly account for the accumulator behaviour.

Combining the equations leads to the following two first order nonlinear differential equations, for the negative piston velocity part. If v < 0:

$$\frac{dP_1}{dt} = (-\dot{x}A_p + k_{d\_cv}A_{cv}\sqrt{\frac{2|P_1 - P_2|}{\rho}} sign(P_2 - P_1))\frac{\beta}{(L_1 + x) \times A_p} \tag{4-5}$$

$$\frac{dP_2}{dt} = \frac{\dot{x}(A_p - A_r) - k_{d\_cv}A_{cv}\sqrt{\frac{2|P_1 - P_2|}{\rho}} sign(P_2 - P_1)}{\frac{(A_p - A_r)^2}{K_a} + (L_2 - x + \frac{P_2 A_f}{K_a})\frac{A_p - A_r}{\beta}} \tag{4-6}$$

The first order nonlinear differential equations for the positive piston velocity part of the solution can be derived using the same method as explained for the negative piston velocity part. If v ≥ 0 the equations become:

$$\begin{aligned}\frac{dP_1}{dt} = (-\dot{x}A_p + k_{d\_cv}A_{cv}&\sqrt{\frac{2|P_1 - P_2|}{\rho}} sign(P_2 - P_1) \\ &+ k_{d\_owv}A_{owv}\sqrt{\frac{2|P_1 - P_2|}{\rho}} sign(P_2 - P_1))\frac{\beta}{(L_1 + x) \times A_p}\end{aligned} \tag{4-7}$$

$$\frac{dP_2}{dt} = (\dot{x}\left(A_p - A_r\right) - k_{d\_cv}A_{cv}\sqrt{\frac{2|P_1 - P_2|}{\rho}}sign(P_2 - P_1)$$
$$+ k_{d\_owv}A_{owv}\sqrt{\frac{2|P_1 - P_2|}{\rho}}sign(P_2 - P_1))/(\frac{\left(A_p - A_r\right)^2}{K_a} + (L_2 - x \tag{4-8}$$
$$+ \frac{P_2 A_f}{K_a})\frac{A_p - A_r}{\beta})$$

The introduced variables are:

A_p	=	Piston head area	[m^2]
A_r	=	Piston rod area	[m^2]
A_{cv}	=	Valve area of control valve	[m^2]
A_{owv}	=	Valve area of one way valve	[m^2]
k_{d_cv}	=	Discharge coefficient of control valve	[-]
k_{d_owv}	=	Discharge coefficient of one way valve	[-]
L_1	=	Length chamber 1	[m]
L_2	=	Length chamber 2	[m]
x	=	Piston displacement	[m]
$\dot{x}$	=	Piston velocity	[m/s]

The total force of the damping device is a result of the pressure difference across the piston head:

$$F_d = P_1 A_p - P_2(A_p - A_r) \tag{4-9}$$

For the complete derivation of the presented differential equation see Appendix D.2. The differential equations of the damping device will be combined with the equations of motion of the HTV-cargo system. The motions of the HTV-cargo system are dependent on the stiffness and damping characteristics of the damping device while the amount of damping and stiffness force of the damping device is dependent on the motions of the system.

Basic damper characteristics

The performance of the basic damping device can be captured in a force versus displacement diagram, which shows the characteristics of the system (see Figure 4-6). In this characteristic, the area of the control valve is assumed to be constant in time. The dimensions of the damping device will be discussed in Appendix D.4. In order to obtain the characteristics of the system the piston is subjected to a forced sinusoidal motion with an amplitude of 0.15 [m] and a frequency of 0.63 [rad/s].

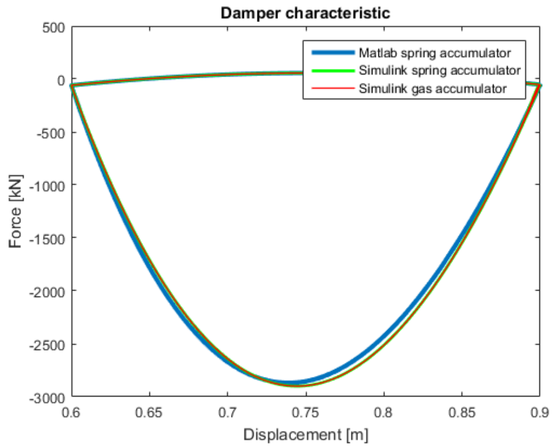

Figure 4-6 Basic damping device characteristic in Force-displacement plot

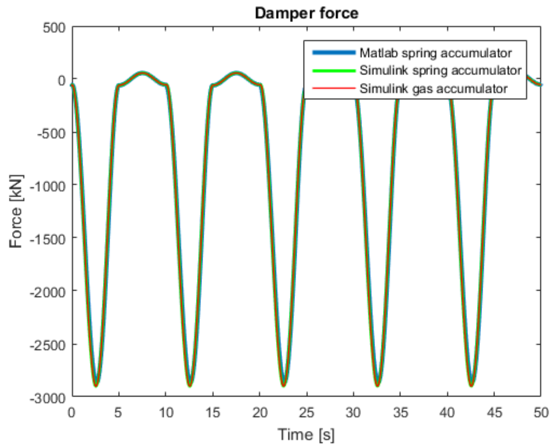

Figure 4-7 Damping force of the basic damping device

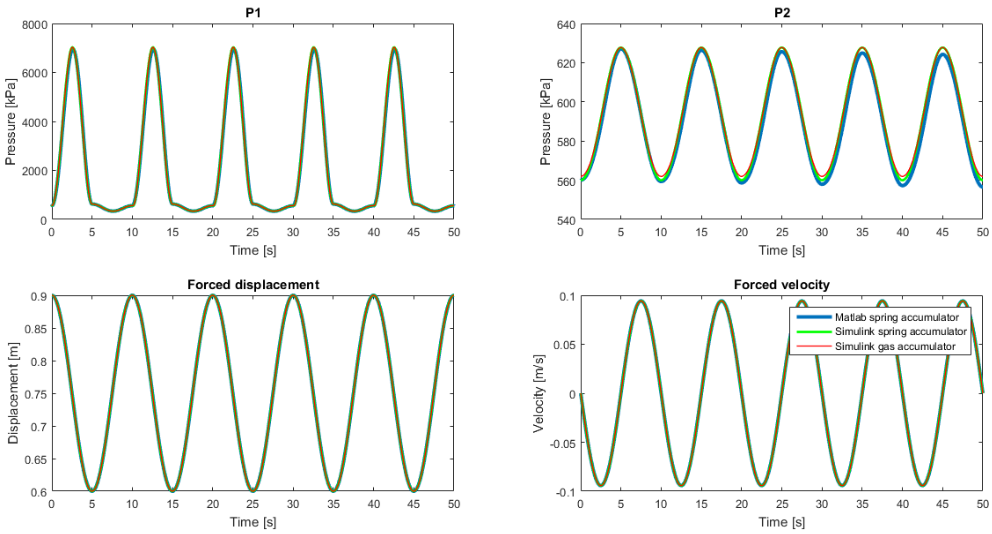

Figure 4-8 Pressures inside the cylinder chambers and piston displacement and velocity of the basic damping device

It can be seen in Figure 4-6 and Figure 4-7 that the curve of the force for the ingoing motion is different from the curve for the outgoing motion of the piston. The system actually shows two operating phases:

1. Phase 1: Piston moves into the cylinder (bottom part of the figures),
2. Phase 2: Piston moves out of the cylinder (top part of the figures).

In phase 1, the piston is pushed back into the cylinder. The fluid is only allowed to flow from chamber 1 to chamber 2 through the control valve. The flow rate through the control valve is dependent on the pressure difference between chambers 1 and 2 (P1-P2 in Figure 4-8). The control valve area is relative small which results in a large pressure difference between the two chambers (see equation (4-2)), causing the output force of the damper to be large.

In phase 2, the piston is pulled out of the cylinder. In this case the fluid is allowed to flow from chamber 2 to chamber 1 through the control valve and the one way valve. This causes the available flow area for the fluid to be larger than during phase 1 and this results in a smaller output force of the damper.

The subplots in Figure 4-8 show the internal pressures in chamber 1 (P1) and chamber 2 (P2) and the induced motion and velocity of the piston. It clearly shows that the pressure differential between chamber 1 and chamber 2 is governed by the pressure in chamber 1. This is logical because chamber 2 is connected to the accumulator which takes up or gives back fluid to chamber 2. In this way the pressure fluctuation in chamber 2 is reduced. On the other hand chamber 1 is not directly coupled to an accumulator; the fluid can only reach the accumulator if it passes the control valve. The compression and decompression of the fluid can give large fluctuations of the pressure inside the chamber.

It is also noted in Figure 4-6 that the damper force is not equal to zero when the piston velocity is zero, this is caused by the accumulator. The accumulator accounts for the fluid volume displaced by the piston rod and helps to prevent cavitation in the system during normal operation. Cavitation occurs when the pressure in the system is below the vapour pressure of the fluid. The accumulator prevents the pressure in the system to go below this pressure limit. The accumulator acts like a spring in the device, seen from a phenomenological perspective and produces an offset in the damper force [16]. The added stiffness causes the system to behave viscoelastic. This explains why the shape of the damper characteristic is not symmetrical. However the amount of stiffness added to the system is small compared to the amount of damping within the system. Therefore the influence of the stiffness is small.

The Simscape model will be used to verify the formulas used in the fluid mechanics model. The Simscape model can be used as a verification tool because Simscape is an independent environment in which the manner of connecting the components determines the algorithm of relations. In this way the user is not allowed to interfere with the relations involved between the components but only with their connections. The user only provides the input for the components. The fluid mechanics model is also used to verify the Simscape model, because it is difficult to see if an error is made within the model. The components can be seen as black boxes and the user is not able to see the calculations which are made within the Simscape model.

The figures show that the MATLAB and Simulink model for a spring charged accumulator produce almost the same results. However it is noticed that the Simscape model is lagging behind on the MATLAB function. This is caused by the simplification of the compressibility of the fluid (bulk modulus) in the MATLAB model. The bulk modulus is a measure of the ability of a substance to withstand changes in volume when it is under pressure. Therefore the bulk modulus is dependent on the pressure and volume (see equation (D-9)). The bulk modulus can be seen as the stiffness of the fluid; the bulk modulus ads stiffness to the system [17]. However for simplicity the bulk modulus is taken as a constant in the MATLAB model while in the Simscape model it is dependent on the volume and pressure. Therefore a small difference in output values occurs between the models. The difference is small because the bulk modulus of oil is already very high causing the effect of a changing pressure and volume to be relatively small.

The effect of the used accumulator on the characteristic behaviour of the system is also investigated, with Simulink. The major difference between a spring charged accumulator and a gas charged accumulator is the method in which the compressive force is obtained. In a spring charged accumulator linear springs are used, causing the system to behave linearly. In a gas charged accumulator gas is used, causing the system to react nonlinear. However it is concluded from the figures that the type of accumulator which is used does not change the characteristic of the system much. This effect is mainly caused by the fact that the pressure fluctuation in the accumulator is relatively small (seeFigure 4-8). As gas charged accumulators are used more often for these kinds of systems, it is decided to implement a gas charged accumulator in the system.

The simulations for the to be used damping device will be done in Simscape, because it is easier to add different components to the device in the Simulink model than in the MATLAB model. Another advantage of using Simscape is that the program uses pre-programmed equations linking the different hydraulic, pneumatic and control differential equations with each other. The above analysis showed that the MATLAB and Simscape models produce the same results.

4.4 Parametric study

A parametric study is conducted to investigate the effects of the pre-load pressure within the system and the fender stiffness on the behaviour of the damping device. It is also investigated what the effect of the changing control valve area is on the characteristics of the relative horizontal motion reduction system.

Pre-load pressure

The total volume of the accumulator is taken as a fixed value and has a capacity which equal to the fluid volume inside the hydraulic cylinder. However, the accumulator influences the output force of the damping device, as could be seen in Figure 4-6. Therefore it is important to investigate the effects of a pre-load pressure inside the system.

The accumulator and chamber 2 of the cylinder are in direct contact with each other. Therefore the pressures in the accumulator and chamber 2 are equal to each other. The initial pressures inside chamber 1 and chamber2/accumulator will be varied to determine the influence of the pressure on the damping device. The initial pressures inside the chambers can be calculated according to Pascal's law [15]:

$$P_1 A_1 = P_2 A_2 \tag{4-10}$$

It is assumed that all the valves, connecting the two chambers, are closed when the pre-pressure is applied to the damping device. In this way the fluid flow between the chambers, caused by the pressure difference between the chambers, is restricted. The resulting forces working on the piston head should be equal such that an equilibrium occurs. The piston head will keep this equilibrium position as long as there is no fluid flow. Therefore equation (4-10) can be used to calculate the initial pressures in the damping device.

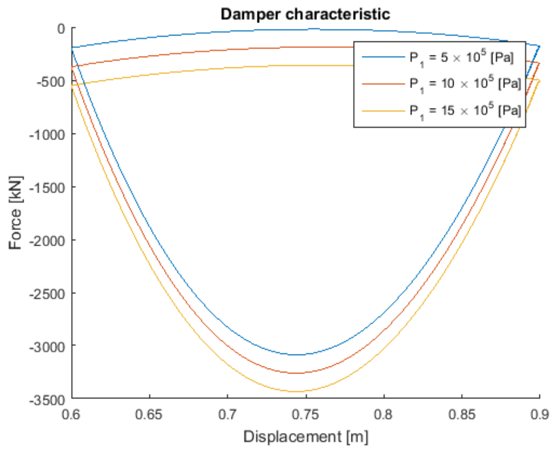

Figure 4-9 Force-displacement plot: influence initial pressure in system

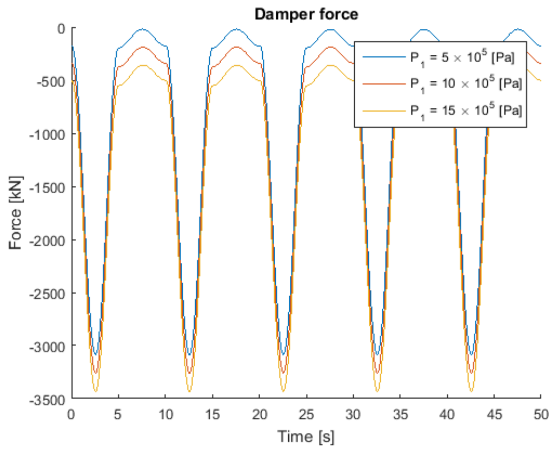

Figure 4-10 Force plot: influence initial pressure in system

The system characteristics are obtained by subjecting the piston to a forced sinusoidal motion with an amplitude of 0.15 [m] and a frequency of 0.63 [rad/s]. The control valve diameter is fixed at 0.025 [m].

Figure 4-9 and Figure 4-10 show that there is a relation between the pre-pressure within the damping device and the value of the force output of the damper. The higher the pre-pressure the larger the force output of the system is. This is caused by the fact that the fluid is allowed to flow from chamber 2 to chamber 1 (and vice versa); the system tries to reduce the pressure difference between the two chambers. The side effect is that the forces working on the piston head are not in equilibrium anymore. The resulting force of chamber 1 will be larger than the resulting force of chamber 2. Therefore the fluid will try to push the piston out of the cylinder, if no external force is working on the system.

Figure 4-9 also shows that the shape of the damper characteristic stays more or less the same for the different pre-pressure values. The amount of work performed by the damping device is dependent on the area within this envelope. Therefore the damping coefficient of the system will not change much. The amount of pre-pressure has an effect on the stiffness of the damping device. The higher the pre-pressure, the larger the stiffness of the device is. However this effect is relative small compared to the effect of the fender stiffness.

The major downside of the pre-pressure is that the piston will be pushed out of the cylinder. It is possible that the piston is pushed out faster than the cargo is moving away of the system and in this way provides extra energy to the motions of the cargo. This is an undesired effect and it is therefore concluded that the pre-pressure within the system should be low enough to prevent the piston from moving back to its outer position too quickly and the pre-pressure should be high enough the prevent cavitation in the system. For this reason the pre-pressure in chamber 1 is set at 5 bar.

Fender

The stiffness of the fender is dependent on the geometry and height of the fender. The stiffness is also highly dependent on the characteristics of the rubber of which the fender is made. Rubber is made up of long chains, which have a degree of cross-linking with their neighbouring chains. The degree of cross-linking determines the elasticity and therefore the stiffness of the fender. The cross linking bonds pull the elastomer back into shape when the deforming force is removed. Therefore the cross linking bonds effect the elasticity of rubber [18].

The function of the fender in the damping device is to pull the piston back to its initial position. However the speed at which the fender returns to its original shape should be slower than the speed at which the cargo is moving away of the damping device. Therefore it is important to investigate the effect, the stiffness of the fender has on the system. The system characteristics are obtained by using the same model as is used to investigate the pre-pressure effects. The pre-pressure within the system will be set at 5 bar.

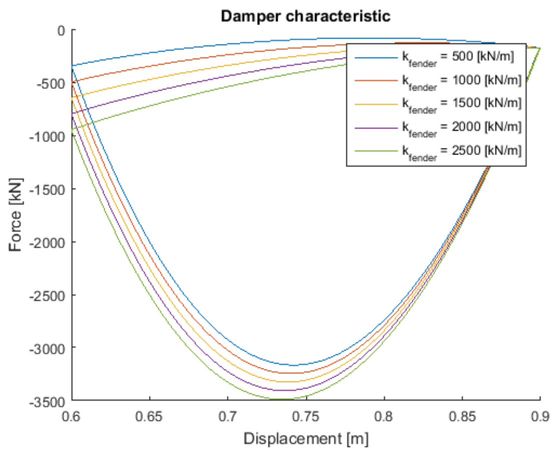

Figure 4-11 Force-displacement plot: influence fender stiffness on system

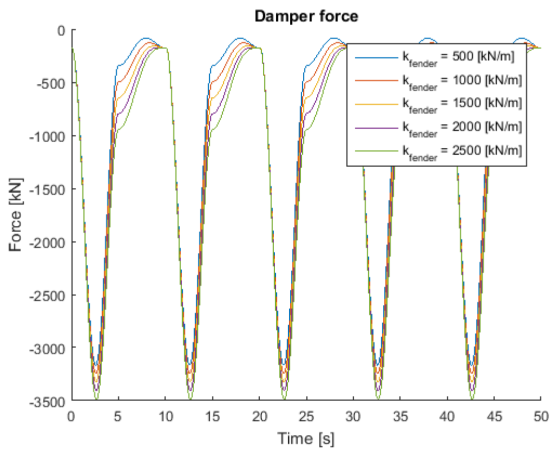

Figure 4-12 Force plot: influence fender stiffness on system

Figure 4-11 and Figure 4-12 show that there is a relation between the stiffness of the fender and the value of the output force of the damper. The fender adds an elastic component to the damping device. The value of the elastic component is dependent on the stiffness and deflection of the fender. The fender is connected to the end of the piston rod and the damper housing; the fender deflection is dependent on the displacement of the piston rod, compared to its initial position. The force which is needed to pull the piston rod back to its initial position is relatively small. This is also caused by the fact that the pre-pressure in the cylinder wants to push the piston rod back to its initial position. To keep the pushing force of the fender low, the stiffness of the fender should be low. Furthermore a large increase in the total system stiffness should be avoided during the second phase of the clamping operation (see Chapter 4.1). Therefore the fender stiffness should be as low as possible.

The height of the fender is dependent on the stroke of the damping device and the initial motions of the cargo before the cargo is connected with the clamping structures. The stroke of the damping device is 1.0 [m], thus the height of the fender should at least be 1.0 [m]. The stiffness of the fender is dependent on its reaction force and its deflection (see Figure 4-14). The figure also shows that the fender behaves non-linear. It is possible to model the fender as a linear spring as long as the deflection of the fender stays below 35 percent. The maximum amplitude of the motions is 0.4 [m] at the start of the loading operation (see Chapter 2.2). Therefore the height of the fender (h_f) should be at least 1.14 [m].

The type of fender that will be applied to this system is a cone fender, because the cone fenders have an excellent life-stability in a harsh marine environment. To fulfil the height requirements, a cone fender with a height of 1.2 [m] will be used for this design case. The fender which produces the lowest possible stiffness is the SCN1200 F0.6 [19]. The reaction force of this fender is 733.9 [kN]. Therefore the stiffness of this fender is 1750 [kN/m].

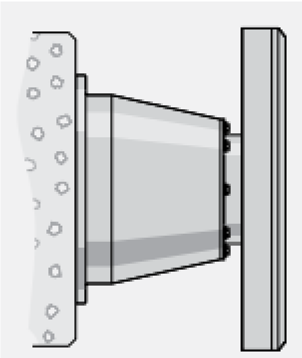

Figure 4-13 Cone fender [19]

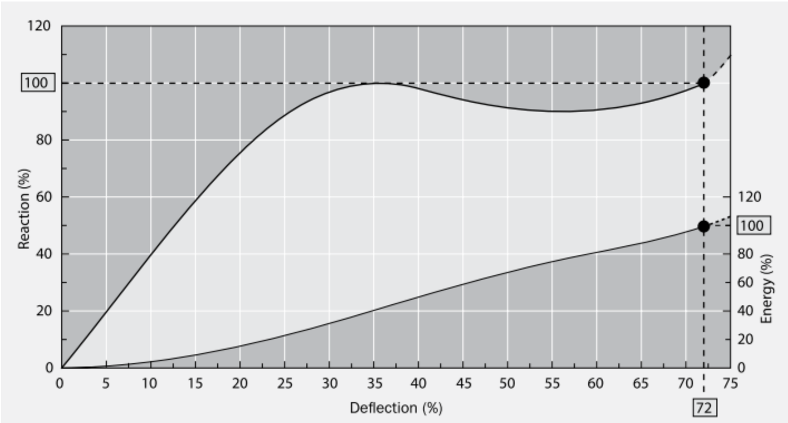

Figure 4-14 Generic curve cone fender [19]

Influence diameter control valve

The amplitude of the relative horizontal motions of the HTV-cargo system is governed by the stiffness and damping coefficients of the cargo handling system. The stiffness of the cargo handling system is composed of the stiffness of the subsystems, the line stiffness' and the damping device stiffness'. The damping coefficient is determined by the damping device.

The damping coefficient of the system is a result of the pressure difference between the two chambers of the cylinder. The magnitude of the pressure difference is dependent on the amount of fluid which can flow from one chamber to the other one. The fluid flow is controlled by the available flow area within the control valve. This area is governed by the diameter of the control valve.

The lines used to position the cargo in between the clamping structures have a linear stiffness. The lines form the standard cargo handling system which will be used to position the cargo in between the clamping systems. The standard cargo handling system will also be used during the loading or discharge operation of the cargo. It is best to keep the mooring lines attached to the HTV and the cargo, from an operational viewpoint. Therefore one part of the stiffness of the clamping system consists of the stiffness of the standard cargo handling system.

It is already known that the stiffness of the damping device is dependent on the pre-pressure within the system and the fender stiffness. These values are fixed. However it is unknown how the diameter of the control valve influences the stiffness of the system. The focus of this section will be on how the diameter of the control valve influences the damping and stiffness characteristics of the new relative horizontal motion reduction system. In order to perform this study, a model of the system is created in Simscape. The system will be subjected to a certain force in order to investigate the influence of the control valve diameter on the system characteristics.

The number of clamping systems applied in this simulation is 5 at each side of the cargo, 10 in total. However the damping device can only take up compression forces. Therefore only 5 systems will take up the motions of the cargo. The total force working on the cargo handling system is approximately 25,000 [kN] (2500 [t]) (see Chapter 2.4). It is assumed that the stiffness of the Dyneema lines is low compared to the stiffness of the clamping system. Therefore the amount of force which will be taken up by the Dyneema lines is negligible. Therefore the amount of force which needs to be taken up by one clamping system is approximately 5000 [kN] (500 [t]), for the static case. The capacity of the normal sea fastenings, which are used to secure the cargo during the voyage, is 200 [t] per sea fastening. The sea fastenings are positioned against the bulkheads of the cargo, as these are the strongest points in the cargo hull. However in some cases sea fastenings with a higher capacity are used. It depends on the characteristics of the cargo hull how much force the system can exert on the cargo hull. It is assumed that the hull of the FPSO MOHO Nord (the base case) can handle the force of 500 [t]. Therefore it is possible to use only 5 clamping systems. If the cargo hull is unable to handle these forces, more clamping systems need to be used. The amount of force per system will be lowered, the force working onto the cargo hull per system is also lower.

The effect of the control valve diameter on the characteristics of the system can be investigated, by looking at the responses of the system in two ways. At first the response of the system can be monitored when it is excited by a regular oscillating force with different excitation frequencies. This analysis provides insight in how the control valve influences the natural frequencies of the system. Secondly a free decay test can be performed. This test shows how the system returns to its equilibrium position after a disturbance is brought into the system. This test gives insight in the stiffness and damping characteristics of the system.

Response to a regular oscillating force

The response of a system to a regular oscillating force for different excitation frequencies can be obtained by two methods: analytically and numerically. In both methods the set of differential equations is solved for the particular solution. The particular solution describes the steady state motions of the system. The set of differential equations consists of the equations of motion for the system and the differential equations which describe the behaviour of the damping system. The particular solution consists of a formula in which the amplitude of the motions is calculated as a function of the excitation frequency.

If the analytical method is used the set of equations is solved in the frequency domain, the used equation will look like equation (B-3). However a frequency domain solution requires a linear or linearized set of equations. The damping device is a nonlinear system; therefore the system needs to be linearized before the frequency domain solution can be obtained. The equivalent stiffness and damping coefficients need to be obtained. The equivalent damping value is obtained by linearizing the nonlinear damping values around the mean motion amplitude. The equivalent stiffness value is obtained by linearizing the nonlinear stiffness values around an operating point.

If the numerical method is used the set of equations is solved in the time domain. In this case simulations will be performed with the same excitation force but for different frequencies. The simulations are performed long enough, so that the system responds in its steady state. Then the maximum amplitude is selected per frequency. These values are plotted and connected with each other in a graph. This is a more straightforward method because no assumptions have to be made in order to solve the problem. Therefore the numerical method will be used to obtain the responses of the system to a regular oscillating force.

Different simulations are performed to investigate the effects of the system characteristics. The simulations are performed for different control valve diameters, different excitation forces, different masses and different cylinder dimensions.

Figure 4-15 shows the response of the system for different control valve diameters. The mass of the cargo is set at 158,000 [t], equal to the actual mass and added mass of the FPSO MOHO Nord. The response of the system is dependent on the amplitude and frequency of the force working on the cargo. The amplitude of the force is set at 1000 [kN]. It is visible in the graph that the motions at the high frequencies are approximately the same, it is therefore decided to stop the simulation at a frequency of 1 [rad/s], for the remaining simulations.

Figure 4-16 shows the response of the system for different excitation forces. In order to see the difference in the response for the different excitation forces, the response is divided by the excitation forces. If the system would be linear, an increase in excitation force, would give a linear increase in the response amplitude and the amplitude over force ratio should be the same. The remaining simulations are shown in Appendix D.5.

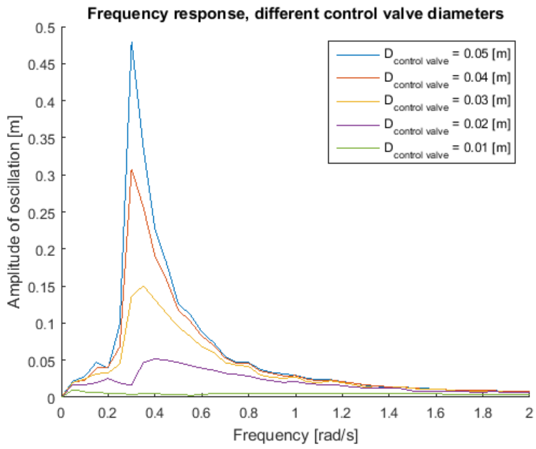

Figure 4-15 Frequency response, different control valve diameters

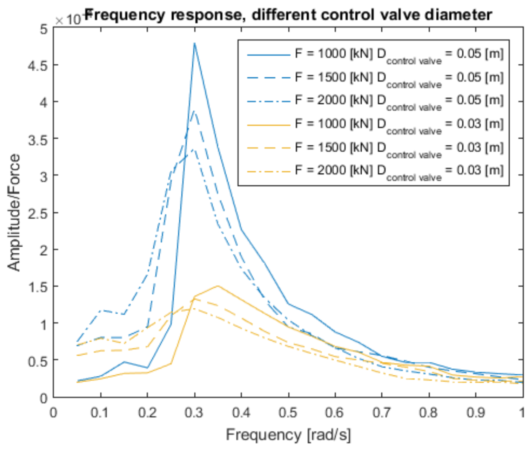

Figure 4-16 Frequency response, different masses

It is concluded that the stiffness and damping values of the system are dependent on the control valve diameter. The amount of damping is increased as the control valve is closed, causing the resonance motions to reduce. The amount of stiffness is also increased as the control valve is closed. This is visible due to the fact that the resonance peak shifts to the higher frequencies. This is clearly shown in Figure 4-15 as the amplitude of the oscillations reduces while the excitation force stays the same. It can be concluded from Figure 4-16 that the system is nonlinear. If the system is linear the responses of the different forces for the same control valve diameter would be the same, as the ratio amplitude over force would be constant. The amplitude of the motions is also dependent on the mass of the cargo. If the mass increases the natural frequency of the system becomes smaller, thus the natural frequency of the system is dependent of the mass of the cargo. This was already known from equation (B-5). It is also found that the piston head area has an effect on the motions of the system. If the dimensions of the piston are smaller, the stiffness and damping in the system are smaller. This is caused by the fact that the amount of damping which the system produces is dependent on the flow rate of the fluid. If the piston area is smaller, larger motions are needed to obtain the same amount of fluid flow. This causes the piston motions to become larger, increasing the cargo motions compared to a cylinder with larger dimensions.

Free decay test

A free decay test is conducted to see how the diameter of the control valve influences the motion behaviour of the HTV-cargo system. A free decay test clearly shows how the oscillations within the system decay, after a disturbance is brought into the system. The free decay test can be used to determine the damping coefficient and stiffness of the system. A free decay test can be used to obtain important design values relatively fast and to investigate the effect of these values on the system. However the system needs to be positioned out of its equilibrium position (excited) in order to analyse how it returns to its equilibrium. Therefore the system is given an initial position and is released of this position. Different simulations are conducted to see the influence of the control valve, cargo mass and cylinder dimensions.

Figure 4-18 shows the response of the system for different control valve diameters. The mass of the cargo is set at 158,000 [t], equal to the actual mass and added mass of the FPSO MOHO Nord. For the other simulations see Appendix D.5.

It is shown in the previous section that the damping device behaves nonlinear. This is caused by the compressibility of the hydraulic oil. Although liquids are almost incompressible, they possess some ability to be compressed. On top of this, the pressurized gas in the accumulator acts like a spring in the system. However the stiffness of the gas is non-linear. Therefore the damping and stiffness characteristics of the system are nonlinear. The equation of motion of the system during the test can be given as:

$$m\ddot{x} + c_1\dot{x} + c_2\dot{x}|\dot{x}| + kx = 0 \tag{4-11}$$

In this equation the following variables are used:

c_1	=	Linear damping coefficient	[Nm/s]
c_2	=	Quadratic damping coefficient	[Nm/s]
k	=	Stiffness of the system	[N/m]
m	=	The cargo + added mass	[kg]

The linear and quadratic damping coefficients can be found by solving the quadratic exponential decrement which envelopes the free decay motions. This can be done in the following way. At first the peak values are obtained. When the decrease of motion amplitude divided by the mean motion amplitude is plotted versus the mean motion amplitude (blue dots) Figure 4-17 is obtained. It is possible to fit a linear line ($y = a + bx$) through the blue dots (yellow line).

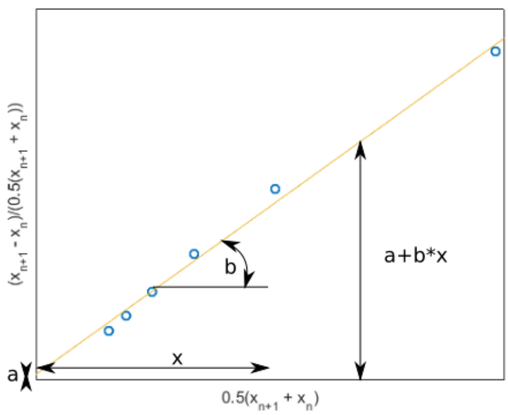

Figure 4-17 Exponential decrement

When the values *a* and *b* are known the damping coefficients can be calculated with the following equations:

$$c_1 = 2a\frac{m}{t} \tag{4-12}$$

$$c_2 = \frac{3}{8}bm \tag{4-13}$$

In a free decay test, the system is moving in its natural period (*t*). If the oscillation period is known it is possible to determine the equivalent stiffness with equation (4-14) [11]. In this way it is possible to gain insight in the behaviour of the equivalent stiffness over the control valve area.

$$t = \frac{2\pi}{\omega_n} \tag{4-14}$$

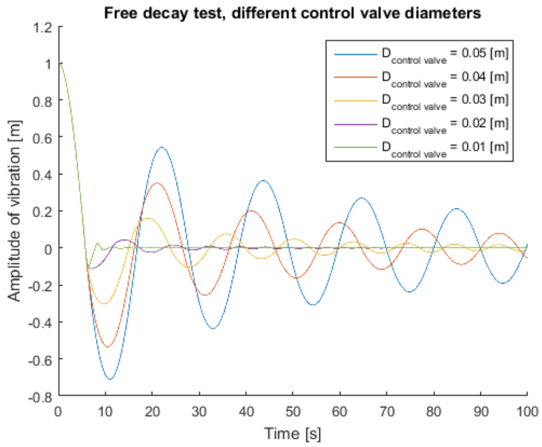

Figure 4-18 Free decay test, different control valve diameters

Table 4-1 Damping coefficient for different control valve diameters

$D_{control\ valve}$ [m]	c_1 [kNs/m]	c_2 [kNs/m]	k_{eq} [kN/m]
0.05	0.89×10^3	44.3×10^3	14.28×10^3
0.04	1.42×10^3	99.7×10^3	18.83×10^3
0.03	2.09×10^3	314.2×10^3	36.63×10^3
0.02	5.14×10^3	1429.3×10^3	128.29×10^3
0.01	18.54×10^3	5296.1×10^3	917.15×10^3

It is clearly visible in Figure 4-18 that the diameter of the valve has a large influence on the stiffness and damping coefficients of the system. This effect is also visible in Table 4-1, the values become larger as the valve diameter decreases. The simulations for the different mass and cylinder dimensions show the same characteristics as were already discussed in the previous section.

It can also be seen in Table 4-1 that the response of the system becomes more and more nonlinear as the valve diameter reduces. The value of c_1 increases at a slower rate than the value of c_2. The effect of the control valve area on the stiffness is the largest when the valve is almost closed. During this phase the fluid is compressed the most. The fluid flow through the control valve is dependent on the control valve area and the pressure difference between the chambers (see equation (4-2)). Due to this pressure the fluid is compressed a little bit and generates a stiffness coefficient. As the hydraulic fluid is almost incompressible, a small compression in volume can result in large forces i.e. a large increase in stiffness. The nonlinearities in the results are not only caused by the working principle of the damping device but also by the layout of the clamping system. The cargo is not constantly attached to the damping device. Therefore the damping and stiffness values of the system are not constant during the operation.

The cargo will experience the stiffness and damping provided by the damping devices when $x < 0$, $v < 0$ and $x > 0$, $v > 0$. In the other cases the stiffness and damping are generated by the standard cargo handling system. Therefore the equation of motion of the system has the form of equation (4-11), but the input values will differ during the operation. In this case no pre-compression of the fender is applied to the clamping system. If a pre-compression is applied the available space for the cargo to move in is reduced. Therefore the characteristics (damping and stiffness coefficients) of the system will change and consequently the response of the system. A pre-compression also reduces or avoids peak pressure loads when an impact occurs between the clamping systems and the cargo (see Chapter 4.5).

4.5 Performance damping device

In this section the performance of the system will be presented and evaluated. Since several simulations, for different environmental conditions, are performed to investigate the performance of the clamping system, not all the simulations will be discussed, as it would be too much. Therefore only the most interesting cases will be pointed out.

Simulation method

The motions of the HTV-cargo system will be simulated for the steady state case. To reach this state quickly, a ramp up function is used; the amplitude of the excitation force grows in time before it reaches its maximum value. During this phase the control valve will be completely opened; adding only a small amount of damping and stiffness to the system. At time 100 [s] the control valve will start to close and at time 700 [s] the control valve will be completely closed (see Figure 4-19). The start and end of this phase will be marked by a line in the simulation figures. From this moment on the cargo will be clamped by the system. As the fluid is not allowed to flow between the 2 chambers within the cylinder, the fluid stiffness will dominate the stiffness of the system. The damping devices which are located at opposite sides of the cargo and work together to reduce the motions of the system are referred to as one system.

The results of the simulations will be compared against the reference case: the MATLAB model of the standard cargo handling system. The simulations will be performed for the excitation force and frequency corresponding to the actual wave forces (the base case). For this scenario the magnitude of the motions will be in the same order of magnitude as the resulting motions found in the AQWA simulations. However also other scenarios will be considered. For these scenarios the excitation force and frequency are arbitrarily selected, in order to investigate the performance of the system in those cases.

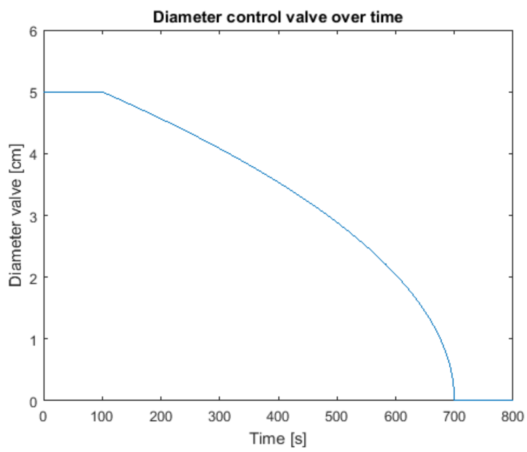

Figure 4-19 Control valve diameter over time

The FPSO Moho Nord will be used as the cargo. The natural frequency of the original mooring system is 0.09 [rad/s], the corresponding natural period is 73.8 [s]. The stiffness of the Dyneema lines is 1150 [kN/m].

Influence of pre-compression of the fender

During the loading operation the damping devices will be pushed against the cargo by the clamping systems. The damping devices can be positioned such that the fenders are touching the cargo without being compressed (at both sides of the cargo), or such that the fenders will be compressed (at both sides of the cargo) (see Figure 4-20).

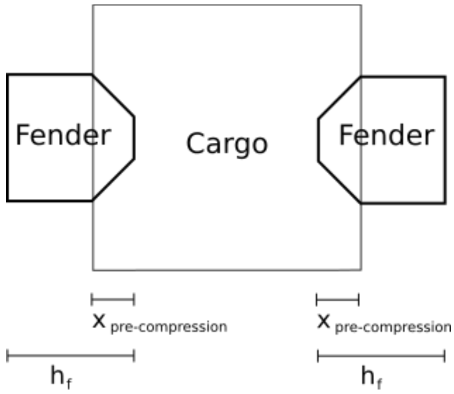

Figure 4-20 Schematic overview of pre-compression of the fender

Before the performance of the system can be compared with the reference case, the effect of the pre-compression of the fenders needs to be investigated. The base case conditions are used to simulate the influence of the pre-compression of the fender. The system has an excitation frequency of 0.63 [rad/s], an excitation force of 25,000 [kN] and 5 damping devices at each side of the cargo. The system will be simulated for the case without a pre-compression and with a pre-compression of 0.1 [m] of the fender.

The maximum response motions of the systems with and without pre-compression are presented in Figure 4-21. It clearly shows that the amplitude of the motions is smaller if a pre-compressed fender is used. This phenomenon can be explained with the help of Figure 4-22 in which the maximum response velocity of the system is presented. The motions and velocities are symmetrical around the x-axis, therefore only the positive directions are shown. The response velocity of the system is higher for the case without a pre-compressed fender. This is caused by the available space the cargo is allowed to move in. This space is smaller when the fenders are pre-compressed then when they are not pre-compressed. The cargo can absorb more energy from the waves in order to generate larger motions. The compressed fenders limit the free space of the cargo; if the cargo moves into the damping devices at one side, the systems at the other side can still, for a certain amount, heave out. Thus if the cargo starts moving to these systems, it will meet them earlier than the non-compressed systems, reducing the amount of energy the cargo can absorb from the waves. The occurring motion and velocity responses are presented in Appendix D.6.

Figure 4-23 and Figure 4-24 show the reaction force of 1 damping device against the piston displacement and the time. It is clearly shown that the forces of the damping device are lower as the velocity of the cargo is lower. The large loads are caused by the cargo hitting the damping device (impact loads). As long as the cargo gets free of the damping devices, impact loads will be present. The amplitude of these loads is dependent on the diameter of the control valve. The more closed the valve is, the more force it takes to move the piston, increasing the impact loads. It can be seen that the damper forces stay constant for the pre-compressed system for t > 600 [s]. From this moment on the cargo stays in contact with the damping devices causing the impact loads to disappear.

Figure 4-25 shows the pressures inside the cylinder. It is clearly shown that the impact loads induce large pressure differences between the chambers of the cylinder and cause the large reaction force of the damping device.

Summarising, the amount of pre-compression of the fender has a large impact on the forces within the system. The more compressed the fender is, the smaller the forces will be. However it is practically impossible to give the fenders large compressions at the start of the operation, because at this phase of the operation relative large motions occur, requiring very large fenders. On top of this, the more the fenders are compressed, the larger their pushing force against the cargo at the start of the operation becomes. The value of this force is also dependent on the stiffness of the fenders. Therefore soft fenders should be used. However if the fenders are too soft, it is possible that they cannot carry their own weight in their compressed shape. This induces the risk of buckling in the piston rod. It is therefore decided to use a pre-compression of 0.1 [m] for this thesis.

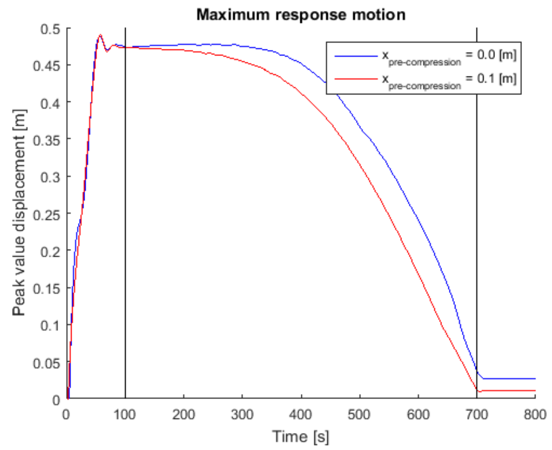

Figure 4-21 Maximum response motion, pre-compression fender

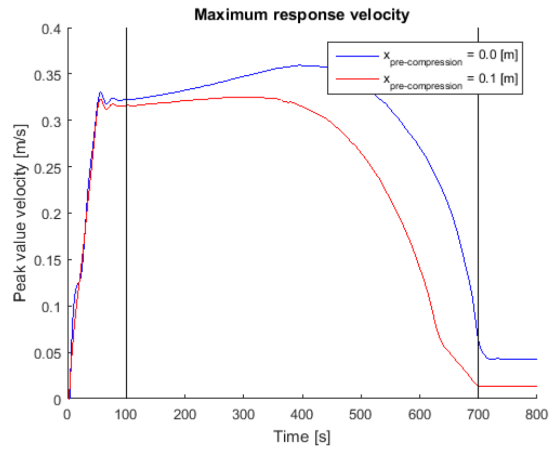

Figure 4-22 Maximum response velocity, pre-compression fender

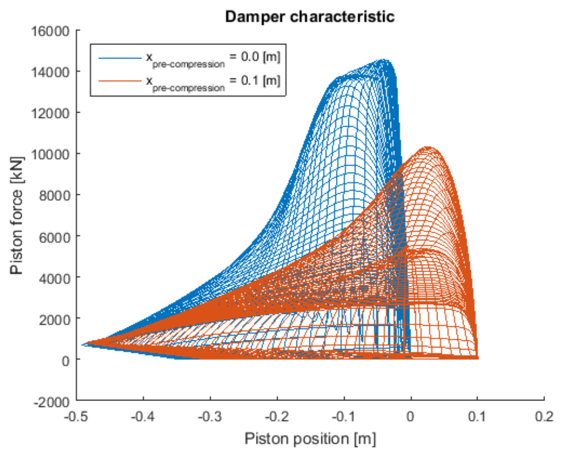

Figure 4-23 Damper characteristic, pre-compression fender

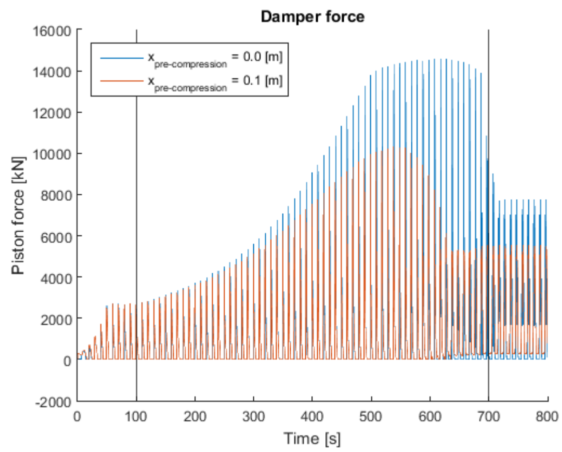

Figure 4-24 Damper force, pre-compression fender

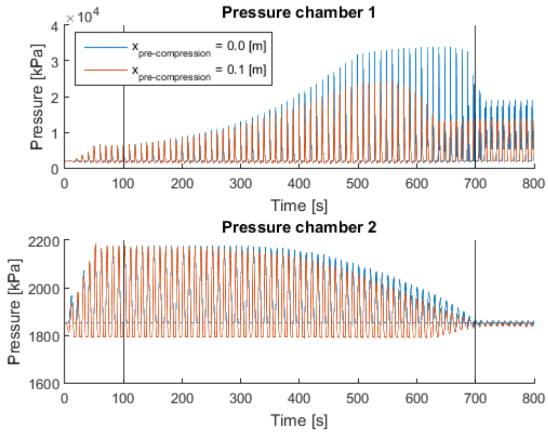

Figure 4-25 Pressure inside the cylinder chambers, pre-compression fender

Base case scenario

In this section a closer look is taken at the response motions for the base case scenario. It is also investigated what the effect of the number of devices and the relief pressure is on the performance of the system. The excitation frequency of the system is 0.63 [rad/s] and the excitation force is 25,000 [kN].

Influence number of devices

In this section, the influence of the number of damping devices on the resulting motions within the HTV-cargo system will be investigated. It will also be investigated how the number of damping devices will influence the output force of 1 damping device.

It is clearly visible that the response amplitudes increased, at the start of the loading operation, by applying the damping devices; the motions have grown with 18% in the case of 5 damping devices and with 23% in the case of 10 damping devices, compared to the present system (see Figure 4-26). This effect is also noticeable for the response velocity (Figure 4-27). However as the control valve within the damping devices is slowly closed over time, the resulting amplitudes of the motions of the cargo are also reduced. As the 10 damping devices are able to generate more damping than 5 systems the motions are reduced faster when 10 damping devices are used instead of 5. The motions of the system are larger at the start of the operation because the addition of the damping devices causes the system to have a higher natural frequency than the standard cargo handling system. This shift in natural frequency causes the response amplitudes in the higher frequency range to grow. This case evolves around the higher frequencies; in this case the response motions of the system are larger than for the present system.

The reaction force of 1 device is lower as more damping devices are used to reduce the motions of the system (see Figure 4-28 and Figure 4-29). The excitation force and the impact loads are divided over more systems, reducing the amount of force one system needs to absorb. It is also visible in the figures that the force output of 1 damper is 10,000 [kN] (1000 [t]), if 5 damping devices are used. This output force is considered to be too high, as it is assumed that the cargo hull can only take up 5000 [kN] (500 [t]). The force output of 1 damper is 5000 [kN] (500 [t]), if 10 damping devices are used. Therefore it is better to use 10 damping devices than 5.

The pressures inside the cylinder are much lower for the case in which 10 damping devices are used than for the case in which 5 damping devices are used (see Figure 4-30). This is mainly caused by the fact that the impact loads are divided over more systems, and lower the load per system.

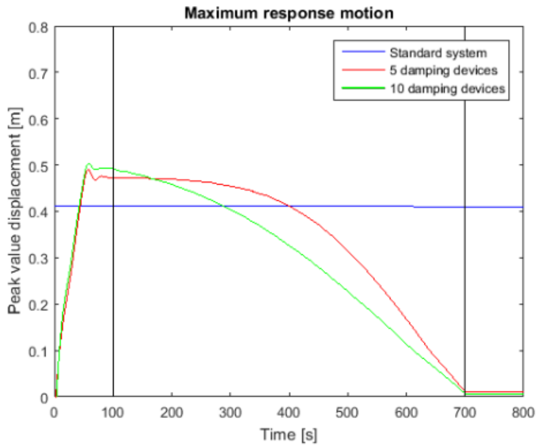

Figure 4-26 Maximum response motion, different amount of damping devices

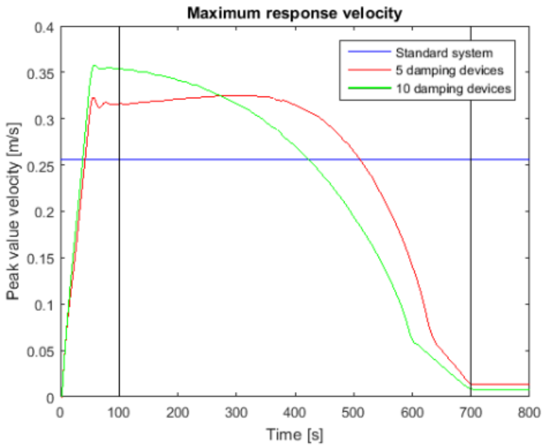

Figure 4-27 Maximum response velocity, different amount of damping devices

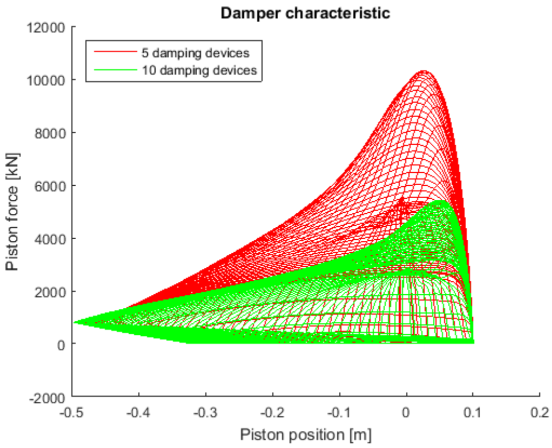

Figure 4-28 Damper characteristic, different amount of damping devices

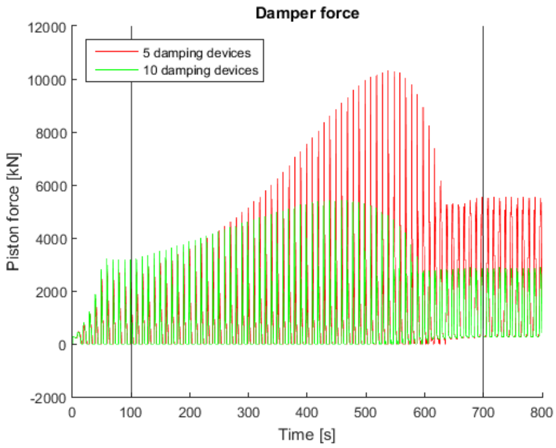

Figure 4-29 Damper force, different amount of damping devices

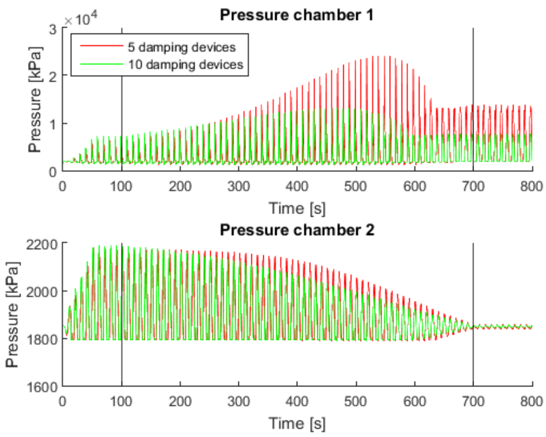

Figure 4-30 Pressure inside the cylinder chambers, different amount of damping devices

Influence relief pressure

In this section the influence of the relief pressure of the damping cylinders on the resulting motions within the HTV-cargo system will be investigated. For these simulations a total of 5 damping devices at each side of the cargo will be used.

Like for the case in the previous section (Influence number of devices), the response amplitudes increased, at the start of the operation, by applying the damping devices (see Figure 4-31). As the control valve is closed over time, the resulting amplitude of the motions is also reduced over time. It is noticed that it takes more time for the amplitude of the motions to decline if the relief pressure of the cylinder is set at a lower value. The response velocity is also larger for the case with the low relief pressure (see Figure 4-32). If the pressure relief valve opens to quickly the motions of the cargo will not be reduced (see Figure 4-31). Therefore the relief value should be above a certain minimum value.

Figure 4-33 to Figure 4-38 reveal how the set value of the pressure relief valve influences the motions in the system. If the pressure in a cylinder chamber becomes too high the pressure relief valve will open, lowering the pressure in the cylinder chamber. However, the force output of the damping device is governed by the pressure difference between the two chambers (see equation (4-9)). The following relation is found between the value of the pressure relief valve and the force output of the damping device; the lower the set value of the pressure relief valve is, the lower the damper force is. However, the motions of the system will not be reduced if the pressure at which the pressure relief valve opens is set too low. In this case the system is not able to generate enough damping. The impact load of the cargo hitting the damping device generates a pressure in the cylinder chamber which is higher than the pressure relief pressure. The pressure relief valve will be opened very quickly after the cargo makes contact with the damping device, causing the amplitude of the response motions to stay the same. This causes large impact loads to occur when the control valve is closed because the fluid can only travel through the pressure relief valve to the other chamber and a large force is necessary to open the relief valve. This problem can be solved by setting the relief pressure such that the pressure relief valve will not open that quickly. However this will still create impact loads because still a force is needed to open the valve. Although the motions will be reduced causing the impact loads also to be lower.

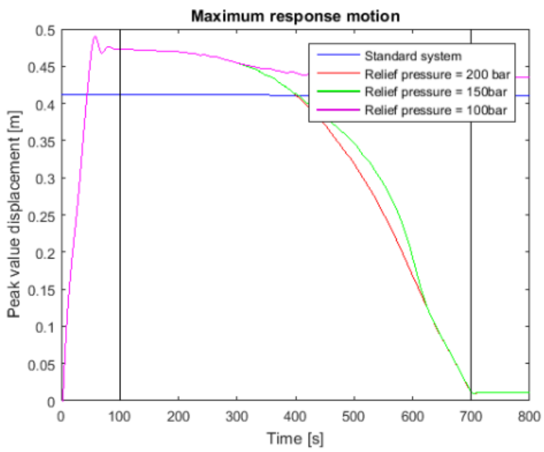

Figure 4-31 Maximum response motion, different relief pressures

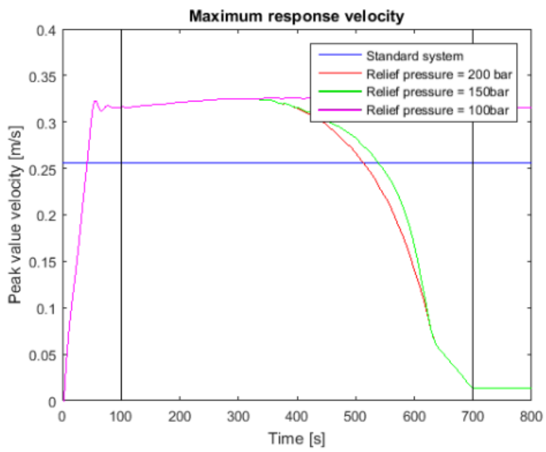

Figure 4-32 Maximum response velocity, different relief pressures

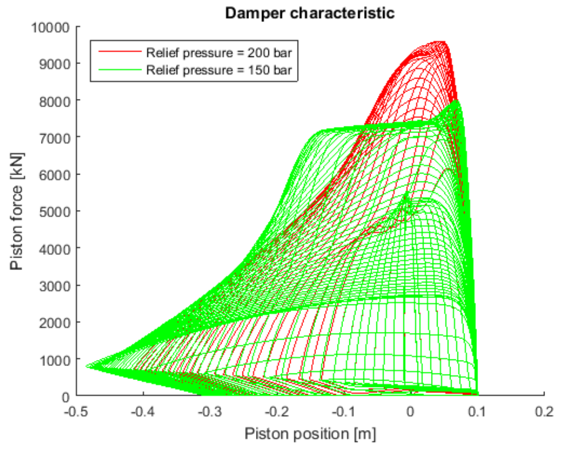

Figure 4-33 Damper characteristic, different relief pressures (200 bar, 150 bar)

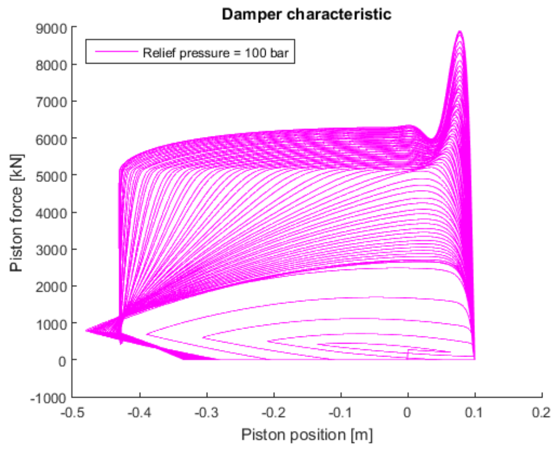

Figure 4-34 Damper characteristic, different relief pressures (100 bar)

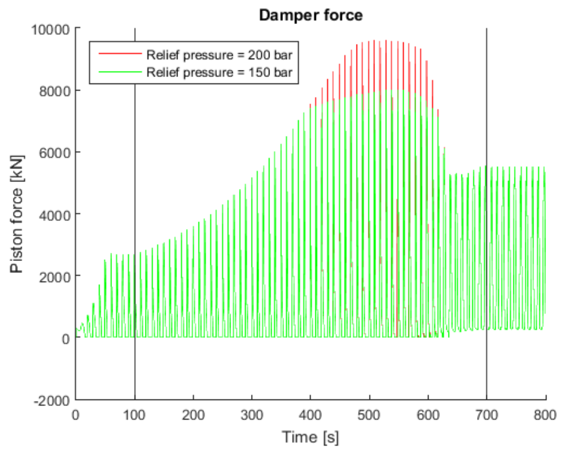

Figure 4-35 Damper force, different relief pressures (200 bar, 150 bar)

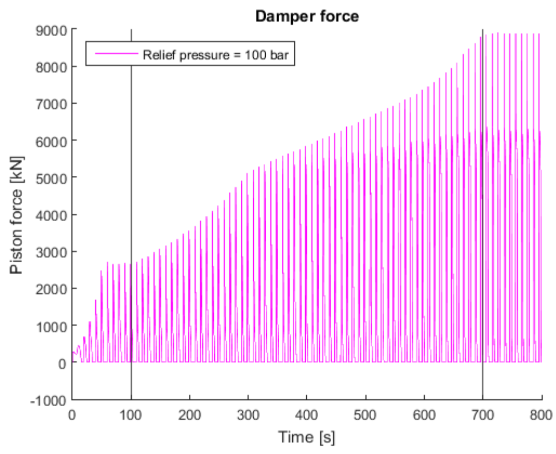

Figure 4-36 Damper force, different relief pressures (100 bar)

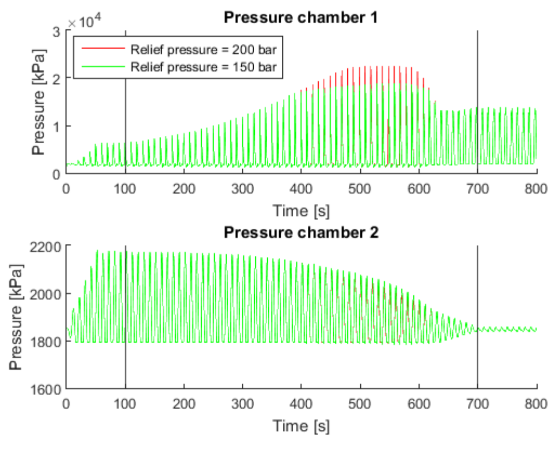

Figure 4-37 Pressure inside the cylinder chambers, different relief pressures (200 bar, 150 bar)

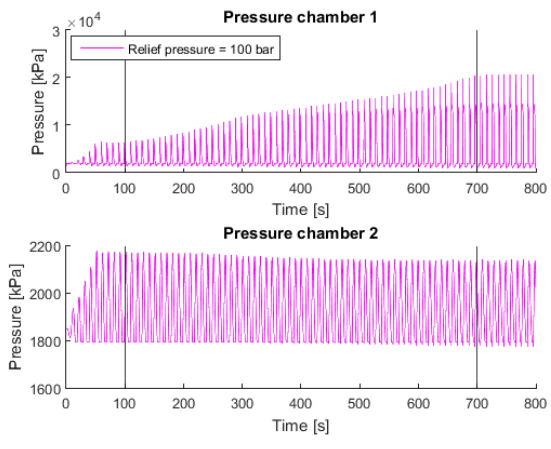

Figure 4-38 Pressure inside the cylinder chambers, different relief pressures (100 bar)

Conclusion

It is shown that for both cases the response motions of the system are reduced and fulfil the design requirements (see Chapter 2.6). However, it is also shown that the output forces of the damping devices need to be controlled such, that they do not damage the cargo hull. It is possible to control these forces by changing the number of damping devices which are used or by changing the value of the relief pressure. The output force of the damping device is influenced the most by the impact loads between the cargo and the damping devices. When the output forces of the damper for the two cases are compared to each other it is concluded that the output force is influenced the most by the amount of damping devices. This can be explained by the fact that the impact load is divided over a larger area when more systems are used, thus lowering the load per device. While if the pressure in the cylinder is controlled, the generated damping by the device is changed, influencing the response motions of the system. The less damping is generated the longer it takes to reduce the motions. However the use of more systems is more expensive then applying a lower relief pressure.

It is therefore concluded that (from an economical perspective) an optimisation has to be done between the relief pressure and the number of systems. This optimisation is dependent on the allowable load on the cargo hull.

Excitation frequency close to the natural frequency of the present system

In this section a closer look is taken at the response motions of the HTV-cargo system when the excitation frequency is close to the natural frequency of the standard cargo handling system. The excitation frequency of the system is 0.1 [rad/s] and the excitation force is 1000 [kN]. For this simulation a total of 5 damping devices at each side of the cargo will be used.

It is noticed that the motions of the standard system show the beating effect, caused by the excitation force being close to the natural frequency of the standard system (see Figure 4-39). The beating effect can also be noticed for the response velocity (see Figure 4-40). Beating occurs when the natural frequency of the system (ω_n) and the frequency of the excitation force (ω) become very close. It can be seen in Figure 4-39 that the oscillations 'beat' with a long period. This period is inverse proportional to the difference between ω and ω_n. The beating only occurs in a system without (which is unrealistic) or with a small amount of damping. The amplitude of this phenomenon is variable in time, as it builds up and dies down continuously [11]. The response motion of the system is reduced by applying the damping devices, the maximum amplitude of the motion when the systems are applied is 0.1 [m]. This is a reduction of 97%. Moreover the beats have disappeared.

The excitation frequency of the system is located at the left side of the natural frequency of the damping device. Therefore the motions of the system are driven by stiffness (see Chapter 2.5). The addition of the damping terms will not have a large effect on the response amplitudes of the system, except if a large amount of damping is added to the system. This is visible in Figure 4-41 and Figure 4-42 as the output forces of the damping device stay constant over the control valve diameter, until the control valve is almost closed. Therefore the motions of the system are reduced by the increase in stiffness and not by the added damping. In this case, it is possible to perform the loading operation without changing the diameter of the control valve.

It is clearly visible that the pressure fluctuations inside the chambers are small compared to the previous cases (see Figure 4-43). The amplitudes of these fluctuations are controlled by the velocity of the moving piston. The piston velocity is relative low, causing low pressure fluctuations, thus dissipating a low amount of energy of the motions.

Summarising, the damping devices are capable of reducing the response motions in the low frequency areas. However the reduction is caused by the stiffness of the devices and not by the generated damping.

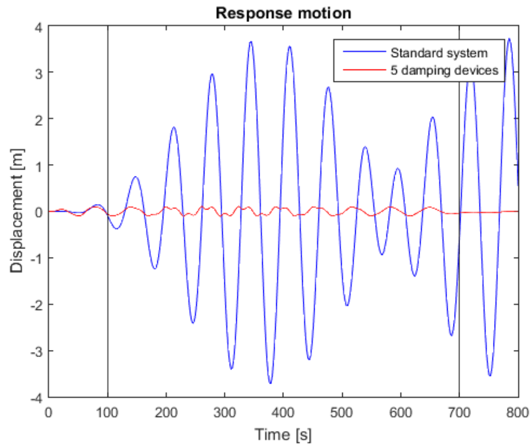

Figure 4-39 Response motion, excitation frequency close to the natural frequency of the standard system

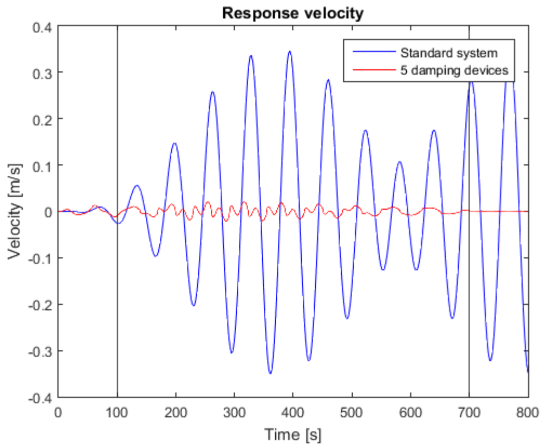

Figure 4-40 Response velocity, excitation frequency close to the natural frequency of the standard system

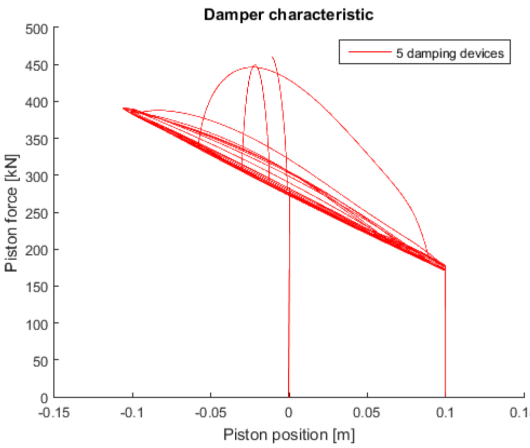

Figure 4-41 Damper characteristic, excitation frequency close to the natural frequency of the standard system

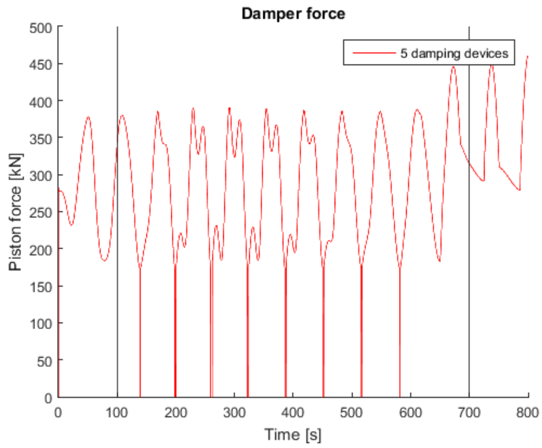

Figure 4-42 Damper force, excitation frequency close to the natural frequency of the standard system

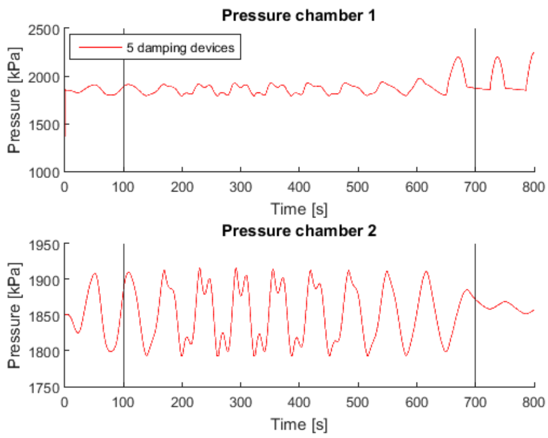

Figure 4-43 Pressure inside the cylinder chambers, excitation frequency close to the natural frequency of the standard system

Excitation frequency close to the natural frequency of the relative motion reduction system

In this section a closer look is taken at the response motions of the system when the excitation frequency is close to the natural frequency of the new relative horizontal motions reduction system. The excitation frequency is 0.3 [rad/s] and the excitation force is 5000 [kN]. For these simulations a total of 5 damping devices at each side of the cargo will be used.

It is noticed that, contrary to the previous case in which the excitation frequency was close to the natural frequency of the present system, the beating effect is absent (see Figure 4-44). This is caused by the damping which is added by the damping devices. The motions increased with 84% at the start of the operation. This effect is also noticeable for the response velocity (see Figure 4-45). However as the control valve is slowly closed over time, the resulting amplitude of the motions of the system are also reduced. Thus even if the system is excited close to its natural frequency, the system is capable of reducing the amplitude of the response in order to fulfil the design requirement of 0.15 [m].

In contrast to the previous cases in which the excitation frequency was located away from the natural frequency of the system, the amount of damping applied to the system immediately has an effect on the obtained response amplitude (see Figure 4-46 and Figure 4-47). It is noticeable that the pressures inside the cylinder behave the same as for the cases in which the excitation frequency was larger than the natural frequency (see Figure 4-48).

Summarising, the damping devices are capable of reducing the horizontal motions of the system, when the excitation frequency of the system is close to the natural frequency of the system. It is even possible to reduce the motions such that they fulfil the design criterion of x ≤ 0.15 [m].

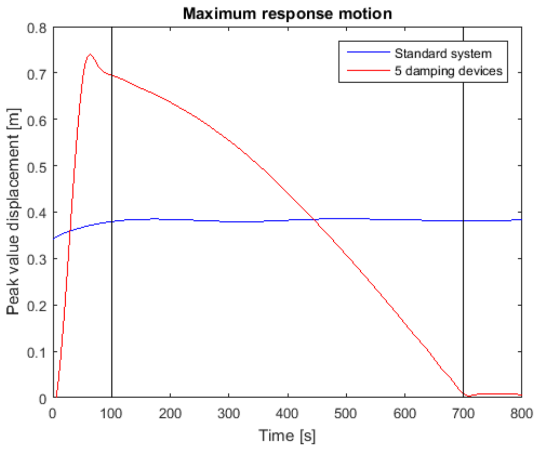

Figure 4-44 Maximum response motion, excitation frequency close to the natural frequency of the new cargo handling system

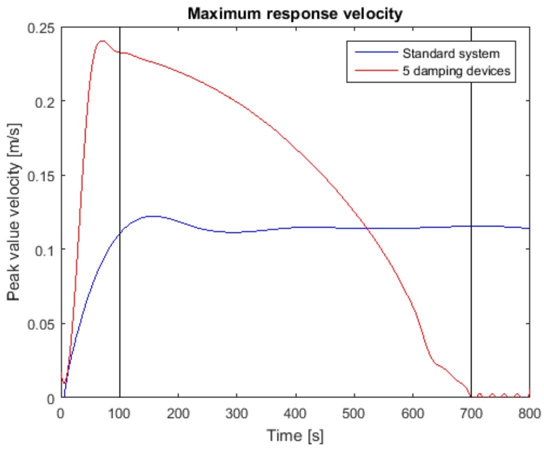

Figure 4-45 Maximum response velocity, excitation frequency close to the natural frequency of the new cargo handling system

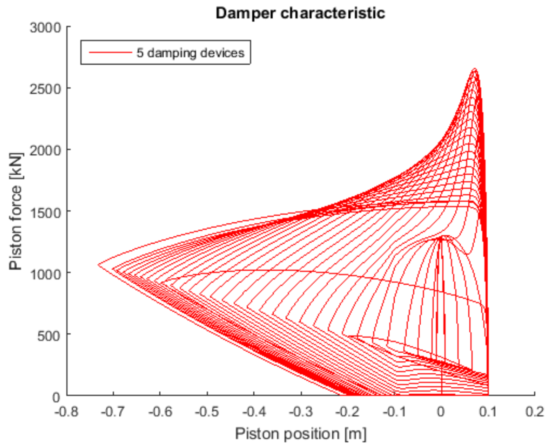

Figure 4-46 Damper characteristic, excitation frequency close to the natural frequency of the new cargo handling system

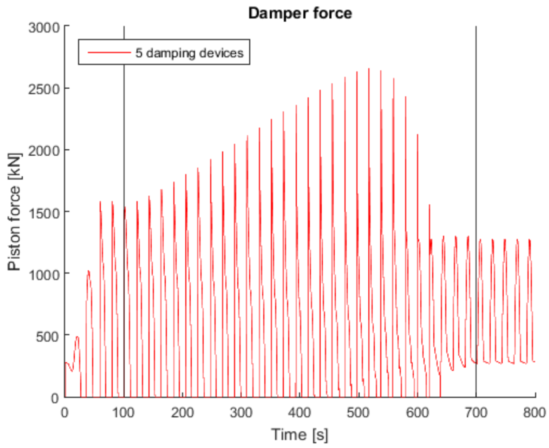

Figure 4-47 Damper force, excitation frequency close to the natural frequency of the new cargo handling system

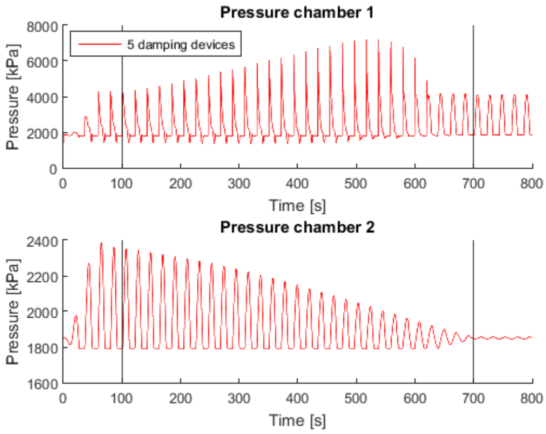

Figure 4-48 Pressure inside the cylinder chambers, excitation frequency close to the natural frequency of the new cargo handling system

Irregular external force signal

In the previous cases, the systems are excited by a regular wave force. However real waves are irregular and cannot be described by a single frequency and amplitude. Thus the actual wave forces on the vessel are also irregular. This section will focus on the responses of the system when the system is excited by an irregular wave force. Only one case will be discussed because the outcomes of the different cases are the same (for the other cases see Appendix D.5).

The irregular wave signal is modelled by the summation of two regular sine-waves, each with its own frequency and amplitude. In reality the wave force consists of a combination of more sine waves. However the purpose of these case studies is to investigate the behaviour of the system if it is excited by an irregular force. Moreover it is important to see if the transient response of the system does not influence the behaviour of the damping device. The transient response occurs when the excitation force is growing in value.

Case 1: $\omega_1 = 0.3$ [rad/s], $\omega_2 = 0.63$ [rad/s]

In this section a closer look is taken at the response motions of the HTV-cargo system when the first excitation frequency is close to the natural frequency of the relative horizontal motion reduction system, if 5 damping systems are used. The second excitation frequency is located away of any natural frequency of the new cargo handling system. The following values are used in this simulation: $\omega_1 = 0.3$ [rad/s], $F_1 = 5000$ [kN], $\omega_2 = 0.63$ [rad/s], $F_2 = 10{,}000$ [kN].

Both the high and low frequent parts of the irregular force can be detected in the response motions (see Figure 4-49). It is also clearly visible that resonance motions occur for the case with 5 damping devices. However the amplitude of the resonance motions is controlled by the amount of damping which is generated by the damping devices. It is also noticed that the smaller the diameter of the control valve becomes, the smaller the amplitude of the motions becomes. Therefore even if one of the excitation frequencies is close to the natural frequency of the system, the system is capable of reducing the relative motions. It is also noticed that the increasing wave force, causing a transient response of the HTV-cargo system, does not influence the performance of the clamping system. The amplitude of the occurring motions still reduces over as the control valve area reduces.

Figure 4-51 and Figure 4-52 also show that the system experiences resonance if 5 damping devices are applied. The piston displacement and corresponding force are much larger than for the case with 10 damping devices, while for the other simulations the piston displacement was, approximately, the same for the 2 system layouts (see Appendix D.5). This is caused by the fact that the system with 5 devices is experiencing resonance and, in the other simulations, the excitation frequencies were located away of the natural frequency of the system. The motions were dominated by mass terms (see Chapter 2.5) and the response amplitude is approximately the same for the two systems in that case. However in the case a system experiences resonance motions, the excitation force pushes the mass into a direction in which it is already moving by itself, increasing the amplitude of the motions. These motions can only be reduced by the addition of damping; therefore the damper force of a system with 5 damping devices is more than twice as big as the damper force of a system with 10 damping devices.

Summarising, the damping devices are capable of reducing the response motions in irregular waves, to the point that they fulfil the design requirement of $x \leq 0.15$ [m].

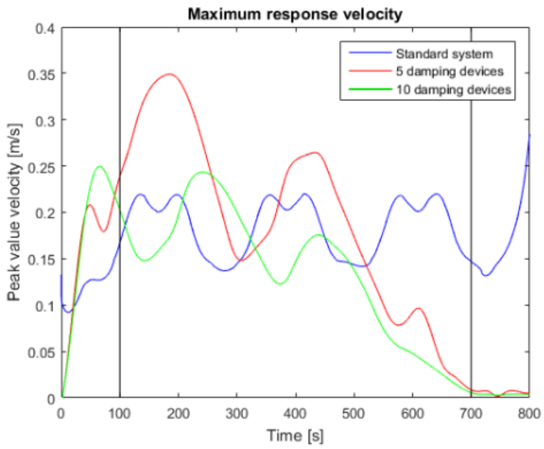

Figure 4-49 Maximum response motion, irregular force case 1

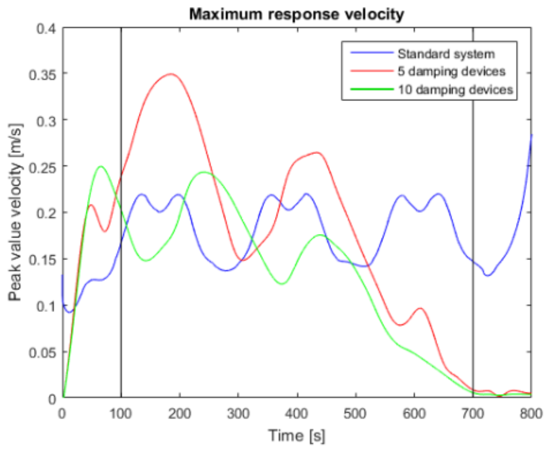

Figure 4-50 Maximum response velocity, irregular force case 1

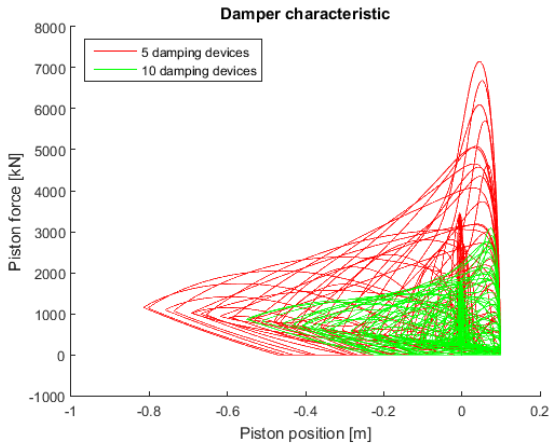

Figure 4-51 Damper characteristic, irregular force case 1

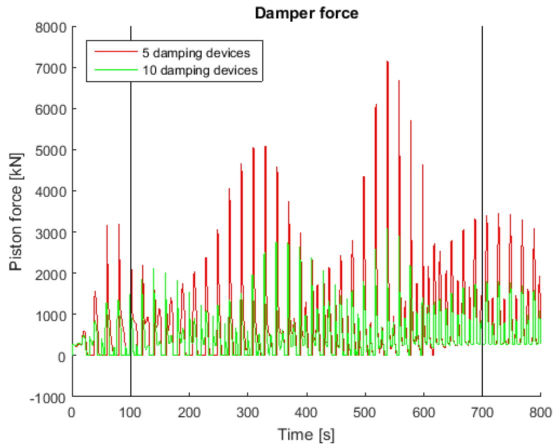

Figure 4-52 Damper force, irregular force case 1

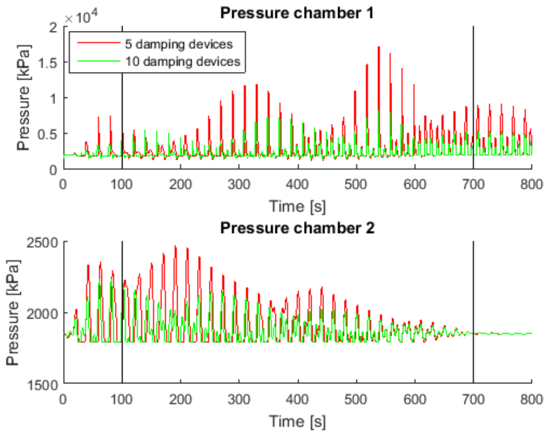

Figure 4-53 Pressure inside the cylinder chambers, irregular force case 1

4.6 Conclusion

In this chapter a clear overview is given of the outlook and working principle of the clamping system. The reduction of the relative horizontal motions is accomplished by a damping device which is mounted on the clamping constructions. This damping device consists of a passive fluid damper combined with a control valve and a fender. The fender is used to absorb the impact loads at the start of the clamp operation and to pull the piston to its outer position during the operation. To keep the impact loads at the start of the operation as low as possible, the stiffness of the fender should be low. The motions of the HTV-cargo system are reduced due to the increasing stiffness and damping coefficients of the system. The damping and stiffness characteristics of the system are determined by the control valve area. The control valve connects the two chambers of the passive fluid damper. The control valve area is reduced over time, increasing the stiffness and damping the system generates. Therefore the natural frequency of the complete system shifts from a low frequency to a high frequency. The motional behaviour of the system changes from a mass driven system to a stiffness driven system.

The performed simulations show that the clamping system is capable of reducing the amplitude of the occurring relative horizontal motions below the set acceptable relative horizontal motion amplitude of 0.15 [m]. This reduction is obtained for a regular and irregular wave spectrum. The transient response of the HTV-cargo system, which is caused by the increase of the irregular wave forces, does not influence the performance capabilities of the clamping system. The system is able to perform in real conditions.

It is also noticed that large impact forces arise in the system. These impact forces occur due to the fact that the cargo is not constantly connected to the clamping system. Therefore the amount of damping and stiffness the HTV-cargo system experiences changes in time. The impact loads occur when the moving cargo hits the non-moving piston rod of the damping device. The amplitude of the occurring impact load is dependent on the relief pressure of the damping device, the amount of damping devices used and the relative velocity of the cargo. If the relief pressure of the device is lowered, the maximum output force of the device is lowered. The pressure in the system and the force output are linearly dependent on each other. Consequently more time is needed to reduce the motions, although the motions will be below the set limit when the control valve is closed. The set value of relief pressure of the valve should be above the pressure which is needed to withstand the static loads occurring from the cargo motions. If the relief pressure is set below this limit the motions will not be reduced. Each clamping device generates the same amount of force to reduce the relative motions of the HTV-cargo system. To lower the occurring forces per device, more devices can be used. The relative velocity of the cargo is dependent on the amount of energy it absorbs from the wave forces, the outgoing piston of the compressed damping device and the distance over which it can accelerate. The stiffness and damping coefficients are negligible in the case the cargo is moving from one damping device to the other. The devices only add damping and stiffness when the cargo is pushing against them. The piston wants to return to its initial position under the influence of the pre-pressure in the system and the stiffness of the fender. The pre-pressure in the system is used to prevent the cavitation in the system. Cavitation occurs when the pressure of the fluid is below its vapour pressure. However the pre-pressure adds a stiffness component to the system just as the fender. If this stiffness component is too large, the piston wants to move faster than the cargo. In this way the device will push the cargo away, increasing the velocity of the cargo. It is also possible to reduce the velocity by decreasing the acceleration distance of the cargo. This can be done by pre-compressing the fender at the start of the operation. Consequently the stroke of the piston is increased limiting the free space of the cargo. If the cargo moves away from one damping device, it meets the other damping device faster, reducing its velocity. However by compressing the fender, the output force to the cargo becomes larger, due to the stiffness of the fender. Consequently the stiffness of the fender should be as low as possible. It is recommended to perform more research to find the optimal combination between the pre-compression of the fender, the maximum allowed pressure and the amount of systems which is used.

The dimensions of the damping device are based on the performed simulations. Therefore the dimensions are not optimal. The dimensions are determined through the simulations because the calculated dimensions showed responses which were not acceptable. The dimensions were increased until the system provided acceptable responses. However the optimal dimensions of the cylinder are dependent on the required force which they need to produce. These forces are dependent on the wave forces and the impact loads. As the system is only simulated for a limited amount of conditions it is advised to increase the number of simulations to include more cases in order to obtain the optimal design which can be used for different types of cargo.

The presented system is only a preliminary design, as the aim of this phase was to investigate the working principle of the system. It is recommended to take this design to the next phase in the design trajectory. The degrees of freedom of the system should be increased from 1, for this design, to 3, to 6 and finally to 12. The design of the damping device needs to be optimised. Furthermore attention should be paid to the clamping structure, on which the damping device is mounted. How does this system looks like and how does it guide the forces into the cargo hull. How is the clamping structure pushed against the cargo: skidding, jacking, pulling with lines. It is also recommended to put Teflon pads between the cargo and the damping device. Large forces are subjected to the system, causing large friction forces between the cargo and the fenders. The friction will damage the rubber of which the fenders are made. Although the friction will have a positive influence on the motions in heave and surge direction, the influence is relatively small.

5 Preliminary design Line tension actuator

This chapter will cover the outlook and performance of the *Line tension actuator*. First the general outlook and working principle of the *Line tension actuator* are discussed. The next section describes the used actuators and the control system. The third section describes the models which are used to analyse the system characteristics. A parametric study is performed to see how the important parameters influence the behaviour of the system. The fifth section provides insight in the performance of the system. Finally a conclusion about the usability of the system is presented.

5.1 Working principle system

The stretch compensation system consists of Dyneema lines and actuators. The actuators are located on the casings of the HTV and the Dyneema lines connect the cargo to the actuators, i.e. the lines connect the HTV and the cargo with each other (see Figure 5-1). The actuator consists of a cylinder filled with a gas or fluid which allows a piston to move (see Chapter 5.2) [20]. The stretch compensation systems are placed at two sides of the cargo; front and aft or portside and starboard side. This is dependent on the cargo which is loaded or discharged onto the HTV.

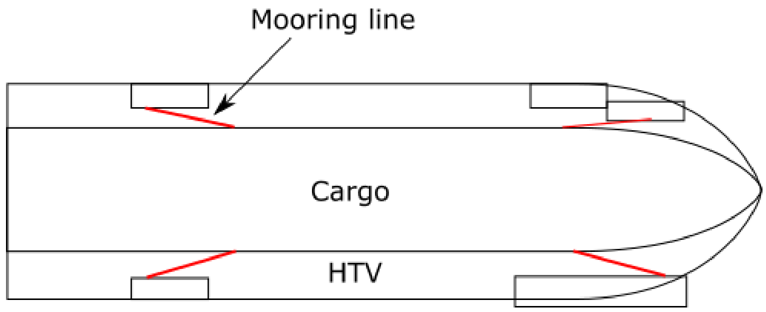

Figure 5-1 System layout

The distance between the HTV and the cargo changes continuously because the HTV and the cargo are both allowed to move freely, since they are not connected rigidly. The wave force working on the HTV-cargo system causes the Dyneema lines to stretch, allowing the HTV and cargo to move relative to each other. The idea is that the stretch compensation system will react to these relative horizontal motions by accommodating the stretch in the Dyneema lines. The principle of the stretch compensation system can be presented with equation (5-1) and Figure 5-2. This means that the actuator should react on the tension (and geometric parameters of the fairleads), where the tension (and change in tension) is a direct result of the relative motions.

$$x = x_p + x_s = 0 \tag{5-1}$$

The used variables are:

x	=	Displacement	[m]
x_p	=	Piston displacement	[m]
x_s	=	Stretch of the Dyneema line	[m]

Therefore the stretch compensation system is a reactive system as it reacts to a change in the line tension. However it is found that the system is unstable. The system reacts to a change in tension the line experiences. The addition of line tension causes the line to stretch. In order to accommodate this stretch the piston needs to heave out. In this way the actuator adds tension to the line, causing the line to stretch even more. In this way the tension in the line keeps rising. Therefore the stretch compensation is unstable and it is considered to be unfeasible.

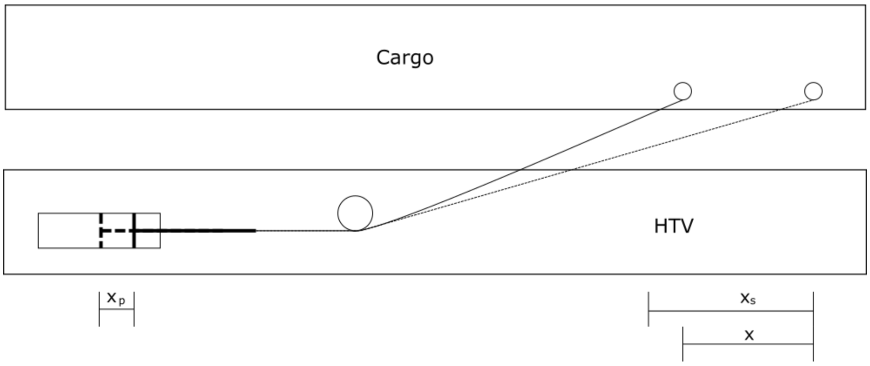

Figure 5-2 Schematic overview relative motions

Therefore another look is taken at the proposed concepts of the conceptual design phase and their basic ideas behind them. Also another look is taken at the previous conducted studies. It is concluded that the principle of reducing the relative horizontal motions by increasing the line tension should be able to work. However instead of a reactive system an active system should be used. In this system the line tension is an actuating force instead of a reaction force. Therefore the design of *Line tension actuator* is proposed.

In 2012 P.S.C. Lee [6] designed an active tension control system which controlled the relative horizontal motions between the HTV and its cargo (see Appendix A.5). The system consists of Tugger winches, Dyneema lines and a control system. The control system consists of measurement equipment and a PID controller. The measurement equipment is used to measure the relative horizontal motion between the HTV and the cargo. This motion is compared with the desired motion of the HTV-cargo system. The difference between this set point and the actual response is fed into the control system. The control system will calculate the amount of line tension which is required to position the cargo at the set point and will allocate this force to the Tugger winches. The Tugger winches will provide this tension to the Dyneema lines and thus to the system. However it was concluded that the Tugger winches were unable to produce the required tensions and the system was considered to be unfeasible.

The calculated tensions can be broken down in three different components:

- The minimal tension in the line. This is a constant tension to prevent the occurrence of slack in the Dyneema lines,
- The tensions in the line which are caused by the stiffness and the motions of the HTV-cargo system ($k \times x$). These tensions will reduce as the motions reduce,
- The control tensions as allocated by the control system. These tensions will increase as the motions reduce. These are the tensions necessary to pull the cargo back to its required position.

Over the course of the design process it turned out that the system designed by Lee in 2012 is based on the same principles as the proposed *Line tension actuator*. The difference is that Lee uses Tugger winches to deliver the required force, while the *Line tension actuator* will utilize actuators based on fluid or pneumatic power. However the designed control system fulfils the design requirements. Therefore this research will focus on the workability of the actuators and not the complete system. The main objective is to find out if the actuators are able to produce the required forces. Therefore the main questions are:

- Is the system able to deliver the required forces?
- Are the occurring pressures realistic?
- Can the required power be delivered?

The calculated tensions obtained by the research of Lee will be used as input values for the *Line tension actuator*.

5.2 System description

The stretch compensation system consists out of an actuator which is connected to the Dyneema line and a control system, which controls the motions of the actuator. The actuator consists out of a piston which is located inside a cylinder. The Dyneema line can be connected to the actuator in two different configurations.

In the first configuration the Dyneema line is connected to the piston (see Figure 5-3). The cylinder is fixed to the HTV and only the piston is allowed to move inside the cylinder. Therefore the piston motions regulate the amount of force the Dyneema line is experiencing. In the second configuration the Dyneema line is connected to the cylinder (see Figure 5-4). The piston rod is fixed to the HTV and the cylinder hull is allowed to move on a track on the HTV. In this case the tension in the line is regulated by the cylinder motions.

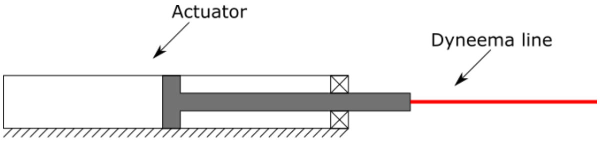

Figure 5-3 First actuator configuration

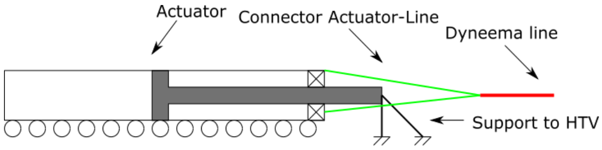

Figure 5-4 Second actuator configuration

The Dyneema lines can only take-up a tension force and not a pushing force. Therefore the tension in the line increases as the line is pulled to the left. The tension in the line is regulated by the pressures inside the two chambers (see Chapter 0). The force-pressure relation is given in equation (5-2).

$$F = PA \tag{5-2}$$

The variables are:

A	=	Area of the piston head in the chamber	[m^2]
F	=	Resulting force on the piston head in one chamber	[N]
P	=	Pressure in the chamber	[Pa]

In the first configuration the tension in the line is regulated by the pressure in the right chamber, while in the second configuration the tension is regulated by the pressure in the left chamber. As the piston head area in the second configuration is larger than that in the first configuration, a lesser increase in pressure is needed in the second configuration, to give the line a certain tension, compared to the first configuration. However in order to move the line over a certain distance, a larger volume of fluid is needed in the second configuration compared with the first configuration. On top of this, the piston experiences a large compression force, in the second configuration. This compression force induces the risk on buckling of the piston rod. It is therefore decided to implement the first configuration in the stretch compensation system.

The control system will control the amount of force the piston produces. This research study will investigate the performance of two different types of actuator systems:

- A hydraulic actuator,
- A pneumatic actuator.

It is important to understand how the different systems work, before the usability of these systems is investigated. The following sections will describe the principles of the actuators and the control system.

Actuator

The outlook of the hydraulic actuator and the pneumatic actuator are similar. The major difference between the two actuators is that one is filled with liquid while the other is filled with gas. In Figure 5-5 a simplified overview of the actuator is shown.

The numbers in Figure 5-5 relate to:

1. Chamber 1,
2. Chamber 2,
3. Piston head,
4. Piston rod,
5. Seal,
6. 3-way directional valve,
7. Hydraulic pump,
   Pneumatic pump,
8. Reservoir.

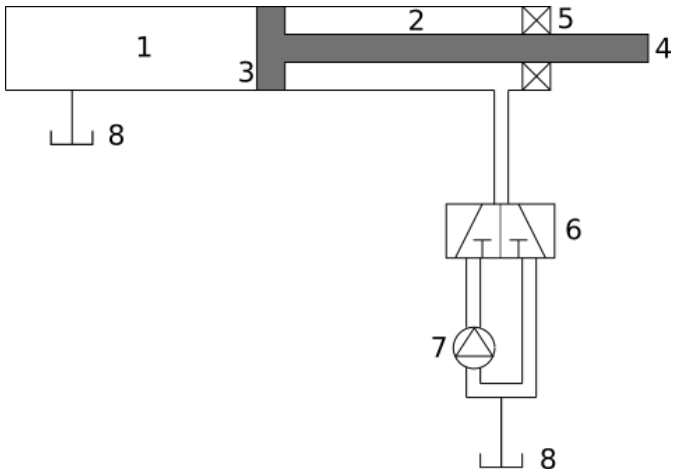

Figure 5-5 Simplified overview actuator

The right hand side of the cylinder is coupled to reservoir (8) by a pump (7) and a 3-way directional valve (6), located in series to each other. The end of the piston rod (4) is connected to the Dyneema line connecting the HTV and the cargo.

The 3-way directional valve is used to control the motions of the piston rod and thus the force. Dependent on the position of the valve, three different fluid flows inside the cylinder can occur. If the valve connects the pump and chamber 2 (2), a high pressurized fluid flow can enter chamber 2. This causes the piston to move to the left. If the valve connects chamber 2 and the reservoir, a fluid flow will leave chamber 2. This will cause the piston to move to the right under the influence of the line force. It is also possible to position the valve such that no fluid flow is allowed. In this case the piston will stay at its current position.

A Dyneema line is only capable of taking up a tension force. Therefore the pressure in chamber 2 should always be equal or larger than the pressure in chamber 1 (1). The line tension causes the fluid to flow from chamber 2 to the reservoir, as the pressure in the reservoir is lower than in the chamber, moving the piston to the right. Hence no additional pressure in chamber 1 is necessary to move the piston to the right. In order to save power, it is decided that no high pressurized fluid flow will be directed to chamber 1.

The pump is used to keep the fluid or gas at a constant pressure. Therefore the pump flow will vary in time as the amount of flow which enters chamber 2 will vary in time. It is assumed that the pump is able to produce a constant pressure, during this study.

Control system

The stretch compensation system will be controlled by a closed-loop control system. The actuator will be equipped with a force sensor to control the piston position, to produce the required force. This signal will be fed into a controller, which compares this feedback signal with the demanded signal to determine the resulting error. This error is defined as the difference between the required signal (F_r) and the obtained signal (F_p). The error is defined as:

$$F_e = F_r - F_p \tag{5-3}$$

Based on the error, the controller produces a command signal which drives the 3-way directional valve, which in turn directs the flow in the actuator. In this way the system will control the force output of the actuator, which is defined by the active tension control system [6]. An overview of this system is shown in Figure 5-6.

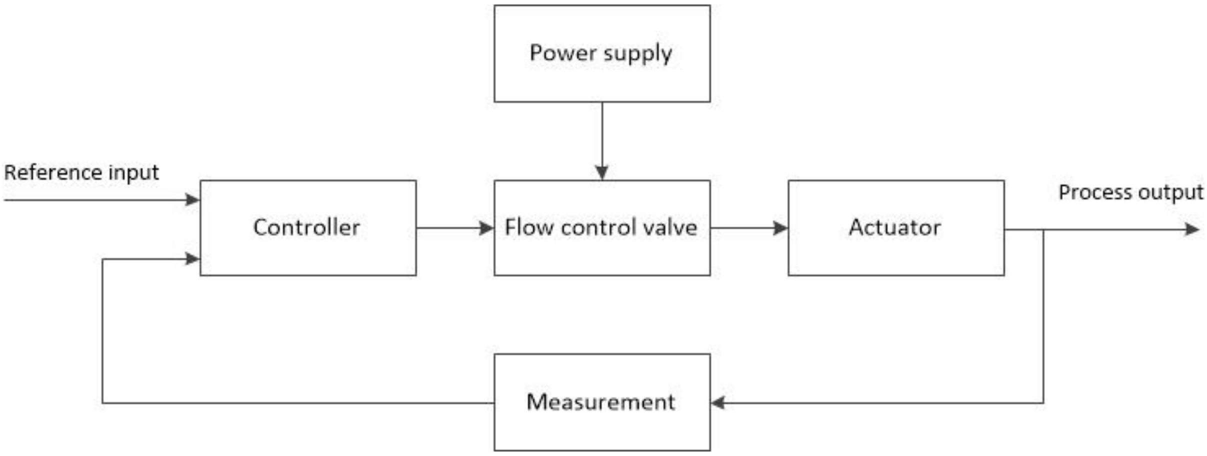

Figure 5-6 Schematic overview control system

Different types of closed loop control systems are utilised. The simplest form of control is on-off. In this control system the system can produce only two output values to the valve: The valve is completely closed or opened. This causes the system to react very aggressive to the continuous changing required force signal. Therefore it is likely that the system reacts very shaky and with large overshoots. Another disadvantage is that the valves wear out very fast due to the very fast opening and closing sequence.

The proportional control system is more complex than an on/off control system. This system is capable of opening the valve such that a proportional fluid flow is accomplished which is able to produce the correct piston force. The controller output is thus proportional to the error signal, i.e. the output of the controller (u_c) is the multiplication of the error (F_e) and the proportional gain (K_p). The larger the value of the gain is, the more aggressive the system will react on the error. The mathematical expression is given in equation (5-4):

$$u_c(t) = K_p \times F_e(t) \tag{5-4}$$

One of the common used controllers is the Proportional (*P*), Integral (*I*), Derivative (*D*) controller. This controller tries to minimize the error over the time by adjusting the control output. The *P* accounts for the present error, the *I* accounts for the values of the error in the past, the *D* accounts for the values of the future error.

In order to investigate the objective for this system (see Chapter 5.1), it is necessary that the delivered force signal of the actuator follows the required force signal calculated by the control system of Lee. It is decided to only use a proportional control system to control the output force of the actuators. Satisfactory conclusions can be made of the simulation results obtained with only a P controller. The characteristics of the system and the performance of the system are clearly visible (see Chapter 5.5). Based on these results it was decided to not optimise the control system further during this thesis. However improvements are possible (PID control, feedforward, feedback). As a consequence more high frequency oscillations in the obtained force signal occur. As well as a steady state error between the required and obtained force.

5.3 Description how the stretch compensation system works

It is important to understand the basic physics of the stretch compensation system, before more complex simulations are performed. Therefore this section will describe the physics of the system. The actuators will be modelled in Simscape, for these models see Appendix E.2.

The amount of force which the actuator delivers to the system can be calculated with equation (5-5), if the pressures inside the actuator chambers are known. The calculation procedure for the pressures will be explained in the succeeding sections. The line force can be calculated with equation (5-6). As a line can only take up tension forces and no pushing forces $F_a \geq 0$. The used axis definition is show in Figure 5-7.

$$F_a = P_1 \times A_p - P_2 \times (A_p - A_r) \tag{5-5}$$

$$F_a = F_{line} = k_{line} \times x_p + k_{line} \times x_c + F_{min} \tag{5-6}$$

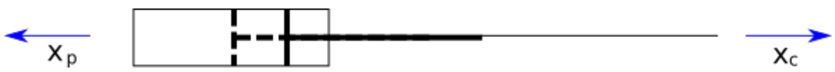

Figure 5-7 Axis definition Line tension actuator

The used variables in the equations are:

A_p	=	Piston head area	[m^2]
A_r	=	Piston rod area	[m^2]
F_a	=	Actuator force	[N]
F_{line}	=	Mooring line force	[N]
F_{min}	=	Minimal tension in Dyneema line	[N]
k_{line}	=	Line stiffness	[N/m]
$P_{1,2}$	=	Pressure in chamber 1,2	[Pa]
x_c	=	Cargo position	[m]
x_p	=	Piston position	[m]

Hydraulic operated system

The principles of fluid mechanics will be used to obtain the formulas which are used in the Simscape model. The derivation of the basic formulas is done in the same manner as for the *Clamping system* (see Chapter 5.3). Therefore only the obtained formulas will be presented for this system.

The liquid flow through the valve is dependent on the position of the spool (see Figure 5-8). The fluid flow through the valve can be calculated according to equation (4-2). The following equations are obtained, in this equation P_s is the supply pressure of the pump and P_r is the return pressure of the reservoir:

If the piston moves to the left, thus has a negative velocity (v ≤ 0), the fluid enters the cylinder:

$$Q_2 = k_d A_v \sqrt{\frac{2 \times (P_s - P_2)}{\rho}} \tag{5-7}$$

If the piston moves to the right, having a positive velocity (v > 0), the fluid flows out of the cylinder:

$$Q_2 = -k_d A_v \sqrt{\frac{2 \times (P_2 - P_r)}{\rho}} \tag{5-8}$$

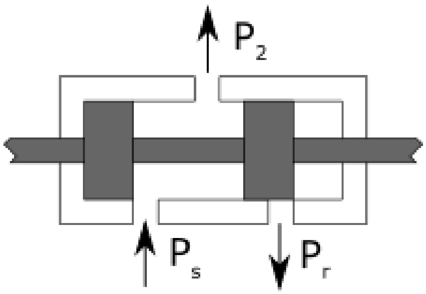

Figure 5-8 Schematic overview of 3-way directional valve

Chamber 1 is connected to a reservoir which is kept under atmospheric pressure. Therefore the pressure in chamber 1 is equal to the atmosphere and will not change during the operation. The following differential equation for chamber 1 is determined:

$$\frac{dP_1}{dt} = 0 \tag{5-9}$$

Re-arranging and combining equations (4-1), (5-7), and (5-8) gives the first order nonlinear differential equation for chamber 2 which controls the piston force.

$$\frac{dP_2}{dt} = (-\dot{x}A_p + Q_2) \times \frac{\beta}{(L_2 - x) \times A_p} \tag{5-10}$$

Pneumatic operated system

The pneumatic models are based on three equations [21]:

- The ideal gas law,
- The conservation of mass,
- The energy equation.

In these equations, it is assumed that: the gas is perfect, the pressures and temperatures in each chamber are homogeneous and kinetic and potential terms are negligible.

The ideal gas law is given as [21]:

$$P = \rho RT \tag{5-11}$$

The mass flow rate continuity equation is given as [21]:

$$\frac{dm}{dt} = \dot{m} = \dot{m}_{in} - \dot{m}_{out} = \frac{d}{dt}(\rho V) = \frac{d\rho}{dt}V + \rho\frac{dV}{dt} \tag{5-12}$$

The energy equation is written as [21]:

$$q_{in} - q_{out} + \gamma C_v(\dot{m}_{in}T_{in} - \dot{m}_{out}T) - \dot{W} = \dot{U} \tag{5-13}$$

The used variables are:

C_v	=	Specific heat at constant volume of gas	[J/kg K]
m	=	Mass gas	[kg]
$\dot{m}_{in,out}$	=	Mass flows entering and leaving the chamber	[kg/s]
P	=	Pressure	[Pa]
$q_{in,out}$	=	Heat transfer terms	[W/m^2]
R	=	Specific gas constant used gas	[J/kg K]
T	=	Temperature	[K]
T_{in}	=	Heat incoming gas flow	[K]
$\dot{U}$	=	Change in internal energy	[J]
V	=	Volume	[m^3]
$\dot{W}$	=	Rate of change in performed work	[Nm]
γ	=	Heat capacity ratio	[-]
ρ	=	Density gas	[kg/m^3]

If it is assumed that the incoming gas flow already has the temperature of the gas in the chamber the energy equation can be written as, by combining equations (5-11), (5-12) and (5-13):

$$\frac{\gamma-1}{\gamma}(q_{in}-q_{out})+\frac{1}{\rho}(\dot{m}_{in}-\dot{m}_{out})-\frac{dV}{dt}=\frac{V}{\gamma P}\frac{dP}{dt} \tag{5-14}$$

The system is used in a very dynamic environment in which the processes occur relatively fast, because the waves have a period of 10 [s]. Therefore the processes within the cylinder also occur relatively fast. For this reason the process is considered to be isentropic. In an isentropic process no heat is transferred, i.e. $q_{in} - q_{out} = 0$. If these conditions are applied to equation (5-14), the following differential equation for the pressure in chamber 2 is obtained, after some re-writing:

$$\frac{dP_2}{dt}=\gamma\frac{RT}{(L_2-x)\times(A_p-A_r)}(\dot{m}_{in}-\dot{m}_{out})-\gamma\frac{P_2}{(L_2-x)}\dot{x} \tag{5-15}$$

Chamber 1 of the pneumatic cylinder is connected to a reservoir which is kept under atmospheric pressure, as well as for the hydraulic system. Therefore the following differential equation for chamber 1 is determined:

$$\frac{dP_1}{dt}=0 \tag{5-16}$$

The used variables are:

A_p	=	Piston head area	[m^2]
A_r	=	Piston rod area	[m^2]
L_2	=	Length chamber 2 at the begin of operation	[m]
x	=	Piston displacement	[m]
$\dot{x}$	=	Piston velocity	[m/s]

By applying the standard equations for a pneumatic mass flow through a valve [21], equations are obtained for the mass flow rate. The used formula is dependent on the type of flow which occurs in the valve. If the upstream to downstream pressure ratio is larger than a critical value (P_{cr}), the flow will attain sonic velocity (choked flow) and will depend nonlinearly on both pressures. If the ratio is smaller than P_{cr} the mass flow depends linear on the upstream pressure. The critical pressure is defined by:

$$P_{cr} = (\frac{2}{\gamma + 1})^{\frac{\gamma}{\gamma - 1}} \tag{5-17}$$

If the piston velocity is smaller than zero (v < 0) and $\frac{P_2}{P_s} \leq P_{cr}$, the gas enters the cylinder:

$$\dot{m}_{in} = k_d A_v \frac{P_s}{\sqrt{T}} \sqrt{\frac{\gamma}{R} (\frac{2}{\gamma + 1})^{\frac{\gamma + 1}{\gamma - 1}}} \tag{5-18}$$

If (v <0) and $\frac{P_2}{P_s} > P_{cr}$, the gas enters the cylinder:

$$\dot{m}_{in} = k_d A_v \frac{P_s}{\sqrt{T}} (\frac{P_2}{P_s})^{\frac{1}{\gamma}} \sqrt{1 - (\frac{P_2}{P_s})^{\frac{\gamma - 1}{\gamma}}} \sqrt{\frac{2\gamma}{R(\gamma - 1)}} \tag{5-19}$$

If (v ≥ 0) and $\frac{P_2}{P_s} \leq P_{cr}$, the gas leaves the cylinder:

$$\dot{m}_{out} = k_d A_v \frac{P_2}{\sqrt{T}} \sqrt{\frac{\gamma}{R} (\frac{2}{\gamma + 1})^{\frac{\gamma + 1}{\gamma - 1}}} \tag{5-20}$$

If (v ≥ 0) and $\frac{P_2}{P_s} > P_{cr}$, the gas leaves the cylinder:

$$\dot{m}_{out} = k_d A_v \frac{P_2}{\sqrt{T}} (\frac{P_r}{P_2})^{\frac{1}{\gamma}} \sqrt{1 - (\frac{P_r}{P_2})^{\frac{\gamma - 1}{\gamma}}} \sqrt{\frac{2\gamma}{R(\gamma - 1)}} \tag{5-21}$$

For the complete derivation of the differential equations see Appendix E.1.

Control system

The control system determines the amount of flow through the 3-way valve. The control system provides an input signal to the valve actuator. The valve actuator controls the position of the spool which determines the valve opening. The input signal to the valve actuator is calculated according to equation (5-4).

The value of the proportional gain can be determined by different methods: empirical and analytical. The analytical method uses the linearized transfer functions of the actuator and the controller. However the actuator is a non-linear system. Therefore the first step is to linearize the system around its operating conditions (set point). When the system is linearized the transfer function can be obtained. If the transfer function is known the proportional controller can be obtained by complex tuning methods or a more traditional one like Ziegler-Nichols. The empirical method determines the value of the gain based on simulations. Multiple simulations will be performed with different gain values. In this way the effect of the value of the gain on the performance of the system can be investigated and the correct gain value is derived. The simulations are performed for a sufficient amount of time, so that the system responds in its steady state. This method is suitable for systems with a varying set point. It is therefore decided to determine the value of the proportional gain empirically.

The flow area of the valve is determined by the spool position of the valve (the grey part in Figure 5-8). The spool position is controlled by the valve actuator. The valve actuator receives the required spool position from the control system. However it takes time to move the valve from one position to another. The response time of the valve is characterized by the time constant. The time constant describes how fast the valve moves from a closed position to an open position. The valve actuator causes a first order lag to occur in the system. This first order lag will have an influence on the settings of the controller, as the controller needs to accommodate this delay in its output signal to be able to control the system correctly. A simple method to model this lag in the system is with the following equation [22]:

$$\dot{x}_{spool}(t) = \frac{u_c(t) - x_{spool}(t)}{T} \tag{5-22}$$

The used variables in the equations are:

$u_c(t)$	=	Required spool position	[m]
$x_{spool}(t)$	=	Obtained spool position	[m]
$\dot{x}_{spool}(t)$	=	Obtained spool velocity	[m/s]
T	=	Time constant	[s]

5.4 Parametric study

The response of the system to a proportional gain is dependent on the system characteristics, the time constant of the valve and the required force. Therefore a parametric study will be conducted to investigate the effects of these properties on the behaviour of the system for certain gain settings. The characteristics of the system consist of the actuator dimensions, the type of actuator (hydraulic or pneumatic) and the stiffness of the Dyneema lines. The dimensions of the system are fixed for all the different cases (see Appendix E.3). The time constant is different for different types of valves, in this stage of the research it is unknown which type of valve will be used. Therefore the effect of the time constant will be investigated. The required force which the actuator needs to produce is also different per loading operation; therefore the effect of the required force in correspondence to a gain value is also investigated.

For the purposes of this analysis it is assumed that the piston delivers a force to a linear spring load which has a certain stiffness (see Chapter 5.5). In this way, it is possible to investigate the response of the actuator. The tension in the line consists of a sinusoidal force which has a certain amplitude ($F_{amplitude}$) and a certain minimal tension force (F_{min}) to prevent slack in the line.

Influence time constant and force

This section focusses on the effect of the time constant in correspondence to a varying line tension. The simulations are conducted for the hydraulic actuator which is equipped with 2 Dyneema lines of 58 [m].

Figure 5-9 shows the steady state error between the required force and the obtained force for the hydraulic actuator. The required forces used for this simulation correspond to the required forces needed for the base case. For the other simulations see Appendix E.4.

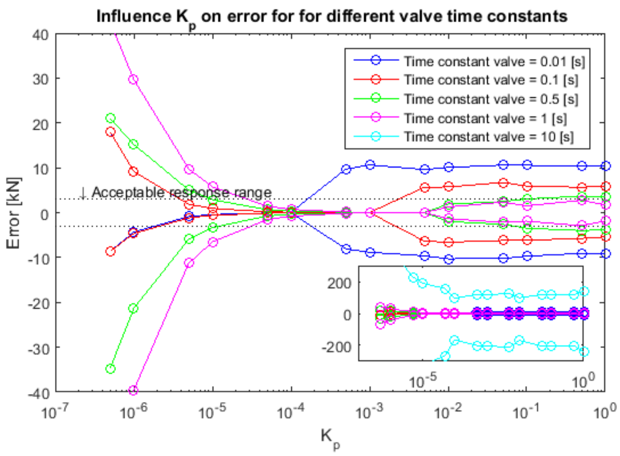

Figure 5-9 Hydraulic system, influence proportional gain, $F_{amplitude}$ = 1500 [kN], F_{min} = -10 [kN]

The value of the proportional gain (K_p) has a large influence on the response of the system. If the K_p value is too small, the system will react very slowly and a phase lag between the required and obtained force occurs, this phase lag determines the magnitude of the error. If the K_p value is too large, the system will react very aggressive. The valve will be opened and closed very fast. In this case the response of the system will have the same characteristics as when an on/off controller is used. Therefore the error will be at a constant level for a certain range of K_p values. The effect of the K_p value is shown in Figure 5-10 and Figure 5-11. In this figures the valve time constant is 0.5 [s].

The value of the valve time constant has an influence on the bandwidth of K_p values in which the system will have almost no phase lag and almost no aggressive response. If this value is small, the bandwidth is very small. The valve reacts fast, which causes the system to produce an overshoot relatively fast. In this case damping should be added to the system and instead of using a simple proportional controller it is better to use, for example, a PID controller. If the time constant is large, the system reacts very slow and thereby will not reach the required force. The slow response of the valve adds damping to the system. It is therefore decided to use a time constant value of 0.5 [s] for this research study.

The magnitude of the force signal also has an influence on the system. The force in the system is accomplished by the pressure in the cylinder (see equation (5-5)). The pressure difference between the cylinder and the atmosphere or pump determines the amount of flow, together with the controller which controls the valve area. If a large force amplitude is required, the difference in the pressures is high. When the valve area is adjusted too slowly (K_p is too low) the velocity of the flow is too low which causes a larger error, due to the larger required velocity. If the valve area is adjusted too fast (high K_p), the velocity of the flow is too high, causing a large error. This causes the errors to have different values while the range of acceptable K_p values stays approximately the same independent of the value of the required force.

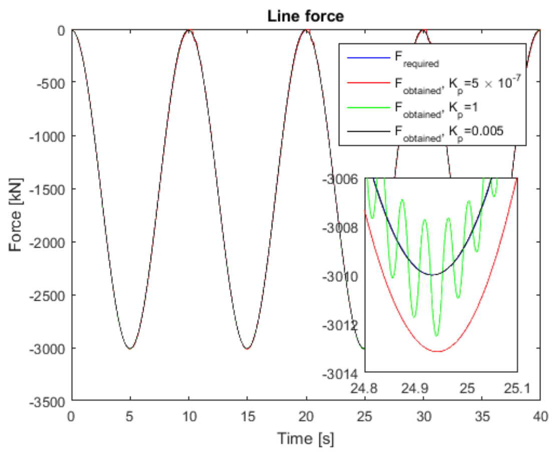

Figure 5-10 Hydraulic system, Line force

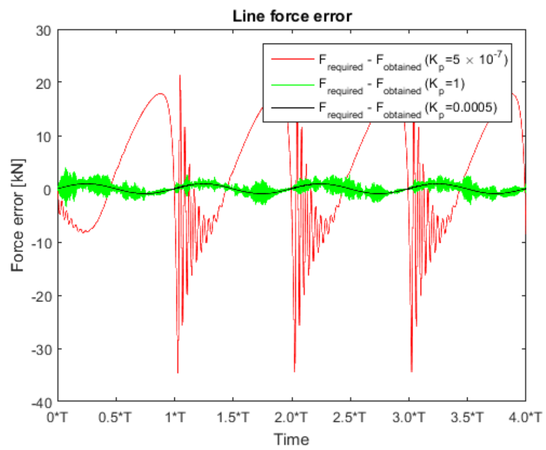

Figure 5-11 Hydraulic system, line force error

Influence line stiffness

This section focusses on the effect of the stiffness of the Dyneema lines on the values of the proportional gain. The simulations are conducted for the hydraulic actuator, the time constant of the valve is taken as 0.5 [s] and the force has the following characteristics: $F_{amplitude}$ = 1500 [kN], F_{min} = -10 [kN].

Figure 5-12 shows the steady state error in the produced line force for the different stiffness's and the different gain values. The figure shows that the influence of the line stiffness on the bandwidth of K_p values which produce an acceptable response is relatively small. However if the system is reacting to slow, the error is larger if the system is equipped with lines which have a low stiffness. In order to obtain the same line tension, the stroke of the piston is larger for low stiffness lines compared to high stiffness lines. Therefore the amount of fluid needed is larger for low stiffness lines. If the control system acts to slow, not enough fluid will flow into the chamber causing a larger error to occur. If the system acts too aggressive (high K_p value) to much fluid will flow into the chamber. A line with a high stiffness will produce a larger tension than a line with a low stiffness. This explains why the errors of the high stiffness lines are larger than the low stiffness lines. This also explains why the bandwidth of the usable K_p values stays approximately the same.

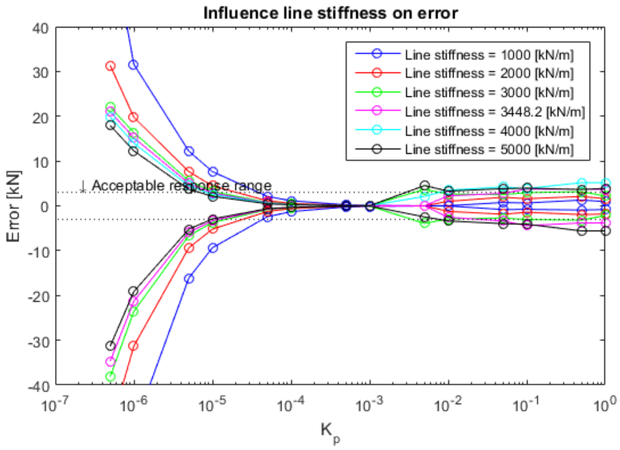

Figure 5-12 Hydraulic system, influence line stiffness on proportional gain

Influence fluid

In this section it is investigated what the effect of the used fluid in the actuator is on the values of the proportional gain. The simulations are conducted for the two different types of actuators, which have a valve time constant of 0.5 [s], are equipped with 2 Dyneema lines of 58 [mm] and a force with the following characteristics: $F_{ampplitude}$ = 1500 [kN], F_{min} = -10 [kN].

Figure 5-13 shows the steady state error in the produced line force for the two systems compared to the different gain values. It is clearly visible that the error for the hydraulic system is smaller than for the pneumatic system, irrespective of the used K_p value. The difference in the error is caused by the difference in compressibility of the liquid and gas. Liquid can be modelled as incompressible and its behaviour is therefore independent of the pressure in the system, while gas is very compressible which causes its behaviour to be very dependent on the pressure.

The isentropic bulk modulus of gas can be calculated as:

$$\beta = \gamma \times P \tag{5-23}$$

If equation (5-23) is substituted into the nonlinear differential equation for the cylinder pressure for liquid (equation (5-10)) the following equation is derived:

$$\frac{dP_2}{dt} = \gamma \frac{P}{(L_2 - x) \times (A_p - A_r)} \mathrm{Q} - \gamma \frac{P}{(L_2 - x)} \dot{x} \tag{5-24}$$

If the change of flow rate (Q) is replaced by the mass flow rate ($\dot{m}$) equation (5-15), the differential equation for the pressure in the pneumatic cylinder, is obtained. These equations show that the behaviour of the gas is dependent on the pressure and heat capacity ratio. The heat capacity ratio is constant for a gas. However the pressure changes during the operation, which causes the fluid to show large compressions and expansions.

The largest error of the pneumatic systems occurs in the outgoing motion of the piston (see Figure 5-14). In the outgoing motion the piston is pulled back by the Dyneema line, during this process the gas flows from the high pressure chamber to a tank with a low pressure. As the line tension reduces, the pressure in the chamber reduces, this causes the gas to expand (see equation (D-5)).

When the tension in the line needs to be increased again, the pressure will go up, and the gas is compressed again. However the entering gas comes from a higher pressure than the gas experiences in the cylinder, therefore the gas is already compressed and a smaller error occurs compared to the outgoing motion. In fact the error of the gas for the ingoing piston motion is reduced by the decompressions of the inflowing gas. Therefore the mean value of the error is not close to zero as for the hydraulic system. A proportional control system will always have a steady state error between the required and obtained signal. Due to the incompressibility of the fluid, the mass of the fluid does not have a large effect on the obtained pressure, i.e. the piston motion, contrary to the mass inflow of the gas. If a high amount of compressed gas enters the chamber, the volume of the gas changes due to the lower pressure inside the cylinder, thereby the position of the piston will be changed and thus the line tension. However the mass of the gas stays the same, it only has a larger volume.

The controller does not take into account the compression and expansion of the gas. This causes the larger error in the outgoing piston motion and a smaller error in the ingoing piston motion. The proportional controller is a linear control system while the compression and expansion of the gas is a nonlinear process. However the effect of the larger error in the outgoing piston motion is small compared to the system dynamics (see Case 2: Pneumatic actuator 1). This also explains why the effects of a good reacting system and an aggressive pneumatic system are not visible in the error; both configurations produce a larger error due to the compressibility of the gas. It is therefore recommended to use the same range of K_p values as for the hydraulic system.

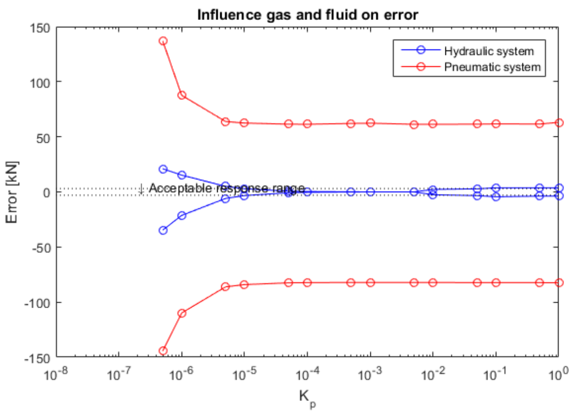

Figure 5-13 Influence of the used fluid on the proportional gain

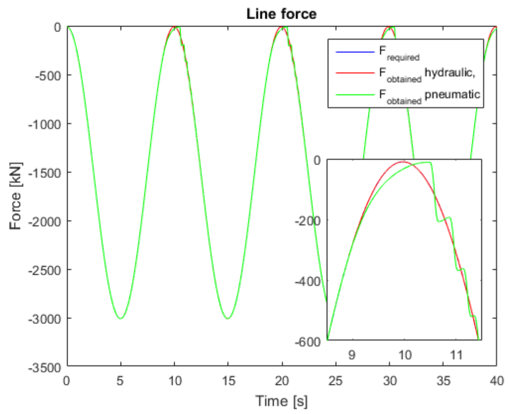

Figure 5-14 Line force of the two different systems

5.5 Simulations system

In this section the performance of two types of actuators will be presented and evaluated. The two actuators will be compared to each other and the best one will be selected for further research. The influence of the used lines on the performance is also investigated. Finally the performance of the complete system is presented.

Simulation method

The active tension control system of Lee [6] is implemented to the base case scenario. The calculated tensions of this system are used as input for the models of the hydraulic and pneumatic actuators. The *Line tension actuator* uses four actuators. Two are positioned at the front of the HTV and two at the back. The layout of the system for the performed simulations is shown in Figure 5-1. Due to this layout only the surge motions will be reduced. The actuators at the portside aft and front and at the starboard side aft and front form 1 system to reduce the motions. It is noticed that the system is symmetrical around the longitudinal axis. This causes the actuators at the front (actuator 1 and 3) to produce the same tensions, just as the actuators aft (actuator 2 and 4). The required line tensions the actuators need to produce is shown in Figure 5-15 and Figure 5-16. However also small motions occur in the sway and heave direction. These motions influence the force the actuators need to produce to reduce the motions in the surge direction. The required tensions are larger compared to when the tensions are only influenced by the relative motions in the surge direction.

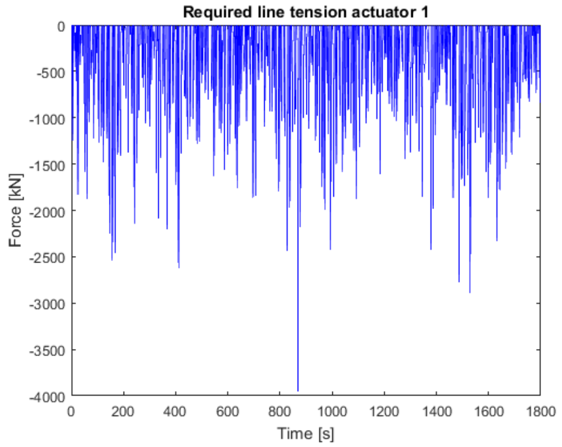

Figure 5-15 Required line tension actuator 1

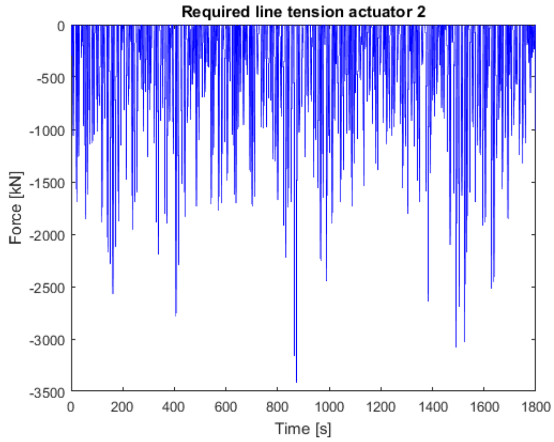

Figure 5-16 Required line tension actuator 2

The produced force of the actuators will be compared with the required tension in order to determine if the system is capable of delivering the required force in the required time, for an operating pressure of 300 bar. It will also be checked what the average and peak flows in the systems are in order to determine if these flows can be delivered by pumps.

Mooring configuration

The cargo will be connected to the actuators by Dyneema lines. The properties of the Dyneema line are given in Table 5-1.

Table 5-1 Dyneema lines

Material	EA [kN]
Dyneema line	100,000

The lines will approach the Dockwise Vanguard almost horizontally with only a small angle, because the clearance between the cargo and the Dockwise Vanguard is small compared to the line length. The cargo will be connected to the actuators with 64 [mm] Dyneema lines. It is assumed that these lines have a safe working load of 2000 [kN], in this thesis. An unwanted situation during the operation occurs if the lines break. In this case the motions of the cargo cannot be controlled anymore, inducing the risk on collisions. However people are also working in the vicinity of the actuators. It is possible that the breaking line hits someone, this should be prevented. To guarantee the operation to be performed safely the safe working load is introduced. This value is obtained by dividing the minimal breaking load of the line through a safety factor. In the offshore industry usually a safety factor of 2 is applied. Different types of lines possess a different safe working load. In this phase of the design process it is not yet exactly known which type of line will be used as this thesis focuses on the workability of the system. It is therefore assumed that the assumption is correct for the purposes of this thesis.

Due to the location of the casings and the shape of the cargo and the location of the bollards on the cargo, the Dyneema lines will have different lengths. The stiffness of the lines is calculated according to equation (5-25) and the stiffness of the lines is presented in Table 5-2.

$$k_{line} = \frac{EA}{L} \tag{5-25}$$

Table 5-2 Mooring line properties

Actuator	Length [m]	Stiffness [kN/m]
1	58	1724.1
2	35	2857.1
3	58	1724.1
4	35	2857.1

Case 1: Hydraulic actuator 1

In this section the performance of hydraulic actuator 1 is investigated. The actuator needs to deliver a force of approximately 4000 [kN]. Therefore the actuator is equipped with 2 Dyneema lines of 58 [m].

Figure 5-17 shows the relation between the system and the proportional gain values. It can be concluded that a proportional gain value of $1{\times}10^{-5}$ gives the smallest line tension error between the required line tension and the obtained line tension. This error is caused by the overshoot in tension produced by the proportional controller. Therefore this system will be modelled for a K_p of $1{\times}10^{-5}$, as this is the best possible fit which can be obtained at this moment.

Figure 5-18 shows a zoomed overview of the required line tension and the obtained line tension of the actuator. Figure 5-19 shows the difference between the obtained and required line tension. It is clearly visible that the error in the line tension is generally in the order of ± 10 [kN]. These errors are caused by a sudden change in the trajectory of the required force. This is shown in Figure 5-21 which is a close-up of the line tension figure around time 159 [s]. The required tension in the line reduces very fast and is then instantly kept at 10 [kN]. This causes the proportional controller to produce an overshoot in the force. In time the error reduces and thereby also the error in the line tension. Also larger errors in the line tension are shown, for example around time 1307 [s] (see Figure 5-22). It is clearly visible that the error occurs at a time when the tension in the line needs to be increased, the error occurs when the trajectory of the required line tension changes abruptly. The more abrupt the motions of the piston need to be changed, the larger the line tension error will be. This can be explained with Figure 5-20 in which the typical motion of the piston is shown. The motions of the piston are dependent on the stretch in the mooring line which in turn is dependent on the distance between the piston and the cargo. However due to the wave forces, the cargo is moving in surge, sway and heave direction. Therefore the piston is always moving, even when the tension in the line is kept stable. The value of the occurring error is dependent on the sharpness of the angles in the motions of the piston. If a sharp angle occurs the error will be large, because the motion of the piston needs to be reverted instantly, while if a soft angle is observed the direction of the motion stays the same, only the piston velocity needs to change. This causes a smaller error because the change in the valve actuator signal is smaller. The stroke of the piston during the operation is 1.3 [m].

The peak flow rate of the fluid during the operation is 0.26 [m^3/s] (see Figure 5-23). The flow rate is dependent on the displacement of the piston which is needed to obtain the required tensions in the lines. The flow rate is also dependent on the velocity in which the required tension needs to be obtained. The power required to produce the $pump_{pressure}$ of 300 bar is calculated by multiplying the $pump_{pressure}$ with the flow rate. In this case it is assumed that the pressure between the pump and the valve is constantly at 300 bar. However in reality the pressure will drop, because fluid flows away. Therefore the calculation of the power is done conservatively. The required power during the operation is shown in Figure 5-24. It is found that the needed peak power 7.9 [MW].

Figure 5-25Figure 5-24 shows the pressure inside the cylinder. It is clearly visible that the pressure in the cylinder stays well below the $pump_{pressure}$. The dimensions of the actuator are chosen such that a relative large difference between the $pump_{pressure}$ and the pressure in the actuator occurs. This pressure difference is needed to be able to change the position of the piston quickly, as the piston velocity is dependent on the flow rate of the fluid, which is dependent on the pressure difference between the pump and the chamber.

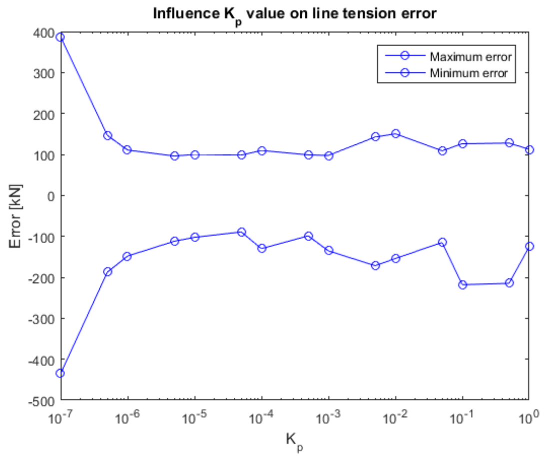

Figure 5-17 Influence proportional gain on line tension error, case 1: hydraulic actuator 1

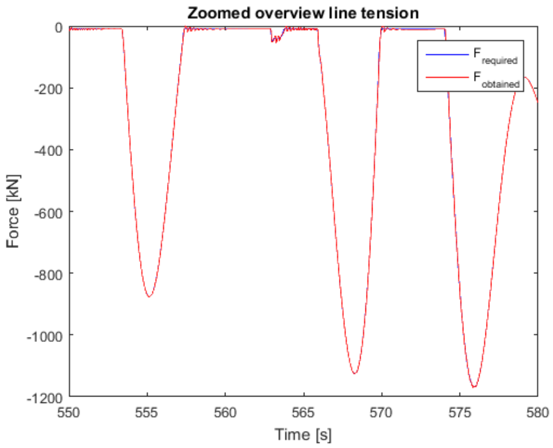

Figure 5-18 Zoomed overview line tension, case 1: hydraulic actuator 1

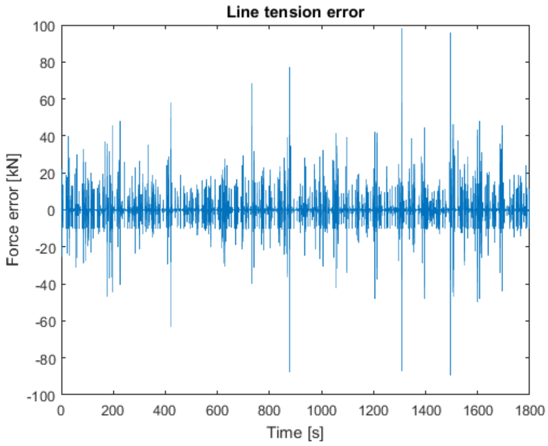

Figure 5-19 Line tension error, case 1: hydraulic actuator 1

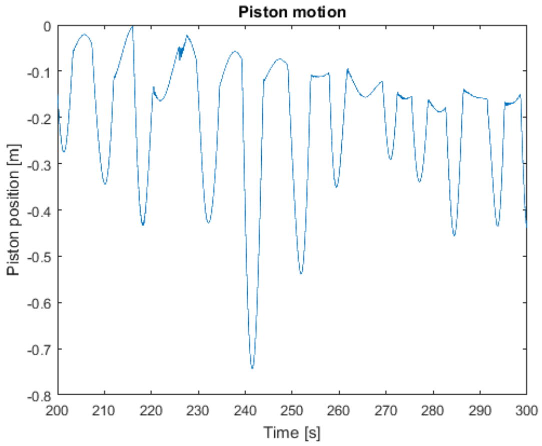

Figure 5-20 Typical piston motion, case 1: hydraulic actuator 1

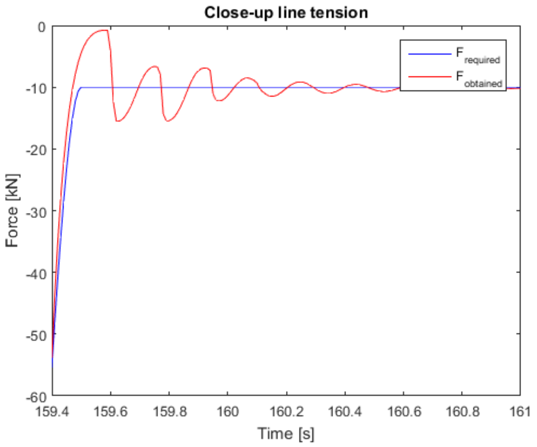

Figure 5-21 Close-up line tension around t = 159 [s] , case 1: hydraulic actuator 1

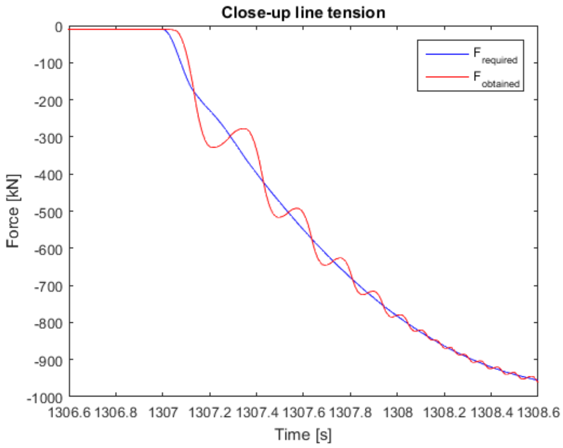

Figure 5-22 Close-up line tension around t = 1307 [s] , case 1: hydraulic actuator 1

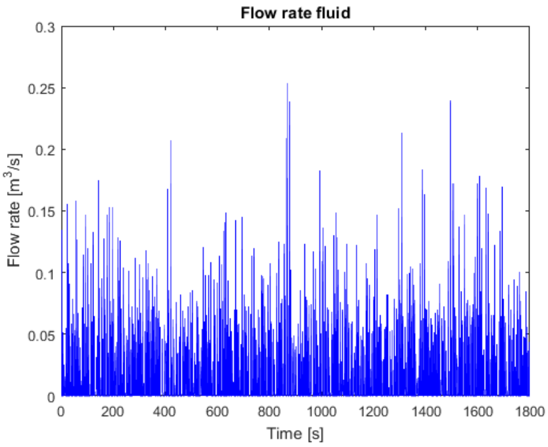

Figure 5-23 Flow rate fluid, case 1: hydraulic actuator 1

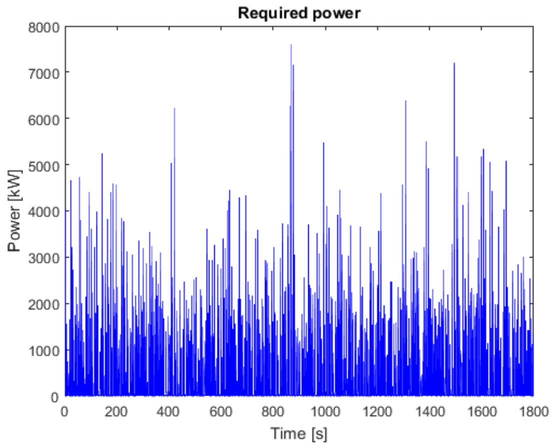

Figure 5-24 Required power, case 1: hydraulic actuator 1

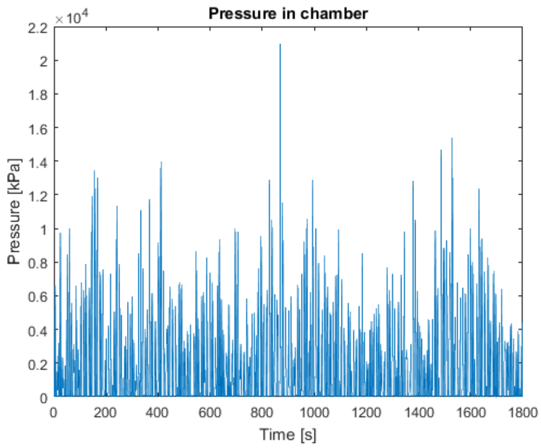

Figure 5-25 Pressure in chamber, case 1: hydraulic actuator 1

Case 2: Pneumatic actuator 1

In this section the performance of pneumatic actuator 1 is investigated. The actuator needs to deliver a force of approximately 4000 [kN]. Therefore the actuator is equipped with 2 Dyneema lines of 58 [m].

Figure 5-26 shows the relation between the proportional gain value and the error which occurs in the system. The values of the upper line occur when the piston heaves out (see Chapter 5.4). It is clearly visible that a gain value of 1×10^{-5} produces the smallest error, therefore it is recommended to use this value.

Figure 5-28 shows the line tension error which occurs. The positive values in the graph occur when the piston needs to move out of the cylinder. These larger errors occur, because the pressure in the chamber reduces and the gas will expand, slowing down the outgoing motion. However the error for the ingoing motion is relatively small compared to case 1. The pressure at the supply side of the valve is higher than the pressure in the cylinder, which is the reason that the gas flows from the pump to the cylinder. However this decrease in pressure, which the gas experiences, causes the gas the expand in the cylinder. Therefore the required line tension is produced relatively fast. This effect is more clearly illustrated in Figure 5-27, which also shows the positive effect of the delay in the obtained line tension. The delay of the obtained force occurs at the lower force range, when the required force is approaching its constant value. The angle at which the obtained force reaches the constant force is less sharp than in the hydraulic system. Therefore the system settles around this value with fewer oscillations compared to the hydraulic actuator.

The required tension and line stiffness is the same as for case 1, therefore the motion of the piston is the same, the piston stroke is 1.3 [m]. Also the pressure inside the cylinder is the same as for case 1 for this reason, as the involved forces are the same. The required flow rate of the gas should be approximately the same as for the hydraulic case. However the mass flow rate will be different as another type of fluid is used in the pneumatic system. The mass flow rate of the gas is dependent on the pressure in the cylinder chamber, as compressed gas is heavier than non-compressed gas. The flow rate of the gas can be calculated by dividing the mass flow rate, out of the pneumatic pump unit, through the density of the gas. The density of the gas can be calculated with equation (5-11). For this calculation it is assumed that the temperature of the gas leaving the pump is constantly 293 [K]. The obtained flow rate is presented in Figure 5-29. The peak flow rate of the fluid during the operation is 0.27 [m^3/s] and the average flow rate is 0.008 [m^3/s]. Therefore it is concluded that the compressibility of the gas has minor influences on the volumetric flow and consequently the power. The amount of power needed to produce the high pressurized gas is calculated with the same method as in case 1. The corresponding peak power in the system is 8.2 [MW] and the corresponding average power is 0.25 [MW] (see Figure 5-29). The largest flow rates occur when the required line tension changes from a stable value to a rising value. In this case the pressure inside the cylinder needs to increase quickly so a larger flow rate is needed.

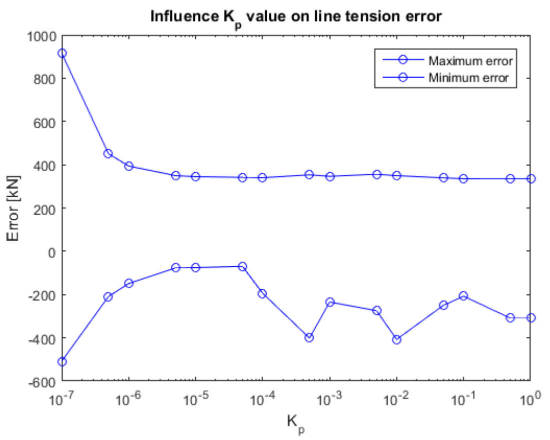

Figure 5-26 Influence proportional gain on line tension error, case 2: pneumatic actuator 1

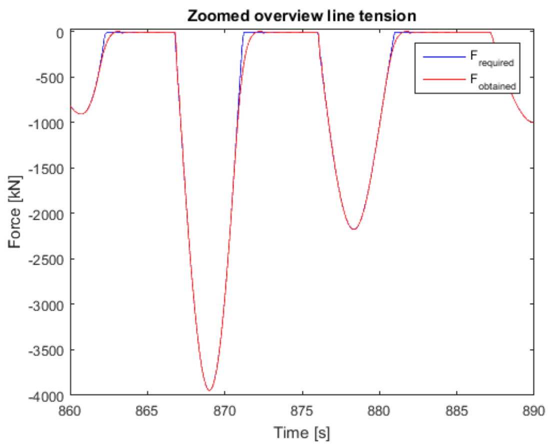

Figure 5-27 Zoomed overview line tension, case 2: pneumatic actuator 1

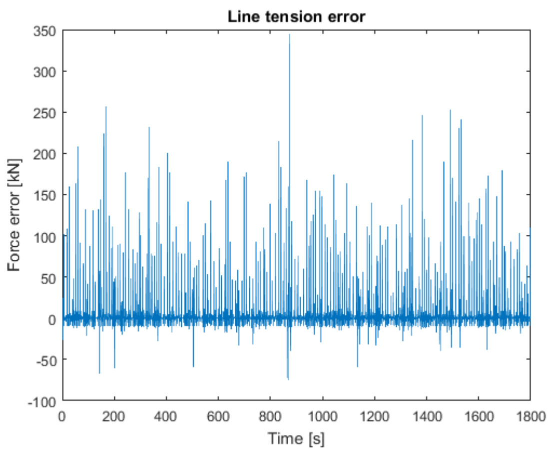

Figure 5-28 Line tension error, case 2: pneumatic actuator 1

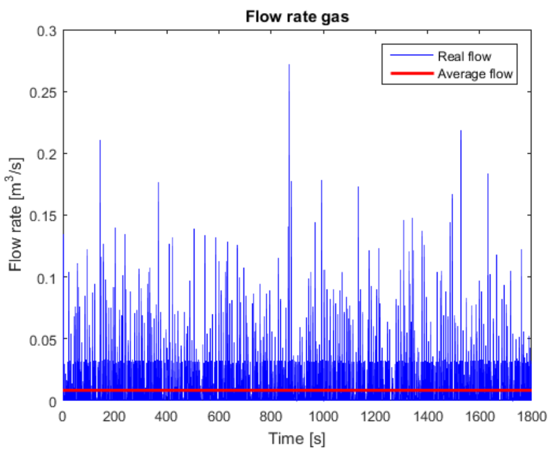

Figure 5-29 Flow rate gas, case 2: pneumatic actuator 1

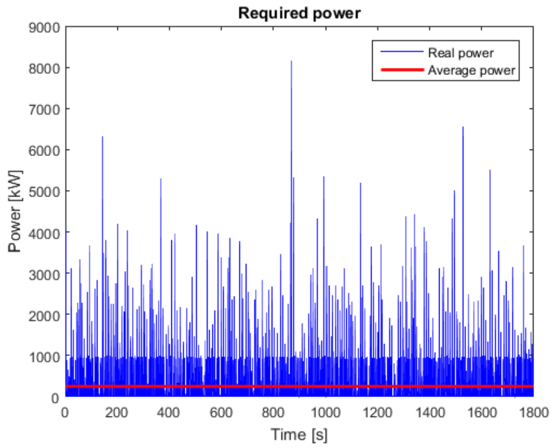

Figure 5-30 Required power, case 2: pneumatic actuator 1

Comparison hydraulic and pneumatic actuator

The simulation results of the hydraulic and pneumatic actuator are compared to each other, in this section. Furthermore the feasibility of the two systems is investigated.

It is noticed that the frequency of the oscillations (caused by the proportional controller) are very high (approximately around 4 Hz see Figure 5-21 and Figure 5-22) in the case of the hydraulic actuator compared to the pneumatic actuator (see Figure 5-27). This is largely caused by the different bulk modulus's of the fluids. Hydraulic fluids have a high bulk modulus while the bulk modulus of gasses is low. Therefore hydraulic fluids are almost incompressible, leading to a minimum of spring action. Pneumatic systems are highly compressible, leading to a large spring action. This causes the pneumatic system to develop a delay to the required force which is a positive development for the 3-way valve because the motions of the valve are reduced. In reality the frequency of these oscillations will be lower as the valve position will be changed at a slower speed. The required force will be calculated ones every second, for example. A discrete signal will be obtained instead of a continuous signal which is used for these simulations. It is fine to use a discrete signal because the motions of the large HTV-cargo system will not be influenced by these high frequent oscillations in the piston position. The error in line tension which is induced by these oscillations is relatively small compared to the required tension.

It is also noticed that the amounts of power the systems need are approximately the same, the compressibility of the fluid has a minor influence on the fluid flow. However the required peak powers need to be delivered to the actuators very fast. Unfortunately no existing hydraulic or pneumatic pumps exists which are capable of doing this. An advantage of the use of a pneumatic system is that stored energy can be used. High pressurised gas is stored in large tanks, which deliver the peak flow rates. These tanks will be filled by a pneumatic pump which pumps an average amount of flow in the cylinder. In this way a lot of power can be saved. It is possible to deliver this average amount of flow.

In the simulations the pressures in the supply and return tank are kept constant, in reality the pressures will change. The pressure in the supply tank reduces when an amount of gas leaves it. In order to still deliver the correct line tension the area of the valve needs to be increased, to deliver the same flow rate at a different pressure. The control system needs to adjust this. However as long as the supply pressure is higher than the cylinder pressure, the system will be able to deliver the correct amount of tension. As the flow leaves the chamber it will enter the return tank. Therefore the pressure in the return tank will rise during the operation. As the system already has problems to deliver the required force at the outgoing motion of the piston, the rise in pressure in the return tank will make this effect worse. Therefore the tank should be large enough to accommodate the inflow without a large pressure increase. Furthermore, the dimensions of the tank should be such that the tank is not vacuumed by the pump unit which pumps the nitrogen from the return tank into the supply tank.

The amount of flow which leaves the tank varies in time and the amplitude of the flows is different over time. It is possible that a cycle with a large flow is followed by a small flow. It is also possible that three large amplitude flows occur before a small amplitude flow occurs. Therefore the flow rate of the pump will be larger than the calculated average flow, because if some large outflows occur after each other, the amount of flow which left the chamber is too large to be refilled in time. The supply pressure becomes too low, causing the system to be unable to reach the required line tension in time. This will have an influence on the relative horizontal motions. Therefore the $pump_{power}$ should be larger than the average amount of power. However it can be less than the maximum amount of power, because the pump does not have to produce the peak flow rates. For this reasons only the pneumatic actuator is considered to be a technical feasible option.

Case 3: Different line stiffness

In this section the effect of the line stiffness on the performance of the system is investigated. Pneumatic actuator 1 is simulated if it is equipped with 2 or 4 Dyneema lines of 58 [m] (stiffness is 3448.2 [kN/m] of 6896.4 [kN/m]).

Figure 5-31 shows a zoomed overview of the required line tension and the obtained line tensions of the two systems. Figure 5-32 shows the line tension error of the two systems. It is noticed that the line tension errors decrease a little if a stiffer mooring configuration is used. It is likely that the overshoot is reduced due to the smaller distance the piston needs to change to provide the required force. The piston stroke is 1.3 [m] for the soft system and 0.74 [m] for the stiff system. It is also concluded that the two systems react a little bit different to the proportional controller as the period of the oscillations in the obtained line tension for the stiff system are a little bit longer than for the soft system.

The change in the needed piston displacement has an effect on the amount of gas flow which is needed to deliver the required force (see Figure 5-33). The gas flow rate for the soft system is 0.008 [m^3/s] and for the stiff system 0.005 [m^3/s]. Consequently also the required power changes (see Figure 5-34). The needed peak power for the soft system is 8.2 [MW] while for the stiff system it is 5.5 [MW]. This effect is also noticeable in the average power, the soft system needs 0.25 [MW], the stiff system needs 0.16 [MW].

As the relation between tension and stiffness is linear (F = kx), it was expected that the needed piston stroke halved. The tension stayed the same and the stiffness became twice as large. Consequently the remaining values also should be halved. However this is not the case which is caused by the relative motions of the cargo. These stayed the same for the two systems. The piston also needs to move to keep the minimum tension in the line, the amplitude of the motions of the piston and the cargo are the same in this case. However in reality the relative motions of the cargo should be reduced for the stiff system. As it is uncertain how the active tension control system of Lee reacts to this change in line stiffness it is decided to keep the cargo motions the same for the two systems. Therefore the needed flow rate and power are larger than expected. The difference in response to the proportional gain also influences the needed flow rate. However this effect is small compared to the relative cargo motions. It is likely that the power needed for the stiff system is half of the power needed for the soft system, if the relative motions of the system are reduced as is likely to happen in reality.

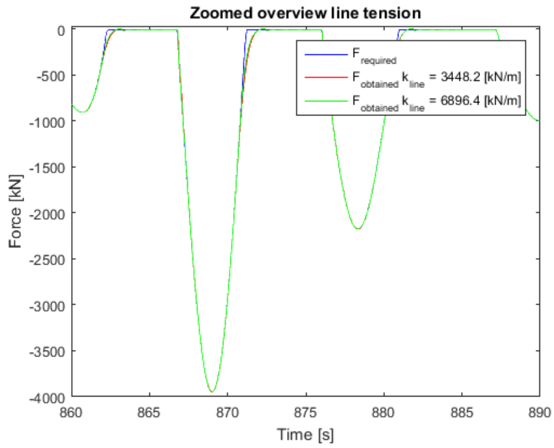

Figure 5-31 Zoomed overview line tension, case 3: different line stiffness

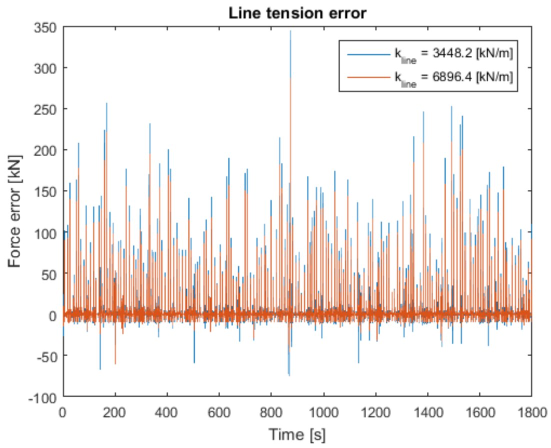

Figure 5-32 Line tension error, case 3: different line stiffness

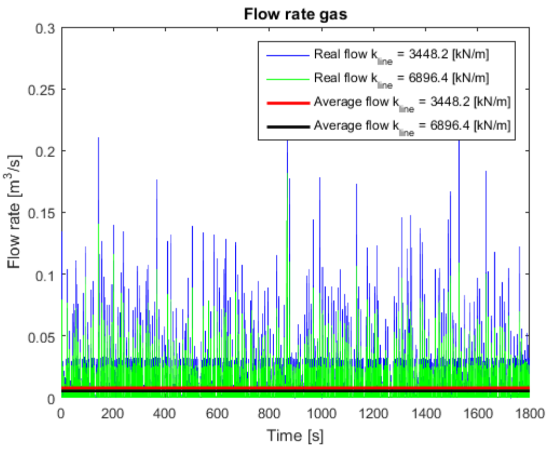

Figure 5-33 Flow rate gas, case 3: different line stiffness

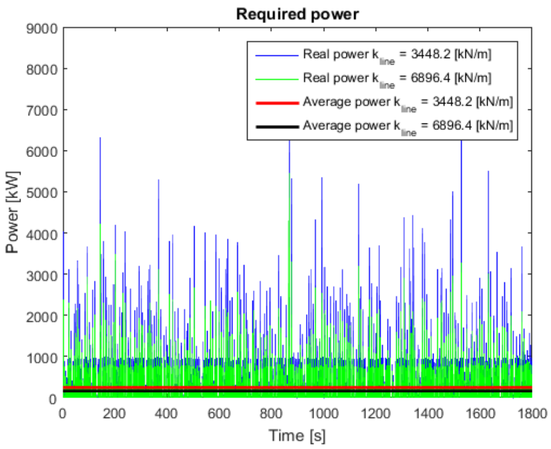

Figure 5-34 Required power, case 3: different line stiffness

Complete system

In the previous sections the difference between a hydraulic and pneumatic operated system and the influence of the line stiffness is investigated. However it is also important to investigate the performance of the complete system, which consists of four pneumatic actuators. Therefore the results of the simulations for actuator 1 and 2 need to be combined. The results for actuator 1 and 3 are similar as is the case for actuators 2 and 4. The *Line tension actuator* system is symmetrical around the longitudinal axis for the simulation case. For the simulation results of pneumatic actuator 2 see Appendix E.5.

In Figure 5-35 a few oscillations are shown for the required and obtained line tension by the *Line tension actuator*. It is noticed that the large differences between the required force and the obtained force occur around the equilibrium position of the cargo. The effect of this line tension error on the dynamics of the HTV-cargo system at this location is negligible. The delay occurs at the moment when the actuator at the other side of the cargo starts to increase the tension in the lines and when the cargo is positioned above the desired location on the deck of the HTV. Furthermore the tension error is small compared to the forces involved in the dynamics of the system. Due to the delay less force oscillations occur in the system, which is positive for the valve. The motions of the valve are reduced which decreases the wearing out of the valve. Therefore the positive effects outweigh the negative effects.

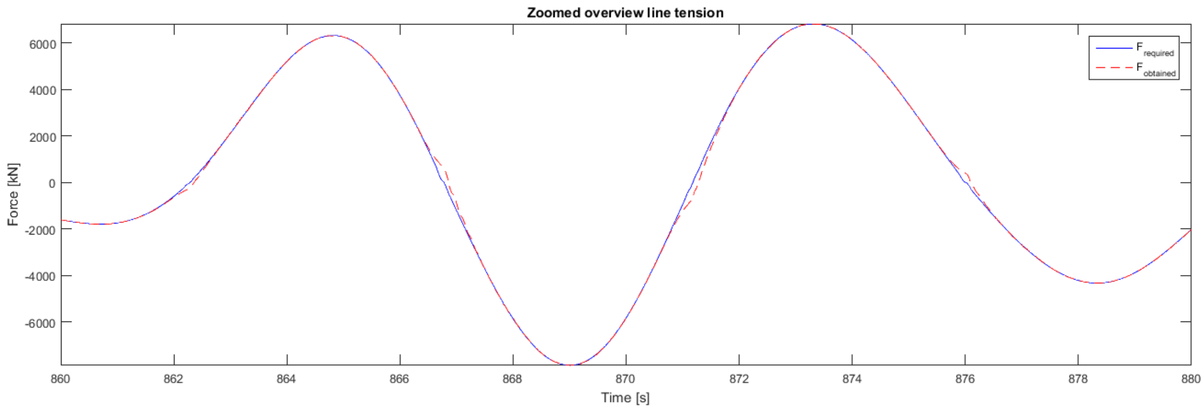

Figure 5-35 Zoomed overview line tension complete system

Another method to reduce the occurrence of these oscillations is by improving the control system. It is recommended to investigate the use of a feed forward and feedback control or a combination of these systems to smoothen the sharp change in the required line tension. A schematic overview of this is provided in Figure 5-36. The green lines represent the required tensions of actuator 1 and 2 as they are in the performed simulations. The red lines represent the required tension when they can be obtained in a more smooth manner. In this case also the allocation of the forces needs to be optimised. If the tension of actuator 1 is decreasing at a changed pace, actuator 2 needs to start pulling earlier to maintain the tension difference correctly. In this way, it is likely that the cylinder will produce the required tension without showing high frequency oscillations.

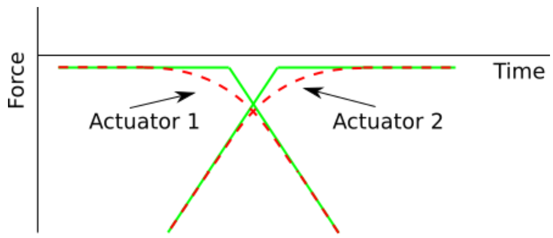

Figure 5-36 Schematic overview smoothening required line tension

The system uses four actuators of which two need to deliver an increasing line tension, while the two other actuators are kept at a constant tension. The peak flow rate is approximately 0.6 [m^3/s] and the corresponding peak power is approximately 17 [MW]. The average flow rate is approximately 0.2[m^3/s] and the average power is 0.55 [MW]. It is possible to acquire a pump which can deliver these average flow rates, therefore the *Line tension actuator* is considered to be technical feasible.

5.6 Conclusion

In this chapter a clear overview is given of the outlook and working principle of the *Line tension actuator*. The reduction of the relative motions is accomplished through an increase in the line tension by an actuator. The lines connect the HTV and the cargo. The actuator consists of a piston which moves inside a cylinder. The amount of tension which the system needs to deliver is calculated by the active tension control system of Lee [6], therefore the focus of this research is on the design of the actuators. Two different kind of actuators are investigated, a hydraulic and a pneumatic one. The actuators are controlled by a proportional control system.

At first the interaction between the controller and the actuator is investigated. In this way the characteristic behaviour is obtained. The controller is connected to a valve actuator which controls the position of the three way directional valve. This valve controls the fluid flow in and out of the actuator, allowing the actuator to produce a certain line tension. The gain value of the proportional controller (K_p) has a large influence on the force the actuator produces. If the gain value is set too low, the system will react with a delay. If the gain value is set too high, the system reacts very aggressively. Therefore the optimum value of the gain needs to be derived. It also takes time to move the valve to another position, this is displayed with a time constant. If this value is low (in the order of hundredths of a second or lower) the valve moves quickly. If this value is high (in the order of seconds) the valve reacts slowly. A slow reacting valve adds damping to the system. Therefore larger K_p values need to be used to obtain the correct force in time. If, on the other hand, the valve reacts quickly, lower values of K_p need to be used because the system reacts relative fast, producing large overshoots. The time constant of the valve and the proportional gain value are interconnected. The required force, the used line stiffness and the type of fluid which is used, do not have an influence on the interaction between the controller and actuator. They do not influence the amount of fluid flow which is able to flow.

The performed simulations show that the fluid flow rates for a hydraulic and pneumatic system are approximately the same. Therefore the needed peak powers of the two systems are also approximately the same; 17 [MW] for the complete system. The peak power for one actuator is approximately 8 [MW]. Unfortunately no existing hydraulic or pneumatic pumps exists which are capable of delivering this. An advantage of the use of a pneumatic system is that stored energy can be used. The high pressurised gas is stored in a pressure bank which delivers the peak flow rates to the actuator. The pressure bank is filled by a pneumatic pump which pumps an average amount of flow in the pressure bank. The pump is working on an average amount of power. The average amount of power needed for the complete system is approximately 0.55 [MW]. It is possible to acquire a pump which is capable of delivering this average flow. Therefore only the pneumatic actuator is considered to be a technical feasible solution.

It is recommended to include the design of a pressure bank in the system. A pressure bank consists of a high pressure supply tank, a low pressure return tank and a pneumatic pump unit which connects the two tanks. Gas will flow in and out of the tanks and this causes the pressure in the tanks to change during the operation. The amount of flow which leaves the tank varies in time and the amplitude of the flows is different over time. It is possible that a cycle with a large flow is followed by a small flow. It is also possible that three large amplitude flows occur before a small amplitude flow occurs. Therefore the flow rate of the pump will be larger than the calculated average flow. If some large outflows occurred after each other, the amount of flow which left the chamber is too large to be refilled in time. This causes the pressure to be too low, causing the system to be unable to reach the required line tension in time. Therefore the $pump_{power}$ should be larger than the average amount of power. A lot of effort should be directed to the design of the return tank. The pressure in this tank cannot become too high because the tension in the line will not be reduced enough. On the other side the dimensions of the tank should be such that the tank is not vacuumed when the nitrogen is pumped out of it to the high pressure tank. It is therefore recommended to investigate the use of an accumulator, which is able to keep the pressure in the return tank constantly at atmospheric pressure.

It is noticed that an error occurs between the required force and the obtained force by the actuator. These errors are smaller for the increasing line tension and larger for the decreasing line tension. The errors are caused by the compressibility of the gas in combination with the used control system. A proportional controller will always have a steady state error. The large compressibility of gasses leads to a spring action in the system. During the increase in tension this effect is favourable as it reduces the error caused by the control system. During the decrease in tension this effect causes the system to show a delay in the obtained tension compared to the required tension. The positive effect of this delay is that the pneumatic system reaches the constant line tensions very smoothly. Therefore fewer oscillations occur in the obtained force, which means that the opening and closing frequency of the valve is less, which is better for the valve. Therefore it is recommended to design a control system which is able to reduce these oscillations by smoothening the required line tension. A few examples of systems which can be used or combined are feedback and feedforward. The feedback system is used to learn from the past and to predict when the sharp angles occur. The feedforward system is used to predict the occurrence of the sharp angles and to anticipate on it. These control parts can be implemented in both the tension control system and the *Line tension actuator*. Further research is needed to determine the best option. Another possibility to reduce the line tension error is to increase the value of the minimum line tension. This causes the pressure difference between the cylinder and the return tank to increase. The pressure difference has a large influence on the flow rate of the outgoing gas. The spring action occurs when this pressure difference is small. It is expected that the spring action reduces when the pressure difference is larger. However more research is needed to confirm this.

The presented system is only a preliminary design, and the simulations are performed for only one case. It is recommended to extend the number of simulations to include more cases to obtain a system which can be used for different types of cargo in different environmental conditions. On top of this the operational side of the system needs to be investigated: How are the Dyneema lines connected to the cargo and the actuator, How is the change in line length due to the de-ballasting process accounted for in the system, How are the minimal tensions in the lines obtained?

6 Conclusion and recommendation

As the exploration areas of large offshore structures are located at more remote locations, Dockwise wants to perform its loading and discharge operations at more harsh locations. However the relative horizontal motions, which occur between the HTV and the cargo, should be within certain limits to ensure safe and reliable operations. Therefore the occurring motions need to be reduced. This resulted in the following objective for this thesis: *To develop a technical feasible system, capable of reducing the relative horizontal motions of the cargo with respect to the HTV during offshore loading or discharge operations.* To obtain this objective different designs are created and simulations are conducted. If these results are carefully examined, conclusions can be drawn and recommendations for future research studies are provided.

6.1 Conclusion

At first the characteristics of the standard cargo handling system were investigated. From the obtained results it was concluded that the standard cargo handling system is unable to reduce the relative horizontal motions such that the cargo is positioned on the correct location. The maximum allowed amplitude of the relative horizontal motions is 0.15 [m]. The motions for the standard cargo handling system are 0.3 [m] in surge direction and 0.4 [m] in sway direction, if the operation is performed in head waves. Therefore it is concluded that the standard cargo handling system is unable of controlling the motions during an offshore loading operation. A new cargo handling system needs to be designed.

During the first phase of the design trajectory multiple solutions were considered. The feasible designs were compared to each other with a Multi Criteria Analysis (MCA). With the help of this analysis it was decided that a C*lamping system* and a *Line tension actuator* are the most likely systems to solve the offshore loading and discharge problem.

The clamping system consists of a clamping structure and a damping device. The damping device consists of a fluid damper combined with a control valve and a fender. The fender is used to absorb the impact loads at the start of the clamp operation and to pull the piston to its outer position during the operation. The control valve is used to change the damping and stiffness characteristics of the system over time. The clamping structure is used to push the damping devices against the cargo. It is found that the system is capable of reducing the amplitude of the relative horizontal motions well below the maximum allowed amplitude of 0.15 [m] in regular and irregular sea states. If is also obtained that the performance of the system is dependent on the mass of the HTV and the cargo, as well as the wave forces. The amount of damping which is generated is dependent on the velocity of the piston of the damping device. The piston velocity is dependent on the relative velocity of the cargo. This causes the system to respond differently for different cases. Finally it is concluded that the dimensions of the damping device are realistic. Therefore the clamping system is considered to be a technical feasible solution.

The *Line tension actuator* consists of a pneumatic or hydraulic actuator and a control system. The principle of this system is to reduce to relative horizontal motions through an increase in the line tension by an actuator. The lines connect the HTV and the cargo. The actuator consists of a piston which moves inside a cylinder. In this thesis the system consists of 4 actuators. It is found that large peak powers are needed to produce the required line tension needed to reduce the relative horizontal motions. The peak power for the complete system is 17 [MW] and 8 [MW] per actuator. Only two actuators are producing the required peak tension at the same time. Unfortunately no existing hydraulic or pneumatic pumps exists which are capable of delivering this. An advantage of the use of a pneumatic system is that stored energy can be used. High pressurised gas is stored in a pressure bank which delivers the peak flow rates to the pneumatic actuator. A pneumatic pump is used to refill the pressure bank with an average fluid flow. The needed power to produce this flow is 0.55 [MW] for the complete system. It is possible to acquire a pump which can deliver this. Finally it is concluded that the dimensions of the pneumatic actuator are realistic. For the before mentioned reasons only the *Line tension actuator* with a pneumatic actuator is considered to be a technical feasible solution.

6.2 Recommendations

As it is expected that the *Clamping system* and the *Line tension actuator* are both able to minimize the relative horizontal motions, it is advised to continue the design process for both systems.

The following recommendations are made for the clamping system:

- The simulations are conducted for a non optimised system, this causes large impact forces to occur in the system. It is therefore advised to investigate the relation between the cylinder dimensions, relief pressure of the cylinder and the pre-compression of the fender. The amplitude of these impact values should be reduced to protect the cargo hull from the system penetrating it.
- The simulations are conducted for a preliminary designed system, which focused on the working principle of the system. It is therefore advised to take this design to the next phase in the design trajectory and to include the design of the clamping structure. The clamping structure forms an important part of the system as the damping device is mounted on it and the clamping structure determines the position of the cargo on the deck of the HTV.
- The simulations for this design are done for a cargo which only had 1 degree of freedom. In reality the HTV-cargo system will have 12 degrees of freedom in which it can move. Depending of the layout of the relative horizontal motion reduction system, the C*lamping system* needs to cope with at minimum half of these degrees of freedom. The system has 6 degrees of freedom in the horizontal plane and the system is operating in the horizontal plane. It is therefore recommended to extent the amount of degrees of freedom in steps to see how the system responds in reality.

The following recommendations are made for the stretch compensation system:

- Improve the control system which controls the amount of line tension and the spool position of the 3-way directional valve. The compressibility of the gas causes a spring action to occur which is most visible when the line tension needs to decrease; a delay in obtained line tension occurs. This causes an error to occur in the line tension. Furthermore a proportional gain will always have a steady state error. Therefore it is advised to implement a control system which removes this steady state error. As the largest error occurs when the line tension approaches the constant line tension value it is advised to investigate the effect of increasing the minimum line tension. However the downside is that less capacity is available to cope with the motions of the cargo. Another option is to include a feedforward and feedback control system in the *Line tension actuator*. In this way it is possible to predict sudden large changes in the force trajectory and to anticipate on it.
- It is also recommended to investigate the use of a pressure bank. The pressure bank is used to create a closed loop pneumatic system and to produce the required peak flows. The power bank is not yet included in the simulations for the stretch compensation system.
- The presented system is only a preliminary design and the performed simulations focused on the working principle of the system. It is therefore recommended to take the design to the next phase in the design trajectory. It is also recommended to include the operational side of the system.

In this research the wind and current loads are neglected; the HTV and the cargo are merely excited by the waves. If these loads are incorporated in further research studies a more realistic model will be obtained.

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Appendix A Background information

This appendix will provide background information to the offshore loading and discharge problem. The first section provides an overview of the types of cargo which are transported by HTVs. The second section explains the sequence of the loading and discharge operations. The third section discusses the supporting structures which guide the horizontal and vertical loads working on the cargo to the hull of the HTV. The fourth section discusses the layout of the standard cargo handling system. The fifth section provides a more elaborate discussion of the previous designed systems.

A.1 Cargo

The scope of this thesis is to design a cargo handling system which can handle all types of cargo during the offshore loading and discharge operations. Therefore a general solution is required. It needs to be known which types of cargo are used offshore and what kind of limitations on the cargo can be expected.

Types of cargo

Dockwise has transported the following types of cargo in the recent years, the cargoes in bold are likely to be loaded and discharged at offshore and inshore locations:

- Barge,
- Crane,
- **Dredging equipment (Cutter suction dredger)**,
- **FPSO**,
- **Floating production unit (FPU)**,
- **Jack-up**,
- Lift boat,
- Multicat,
- Module for onshore industrial project (LNG, Mining),
- Navy vessel (Frigate, Submarine),
- **Semi-submersible (Drilling rig, Production unit, Storage unit)**,
- Ships,
- **Spar**,
- Topside,
- **Tension-Leg Platform (TLP)**.

It is important to know what the dimensions and weights of the different types of cargo are, to confirm that the Dockwise Vanguard (the largest HTV currently available) is able to transport these cargoes. The dimensions are gathered from different sources.

FPSO

For the FPSO case (see Figure A- 2) it is important that the width of the FPSO is smaller than the width of the Dockwise Vanguard, because FPSOs are large structures which can hang over the bow and stern of the Dockwise Vanguard. This is only possible when the cargo is positioned in between the casings and the accommodation block. Furthermore the Dockwise Vanguard should be able to carry the FPSOs. Therefore the limiting factors for transportation are the width and the weight of the FPSOs.

In Figure A- 1 the weight of the FPSOs plotted is against the width of the FPSO, also the carrying capacity of the Dockwise Vanguard is plotted against the maximum deck width of the Dockwise Vanguard. As can be seen in the plot, the Dockwise Vanguard is able of transporting most of the FPSOs which are currently in operation.

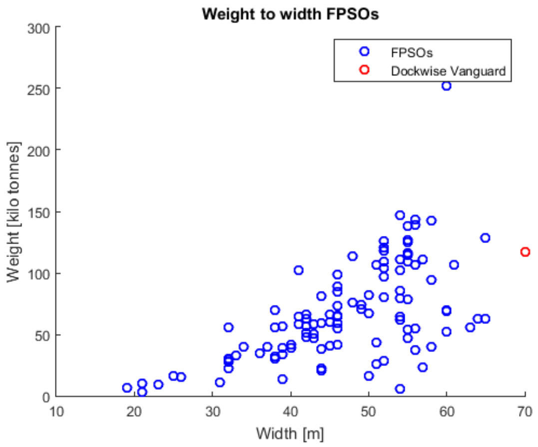

Figure A- 1 Weight to Width of FPSOs in operation [23]

Figure A- 2 FPSO on board of the Dockwise Vanguard [24]

Spar

Spars (see Figure A- 4) can be longer than the Dockwise Vanguard. Therefore the same applies for the Spar as for the FPSO case.

As can be seen in Figure A- 3 is the Dockwise Vanguard capable of transporting almost all the spars which are currently in operation.

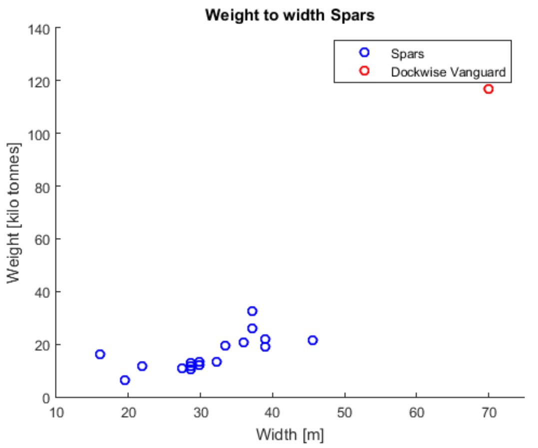

Figure A- 3 Weight to Width of spars in operation [25]

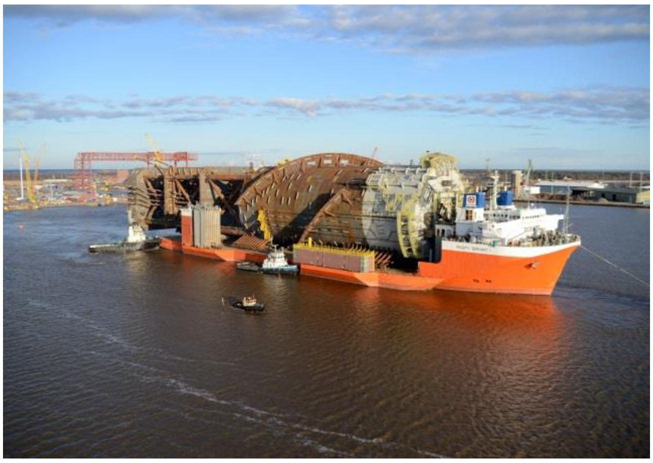

Figure A- 4 Spar on board of the Mighty Servant 1 [24]

Jack-up

Jack-ups (see Figure A- 6) are often triangular shaped structures, which implies that the length is almost equal to the width of the structure. Therefore it is possible that the width of the structures is wider than that of the deck of the Dockwise Vanguard. However the structures may hang over the sides of the deck. This means that only the deck space between the aft casings and the accommodation block can be used. The available deck length is 155.5 meter.

In Figure A- 5 the weight of the jack-ups is plotted against the length of the jack-ups, also the carrying capacity of the Dockwise Vanguard is plotted against the maximum deck width of the Dockwise Vanguard. As can be seen in the plot the Dockwise Vanguard is able to transport all the spars which are currently in operation.

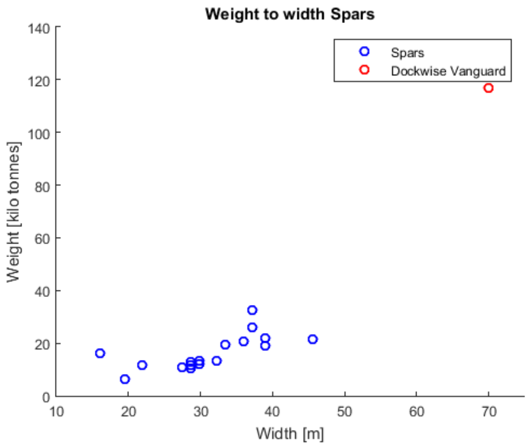

Figure A- 5 Length of jack-ups in operation [26]

Figure A- 6 Jack-up on board of the Transshelf [24]

Semi-sub

Semi-subs (see Figure A- 8) can have a larger width than the deck width of the Dockwise Vanguard. Therefore the same applies for the semi-sub as for the jack-up.

As can be seen in Figure A- 7 the Dockwise Vanguard is capable of transporting almost all the semi-subs which are currently in operation.

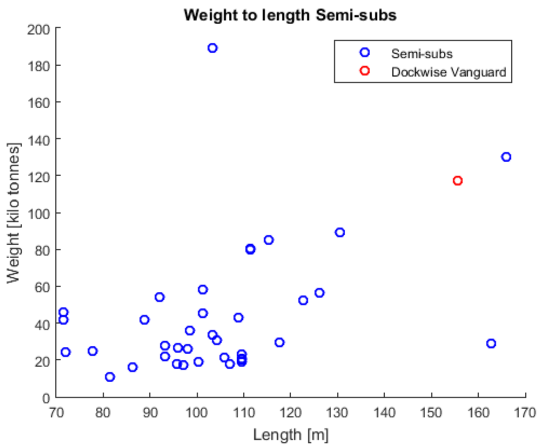

Figure A- 7 Weight to length of semi-subs currently in operation [27]

Figure A- 8 Semi-sub on board of the Blue Marlin [24]

TLP

TLPs (see Figure A- 10) can also be wider than the deck of the Dockwise Vanguard. Therefor the same applies for the TLPs as for the semi-subs and the jack-ups.

As can be seen in Figure A- 9 the Dockwise Vanguard is capable of transporting almost all the TLPs which are currently in operation.

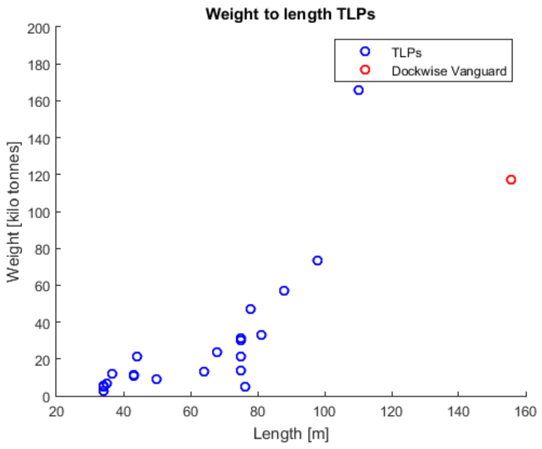

Figure A- 9 Weight to length of TLPs currently in operation [28]

Figure A- 10 A TLP [29]

Limitations

Each type of cargo provides a number of limitations for the cargo handling system and the supporting layout. These limitations consist of protrusions and ambiguities in the drawings of the cargo. The drawings are provided by the owner of the cargo, but sometimes changes are made to the design of the cargo during maintenance projects and the drawings are not adjusted to these changes.

The following protrusions can be found on the structures:

- Anchors,
- Anchor attachment points,
- Anodes,
- Bilge keels,
- Boat landing points,
- Inclined sides,
- Mud mads,
- Pipes,
- Propellers,
- Rudders,
- Spiral strakes,
- Spud cans,
- Stairs,
- Turrets (attachment points),
- Water inlet + outlet points.

A.2 Loading and discharge operations

Loading and discharge operations follow a specific sequence, which is shown in the simplified overview in Figure A- 11.

Loading operation

A loading operation starts with the preparation of the HTV and the cargo (see Figure A- 12). The cribbing beams and the guideposts are positioned at the predetermined locations before the HTV is submerged to its loading draft. The cargo is also submerged to its loading draft (see Figure A- 13). Furthermore the weather forecasts are checked, once again, to ensure a safe operation. When the green light is given for the operation, the cargo is towed aside of the HTV by tugs. Now the actual loading operation begins and the cargo handling system is connected to the cargo. The cargo handling system is used to position the cargo above the cribbing beams and to keep the cargo at that location (see Figure A- 14). When the cargo is positioned the de-ballasting of the HTV can start (see Figure A- 15). The de-ballasting of the HTV is the most critical part of the operation. The maximum allowable relative horizontal motions in the system are determined by the cribbing beam width. If the cargo is not positioned correctly on the cribbing beams, loads can occur at unwanted locations. When the cargo is completely supported by the cribbing, the tugs are disconnected from the cargo. After the de-ballasting phase, it is verified that the cargo is located on the right spots on the cribbing beams. If the cargo is not placed within the limits the loading operation needs to be done again. If the cargo is placed correctly, the HTV will prepare for its voyage and the cargo will be secured to the deck with sea fastenings (see Figure A- 16).

Discharge operation

A discharge operation is almost the same as a loading operation, but then in reverse. The cargo and the HTV are prepared, the sea fastening is removed and the HTV is pre-ballasted. The weather forecasts are checked to ensure the operation to take place within the pre-set limits. The cargo handling system is (re-)connected to the cargo and is used to keep the cargo at its position. The tugs are connected to the cargo and then the HTV is ballasted. Just as for the loading operation, the ballasting phase of the HTV is the most critical part of the discharge operation. As the cargo gets loose from the cribbing beams, transient motions will occur. These motions need to be kept small in order to prevent the cargo from hitting the cribbing beams at its weak spots, in this way the cargo could be damaged. When the clearance between the HTV and the cargo is large enough the cargo will be discharged. The cargo handling system will be disconnected and the cargo is towed away. After this the HTV is de-ballasted and can prepare for the next trip. The cargo is towed to its final destination.

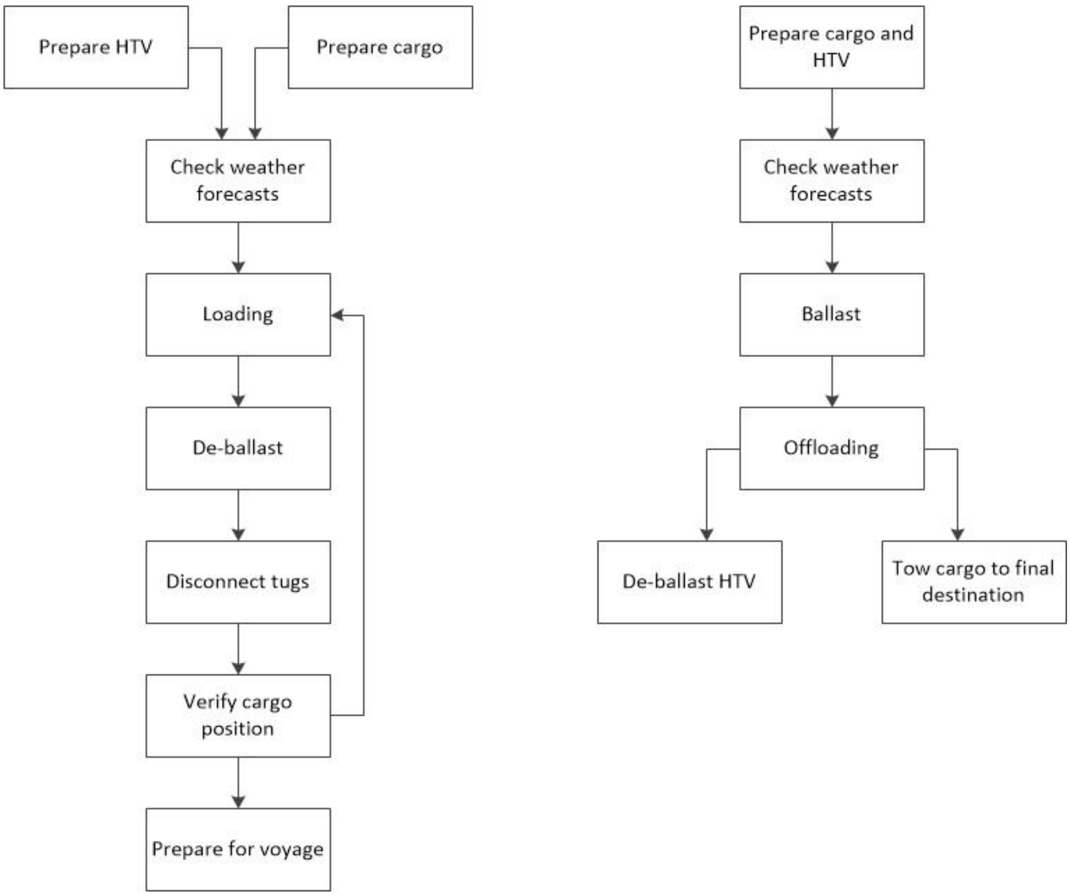

Figure A- 11 Loading and discharge sequence

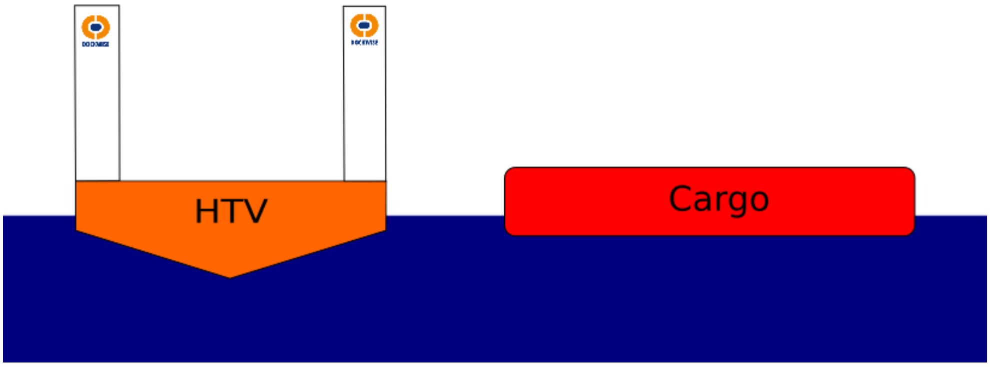

Figure A- 12 Preparation of HTV and cargo

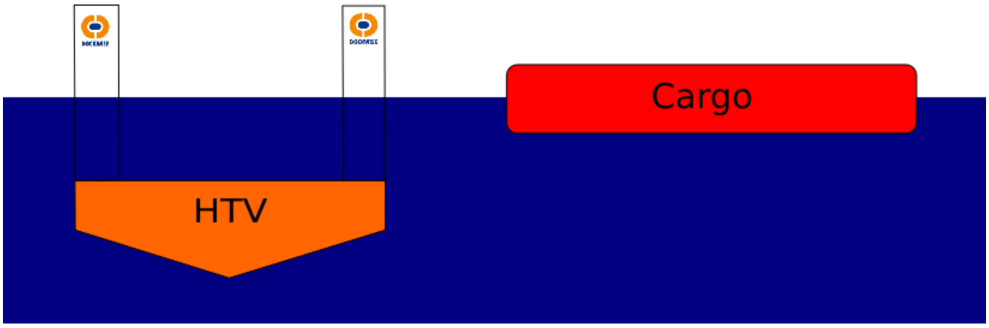

Figure A- 13 Submerging to loading draft

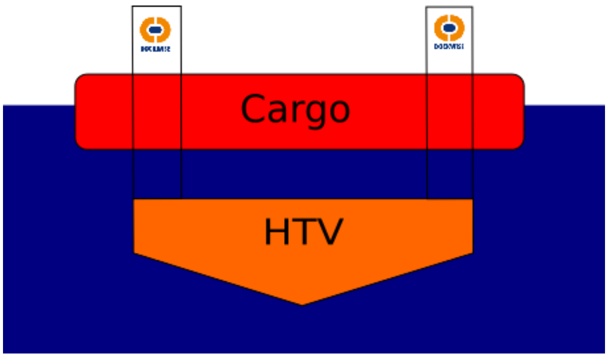

Figure A- 14 Positioning of the cargo

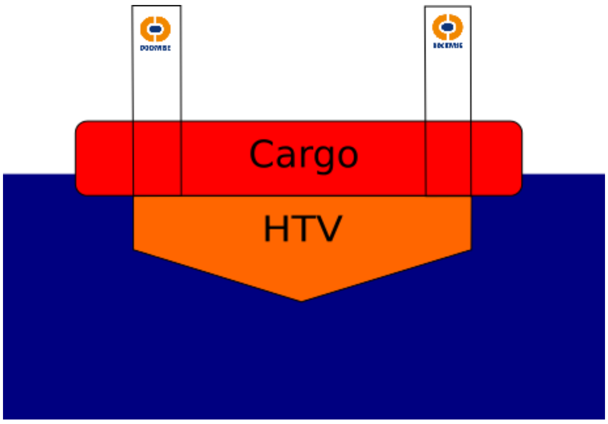

Figure A- 15 De-ballasting of HTV

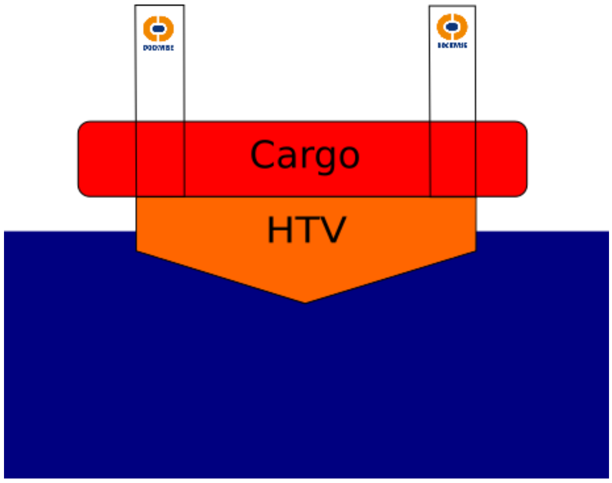

Figure A- 16 HTV ready to sail

A.3 Supporting structures

The cargo will induce loads onto the HTV during the voyage. Two different types of equipment are currently used to guide these vertical and horizontal loads correctly into the hull of the HTV: cribbing and sea fastening.

Cribbing

Cribbing is used to support the heavy structures in the vertical direction, during the voyage. Cribbing are the wooden beams that are positioned on the deck of the HTV (see Figure A- 17) and usually have dimensions of 300 [mm] by 300[mm], however the beams can be placed on top of each other, increasing the height. It is not possible to place the structures directly on the deck of the HTV because the cargo can have protruding parts which prohibit a stable positioning of the cargo. The cribbing is used to place the structures stable on the deck, because openings can be made in the cribbing layout for the protruding parts. Furthermore the cribbing is used to transfer the loads from the cargo into the vessel hull. In this way the loads are distributed evenly over the deck of the HTV and peak loads are avoided.

Figure A- 17 Cribbing layout on a vessel deck

Sea fastening

During the voyage the cargo is subjected to horizontal loads, resulting from the environmental conditions and the motions of the HTV. Sea fastening is used to prevent the cargo from sliding over the cribbing beams, as a result of the horizontal loads. Sea fastening consists of structures which are placed against the sides of the cargo and welded to the deck of the HTV. In this way the sea fastening transfers the horizontal loads from the cargo to the hull of the HTV.

A.4 Cargo handling system

The cargo handling system is the system which is used to manage the horizontal relative motions and to position the cargo at the correct location. The system consists of different types of equipment which together form the cargo handling system.

The standard cargo handling system is the system presently used. It consists of the following equipment:

- **Tugger winches**, these winches are used to control the position of the cargo. The winches can lengthen or shorten the line, which connects the HTV and the cargo to each other, moving the cargo around the deck of the HTV,
- **Dyneema lines**, these lines are as strong as steel, but weigh much less. These lines are also easier to handle than steel lines. The lines are used to connect the cargo and the HTV,
- **Fairleads**, these devices are used to guide the lines and prevent them from rubbing over the edge of the casing where the winches are located. Fairleads typically consists of rollers and/or sheaves,
- **Guide posts**, these devices are used to guide the cargo to the correct location above the cribbing; they are not used to stop the horizontal movements of the cargo [1]. Therefore they can only manage a small load; they merely serve as a visual reference.

A.5 Previous research

In the recent years several studies were performed to develop the optimal cargo handling system. This section gives an overview of the most important studies.

Active tension control system

In 2012 P.S.C. Lee [6] conducted a survey to design an active cargo handling system to control the relative horizontal motions between an HTV and its cargo. To be able to make this design, the Dockwise Vanguard was chosen as the transportation ship and a fully integrated semi-submersible production platform as its cargo (see Figure A- 18). This configuration was chosen to carry out the study, but if the investigated solution works it should not be limited to this configuration. The target environment for this case was North West Australia and the loading/discharge operation is performed in beam waves.

Figure A- 18 Base case scenario [6]

The proposed cargo handling system is an active tension control system. A Proportional-Integral-Derivative (PID) controller is designed, to actively control the motions of the HTV-cargo system. The PID controller calculates the amount of force required to position the cargo on the right place on the HTV. The calculated forces are converted to tensions which will be allocated in the Dyneema lines by the Tugger winches [6].

A PID controller is a closed loop system (see Figure A- 19). In contrary to open-loop control systems, the difference between the set point and the actual response provides a feedback to the controller which can then be adjusted to reach the desired state [6].

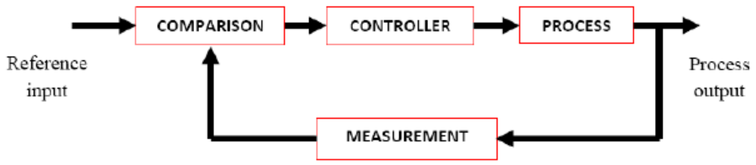

Figure A- 19 Schematic overview closed loop system [6]

A PID controller reacts upon the error of the system using three types of responses:

- Proportional
- Integral
- Derivative

The proportional component reacts to the difference between the set point and the actual response. The proportional component is comparable to changing the stiffness of the mooring system. However there is a difference between changing the stiffness of the mooring lines and using a P controller (proportional controller). If a P controller is used, a type of frequency-dependent filter can be implemented, to design the controller to only react to motions within a certain frequency range. Furthermore the proportional component is largely responsible for the response time and sensitivity of the control system [6]. The integral component reacts to the accumulated error over time; it corrects the steady-state error which is not corrected by other components of the PID controller [6]. The derivative component reacts to the rate of change of the error. The derivative component is comparable to adding a damper to the system. High frequency motions are overdamped, because these motions generate a lot of waves. This means that a lot of extra damping is required to reduce these motions. Low frequency motions are not damped, the addition of damping is more effective in this case [6].

In the ideal situation, the actual system responses are the same as the simulated responses. However in the real situation there are small differences between the systems. These differences are caused by various sources of inaccuracies such as measurement noise and system noise. A filtering technique is required to prevent these inaccuracies from severely impairing the abilities of the controller. Therefore a Kalman filter is designed to achieve this in a recursive manner, by using input data and measures with noise to produce a statistically optimal prediction of the system response [6].

The following conclusions were drawn from this research project:

- The high frequency and low frequency relative motions are effectively reduced using this controller. There exists some remaining motion in the high frequency region but this is less than without the controller. However the required tension in the Dyneema lines to provide the control forces is extremely high. On top of this the required tension speed of the winches is also extremely high [6].
- It is also possible to control only the low frequency relative motions. It is found that the effectiveness of the low frequency controller depends heavily on the type of wave spectrum, because the frequency characteristics of the relative horizontal motions are directly dependent on the type of wave spectrum (wind or swell waves). The required tensions for this control system are much lower than those of the previous system and therefore the tensions are more realistic [6].
- The stiffness of the cargo mooring lines does not have a significant effect on the motion and tension behaviours [6].
- The pretensions in the cargo mooring lines need to be sufficiently high to prevent slack in the mooring lines [6].

ShoreTension

In 2015 A.C.M. Vreeburg [7] conducted a survey to investigate if ShoreTension could be used to minimize the relative horizontal motions between the ship and its cargo. For this survey the Dockwise Vanguard was chosen as the transportation ship and a semi-submersible ring floater as its cargo. This configuration was chosen to carry out the survey, but the outcomes are not limited to this configuration. The system is designed for a chosen maximum wave height and wave period. The ship is positioned in head waves.

ShoreTension (see Figure A- 20) is a hydraulic mooring system for large seagoing vessels and is developed by the Royal Boatmen Association Eendracht (KRVE), together with the Port Authority of Rotterdam. ShoreTension is designed to absorb the motions of a moored vessel caused by external loads [7].

Figure A- 20 ShoreTension [7]

ShoreTension is placed between two bollards on the quayside along the moored vessel. One end of the system is fixated to one bollard, while the other moveable end of the system is connected to the vessels mooring line, guided via a sheave on the second bollard [7]. A moored vessel starts to move due to the tension difference between the different mooring lines, while exposed to external loads. These tension differences cause large forces in the mooring lines, which can lead to line snapping. ShoreTension exerts the same, constant tension on all lines and therefore neutralizes the movements of the vessel [7]. ShoreTension is a passive mooring system because it does not need any external energy during operation. It is activated by a hydraulic system to select the right settings before the operation starts [7].

The ShoreTension system consists of a piston which is able to move in and out a hydraulic cylinder. If the tension in a mooring line becomes larger than a certain pre-load pressure in the system, the piston moves out. If the tension gets lower than the pre-load pressure the piston moves back in [7].

Different simulations were performed to investigate the behaviour and working principle of the system. The following conclusions could be drawn of these simulations:

- Only using ShoreTension results in the cargo finding a new equilibrium position. The ShoreTension cylinders at one side of the ship do not heave back in, while the cylinders at the other side of the ship do not give out. This is caused by the movements of the cargo, which is constantly pushed in one direction. The forces working on ShoreTension are balanced before the ingoing or outgoing motion of the piston is realised. The cargo drifts of, with every loading of waves, until the pistons reach the end or the beginning of the cylinder’s stroke. ShoreTension stops interacting and further movements are counteracted by the mooring lines [7].
- If ShoreTension is installed along the original or soft mooring configuration, the mooring configurations will prevent the cargo from drifting. This indicates that ShoreTension should always be applied together with an existing mooring system. The original mooring configuration keeps the cargo on its equilibrium position and is therefore the preferred cargo handling system [7].
- ShoreTension reduces the energy level of the relative horizontal motions at low frequencies, while slightly increasing the energy level at higher frequencies [7].

- If infinitely stiff mooring lines are connected to the ShoreTension system, alongside the original mooring configuration, the surge and sway motions show a more centred path; as long as the pistons do not approach the end of their strokes. The simulations give the impression that increasing the stiffness of the mooring lines; results in smaller horizontal motions [7].

Also a closer look was taken at the working of the ShoreTension systems along the original mooring configuration. The following was noticed:

- If stiff lines are used for the ShoreTension system, the stiffness of the total system grows. As a result the natural frequency of the system shifts away from the excitation frequency and the resulting response amplitude decreases [7].
- ShoreTension adds damping to the system, and thus dissipates energy, by moving out- and inwards [7].
- ShoreTension performs best with short unstretched mooring lines, because the resulting stiffness of these lines is high [7].
- ShoreTension is capable of reducing the response motions when one of the exciting frequencies of the signal coincides with the natural frequency of the original system [7].

Therefore the following conclusion could be drawn of this study:

ShoreTension is capable of reducing the motions which are excited in the natural frequency region of the standard cargo handling system. However ShoreTension is incapable of reducing the motions which are excited away of the natural frequency [7].

A.6 Design trajectory

In this section a more elaborated description of the design trajectory is given.

Analysing phase

The first step in the design process consists of analysing the offshore loading and discharge problem. Therefore the behaviour of the HTV-cargo system needs to be analysed, this is done with a multi-body hydrodynamic model. The results of the simulations performed for this model provide valuable insight in the interaction between the vessels, cargo handling system and the environment. The outcome of this analysis can be used to define the design requirements and to determine the design approach and what types of models are needed to design a new relative horizontal motion reduction system.

Conceptual design phase

After the problem and the design requirements are defined clearly, the next phase can be entered. The purpose of this phase is to define a suitable concept which is able to solve the offshore loading and discharge problem. The concept of diverging and converging will be used to find the solution. The diverging part consists of a brainstorm session, in cooperation with engineers from different disciplines, to find as much possibilities as possible, to solve the problem. At the end of the brainstorm session a quick feasibility analysis will be conducted to remove all the unrealistic ideas from the list with possibilities. The remaining ideas will be investigated further and eventually another feasibility analysis will be conducted to find the feasible solutions. The feasible solutions will be elaborated further on and in the end a Multi Criteria Analysis (MCA) will be conducted to find the most favourable concepts. Two concepts will be selected for further development.

Preliminary design phase

During this phase, a more comprehensive design of the selected concepts of the previous phase will be developed. The preliminary dimensions of the systems will be determined. Furthermore the working principle of the systems will be defined as well as their dynamic behaviour and their ability to reduce the relative horizontal motions of the system. The design process is an iterative process, in which many issues and decisions influence each other. Also new insights arise during the design process of a system which can cause the reconsideration of earlier made design choices. Sometimes it is best the take a step back in the process the come up with the best solution.

Final design phase

The feasible designs of the previous phase will be developed further during this phase. This phase consists of multiple sub-phases in which the design is getting more and more detailed. During this phase, also the costs become of more importance. At the end of each sub-phase the designs will be checked against the design requirements. The design requirements also evolve during the design process. At the end of the design trajectory it is decided if the design is going to be used.

Appendix B Analysing phase

This appendix will provide additional information about the working principle of the standard cargo handling system. Furthermore insight is given in the global methods which can be used to solve the offshore loading and discharge problem. In the first section additional information about the system description is given. The second section provides the basic theory which is used to calculate the amplitude of the relative horizontal motions. In the third section the simple mass-spring-dashpot model is presented, which can be used to simulate a simplified HTV-cargo system. In the fourth section the derivation of the design values is explained. In the fifth section additional information about the design considerations is presented.

B.1 System description

This section provides additional information about the Dockwise Vanguard and the environmental conditions for which the simulations will be conducted.

Dockwise Vanguard

The vessel currently suitable for the transportation of large FPSOs is the Dockwise Vanguard, which is the largest semi-submersible HTV available. The vessel has a bowless design which allows the vessel to transport cargoes longer or wider than the deck; the cargo is allowed to hang over the bow, sides and stern. Therefore the accommodation and bridge are located at the far starboard side of the vessel. To provide enough buoyancy when the vessel is submerged, the vessel is equipped with 4 casings. These casings are moveable to allow an optimal use of the deck capacity.

Cargo mooring winch specification

A total of 8 winches are installed on the casings and the accommodation block to handle the cargo, 4 winches aft and 4 winches front. The specification of the winches is given in Table B- 1.

Table B- 1 Cargo mooring winch specification

Rated pull (first layer)	350 [kN]
Rated speed (first layer)	15 [m/min]
Loose rope speed (first layer)	30 [m/min]
Drum brake holding force	1120 [kN]

Deck strength

The Dockwise Vanguard is designed to accommodate ultra-heavy cargoes. However the vessel transports different kind of cargoes which have different dimensions. The overhanging parts of the cargo can generate higher local stresses in the deck. Therefore some parts of the deck are strengthened (see Figure B- 1).

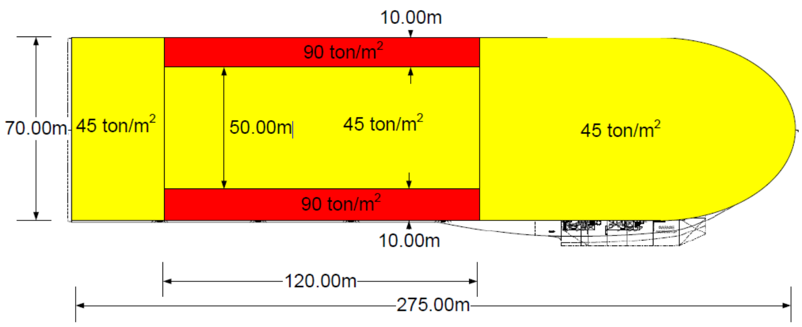

Figure B- 1 Maximum allowable deck load [3]

Other data
Other important characteristics of the Dockwise Vanguard are:

- 4 main ballast pumps with a capacity of 5300 [m^3/hour],
- Total installed power of 28,500 kW.

Environmental conditions
The following steps are performed to find the governing wave characteristics (H_s and T_p) for the JONSWAP swell wave spectrum. Multiple AQWA runs are performed in which the governing values of the variables of the wave spectrum (H_s and T_p) are varied while the HTV and cargo configuration is kept the same. When these runs are completed, the maximum amplitudes for the different wave characteristics are plotted for the relative surge and relative sway direction (see Figure B- 2). Figure B- 2 shows that a wave spectrum with an H_s of 2 [m] and a T_p of 10 [s] is governing for the relative horizontal motions.

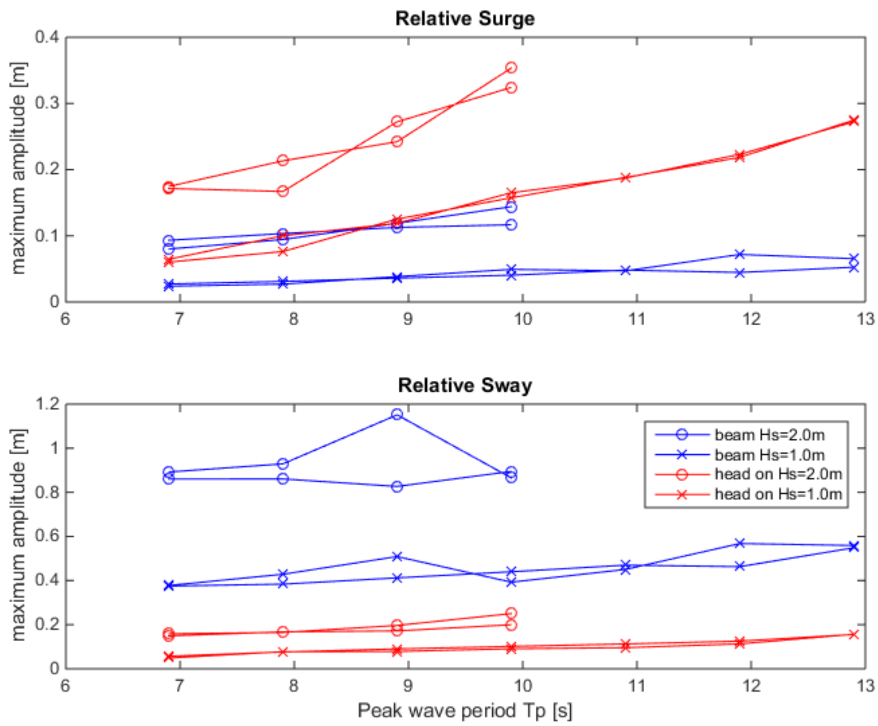

Figure B- 2 Maximum amplitudes for different wave characteristics

B.2 Theoretical background oscillation response
Waves and currents in the seas and oceans cause floating elements and submerged structural elements to start moving. Oscillations are motions which repeat themselves after a certain time interval. It is necessary to examine these oscillations in order to analyse the oscillatory motions of bodies and forces associated with them [11]. An oscillating system often consists of the following parts:

- A mass, storing the kinetic energy [11],
- A spring, storing the potential energy [11]
- A damper, dissipating energy [11].

Response of a system under a harmonic force
The system responds to a harmonic load which in principle has a time range from $-\infty \leq t \leq \infty$. However in the simulations of a dynamic system the following situation will exist, there is no load for $t < t_0$ while there is a load for $t > t_0$. Therefore the system will show two types of responses for the simulations: a transient response

shortly after the beginning of the load and a steady-state response to the long duration (as the transient response is damped out) [30]. As the loading and discharge operations take place in a dynamic environment, where the system is always experiencing a harmonic load, only the steady-state solution is of importance for the system dynamics in this thesis.

The oscillations of a linear damped system with a single degree of freedom under the action of an external force, around its equilibrium position, are described by the equation of motion [11].

$$m\ddot{x} + c\dot{x} + kx = F_0 \sin(\omega t) \quad \text{(B-1)}$$

The steady state response of the system is described by a harmonic function.

$$x = Xsin(\omega t - \varphi) \quad \text{(B-2)}$$

In this equation, the response amplitude X and the phase lag φ are calculated with the following equations:

$$X = \frac{F_0}{\sqrt{(k - m\omega^2)^2 + c^2\omega^2}} \quad \text{(B-3)}$$

$$\varphi = \arctan(\frac{c\omega}{k - m\omega^2}) \quad \text{(B-4)}$$

The response amplitude is very often plotted against the excitation frequency to visualise its dependency. If the excitation frequency equals the natural frequency, the amplitude of the response will become infinitely large, if no damping is applied. The force is capable of pushing the mass always in the direction, in which it is already moving by itself, increasing the amplitude of the motions to infinity. This phenomenon is called resonance. The undamped uncoupled natural frequency is calculated with the following formula:

$$\omega_n = \sqrt{\frac{k}{m}} \quad \text{(B-5)}$$

The used variables in the mentioned equations are:

c	=	Damping coefficient	[Ns/m]
F_0	=	Force amplitude	[N]
k	=	Stiffness	[N/m]
m	=	Mass	[kg]
t	=	Time	[s]
X	=	Response amplitude	[m]
x	=	Displacement	[m]
$\dot{x}$	=	Velocity	[m/s]
$\ddot{x}$	=	Acceleration	[m/s^2]
φ	=	Phase lag	[^{o}]
ω	=	Excitation frequency	[rad/s]
ω_n	=	Natural frequency	[rad/s]

Damping

Damping decreases the response amplitude of the system in its natural frequency region significantly; therefore damping devices are an important feature in oscillating systems. For a linear damped system, under a harmonic force, the amount of energy dissipated by the damper is equal to the work done by the damping force over the displacement [14]:

$$W_d = \oint F_d dx \tag{B-6}$$

$$W_d = \pi c \omega X^2 \tag{B-7}$$

The used variables in the mentioned equations are:

c	=	Damping coefficient	[Ns/m]
F_d	=	Damping force	[N]
W_d	=	Work performed by damper	[J]
X	=	Response amplitude	[m]
ω	=	Excitation frequency	[rad/s]

The damping coefficient of a system can be obtained from a free decay test. The term free decay is used to indicate that there is no external force causing the motion; the motion is primarily the result of the system not being in its equilibrium position. The free decay test can be conducted in the following manner:

- The system is pulled out of its equilibrium position,
- The system is released form this position,
- The response of the system is measured while it returns to its equilibrium position.

An example of the response of a system in a free decay test is shown in Figure B- 3. The response of the system, how oscillations in a system decay after a disturbance is brought into the system, can be described by the damping ratio (ζ). The damping ratio represents the amount of damping within the system. The damping ratio is calculated according to the following formula [31]:

$$\zeta = \frac{c}{c_{crit}} \tag{B-8}$$

$$c_{crit} = 2\sqrt{km} \tag{B-9}$$

The used variables in the mentioned equations are:

c	=	Actual damping coefficient	[Ns/m]
c_{crit}	=	Critical damping coefficient	[Ns/m]
k	=	Stiffness of system	[N/m]
m	=	Mass	[kg]
ζ	=	Damping ratio	[-]

The system can respond in 4 different ways to the disturbance to its equilibrium position:

- Undamped ($\zeta = 0$), the solution is an undamped simple harmonic oscillation,
- Underdamped ($0 < \zeta < 1$), the solution is an exponentially decaying harmonic oscillation,
- Critically damped ($\zeta = 1$), this is the border between the underdamped and overdamped cases,
- Overdamped ($\zeta > 1$), the solution is a decaying exponential without an oscillation.

It is shown in Figure B- 3 that the time-displacement curve of the undamped response can be enveloped by an exponential function. This function is called the amplitude of vibration and can be used to determine the amount of damping within the system [11].

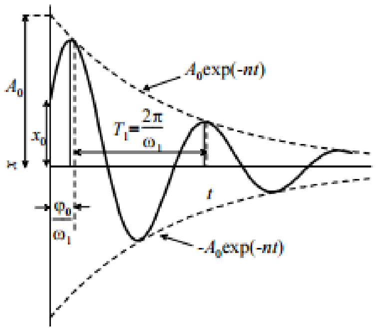

Figure B- 3 Free vibration, underdamped system [11]

The formula of the exponential function is:

$$x = \pm A_0 \exp(-nt) \tag{B-10}$$

$$2n = \frac{c}{m} \tag{B-11}$$

In this formula is A_0 a constant value to start the function.

B.3 Design values

This section explains how the excitation force, added mass and damping values for the MATLAB model are determined.

MATLAB mass-spring-dashpot model

The MATLAB mass-spring-dashpot model of the HTV-cargo system is schematically shown in Figure B- 4.

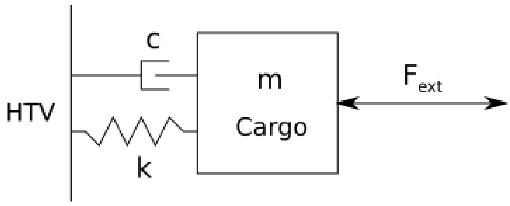

Figure B- 4 MATLB mass-spring-dashpot model

The equation used to solve this model is:

$$m\ddot{x} + c\dot{x} + kx = F_{ext} \tag{B-12}$$

F_{ext} represents the external forces working on the system, which induce the system to move. F_{ext} is made up of the wave forces and the hydrodynamic coupling terms between the HTV and the cargo (added mass and damping). The hydrodynamic coupling terms are frequency dependent; normally they are situated in the left side of equation (B-12). However in this model they are taken as constants. Therefore the right side of equation (B-12) needs to be adjusted, allowing the motions to have the same amplitude as in the AQWA model.

F_{ext} is calculated according to equation (B-13). The force follows a sine-signal with a specified amplitude of frequency, because the wave forces are included in this force.

$$F_{ext} = F_0 \sin(\omega t) \tag{B-13}$$

The model is evaluated by an ODE solver. Therefore the equation of motion is rewritten as a set of equivalent first order differential equations, such that they can be inserted into the ODE. The ODE solver which is used, is ODE45.

Added mass and damping

A low speed manoeuvring model is one which is suitable to describe the motion of a ship in calm water. This model is frequency independent and kept as basic as possible. Unfortunately it is not possible to properly decouple the coefficients of the model from the motions itself. However it is common to do this, in order to keep the models at the start of a design trajectory as simple as possible. The most common way is to use the low frequency added mass and damping terms at the start of a design trajectory [10].

(Ross et al. 2006) show that this low frequency model can be combined with the Cummins equations [10]. The low frequency model will give a good approximation of the frequency independent values for the added mass and damping. The low frequency model is frequency independent and is valid for dynamic positioning and manoeuvring. This is exactly the job description of the relative horizontal motion reduction system. The model consists of the following sub-models [10]:

- Zero frequency model for surge, sway and yaw,
- Natural frequency model for heave, roll and pitch.

In the surge and sway direction the added mass values at $\omega = 0$ should be used. Therefore the added mass used in the MATLAB simulations is: 7500 [t] for the surge direction and 80,000 [t] for the sway direction (see Figure B- 5). The amount of damping is zero for both directions (see Figure B- 6).

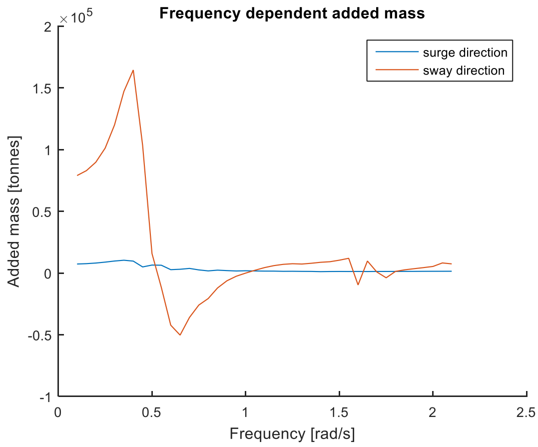

Figure B- 5 Frequency dependent added mass

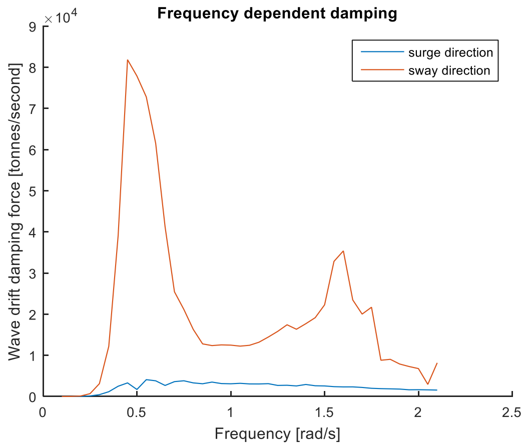

Figure B- 6 Frequency dependent damping

Wave excitation force

The relative motions of the HTV-cargo system are known from the AQWA simulations. With the mass, added mass, damping and stiffness also known, it is possible to calculate the corresponding excitation force for the MATLAB model.

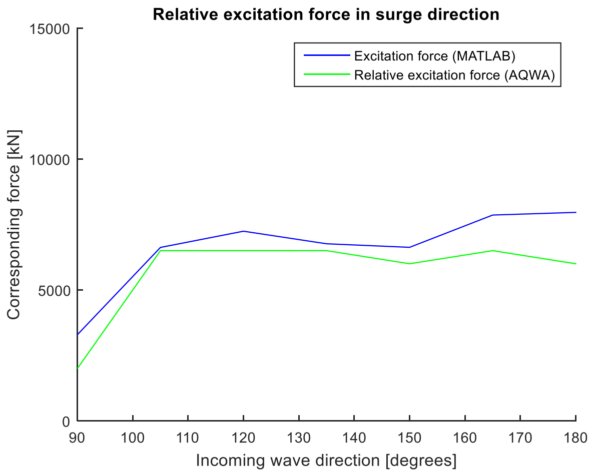

Figure B- 7 Relative excitation force in surge direction

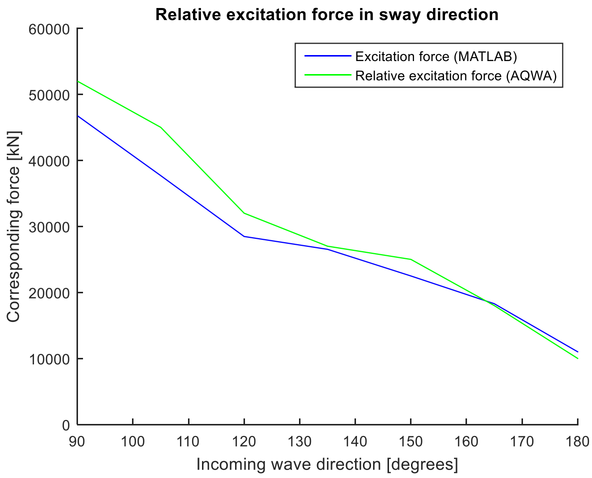

Figure B- 8 Relative excitation force in sway direction

It can be concluded from Figure B- 7 and Figure B- 8 that the values of the excitation force in the AQWA model and the MATLAB model are almost the same.

As can be concluded from Figure 2-8 and Figure 2-9 the forces working on the system are symmetric for all four quadrants of the compass rose. Therefore the excitation forces are only shown for the quadrant which ranges from beam waves (90 degrees wave direction) to head waves (180 degrees wave direction).

The loading and discharge operations of an FPSO will be performed in head waves. In this configuration the risk of a collission between the HTV and the cargo will be the smallest. However it is nearly impossible to keep the heading of the vessel exactly in head waves. Therefore it is assumed that the heading of the vessel will vary between 150 degrees and 210 degrees. Therefore the largest excitatoin force in this range will be used for the design of the cargo handling systems. The characteristic excitation forces are: F_0 = 8000 [kN] for surge direction and F_0 = 25,000 [kN] for sway direction.

B.4 Design considerations

It is possible to reduce the motions of a dynamic system by changing the stiffness of the system or by adding damping to the system. The effect of this on the present system will be explained in the following sections.

Stiffness

It is possible to connect the cargo and the HTV to each other by a stiff connection. However the chance on resonance motions should be minimized. Therefore the final stiffness of the system should be such that the natural frequency of the system is outside the excitation frequency domain. The excitation frequency domain ranges from 0 [rad/s] till approximately 1.5 [rad/s]. If no extra damping is applied to the system, the resonance motions will not be reduced, if they occur. Therefore it is determined that the natural frequency of the system should be larger or equal to 2 [rad/s].

The required stiffness for the cargo handling system can be calculated according to equation (B-5). The required stiffness in the surge direction is k_{surge} = 344,000 [kN/m] and the required stiffness in the sway direction is k_{sway} = 634,000 [kN/m].

Damping

Another possibility is to add damping to the system, in order to reduce the relative horizontal motions. The amount of damping necessary to reduce the relative horizontal motions can be calculated according to equation (B-3), for this case it is assumed that the stiffness of the system is not changed.

The critical damping in surge direction is $c_{crit,surge}$ = 37,417 [Ns/m] and the critical damping in sway direction is $c_{crit,sway}$ = 28,655 [Ns/m]. The damping ratio is calculated for different relative motion restrictions according to equation (B-8)(see Figure B- 9 and Figure B- 10).

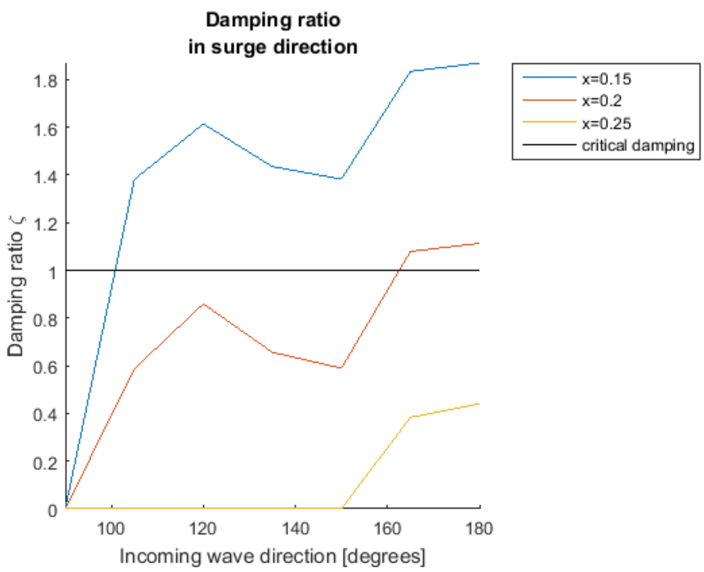

Figure B- 9 Damping ratio in surge direction for different relative motion restrictions

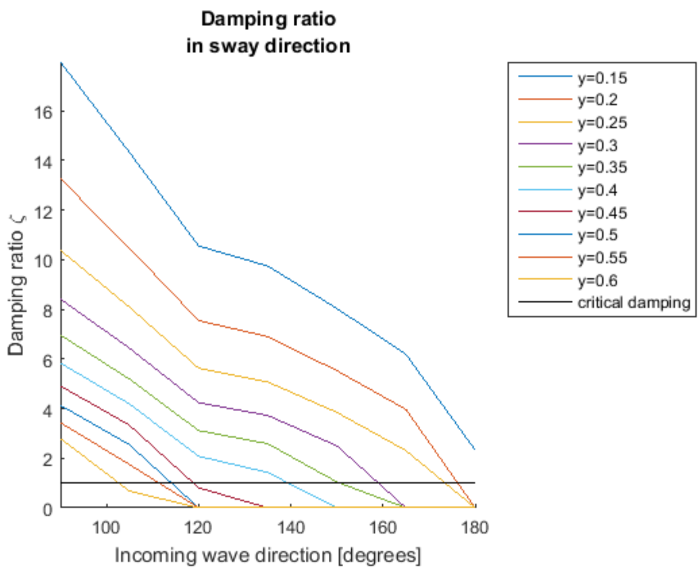

Figure B- 10 Damping ratio in sway direction for different relative motion restrictions

Appendix C Conceptual design phase

This appendix will provide additional information about the development of the concepts which are able to solve the offshore loading and discharge problem. The first section deals with the possible design directions and their respective concepts which are found during the brainstorm session. The second section will give an overview of the criteria and the corresponding weight factors used in the MCA.

C.1 Design directions

In order to come to a well thought out design, all the possible solutions need to be brought forward. During a brainstorm session with engineers, from different disciplines, six design directions were determined. For each design direction multiple possibilities were found (see Figure C- 1). The different possibilities within the design direction can be combined to form other possibilities. Therefor every design direction provides multiple solutions to solve the offshore loading and discharge problem. In the next sections the different design directions and their respective solutions will be discussed.

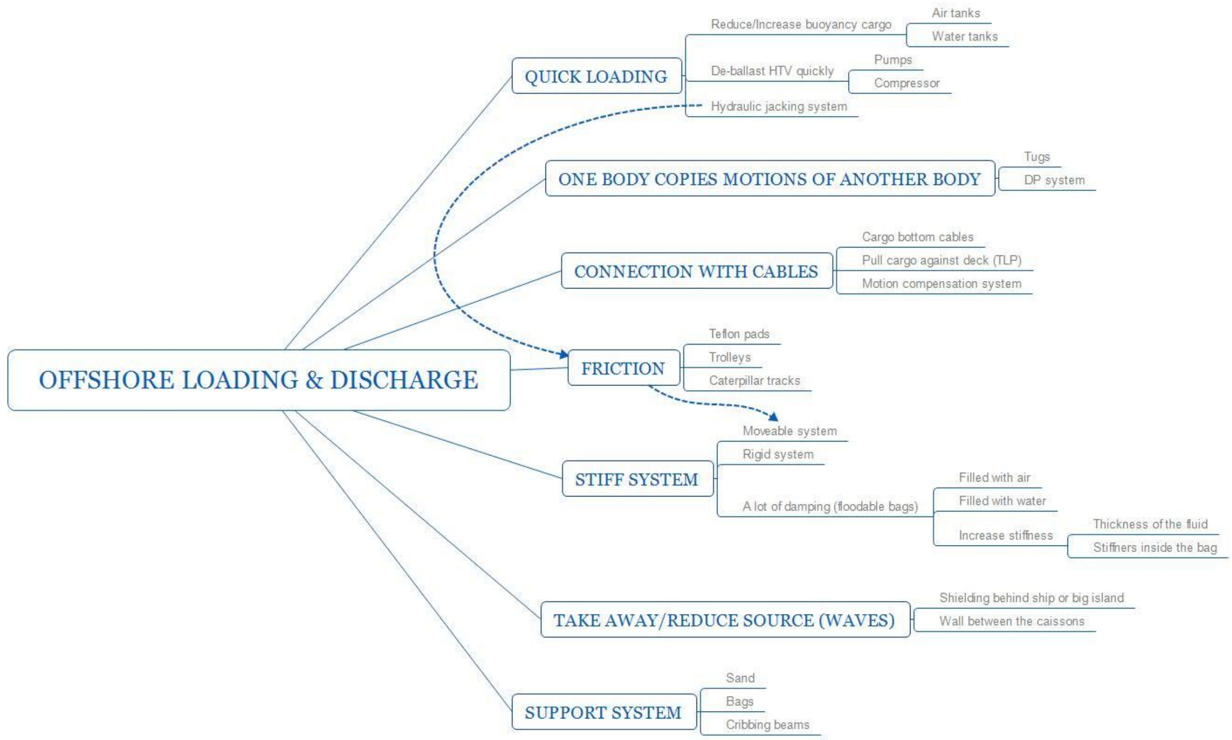

Figure C- 1 Design possibilities

Design direction 1: Quick loading

The principle behind quick loading is to establish a vertical touch down during a calm period of the sea state; this must be done within a few (2 to 3) wave periods (20-30 [s]), when the relative horizontal motion is expected to be within the relative horizontal motions criteria. The principle is based on the fact that waves are travelling in wave groups, which have different wave heights and periods. This means that the corresponding relative motions between the cargo and the HTV will also differ per wave group.

The relation between the wave amplitude and the corresponding relative horizontal motion of the system is known. It is therefore known for which sea states the cargo handling system is able to perform its job. The 3 hour sea state for which the motions are within the design limits (see Chapter 2.6) is called a calm sea state. This calm sea state can be used as a reference to analyse how much calm periods are located within a different sea state, for which the relative horizontal motions are outside the design limits. It is possible to perform the loading and discharge operation in these sea states during a calm period.

The duration and number of these calm periods, for the different sea states, can be determined statistically. This can be combined with a prediction of the characteristics of the incoming waves just before the operation starts. At the Delft University of Technology, a PhD study is in progress which focusses on the prediction of the incoming waves [32]. The aim of this study is to predict the wave elevation some 60 to 120 [s] ahead. Timing of the operation is essential. It is possible that the cargo is not located correctly above the supporting structure and needs to be positioned correctly before the de-ballasting procedure can start, this will take a couple of seconds, reducing the time available for the operation. It is also possible that the cargo is placed incorrect on the supporting structure and that the loading operation needs to be done again. In this case it is important that the cargo is released from the supporting structure during the calm period in order to perform the operation safely. Therefore it is assumed that the quick loading operation needs to be performed within a few wave periods.

A quick loading operation can be accomplished by roughly three methods:

- Reduce the buoyancy of the cargo very quickly,
- De-ballast the HTV very quickly,
- A hydraulic jacking system (see Design direction 4: Friction)

Reduce the buoyancy of the cargo very quickly

The buoyancy of the cargo can be reduced by attaching buoyancy tanks to the cargo, which decrease the draft at the start of the operation. When the distance between the deck of the HTV and the keel of the cargo is very small, the tanks are disconnected from the cargo or valves are opened to fill the tanks. Consequently the cargo quickly increases draft.

De-ballast the HTV very quickly

The HTV can be de-ballasted quickly by installing very powerful pumps which can transport a lot of water per minute. In this way the ballast tanks can be filled or emptied very quickly. Another possibility is to install powerful compressors which will fill the ballast tanks with air. The air will push the water out of the ballast tanks.

Reduce the buoyancy of the cargo and de-ballast the HTV very quickly

In this solution, the buoyancy of the cargo is reduced while, at the same time, the HTV is de-ballasted. In this way the HTV is moving up while the cargo moves down and the time needed for the operation is further reduced.

Feasibility

The cut water plane area for the Dockwise Vanguard is about 1275 [m^2]. The clearance between the top of the supporting structure and the keel of the cargo is approximately 2 [m] at the start of the operation (see Chapter 2.6). To overcome this clearance, about 2500 [m^3] of ballast water needs to be pumped out of the ballast tanks. On top of this, ballast weight needs to be removed such that enough support load is created to overcome the horizontal excitation loads.

It would take at least 7 minutes to overcome the clearance of 2 [m], if the main ballast pumps of the Dockwise Vanguard are used. This is considered to be too slow. A faster method would be to have quick de-ballast tanks above the submerged waterline, and as such let gravity do the work; time may be reduced to order of 1 to 2 minutes depending on the diameter and number of outlet valves. For quick ballasting these tanks should be below the waterline, such that they can be filled quickly. Another problem arises from the fact that, due to its size, the Dockwise Vanguard is very inert thus slowing down the time in which it can float upwards.

The cut water plane area for the cargo is much larger than for the Dockwise Vanguard. Therefore a lot more ballast water is needed to increase the draft of the cargo. A lot more buoyancy tanks and/or pumps are needed to accomplish the quick loading operation if the draft of the cargo needs to be changed.

As shown by P. Naaijen, in his PhD study at the TU Delft, real time wave prediction can support motion critical operations [32]. The available operation time can be timed when the wave elevation is forecasted, based on a measurement of the wave field at an appropriate distance.

Time intervals can be determined where the peak and trough values are lower than the maximum design wave amplitude, for different sea states (see equation (C-1)) [33].

$$|a_{operation}| \leq 0.5 \times H_{s,design}\sqrt{0.5 \times \ln N} \tag{C-1}$$

$$N = \frac{t}{T_p} \tag{C-2}$$

The used variables in the mentioned equations are:

$a_{operation}$	=	Wave amplitude at which the operation can be performed	[m]
$H_{s,design}$	=	Significant design wave height	[m]
N	=	Number of waves	[-]
t	=	Sea state duration	[s]
T_p	=	Wave period	[s]

The time intervals will start or end at the mean time between a successive peak and trough (or vice versa). This is illustrated in Figure C- 2. Also, a histogram of time intervals with different durations is determined, see Figure C- 3 and Figure C- 4.

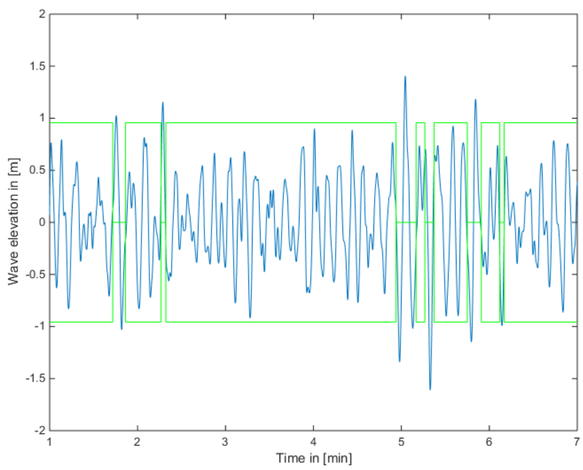

Figure C- 2 Time intervals

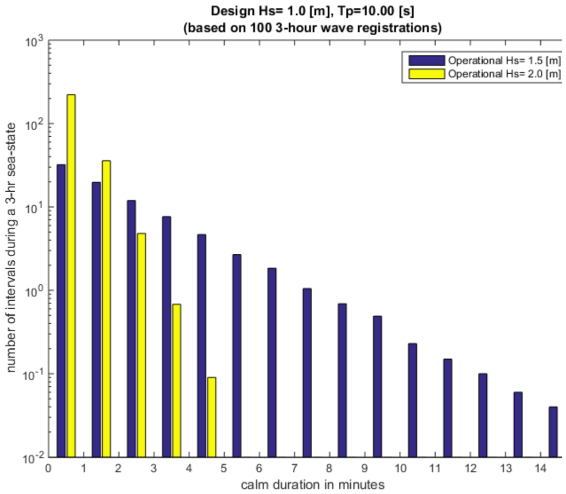

Figure C- 3 Time intervals histogram

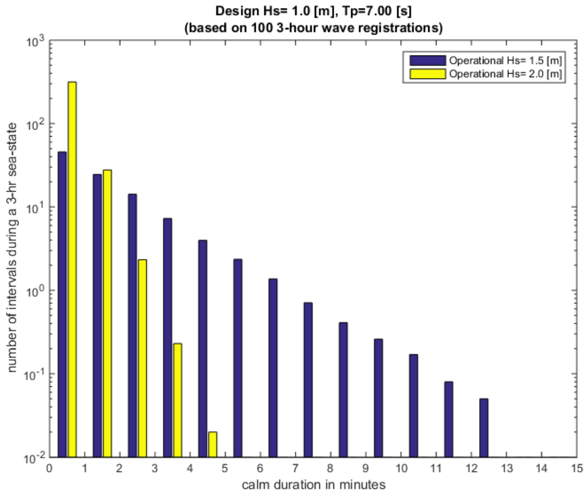

Figure C- 4 Time intervals histogram

For an Operational sea-state of H_s =1.5 [m] with T_p =10.0 [s], for about 130 minutes (out of 180 minutes) the events longer than 2 minutes will have a wave elevation which will be equivalent to a H_s = 1.0 [m] sea-state. For an operational sea-state of H_s=2.0 [m], this is valid for about 15 out of 180 minutes. Table C- 1 indicates the number of events with a duration longer or equal to 2 minutes.

Table C- 1 Number of calm events per sea state

H_s [m]	1.5		2.0	
T_p [s]	Duration [min.]	No. of events	Duration [min.]	No. of events
7.0	115.4	31	6.7	2
10.0	129.7	31	14.8	5

Advantages and disadvantages

The advantages and disadvantages for this concept are discussed below.

Advantages

- The system is passive when the tanks are drained above the waterline (except for the opening and closing of the valves),
- The drainage speed can be increased when compressed air is used,
- The used technology is known (using valves),
- Simple steel structures are used,
- Maybe it is possible to integrate the system in the existing ballast tanks.

Disadvantages

- Water needs to be pumped around,
- Tanks need to be attached/added to the cargo,
- A system is required which monitors and predicts the incoming wave groups,
- It is not sure if controllability is ensured.

Conclusion

The solutions presented for this design direction are very interesting. However it is concluded that this design direction can only be used to widen the operational limitations; the operation can be performed during calm periods in a harsher sea state. It is also concluded that the proposed systems which use additional ballast tanks are operating too slow. The systems need at minimum 1 to 2 minutes to overcome the clearance of 2 meters while it is necessary to overcome this clearance in 30 seconds. It is therefore not possible to use these systems.

A hydraulic jacking system uses hydraulic cylinders to overcome the clearance and the horizontal excitation loads. If the valves of the cylinder are opened the cylinder will heave back in, thus releasing the cargo. It is possible to perform this operation within the 30 [s] limit. Therefore this system is a viable option. However this system will be combined with the standard cargo handling system. It is shown in Chapter 2.2 that the relative horizontal motions of the standard cargo handling system are outside the design limits for the designed sea state. Therefore another relative motion reduction system is needed which can always operate within the design sea state. The to be designed cargo handling system can be combined with this design direction in order to widen the operational window.

Design direction 2: One body the copies motions of another body

The principle behind one body copies the motions of another body is that the motions of the two bodies are exactly the same, the relative motions are zero. Therefore a solution is needed which allows one body to copy the motions of the other body. The HTV will be moored by a soft mooring system (see Chapter 2.1) and therefore the cargo should copy the motions of the HTV. The Dockwise Vanguard will be moored because it is, from an operational viewpoint, impractical to moor the cargo and let the HTV copy the motions of the cargo.

The motions can be copied by multiple systems:

- Tugs,
- Dynamic Positioning (DP) system.

Tugs

At this moment tugs are used to tow the cargo to the HTV before the cargo is connected to the standard cargo handling system. It can be possible to use the tugs, to let the cargo copy the motions of the HTV and position the cargo correctly onto the supporting structure.

DP system

Dynamic positioning is a method to automatically maintain a vessel on its position by use of thrusters. Position reference sensors and actuators provide the system with information about the vessel's position and the magnitude and direction of the environmental forces affecting its position [34]. Two types of DP systems are used these days to keep the vessel at its location:

- Propeller driven systems (Azimuth thrusters),
- Water jets.

However the reaction time of a DP system using Azimuth thrusters is considered to be too large, allowing the motions within the system to become too large. Therefore hydraulic jets are used to generate the force to move the cargo, instead of propellers. The basic idea is that water jets can react faster than propellers or Azimuth thrusters.

Water jets

This is a relatively new type of thruster. The principle is simple, being a pump that draws water in through the intake at the bottom of a vessel, accelerates it and directs it astern (see Figure C- 5) [35]. The jet is able of steering the wake generated by the pump over 360 degrees [34]. The response times for the actuators on a water jet in order to move the reverse duct and the steering deflector to new positions are as follows:

- Steering deflector, full port to full starboard: 3 seconds,
- Reverse duct, fully raised to fully lowered: 4.5 seconds.

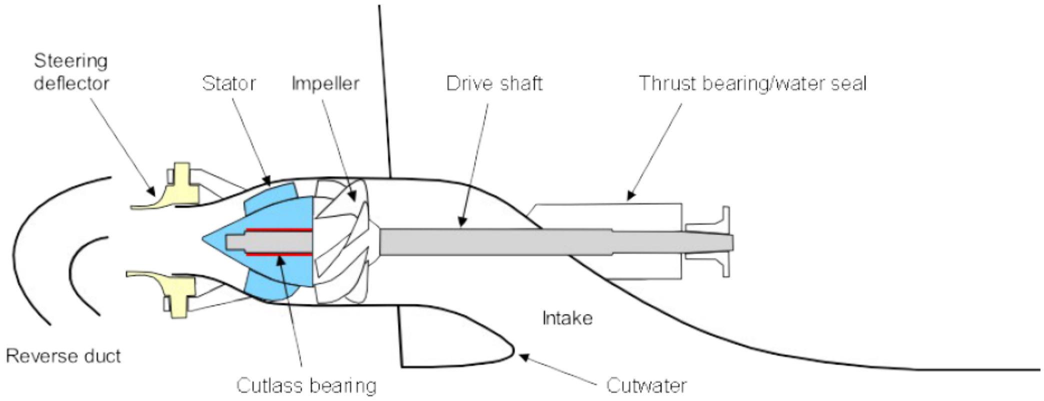

Figure C- 5 Water jet key components [35]

Feasibility

Tugs are operated manually and not automatically, meaning that it is difficult to control the amount of thrust the tugs need to produce to keep the cargo on the same location. The tugs are also not connected to the position measurement system which is needed onto the cargo in order to place it correctly onto the supporting structure. Therefore this system needs to be connected to the tugs again for every loading or discharge operation, which is not very economical. On top of this the tugs are connected to the cargo with long tow lines which makes this system a soft system. This induces another problem, because a soft system is not capable of reducing the motions such that they are within the design requirements (see Chapter 2.2).

The problem with the water jets is that they need to be attached to the cargo below the waterline because they retract the used water underneath them. This causes the water jets to become large units. This is not very practical. Furthermore the efficiency as a thrust generator is low and therefore the water jet is not used for high performance applications (like this operation) [34]. On top of this the reaction time of the system is considered to be too long.

Therefore the solutions mentioned in this design direction are considered to be unfeasible and this design direction is therefore not able to reduce the relative horizontal motions of the system.

Design direction 3: Connection with lines

The principle behind a connection with lines is to establish a connection between the HTV and the cargo which is able to allow a small amount of motion between the vessels, the motion should still be small enough in order to fulfil the design requirements. In this way it is possible to lower the loads working on the motion reduction system compared to a rigid connection. Furthermore line systems are easier to install on the HTV because the used structures are likely to be smaller than rigid structures. On top of this line systems are easier to adjust to different types of cargo than rigid structures.

Different types of line connection layouts are possible:

- Cargo bottom lines,
- Pull the cargo against the deck of the HTV,
- A stretch compensation system.

Cargo bottom lines

The underlying idea is that the top of the cargo is free to move but the keel should be placed correctly onto the supporting structure and is therefore not allowed to move freely. The movements of the keel need to be reduced. Winches and lines are used in this solution. The winches are located on the deck of the HTV and are connected to lines (see Figure C- 6). These lines are attached to the side of the cargo opposite of the winch; the line of the portside winch is connected to the starboard side of the cargo. The line is connected at the top of the cargo, because the bollards are located at the top and these attachment points are located above the waterline, in this way the system is better manageable. At the sides of the keel, cribbing wood is put in between the line and the hull of the cargo to guide the loads over a larger area into the hull of the cargo and to guide the lines to the winches. To prevent the lines from cutting through the hull of the cargo, the lines can be covered with steel sheets.

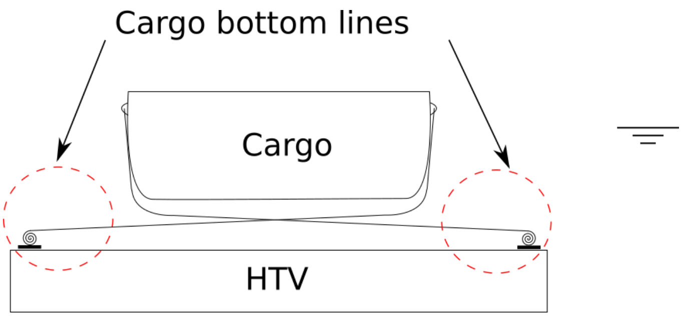

Figure C- 6 Cargo bottom lines, schematic cross section

Feasibility

The system is capable of controlling the relative horizontal motions in the transverse direction. The system is able of controlling the relative horizontal motions in the surge direction. However the system is incapable of reducing the pitch motions due to the long lines and the layout of the system.

The stiffness of the lines depends on their length and thickness. The stiffness of the total system depends on the stiffness of all the elements which combined form the system stiffness. Since long lines are used, already a stiff line type is needed, only for the line to carry its own weight, causing this system to be stiffer than the standard cargo handling system.

The required stiffness can be accomplished by using very stiff lines. However long and also stiff lines are relatively thick and therefore not easy to connect to the cargo. It is likely that the lines are too heavy to manage them manually. Thus instead of positioning the winches on the deck of the HTV it is better to position them on the cargo, where they can be used to pick up the thick lines from the deck of the HTV and to bring tension in the lines. The required stiffness can also be accomplished by using a lot of less stiff lines. These lines are easier to handle and thus easier to connect to the cargo. The downside is that also more winches or pulleys are needed. The lines can be arranged in groups of, for example, 4 lines. From an operational viewpoint it is also in this case better to position the winches on the deck of the cargo in order to protect the winches against the sea water. Otherwise the winches are located in the water and this will increase the amount of maintenance needed for the system.

Pull cargo against deck

The idea for this concept is derived from a TLP. The pretension in the mooring lines will generate a horizontal restoring force which reduces the horizontal motions of the cargo. The closer the keel of the cargo gets to the deck of the HTV, the shorter the length of the lines from the keel to the deck becomes. As a consequence, the possible amplitude of the relative horizontal motions becomes smaller. This makes it possible to place the cargo correctly onto the supporting structure. The winches are located at the top of the casings and the lines are connected to the keel of the cargo through a block (see Figure C- 7).

The pretension in the lines is created by the changing buoyancy of the cargo and the HTV. As the water plane of the cargo is larger than that of the HTV, only the accommodation block and the casings will stick out of the water when the HTV is submerged, the cargo draft will increase less than the HTV draft will reduce.

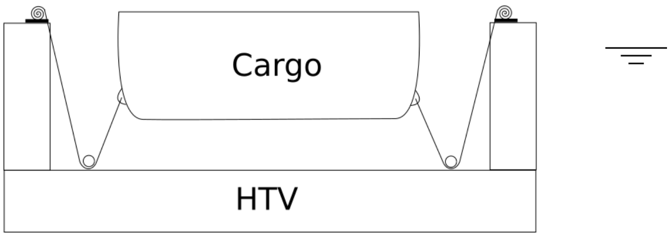

Figure C- 7 Pull the cargo against the deck of the HTV, schematic cross section

Feasibility

This concept is capable of reducing the relative horizontal motions in the transverse and longitudinal direction. However a lot of force is needed to create the necessary pretension in the lines to provide the restoring force to keep the cargo at its location. Powerful winches are needed to create this pretension in combination with heavy lines which are able to handle the corresponding forces. These forces are located at a few small areas on the HTV and the cargo, the connection points of the system on the cargo, the connection points of the blocks on the HTV and the winches.

Another drawback of this system is that the connection points on the cargo need to be welded to the cargo hull. If the cargo is operating at sea and needs to be loaded onto the HTV at sea, this has to be done at sea. As can be seen in Figure C- 7, the connection points are located beneath the waterline. Apart from the fact that this makes it difficult to connect the lines to the attachment points, it is difficult to weld underneath the waterline. Furthermore it is unknown what is located inside the cargo at the locations of the attachment points. If for example, a fuel bunker is located at the inside of the hull, it can be dangerous to weld the attachment points at that location to the hull. Therefore this solution is considered to be unfeasible.

Stretch compensation system

The principle of a stretch compensation system is to use the stretch in the lines to transfer the wave energy working on the cargo into an energy absorber. Therefore the tension in the lines is regulated by a hydraulic system. The idea is derived from the heave compensation systems used to place the subsea structures in the offshore industry. However instead of keeping just one structure motionless two structures need to be kept motionless relative to each other.

The cargo is connected to the HTV by a mooring line configuration. The stretch compensation system is located in between the winch and the cargo. The system consists of a hydraulic cylinder and a few blocks (see Figure C- 8). The blocks guide the line over the hydraulic cylinder. The stretch in the mooring lines, caused by the wave forces, is compensated by the movements of the cylinder. The cylinder manages the vertical length of the mooring layout. In this way the cargo can be kept at the same relative horizontal position, because the horizontal length of line stays the same.

In order to reduce the motions it is necessary that the system is able to react very fast and is able to create a very large force within a few seconds [6]. Therefore energy needs to be stored within the system which can be used to create the compensation forces, necessary to move the cylinder out, within seconds.

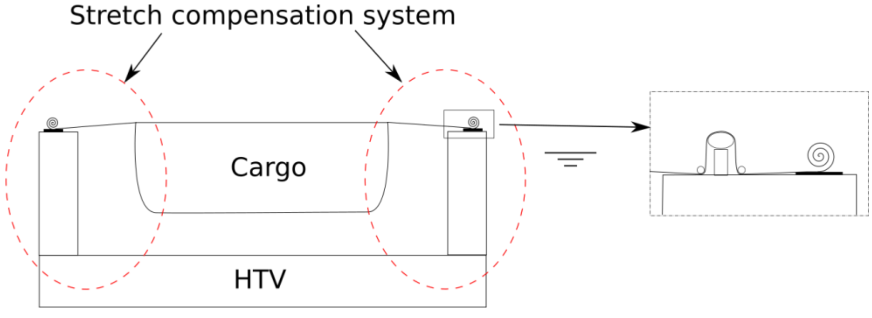

Figure C- 8 Stretch compensation system, schematic cross section

Feasibility

This system is capable of controlling the relative horizontal motions in both the transverse and longitudinal direction, depending on the angle between the mooring line and the cargo. The system layout can easily be adapted to the dimensions of different types of cargo. Due to the fact that lines are used whose lengths can easily be changed.

The amount of stretch in the lines is dependent on the stiffness and length of the lines. The larger the stiffness, the smaller the allowed stretch within the lines. The stiffer the lines are, the lesser the system has to compensate for the motions, the smaller the motions of the cylinder can be. However this says something about the properties of the motions and not about the required force for the cylinder, to heave out to compensate the stretch in the lines. It is likely that a lot of power is needed to move the cylinder.

This system is an active system and therefore a control system is needed to operate the system. This control system needs information about the positions of the HTV and the cargo in order to determine the relative position of the cargo. Thus position measurement equipment needs to be installed on both vessels. The control system determines how much the cylinders need to heave out or heave in, in order to compensate the stretch of the mooring lines and the amount of force needed to do this.

Conclusion

The presented concepts for this design direction are very interesting. However it is concluded that the concept to pull the cargo against the deck of the HTV is unfeasible. The *Cargo bottom line* option is feasible but is only able to control the motions in the transverse direction, thus it should be combined with a system that controls the motions in the longitudinal direction. The *Stretch compensation system* is able to control the motions in both transverse and longitudinal direction.

Design direction 4: Friction

The principle behind the use of friction is to slowly add an energy absorber to the system in order to reduce the relative horizontal motions. The idea is that the induced friction can be used to overcome the horizontal excitation loads. If the friction components are combined with a jacking system it is possible to use the system to overcome the clearance between the cargo and the HTV and to place the cargo on the jacking system. The jacking system can then be used to position the cargo correctly onto the supporting structure.

Different support systems which induce friction in the system are possible:

- Teflon pads,
- Trolleys,
- Caterpillar tracks.

Teflon pads

In this concept the friction plates will be mounted onto a jacking system which is placed on Teflon pads. The cargo will be placed on the jacking system before it is placed on top of the supporting structure (see Figure C- 9). The friction plates will absorb a portion of the kinetic energy of the cargo and the HTV. Consequently the remaining energy for the motions is lower and thus the magnitude of the relative motions will be lower.

The cargo will be positioned above the HTV and when the clearance between the cargo and the HTV is small enough, the jacking system will push the friction plates to the keel of the cargo. The jacking system consists of cylinders which control the vertical height of the system. The cylinders can be pushed out faster than the HTV can be de-ballasted and therefore this system is also a quick loading system (see Design direction 1: Quick loading). The cylinders will push with a certain force to prevent the pistons from sliding back in. Furthermore the cylinders will induce a vertical force between the HTV and the cargo which will generate the friction force. Examples of jacking systems are the Unideck system and the Smart-Leg system [12].

The jacking system will be placed on a Teflon pad and horizontal hydraulic cylinders will connect the system and the deck of the HTV. During the first phase, where the amount of friction between the HTV and the cargo is built up, the jacking system is free to follow the motions of the cargo. When the friction force is large enough to overcome the horizontal excitation forces, the stiffness of the horizontally located hydraulic cylinders will be increased in order to restrain the relative horizontal motions. From this moment on, it is possible to position the cargo correctly above the supporting structure. This is done by exerting a horizontal force on the system with the help of the horizontal cylinders. Due to the Teflon pads, the friction between the jacking structure and the deck of the HTV is very low. Therefore a relative low force is needed to position the cargo correctly above the supporting structure. If the cargo is placed correctly above the supporting structure, the vertical positioned hydraulic cylinders will slide back in, after which the cargo is placed onto the supporting structure.

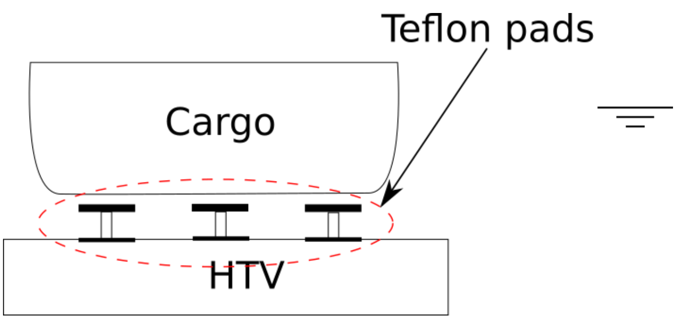

Figure C- 9 Teflon pads, schematic cross section

Feasibility

This solution is capable of reducing the relative horizontal motions in both the transverse and longitudinal direction. The system can easily be adapted to different cargo layouts, because the systems are relatively small and thus simple to reposition.

The system is completely located beneath the waterline, which makes it impossible to repair or check elements of the system when the HTV is in the de-ballasted position. It is therefore important that the system is fully redundant; if one jacking system fails, the other jacking systems can still finish the operation or the operation can be abandoned safely.

This system is an active system and therefore a control system is needed to control the system. In order to position the cargo correctly onto the supporting structure, the relative position of the cargo to the HTV needs to be known. Position measuring equipment needs to be present on the HTV and the cargo. It is necessary to measure the pushing force of the vertical cylinders onto the cargo to check if the system created the correct amount of friction between the HTV and the cargo. With this information, the control system is able to determine how much the cylinders in this system need to heave out or in, to be able to position the cargo correct onto the supporting structure.

Trolleys

The principle of this solution is the same as for the *Teflon pads* solution. Therefore this system is also almost the same as the *Teflon pads* solution, however instead of sliding the system over the deck, the system is driven over the deck (see Figure C- 10). The jacking system is placed on balls which allow the system to move in every horizontal direction. The cargo is positioned correctly above the supporting structure by exerting a horizontal force on the system with the help of horizontally placed hydraulic cylinders. If the cargo is placed correctly above the supporting structure, the vertical positioned hydraulic cylinders will slide back in, after which the cargo is placed on the supporting structure.

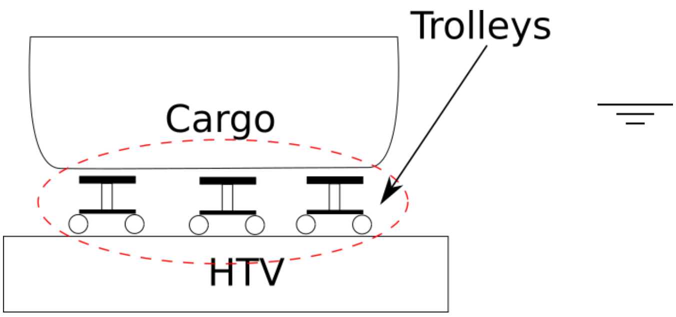

Figure C- 10 Trolleys, schematic cross section

Feasibility

The overall principle for this system is the same as for the Teflon pads system, therefore it is concluded that this system is also feasible. However the maintenance costs for this system will be higher than for the Teflon pads system because the balls are more maintenance sensitive than the Teflon pad. It is likely that the balls will wear of faster and are more sensitive for protrusions which can be located in between the balls.

Advantages and disadvantages

The advantages and disadvantages for this concept are discussed below.

Advantages

- The system can position the cargo correctly in all the horizontal directions,
- The system is relatively easy to manage,
- The system is relatively simple to position on the deck of the HTV.

Disadvantages

- It is an active system, it is not sure if it is possible to control the motions if the system fails,
- A larger number of systems is needed, because the free area under the keel of the cargo is small due to the protrusions,
- The system is maintenance sensitive because it is located in a very corrosive environment,
- The system is operating just above the supporting structure.

Caterpillar tracks

The principle for this system is the same as for the *Teflon pads* and T*rolley* systems, reduce the motions by inducing a friction force in the system. In this concept Caterpillar tracks are mounted on top of a jacking system (see Figure C- 11). The caterpillar tracks are rotating around the horizontal axis and in this way they induce a friction force between the HTV and the cargo. First the cargo is placed on top of the caterpillar tracks and therefor the caterpillar tracks are higher than the supporting structure. When the wave forces are no longer causing the cargo to move relatively to the HTV, the caterpillar tracks can be used to move the cargo around and position it correctly above the supporting structure. If the cargo is positioned correctly, the height of the caterpillar tracks will be reduced and the cargo is placed on the supporting structure.

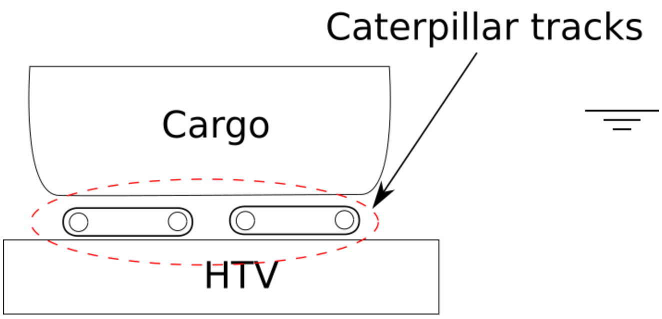

Figure C- 11 Caterpillar tracks, schematic cross section

Feasibility

This solution is capable of reducing the relative horizontal motions in the transverse or the longitudinal direction, depending on the orientation of the system. If multiple structures are used, which are oriented in the transverse or the longitudinal direction (the complete system is oriented in two directions), the system is capable of reducing the relative horizontal motions in both the transverse and longitudinal direction.

However it is unlikely that the caterpillar tracks can induce the same amount of friction as the *Teflon pads* and T*rolley* systems. Furthermore a lot of force is required to be able to move the cargo in order to position it correctly. Therefore powerful electro motors are necessary which drive the caterpillar tracks. Another downside of this system is that all the important elements are located beneath the waterline. Thus it is difficult to repair or operate parts of the system manually in case of defects in the system. Therefore this system is considered to be unfeasible.

Conclusion

The concepts presented for this design direction are very interesting. However it is concluded that the concept in which caterpillar tracks are used is considered to be unfeasible. The *Teflon pad* and *Trolle*y systems are able to control the relative horizontal motions in both the transverse and longitudinal direction and are feasible concepts.

Design direction 5: Stiff system

The principle behind a stiff system is to establish a connection between the HTV and the cargo which reduces the relative horizontal motions between the cargo and the HTV till zero. In order to accomplish this, different principles can be used (see Figure C- 1). A moveable system is a system, whose layout is changed in time. It is important that the chance on resonance motions is reduced as much as possible, or that the system is able to handle the resonant motions (see Chapter 2.5). A rigid system is a passive system which has the same layout during the complete operation and by its shape reduces the relative motions of the system. The idea behind a system which has a lot of damping capabilities is that the kinetic energy, induced by the wave force on the cargo, is absorbed by the cargo handling system. In this way the relative horizontal motions are reduced and it is possible to position the cargo correctly onto the supporting structure.

Different stiff systems are possible:

- Clamping,
- A converging vertical system,
- Sea fastening,
- Catcher,
- Floodable bag,
- Floodable bag as fender.

Clamping

A stiff connection can be utilised by a moveable structure like the system in Figure C- 12. The HTV and the cargo are connected with each other by this system which is placed on the deck of the HTV and will clamp the cargo in between the structures. In this way a stiff connection between the HTV and the cargo is accomplished.

The proposed system consists of a stiff part and a moveable part (see Figure C- 12), and is located at opposite sides of the cargo. The moveable part is attached to the stiff part by a hinge at the deck of the HTV and by a hydraulic cylinder at the top of the stiff structure (the blue line in Figure C- 12). At the start of the operation the cargo is free to move in between the structures. However in time the hydraulic cylinder will start to move out, reducing the free space between the cargo and the structures. During this period the stiffness of the system is very low. If the clamping system is attached to the cargo at both sides, the motions of the cargo are restrained and the system needs to increase its stiffness. This is done by increasing the pushing force of the hydraulic cylinders. The system changes from a mass driven system into a stiffness driven system (see Chapter 2.5). However it is possible that the natural frequency of the system will be the same as the wave frequency, at a certain moment. Therefore it is necessary to add damping to the system to reduce the resonant motions. The damping systems can also be used to absorb the impact loads which are induced onto the clamping system as the moveable arms are approaching the cargo in order to clamp the cargo.

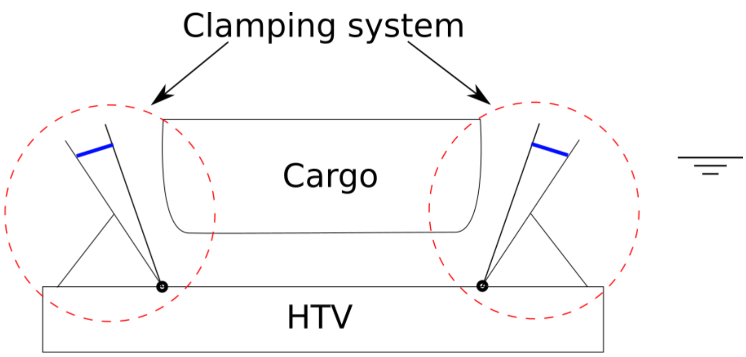

Figure C- 12 Clamping system, schematic cross section

Feasibility

This concept is capable of reducing the relative horizontal motions in the transverse or longitudinal direction, depending on the direction in which the system is working. The moveable arms can only clamp the cargo in one direction; it is likely that, in the other direction, the cargo is hanging over the side of the HTV, making it impossible to clamp the cargo in that direction. Furthermore it is uncertain if the system is able to create enough friction between the cargo and the system to sufficiently reduce the relative horizontal motions in the other direction. However combined with another system this system is able to work.

The system will restrain the relative horizontal motions of the cargo at the keel level. Therefore the system will be connected to the cargo at an approximate height of 6 [m] (see Chapter 2.6). Therefore the system height will be around 6 [m] but this will follow from a more detailed design.

The system will partly be an actively controlled system, because the moving parts of the system need to be pushed against the cargo. This is the active part. The damping systems used to absorb the impact loads can be passive. It is therefore important that the system is fully redundant. The most critical elements of the system are ideally located above the waterline allowing an inspection or to be able to quickly repair the system if the system fails. The system will be controlled by a control system which determines the stiffness of the system and the speed at which the arms of the system propagate towards the cargo. The control system determines how fast the required stiffness is reached. This should be done fast enough to reduce the amplitude of the resonant motions but slow enough to reduce the impact loads onto the cargo if the moveable parts touch the cargo.

Converging vertical system

In this solution a rigid structure is used to place the cargo correctly on the supporting structure. Rigid structures will be placed at opposite sides of the cargo on the deck of the HTV; the cargo will be located in between the structures (see Figure C- 13). The basic idea of this solution is that the available space, wherein the cargo is allowed to move, is slowly reduced over the clearance height between the keel of the cargo and the top of the supporting structure. Therefore the possible relative horizontal motions between the cargo and the HTV will reduce until they are within the design requirements (see Chapter 2.6).

The sides of the structure, which are at the inner side of the system (pointing towards the cargo), will have a sloping angle. In this way it is possible to use a passive system to position the cargo correctly on the supporting structure. During the de-ballasting phase the amount of water above the HTV's deck will be reduced. The width of the converging vertical system, at the waterline, will become bigger, because the system will rise out of the water. In this way the free space between the structures is reduced.

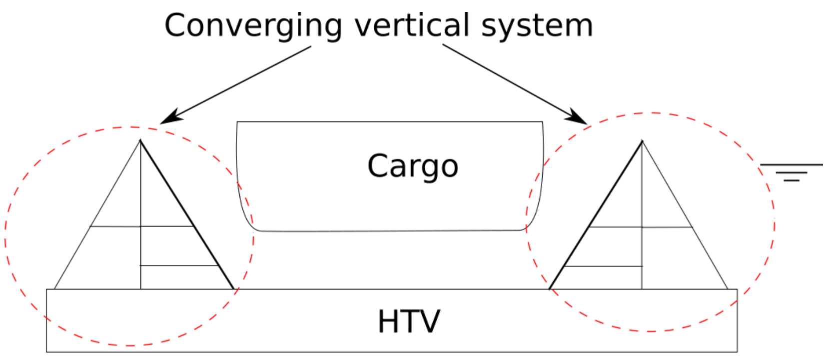

Figure C- 13 Converging vertical system, schematic cross section

Feasibility

This concept is capable of reducing the relative horizontal motions in the transverse or longitudinal direction, depending on the outlook of the operation. It is possible that the cargo is hanging over the side of the HTV in one direction, making it impossible to place these systems in that direction. Therefore this system should be combined with another system working in the other direction, to reduce the relative horizontal motions in both the transverse and longitudinal direction.

The system will restrain the relative horizontal motions of the cargo at its keel level, like with the clamping system. Therefore the height of the system will be approximately 6 [m] (see Chapter 2.6).

The impact loads, which the cargo induces on the system, are also an important aspect of this solution. The motions of the cargo are limited by the structures, a collision between the structures and the cargo will occur. This will give local loads on the structure and hull of the cargo. These loads need to be absorbed by some kind of damping system in order to reduce the magnitude of them. Therefore fenders need to be attached to the sloped sides of the structure.

Sea fastening

This solution uses the sea fastening as a bumper system during the offshore loading and discharge operations. Sea fastening is already used during the voyage to secure the cargo from moving. Normally the sea fastening is placed after the cargo is correctly positioned on the supporting structure. For this solution the sea fastening will be placed on the deck of the HTV before the operation starts. The sea fastening can be used in three different configurations, which will be discussed below.

In the first configuration the sea fastening is welded to the deck of the HTV (see Figure C- 14). The sea fastening is equipped with hydraulic cylinders which are able to absorb the kinetic energy, resulting from the motions of the cargo and the HTV. By pushing out the cylinders, the motions of the cargo will be restricted and the cargo is positioned at the correct location on the supporting structure.

In the second configuration the sea fastening is used as a sliding system (see Figure C- 15). The sea fastening is placed onto a track and is able to move horizontally on this track with the help of hydraulic cylinders. The impact forces work directly on the sea fastenings and are guided to the hydraulic cylinders through the sea fastenings.

The third configuration is a combination of the traditional sea fastening configuration and a sliding sea fastening configuration (see Figure C- 16). At one side of the cargo, the sea fastening is welded to the deck of the HTV, thus the sea fastening has the same function as a fender construction during the current operations. At the other side the sea fastening is able to move in the transverse direction or the longitudinal direction, depending on the layout of operation. In this way the cargo will be clamped in between the sea fastening and is guided to its correct position on the supporting structure.

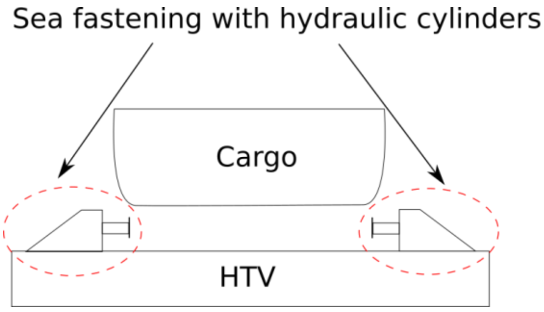

Figure C- 14 First configuration, schematic cross section

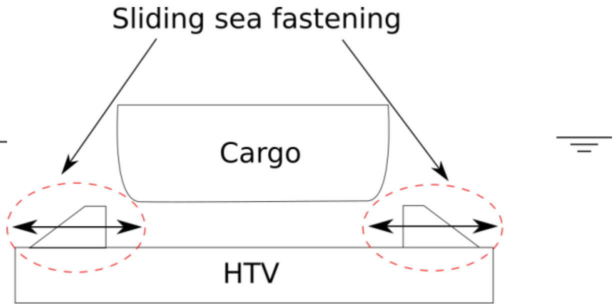

Figure C- 15 Second configuration, schematic cross section

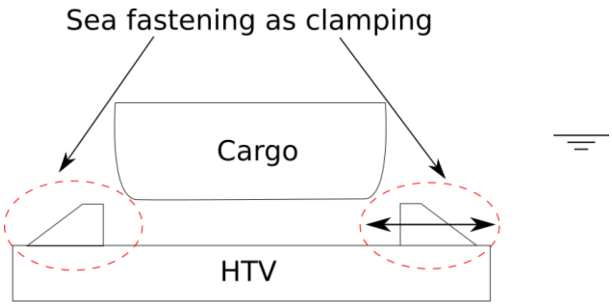

Figure C- 16 Third configuration, schematic cross section

Feasibility

This solution is capable of reducing the relative horizontal motions in the transverse or longitudinal direction, depending on the layout of the operation. The sea fastening is only positioned in one direction because it is possible that the cargo is hanging over the sides of the HTV in the other direction. Making it impossible to place sea fastening in that direction. Therefore this system should be combined with another system working in the other direction, in order to sufficiently reduce the relative horizontal motions in both transverse and longitudinal direction. In order to reduce the relative horizontal motions, the motions of the cargo will be restrained by the sea fastening. Therefore collisions between the cargo and the sea fastening will occur. It is necessary to equip the system with fenders in order to reduce the occurring impact loads on the cargo.

In the first configuration the traditional sea fastening is used and equipped with hydraulic cylinders. The traditional sea fastening is a little higher than the supporting structure; therefore the system is working just above the supporting structure. A lot of sea fastening structures are used to secure the cargo, thus a lot of hydraulic cylinders are used. This makes this configuration expensive to operate and therefore this configuration is considered to be unfeasible.

The second configuration also uses the traditional sea fastening, which operates just above the supporting structure. It is difficult to ensure enough controllability over the motions of the cargo if the system is acting just above the supporting structure. This can be resolved by increasing the sea fastening height but then the principle of the system becomes the same as the *clamping* system, which is already discussed. Therefore this configuration is also considered to be unfeasible.

The third configuration also uses the traditional sea fastening system. In order to increase the controllability of the system, this systems height also needs to be increased. The difference with the *clamping* system is the layout of the system. In the *clamping* system the structures at both sides of the cargo have moveable parts while this system has moveable parts at only one side of the cargo. Therefore this system is different than the *Clamping* system and is considered to be feasible.

Advantages and disadvantages

The advantages and disadvantages for this concept are discussed below.

Advantages

- The forces working on the system will slowly increase, except the impact loads,
- It is possible to design the system such that if the control system is not working the system will react as a stiff structure,
- No extra attachments to the cargo are needed,
- A part of the system is simple to construct.

Disadvantages

- The system can control the motions only in one direction, another system is needed to control the motions in the other direction,
- It is an active system, if the control system does not work correctly the motions can become out of control.

Catcher

In this concept the relative horizontal motions are restricted by establishing a fixed horizontal connection between the HTV and the cargo (see Figure C- 17). The cargo is still free to move up and down but keeps its horizontal position. The effect of the fixed horizontal connection is that the cargo is guided to its correct location on the supporting structure.

The idea is derived from the Leg Mating Units (LMU) which are used for float-over operations. This system is generally located at the interface between the jacket and the deck of the topside. The LMU performs 4 main functions during the mating phase [36]:

- Centering of the deck legs during first phase of lowering [36],
- Reducing the vessel/deck motions during lowering [36],
- Reducing the impact loads between deck and jacket [36],
- Providing final accurate positioning of the deck legs onto the jacket pile [36].

The proposed system needs to perform the same main functions as the LMU. However the difference with float-overs, where only one body moves relative to the other body, is that in this case two bodies are moving relative to each other. Therefore the LMU system needs to be adjusted in order to fulfil these tasks for the offshore loading and discharge problem.

The system consists of two parts. The first part is a stiff pile which is welded to the deck of the HTV. The other part of the system is a catcher and is welded to the side of the cargo. The base of the catcher has the shape of a funnel, while the rest of the catcher consists of a hollow pipe. During the operation the funnel will be located above the pile. The motions of the cargo will be slowly reduced by the catcher because the funnel will guide the pile to the inside of the catcher. In this way the stiff horizontal connection is established.

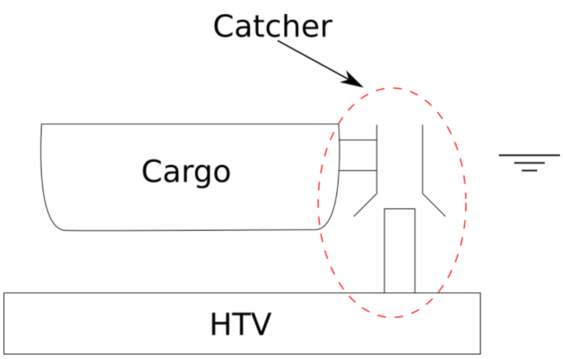

Figure C- 17 Catcher, schematic cross section

Feasibility

This concept is capable of reducing the relative horizontal motions in both the transverse and longitudinal direction. The system can be used on different types of cargo because it is a very simple system.

A major drawback of this system is that the catcher needs to be welded to the hull of the cargo in order to make the system work. It is not always known what is located at the inside of the hull, at the location where the catcher should be mounted. This can make the welding operation a dangerous operation (see Design direction 3: Connection with lines). The catcher also needs to be welded to the cargo at sea, for example because the cargo is operating at sea for already a few years and needs to be transported back to shore. This can be undesirable because the cargo will experience motions like roll and pitch which can make it difficult to weld the catcher correctly to the hull of the cargo.

Another aspect is that the loads working on the cargo are guided to the HTV by the system. The bottom of the catcher is very wide compared to the top of the catcher. It is therefore likely that steel sheets are welded to the cargo to which the catcher is attached. This causes the contact area between the catcher and the cargo to be relatively small. Relatively large local forces are induced into the cargo hull and it is not certain if the cargo hull can withstand these local forces. Therefore this solution is considered to be unfeasible.

Floodable bag

The underlying idea of this concept is to absorb all the kinetic energy of the cargo which is induced on the cargo by the environmental forces. In this way the motions of the cargo will be reduced. Two huge floodable bags will be used to absorb the kinetic energy of the cargo. The bags will be positioned alongside two opposite sides of the cargo. The two bags will have a round shape. In this way the available space, in which the cargo is free to move, will be reduced as the deck of the HTV will move upwards, if the HTV is de-ballasted (see Figure C- 18). Therefore the principle is the same as for the *Converging Vertical System*.

The big advantage of this system, compared to the other systems, is that the protrusions on the cargo do not form a major obstacle to use this system. The bags are made of rubber and filled with air or water, which makes it possible for them to be locally compressed. In this way the bag can be formed around the protrusions and divide the occurring loads over the complete length of the cargo hull.

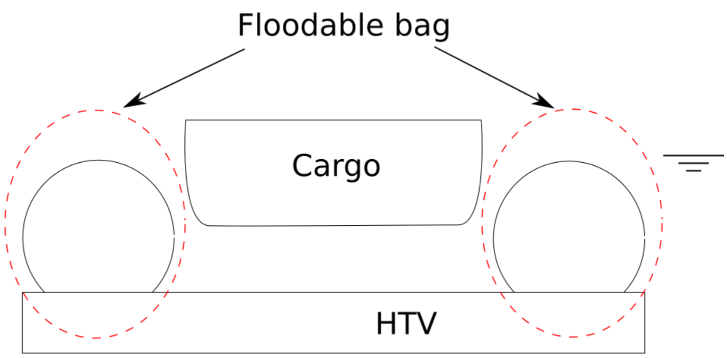

Figure C- 18 Floodable bag, schematic cross section

Feasibility

This concept is capable of reducing the relative horizontal motions in the transverse or longitudinal direction, depending on the layout of the operation. The bags are positioned in only one direction because it is possible that the cargo is hanging over the sides of the HTV in the other direction. Therefore this system should be combined with another system working in the other direction, to reduce the relative horizontal motions in both transverse and longitudinal direction.

Due to the shape of the system, the cargo will be restrained in its motions at its keel level at the start of the de-ballasting process. Therefore the bags need to have an approximate height of 6 [m] (see Chapter 2.6). However it is virtually impossible to make bags which are this big and keep their perfectly round shape, otherwise the height of the bags will be lower and thus the system will not function optimally. During the operation the bags will be formed around the protrusions from time to time. If the rubber cover of the bag is thick enough it is possible that the bags will not be damaged by the sharp edges of the protrusions of the cargo. However during the operation the cargo will be lowered down for another two meters. It is likely that the rubber cover is not capable to cope with these locally induced high tensions in the cover, caused by the sharp edges of the protrusions. As a consequence the bag will be ruptured and cannot be used in the operation. Therefore this solution is considered to be unfeasible.

Floodable bag as fender

The underlying idea of this concept is the same as for the *Floodable bag*, to absorb all the kinetic energy of the cargo. Floodable bags, which are used as fenders, will be attached to a stiff structure. In this way multiple smaller bags can be used and it is less likely for the bags to lose their shape.

The stiff structures will be positioned at opposite sides of the cargo (see Figure C- 19). The structures will be placed on the deck of the HTV such that the motions of the cargo are restrained and fulfil the design requirements (see Chapter 2.6). This system looks like the standard cargo handling system. The difference is that instead of using traditional fenders, floodable bags are used. If the bags are connected to pumps and valves, it is possible to regulate the damping characteristics of the bags. In this way the system can be adjusted to different types of cargo and environmental conditions.

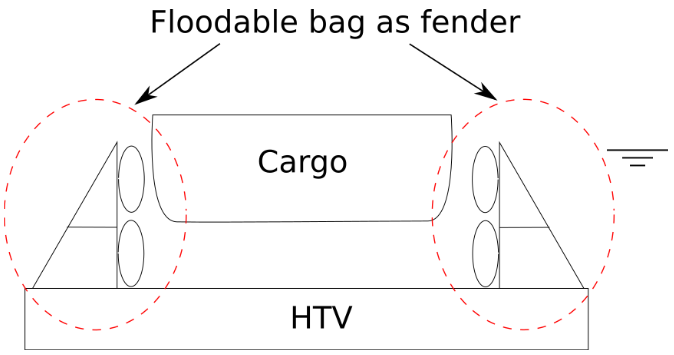

Figure C- 19 Floodable bag as fender, schematic cross section

Feasibility

This concept is capable of reducing the relative horizontal motions in the transverse or longitudinal direction, depending on the layout of the operation. The bags are positioned in only one direction because it is possible that the cargo is hanging over the sides of the HTV in the other direction. Therefore this system should be combined with another system working in the other direction, to reduce the relative horizontal motions in both transverse and longitudinal direction.

However in order to place the cargo correctly on the supporting structure, the structures need to be positioned close to the supporting structure to sufficiently reduce the relative horizontal motions of the cargo. Therefore the standard cargo handling system needs to place the cargo precisely in between the structures. The standard cargo handling system is not capable of doing this. It is possible to make the initial clearance between the structures and the cargo wider. However in this case the bags need to be stiffened in order to increase in size and keep their shape during the loading or discharge operation. This is not possible and therefore this solution is considered to be unfeasible.

Conclusion

The concepts presented for this design direction are very interesting and look promising. However it is concluded that the *Catcher*, *Floodable bag* and *Floodable bag as fender* concepts are unfeasible. The *Clamping*, *Converging vertical system* and *Sea fastening* solutions are feasible but are only able to control the relative motions in one direction, they should be combined with a system that controls the relative motions in the other direction.

Design direction 6: Take away /Reduce source (waves)

The principle of this design direction is to take away/reduce the source (waves). If the wave heights and periods are smaller, the corresponding forces working on the connection between the cargo and the HTV and the corresponding relative horizontal motions will be smaller. For this reason the loading and discharge operations are performed at harbours and sheltered locations at this moment.

The following methods can be used to reduce the waves:

- Perform the operation in the shelter of an island or big ship, downwind of an object are the wave heights lower (see Figure C- 20).
- Build walls between the casings, the waves in between the casings will be lower than the waves surrounding the HTV.

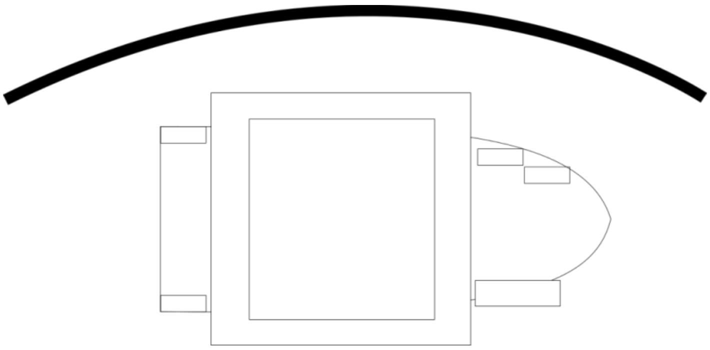

Figure C- 20 Shelter behind an island, schematic top view

Feasibility

It is not possible to perform the operation in the shelter of an island, as per problem statement this is the inherent issue of operating at remote areas. It is possible to perform the operation in the shelter of a huge ship. However it is not very economical to rent a huge ship in order to create a sheltered area. If walls are built between the casings of the HTV a big problem arises, the HTV will not be able to transport all kinds of different cargo, anymore. Furthermore the environmental loads will have a bigger influence on the motions of the HTV, causing the HTV to have larger horizontal motions. It is therefore concluded that this design direction is not feasible, mainly because it is impossible to take the environmental conditions away while operating in the middle of a sea or ocean.

Design direction 7: Support system

This design direction focuses on the layout of the supporting structure. It is based on the possibility that it is impossible to design a cargo handling system which is able to get the relative horizontal motions to fulfil the design requirements (see Chapter 2.6), which are based on the standard cribbing beam dimensions. Therefore different kinds of support structures are considered of which the width can be adjusted. If a support system is used which has a bigger width, it is easier to place the cargo on the support structure. The wider support structure has more places to support the strong points of the cargo.

The following support systems can be considered:

- Place the cargo on sand (see Figure C- 21),
- Place the cargo on floodable and/or air bags (see Figure C- 22),
- Widen the cribbing layout by placing multiple cribbing beams next to each other (see Figure C- 23).

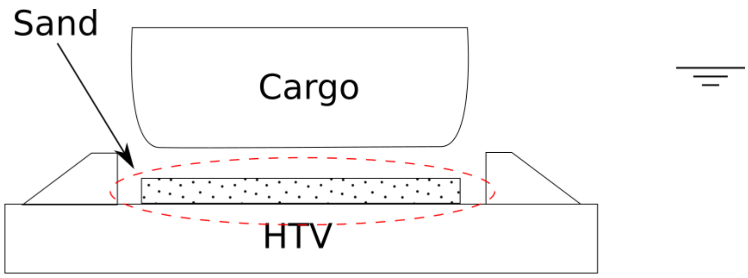

Figure C- 21 Sand as support system, schematic cross section

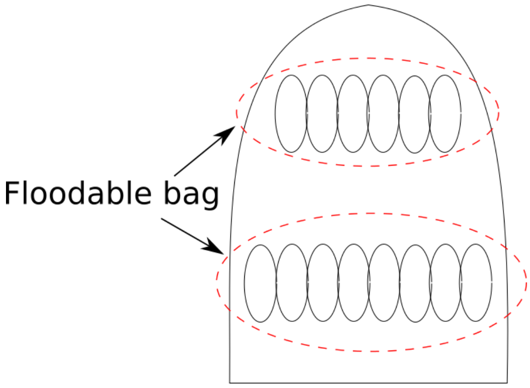

Figure C- 22 Floodable bags as support system, schematic top view

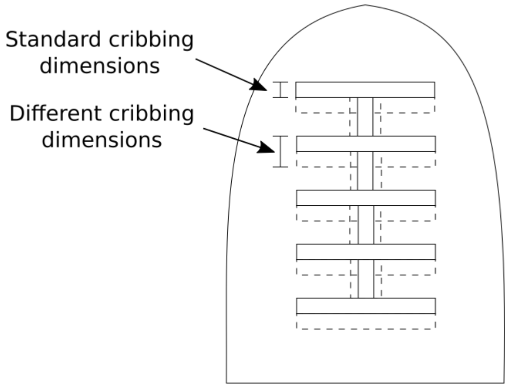

Figure C- 23 Change the cribbing dimensions, schematic top view

Feasibility

It is possible to use sand as the supporting structure. In this case the only limiting factors for the allowed magnitude of the relative horizontal motions are the dimensions of the HTV. However the problem with sand in a wave sensitive area is that the waves are capable of washing the sand away of the deck of the HTV. Another problem is that sand is incompressible. Therefore all the weight of the cargo is transported to the sand through the protrusions which are located underneath the cargo. This is an undesirable situation and therefore this solution is considered to be unfeasible.

Bags can be used to overcome the problems involved in the use of sand as a support system. The deck of the HTV will be covered with bags. In this way the cargo can be placed everywhere on the deck of the HTV. The cargo is positioned onto the bags and the bags underneath the protrusions can be emptied, thus the cargo will only rest on the strong points of the keel. The bags can also be used to absorb the vertical impact loads between the cargo and the HTV. However this is outside the scope of this thesis.

Another option is to use the traditional cribbing layout. The available width of this system can be changed by laying multiple cribbing beams next to each other instead of using only one cribbing beam.

Conclusion
The solutions presented for this design direction are very interesting. However it is concluded that these solutions are only usable if the relative horizontal motions are already, to a certain amount, reduced by the relative horizontal motion reduction system. These solutions only widen the allowable relative motions of the cargo and the HTV, but are not reducing the relative horizontal motions. Therefore these solutions can only be used in combination with another system which is already able to reduce the relative horizontal motions compared to the standard cargo handling system.

Conclusion
The conducted analysis showed that the design directions: *One body copies the motions of another body* and *Take away/Reduce motions* are not feasible to solve the offshore loading and discharge problem. The design directions *Quick loading* and *Support system* provide solutions which can be used if the relative horizontal motions are, to a certain amount, reduced by the cargo handling system. The design directions *Connection with lines, Friction* and *Stiff system* provide solutions which, in some cases combined with another solution, prove to be feasible concepts to solve the offshore loading and discharge problem.

Although all the mentioned concepts started off as individual designs, based on a different basic idea, it is concluded that some of the concepts turned out to be almost the same. Therefore it is decided, for the remaining of the design process, to combine some concepts in one concept.

The found concepts are:

- Concept 1: Cargo bottom lines,
- Concept 2: Stretch compensation system,
- Concept 3: Teflon pads (combined with Trolleys),
- Concept 4: Clamping (combined with Sea fastening),
- Concept 5: Converging vertical system.

C.2 Criteria and weight factors
This section provides a description of the criteria which are used in the MCA. This section also explains how the weight factors for the different criteria are determined.

Criteria
The following criteria are defined.

Construction	Technical challenges faced during the construction phase
Installation	The challenges and amount of work to install the system on the HTV
Maintenance	The amount of maintenance involved per component
Reuse/adjustability	What is the difficulty involved in using the system another time for another type of cargo again
Attachments to cargo	Changes to the cargo required to connect the cargo handling system to the cargo
Complexity	The amount of systems needed to accomplish the job
Efficiency	The amount of work needed to connect the system to the cargo
Energy consumption	The amount of energy which is required to let the system work
Controllability	How well can the system control the motions of the cargo, or locate them on the correct spot on the supporting structure
Safety cargo	Protection of the cargo against damage
Fail safe/Redundancy	Protection against damages, errors, accidents, etcetera

Weight factors

Some criteria are more important than other criteria. Therefore the criteria have different weight factors. The weight factors are determined using a cross criteria analysis (see Table C- 2). In the first row and the left column all the criteria are listed. In order to determine the weight factor per criterion, it is necessary to determine which of two criteria has more priority. This determination is based on the criteria in the rows. Consequently it is per column determined which criterion has the most priority. The most important criterion gets a 1, the other scores a 0. Eventually the weight factors follow from this schedule.

Table C- 2 Weight factors

	Construction	Installation	Maintenance	Reuse/adjustability	Attachments to cargo	Complexity	Efficiency	Energy consumption	Controllability	Safety to cargo	Safety/Redundancy	sum	weight factor
Construction	x	0	0	0	0	0	1	1	0	0	0	2	0,04
Installation	1	x	0	0	1	0	0	1	0	0	0	3	0,05
Maintenance	1	1	x	0	1	0	1	0	0	0	0	4	0,07
Reuse/adjustability	1	1	1	x	1	0	0	1	0	0	0	5	0,09
Attachments to cargo	1	0	1	1	x	0	0	1	0	0	0	4	0,07
Complexity	1	1	1	1	1	x	1	1	0	0	0	7	0,12
Efficiency	0	1	0	1	1	0	x	1	1	0	0	5	0,09
Energy consumption	0	0	1	0	0	0	0	x	0	0	0	1	0,02
Controllability	1	1	1	1	1	1	0	1	x	1	0	8	0,14
Safety to cargo	1	1	1	1	1	1	1	1	0	x	0	8	0,14
Safety/Redundancy	1	1	1	1	1	1	1	1	1	1	x	10	0,18

Appendix D Preliminary design Clamping system

This appendix will provide additional information about design of the clamping system. The first section explains the definition of fluid power. The second section presents the derivation of the equations which are used to simulate the system. The third section describes the used Simscape models. The fourth section presents the dimensions of the damping device. The fifth section provides additional simulation results for the parametric study. The sixth section presents additional results of the performance of the clamping system.

D.1 Fluid power

In fluid power engineering a pressurized fluid is used to perform some form of work. The term fluids can refer to either liquids or gases, and fluid power is the term used for hydraulics and pneumatics. Each type of fluid has its own characteristics and is therefore applied in different systems. A liquid is almost incompressible, especially compared to gasses, and it has a good dynamic response. However it is only able to operate in a confined temperature range specified by the fluid. A gas is highly compressible and is able to operate at high temperatures, however it has a limited dynamic response [15].

The basic properties of a fluid are the density ρ, a substance mass per unit volume, and the viscosity ν, a measure of a fluid's resistance to flow. The compressibility is also fluid specific but is different for liquids and gasses [15].

The relationship between compressibility and pressure of a fluid is dependent on the thermodynamic nature of the process: isothermal or isentropic. A process is isothermal when the process takes place without a change in temperature, this occurs when the fluid is allowed to compress or expand freely, without performing external work. An isothermal process is only of significance for slow compression processes [17]. A process is isentropic if there is no heat loss, in the system, during the process. These conditions are met in practice in rapid processes [15].

Compressibility of a liquid

Although liquids are often treated as incompressible, all liquids possess some degree of compressibility. The compressibility of a liquid is described by its bulk modulus β, the resistance of a liquid to uniform compression. It describes the ratio of the indefinitely small pressure increase to the resulting relative decrease of the volume. As a result, a liquid with a high bulk modulus experiences almost no decrease in volume while a liquid with a low bulk modulus experiences a lot of decrease in volume, while they experience the same amount of pressure increase [15].

The isothermal bulk modulus is calculated with [17]:

$$\beta_t = -V\frac{dP}{dV} \tag{D-1}$$

As isothermal conditions apply to very slow processes, a more detailed value of compressibility is given with the isentropic bulk modulus [17]:

$$\beta_s = \gamma\beta_t \tag{D-2}$$

$$\gamma = \frac{C_p}{C_v} \tag{D-3}$$

The used variables in the mentioned equation are:

C_p	=	Specific heat at constant pressure of the liquid	[J/kg K]
C_v	=	Specific heat at constant volume of the liquid	[J/kg K]
P	=	Pressure	[Pa]
V	=	Volume	[m^3]
β_s	=	Isentropic bulk modulus	[Pa]
β_t	=	Isothermal bulk modulus	[Pa]
γ	=	Heat capacity ratio of liquid	[-]

However the bulk modulus is usually assumed a constant when working at the ordinary rate of pressures and temperatures [37].

Compressibility of gas

Boyle's law is applicable for an ideal gas in a closed system under isothermal conditions. The law can be stated as follows: if a given mass of gas is compressed or expanded at a constant temperature, then the absolute pressure is inversely proportional to the volume [15]:

$$PV = K_t \tag{D-4}$$

Isentropic compression or expansion of the gas is approximated in practice by the rapid compression or expansion of the gas. In this case the relationship between the pressure and the volume of a gas changes to [15]:

$$PV^{\gamma} = K_s \tag{D-5}$$

$$\gamma = \frac{C_p}{C_v} \tag{D-6}$$

The used variables in the mentioned equation are:

C_p	=	Specific heat at constant pressure of the gas	[J/kg K]
C_v	=	Specific heat at constant volume of the gas	[J/kg K]
P	=	Pressure	[Pa]
V	=	Volume	[m^3]
K_s	=	A constant representing isentropic conditions	[-]
K_t	=	A constant representing isothermal conditions	[-]
γ	=	Heat capacity ratio of gas	[-]

The constants K_s and K_t differ from system to system.

It can be concluded from equation (D-4) that if the pressure of a gas increases, the volume of the gas decreases; as the product of the two quantities remains constant. As a result the gas is compressed with increasing pressure and it expands with decreasing pressure.

In practice, most compression and expansion processes tend to follow the isentropic relationship because most of the processes occur very rapid.

D.2 Fluid mechanics based model

The principles of fluid mechanics can be used to develop a simplified model to describe the dynamic behaviour of the damping device.

The general mass flow rate continuity is used to describe the damper behaviour [14]:

$$\frac{dm}{dt} = \frac{d}{dt}(\rho V) = \rho Q \tag{D-7}$$

The centre portion of equation (D-7) can be expanded to [14]:

$$\frac{d}{dt}(\rho V) = \rho\frac{dV}{dt} + V\frac{d\rho}{dt} = \rho\frac{dV}{dt} + V\frac{d\rho}{dP}\frac{dP}{dt} = \rho_i Q \tag{D-8}$$

The used variables are:

m	=	Mass fluid	[kg]
P	=	Pressure within fluid	[Pa]
Q	=	Flow rate into the chamber	[m^3/s]
t	=	Time	[s]
V	=	Volume	[m^3]
ρ	=	Density fluid	[kg/m^3]

The definition of bulk modulus gives [37]:

$$\beta = -V\frac{dP}{dV} \tag{D-9}$$

$$\beta = \rho\frac{dP}{d\rho} \tag{D-10}$$

If equations (D-8), (D-9), (D-10) are combined the following formulae are derived:

$$\rho(\frac{dV}{dt} + \frac{V}{\beta}\frac{dP}{dt}) = \rho Q \tag{D-11}$$

$$\frac{dV}{dt} + \frac{V}{\beta}\frac{dP}{dt} = Q \tag{D-12}$$

Equation (D-12) is the general mass flow rate continuity equation in which the boundary deformation (first term left side) and fluid compressibility (second term left side) are accounted for.

The flow rate into the chamber is calculated according to the following formula:

$$Q = k_d A_v v_v \tag{D-13}$$

From Bernoulli's equation follows that the average velocity of a fluid through a valve is equal to equation (D-14) [15]. This equation utilizes the conservation of energy, mass flow rate continuity, and an incompressible, inviscid fluid is assumed. In an inviscid fluid, the pressure forces on the particles dominate over the viscous forces [14].

$$v_v = \sqrt{\frac{2\Delta P}{\rho}} \tag{D-14}$$

The introduced variables are:

A_v	=	Valve area	[m^2]
k_d	=	Discharge coefficient valve	[-]
v_v	=	Fluid velocity through valve	[m/s]

The mass conservation equations become:

$$\frac{dV_1}{dt} + \frac{V_1}{\beta_1}\frac{dP_1}{dt} = k_d A_v \sqrt{\frac{2|P_1 - P_2|}{\rho}} sign(P_2 - P_1) \tag{D-15}$$

$$\frac{dV_2}{dt} + \frac{V_2}{\beta_2}\frac{dP_2}{dt} = - k_d A_v \sqrt{\frac{2|P_1 - P_2|}{\rho}} sign(P_2 - P_1) \tag{D-16}$$

For sake of simplicity, the accumulator is modelled as a spring which is connected to an accumulator face plate. The spring stiffness and the face plate area can be adjusted to properly account for the accumulator behaviour.

The spring force of the accumulator can be calculated with [14]:

$$K_a x_a = P_2 A_f \tag{D-17}$$

$$x_a = \frac{P_2 A_f}{K_a} \tag{D-18}$$

The introduced variables are:

A_f	=	Accumulator faceplate area	[m^2]
K_a	=	Accumulator spring stiffness	[N/m]
x_a	=	Accumulator faceplate displacement	[m]

The volume of the fluids in the cylinder chambers, and their derivatives, can be calculated with the following formulas:

$$V_1 = (L_1 + x)A_p \tag{D-19}$$

$$\frac{dV_1}{dt} = \dot{x}A_p \tag{D-20}$$

$$V_2 = (L_2 - x + x_a)(A_p - A_r) \tag{D-21}$$

$$V_2 = (L_2 - x + \frac{P_2 \times (A_p - A_r)}{K_a})(A_p - A_r) \tag{D-22}$$

$$\frac{dV_2}{dt} = -\dot{x}\left(A_p - A_r\right) + \frac{dP_2}{dt}\frac{(A_p - A_r)}{K_a}(A_p - A_r) \tag{D-23}$$

The introduced variables are:

A_p	=	Piston head area	[m^2]
A_r	=	Piston rod area	[m^2]
L_1	=	Length chamber 1 at the begin of operation	[m]
L_2	=	Length chamber 2 at the begin of operation	[m]
x	=	Piston displacement	[m]
$\dot{x}$	=	Piston velocity	[m/s]

The combination of all the above equations leads to the differential equations which are presented in Chapter 4.3.

D.3 Simscape model

Two different Simscape models are created in order to mimic the system characteristics. One for the basic damping device and one for the designed damping device.

Basic damping device

A SimHydraulics model of the basic damping device is created in Simcape (see Figure D- 1). The two chambers of the double acting hydraulic cylinder are connected to each other by a one way valve and a fixed orifice. The fixed orifice represents the control valve.

A gas-charged accumulator is used to take up the piston rod space, if the piston is pushed inside the cylinder. This is different compared to the *fluid mechanics model* where a spring-charged accumulator is used.

The hydraulic fluid block specifies the characteristics of the fluid used in the system, like the density, viscosity and bulk modulus.

The model also consists of parts of the Mechanical Translational domain besides the hydraulic components. The double acting hydraulic cylinder possesses two mechanical ports: C and R. These ports are used to determine the motions and forces corresponding to the pressures inside the cylinder. The C port represents the backend of the cylinder while the R port represents the piston. In the Simscape model, the C port will be connected to a mechanical reference point, such that the system stays in place. The R port will be used to connect the system to the induced force or velocity which is working on the system. With this model the characteristics of the basic damping device can be defined.

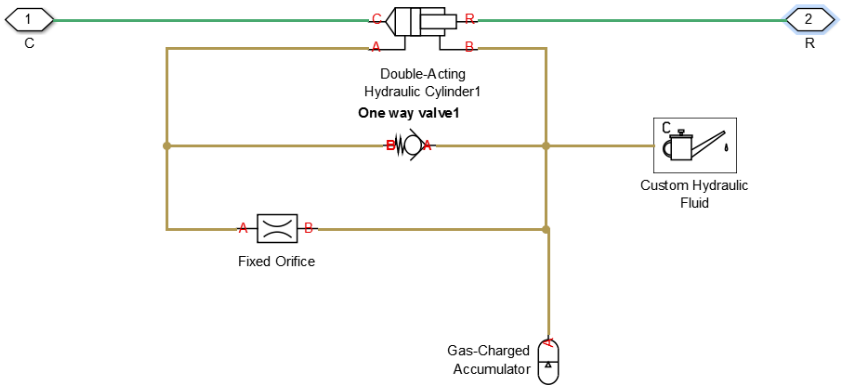

Figure D- 1 SimHydraulics model of the basic damping device

Damping device

A Simhydraulics model of the designed damping device is also created in Simscape (see Figure D- 2). The model consists of the model of the basic damping device to which certain features are added: a translational spring, a pressure relief valve and a hydraulic resistive tube.

The translational spring represents the fender, which is used to pull the piston rod back to its initial position.

The hydraulic resistive tube accounts for all the resistance which is present within the damping device. The hydraulic fluid passes corners, bends and other resistive components and this causes friction between the fluid and the walls of the system; the system experiences some amount of resistance.

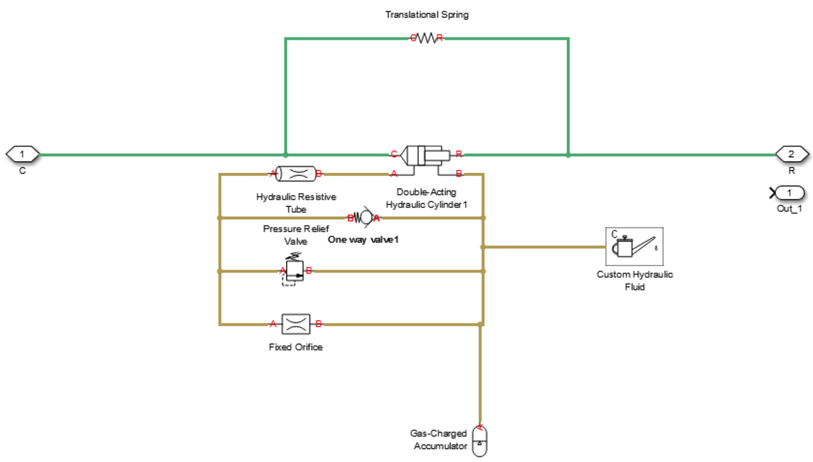

Figure D- 2 SimHydraulics model of the basic damping device

D.4 Dimensions

Before the Simscape simulations can be conducted, the system needs to be initialized. This procedure consists of defining the dimensions and the initial conditions of the damping device. The involved variables will be discussed in the following sections.

Fixed variables

The following dimensions and conditions of the damping device are considered to be fixed. The values of these parameters stay constant for all the loading and discharge operations which the system will perform.

At the start of the design process the diameter of the rod was calculated according to the formula of Euler for buckling (see equation (D-24)). The piston rod is subjected to large compression forces and therefore buckling needs to be taken into account.

$$F_c = \frac{\pi^2 EI}{(KL)^2} \tag{D-24}$$

The used variables in the equation are:

E	=	Modulus of elasticity	[N/m^2]
F_c	=	Critical force	[N]
I	=	Area moment of inertia	[m^4]
K	=	Effective length of the column factor	[-]
L	=	Unsupported rod length	[m]

The diameter of the cylinder is based on the maximum force and allowed pressure inside the cylinder. The maximum force on the cylinder was taken to be 5000 [kN], it is assumed that the cargo is clamped by 5 systems at each side of the cargo. The maximum force which the system experiences, if the cargo is connected rigidly to the HTV is 25.000 [kN] (see Chapter 2.2); for one system it is 5000 [kN]. The maximum allowed pressure in the system is set at 300 bar. In this way a rod diameter of 0.15 [m] and a cylinder diameter of 0.5 [m] were obtained. However during the simulations it was found that the forces which the system experiences are larger than 5000 [kN], due to impact forces which arise in the system when the cargo makes contact with the damping devices (see Chapter 4.5). Therefore the dimensions of the rod and the cylinder were increased such that the system is able to perform its duty. The fixed dimensions are presented in Table D- 1.

Table D- 1 Fixed properties of the damping device

	Damper cylinder dimensions		
L_1	Length chamber 1 (from piston head to fluid inlet)	0.9	[m]
L_2	Length chamber 2 (from piston head to fluid inlet)	0.1	[m]
S	Piston stroke	1	[m]
D	Diameter cylinder	0.75	[m]
D_r	Diameter piston rod	0.35	[m]
	Control valve		
k_{d_cv}	Discharge coefficient control valve	0.7	[-]
D_{cv}	Diameter control valve	0.05-0.00	[m]
	One way valve		
k_{d_owv}	Discharge coefficient one way valve	0.7	[-]
D_{owv}	Diameter one way valve	0.05	[m]
	Pressure relief valve		
k_{d_prv}	Discharge coefficient pressure relief valve	0.7	[m]
D_{prv}	Diameter pressure relief valve	0.05	[m]
P_{prv_set}	Pre-set pressure level at which the pressure relief valve opens	30 x 10^6	[Pa]
P_{prv_reg}	Pressure increase over the pre-set level needed to fully open the valve	5 x 10^6	[Pa]

Hydraulic fluid

The hydraulic fluid used in the damping device is Shell Tellus S2 V32. The properties of this fluid are given in Table D- 2, for a temperature of 15 degrees Celsius and a mean working pressure of 18.5 mega pascal [38].

Table D- 2 Properties of Shell Tellus S2 V32

	Hydraulic fluid		
ρ	Density	872	[kg/m^3]
v	Viscosity	100	[cSt]
r_v	Volume of entrained air	0.0001	[-]
γ_f	Heat capacity ratio	1.17 x 10^6	[Pa]
β_t	Isothermal bulk modulus	19 x 10^8	[Pa]
β_s	Isentropic bulk modulus	22 x 10^8	[Pa]

Resistance

The two chambers of the hydraulic cylinder and the valves are connected with each other by tubes. This forms the complete system where the fluid is flowing through. However every hydraulic system is subjected to some resistance as the fluid needs to flow through valves, bends, corners, etcetera. The involved friction causes a pressure loss, which can be calculated with the *Darcy-Weisbach* equation [37]:

$$\Delta P = f\rho \frac{L_t}{D_t}\frac{v^2}{2} \tag{D-25}$$

The used variables in the equation are:

f	=	Friction coefficient	[-]
D_t	=	Hydraulic diameter tube	[m]
L	=	Length tube	[m]
P	=	Pressure	[Pa]
v	=	Velocity fluid	[m/s]
ρ	=	Density fluid	[kg/m^3]

It follows from equation (D-25) that the pressure loss is proportional to the square of the fluid velocity, which is equal to the piston velocity. A large piston velocity will give a large pressure loss while a small piston velocity will give a little pressure loss. The friction coefficient is dependent on the type of flow which exists within the system: laminar, turbulent or a transition from laminar to turbulent flow.

The friction coefficient for laminar flow (Reynolds number (Re) smaller or equal to 2000) can be calculated with the following equation [37]:

$$f = \frac{64}{Re} \tag{D-26}$$

In this formula, the number 64 represents the shape factor for a tube with a circular cross section. The shape factor is depended on the shape of the cross section the fluid is flowing through and is based on experimental data and theory.

The friction coefficient for tubular flow (Re bigger or equal to 4000) can be calculated according to the White-Colebrook equation [37]. However in this equation the friction coefficient needs to be calculated iteratively which is very time consuming. It is also possible to approximate the friction coefficient with the Haaland approximation, which is used in Simscape [39]. The used values are presented in Table D- 3.

The friction factor for the transition flow (2000 < Re < 4000) is approximated with the linear interpolation between the extreme points of the regimes [39].

Table D- 3 Properties of the tube connecting the two chambers

Resistance			
D_t	Diameter of the tube	0.05	[m]
h_r	Roughness height	1.5×10^{-5}	[m]

D.5 Parametric study

This section presents the simulation results of the different performed parametric studies.

Frequency response

Figure D- 3 shows the response of the system for different masses. The excitation force is set at 1000 [kN]. The control valve is set at 0.05 [m] and 0.03 [m]. Figure D- 4 shows the response of the system for different cylinder dimensions. The used dimensions are based on the case in which the system performs its task and the one in which the impact loads were too big for the system to perform its task. However the value of the excitation force is set at 1000 [kN] and the system is capable to cope with the impact forces which arise in this case.

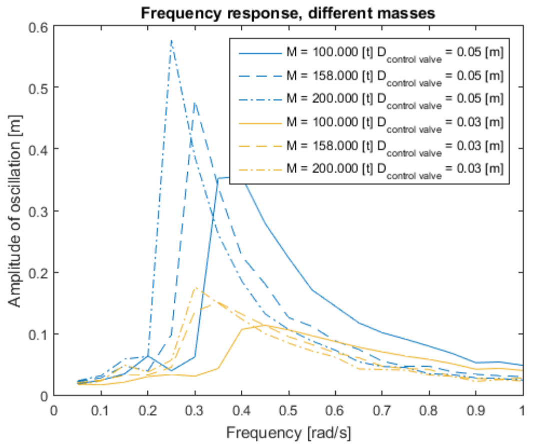

Figure D- 3 Frequency response, different masses

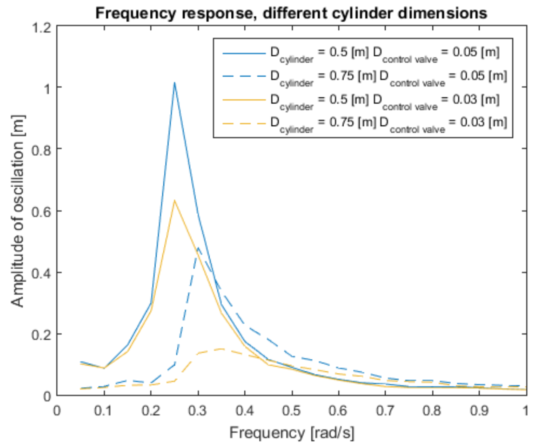

Figure D- 4 Frequency response, different cylinder dimensions

Free decay test

Figure D- 5 shows the free decay test for different masses for two different control valve diameters. Figure D- 6 shows the free decay test for different cylinder dimensions for two different control valve diameters.

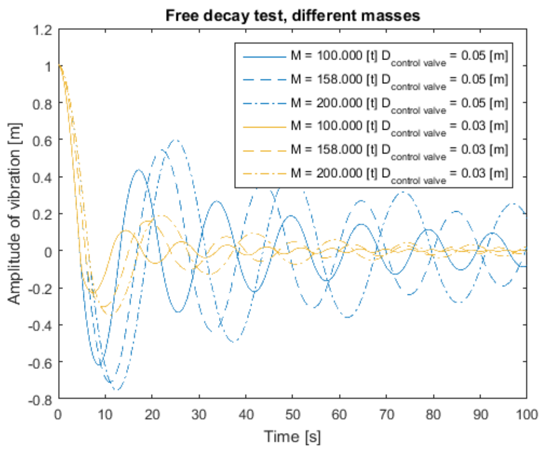

Figure D- 5 Free decay test, different masses

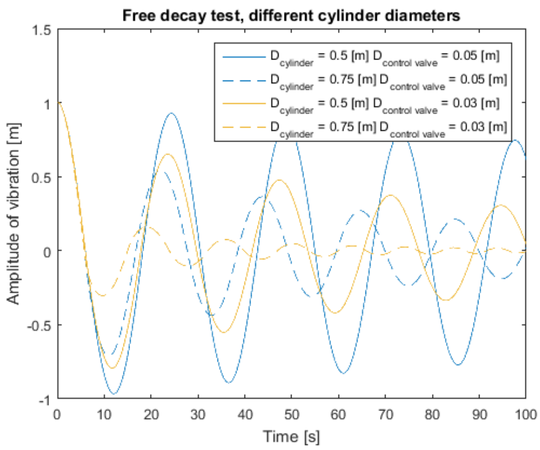

Figure D- 6 Free decay test, different cylinder diameters

D.6 Performance damping device

This section presents the additional simulation results for the different performed performance simulations of the clamping system.

Influence of pre-compression of the fender

Figure D- 7 and Figure D- 8 show the motions and velocities which occur in the system.

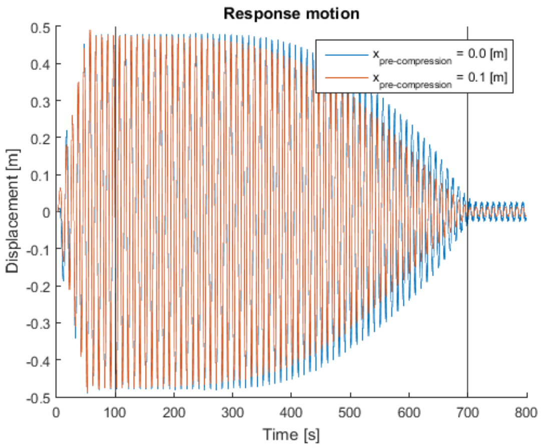

Figure D- 7 Response motion, pre-compression fender

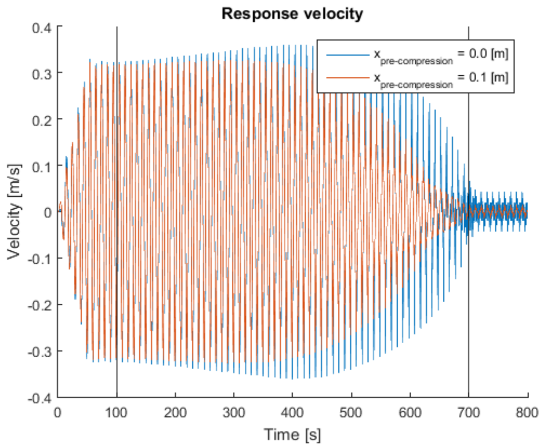

Figure D- 8 Response velocity, pre-compression fender

Base case scenario

In this section the additional results of the base case scenario are presented.

Influence number of devices

Figure D- 9 and Figure D- 10 show the motions and velocities which occur in the system.

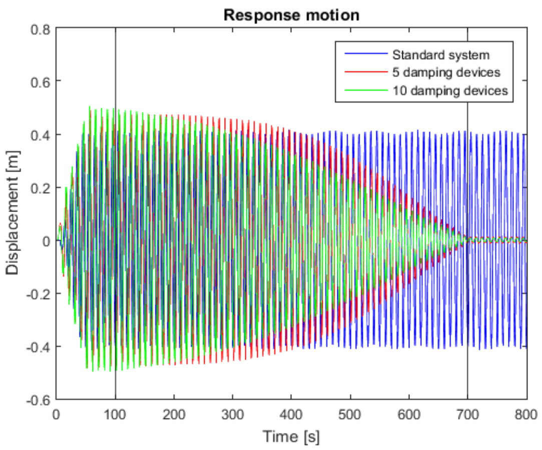

Figure D- 9 Response motion, different amount of damping devices

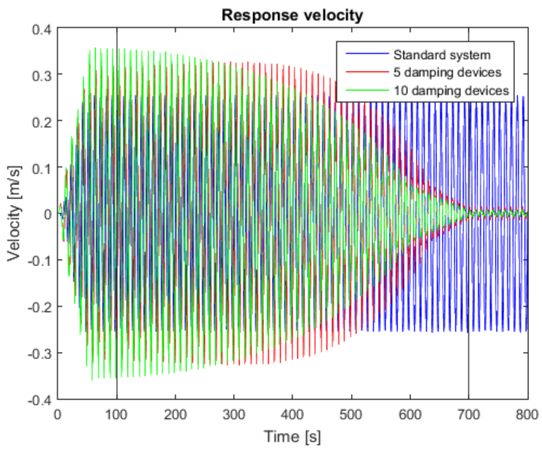

Figure D- 10 Response velocity, different amount of damping devices

Influence relief pressure

Figure D- 11 to Figure D- 14 show the motions and velocities which occur in the system.

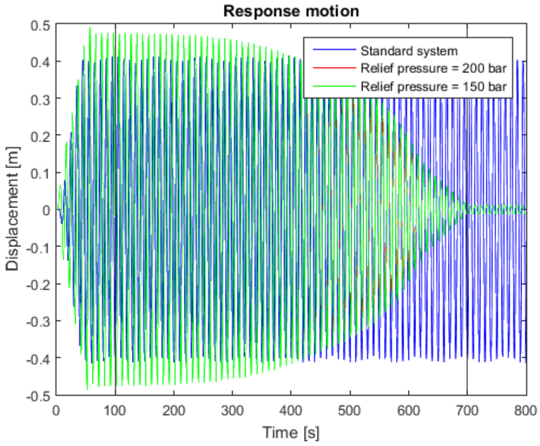

Figure D- 11 Response motion, different relief pressures (200 bar, 150 bar)

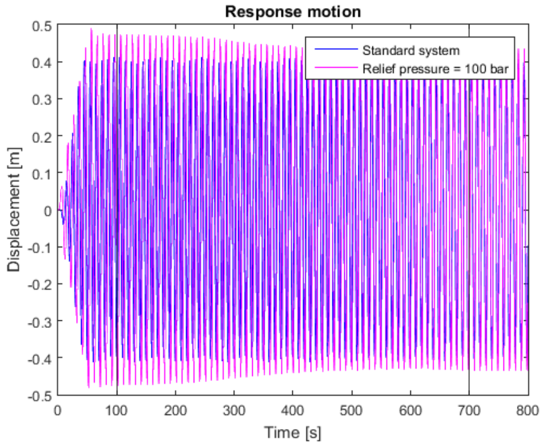

Figure D- 12 Response motion, different relief pressures (100 bar)

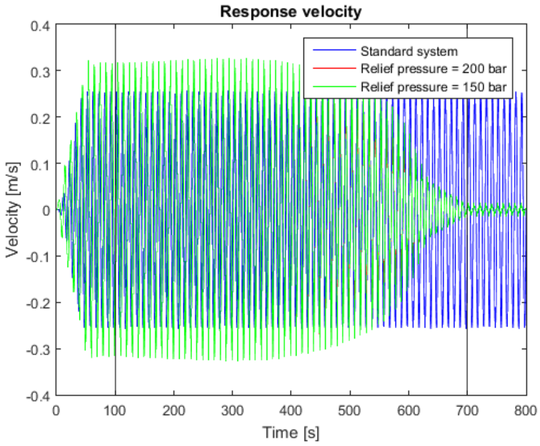

Figure D- 13 Response velocity, different relief pressures (200 bar, 150 bar)

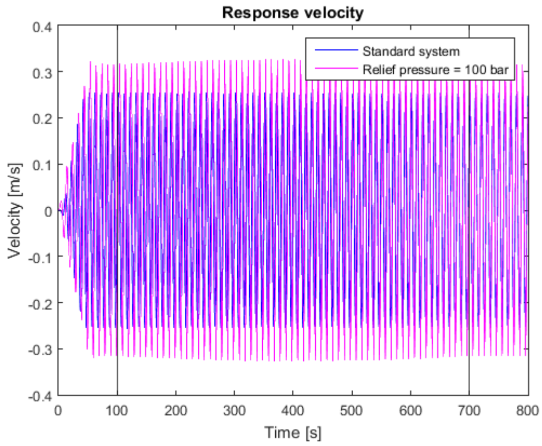

Figure D- 14 Response velocity, different relief pressures (100 bar)

Excitation frequency close to the natural frequency of the relative motion reduction system

Figure D- 15 and Figure D- 16 show the motions and velocities which occur in the system.

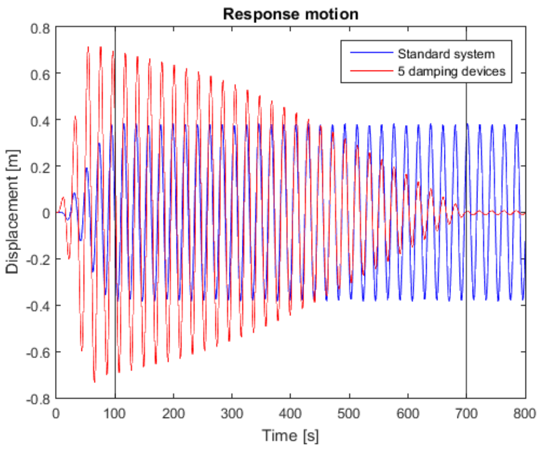

Figure D- 15 Response motion, excitation frequency close to the natural frequency of the new cargo handling system

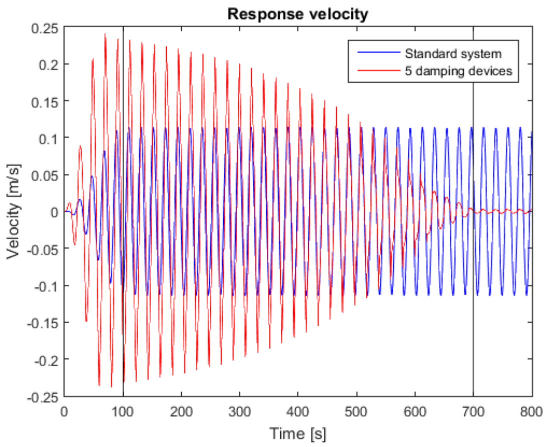

Figure D- 16 Response velocity, excitation frequency close to the natural frequency of the new cargo handling system

Irregular external force signal

This section will focus on the responses of the system when the system is excited by an irregular wave force.

Case 1: ω1 = 0.3 [rad/s], ω2 = 0.63 [rad/s

Figure D- 17 and Figure D- 18 show the motions and velocities which occur in the system.

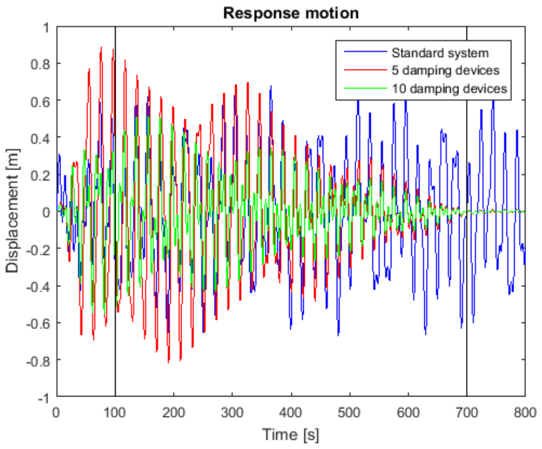

Figure D- 17 Response motion, irregular force case 1

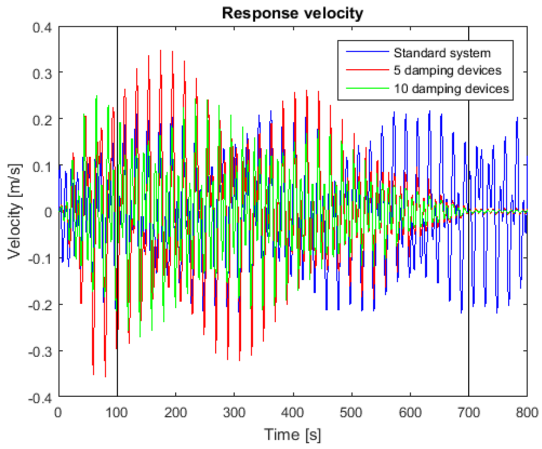

Figure D- 18 Response velocity, irregular force case 1

Case 2: ω_1 = 0.1 [rad/s], ω_2 = 0.63 [rad/s]

In this section a closer look is taken at the response motions of the system when the first excitation frequency is close to the natural frequency of the standard cargo handling system. The second excitation frequency is located away of any natural frequency of the systems. The following values are used in this simulation: ω_1 = 0.1 [rad/s], F_1 = 500 [kN], ω_2 = 0.63 [rad/s], F_2 = 25,000 [kN].

Both excitation frequencies are clearly present in the response of the standard system, although the beating effect is governing (see Figure D- 19). The beats are the effect of the fact that the first excitation frequency, ω_1 = 0.1 [rad/s], is close to the natural frequency of the present system. By applying the damping devices, the amplitude of the response motions of the system is reduced by 77% in the case of 5 damping systems and 75% in the case of 10 damping devices, at the start of the operation. The motions are reduced even more as the control valve within the damping device is closed over time. Moreover the beats have disappeared.

It is clearly visible that the damping devices generate a lot of force and absorb a lot of energy, this is visible through the large force envelope in Figure D- 23. The damping force of a damping device is dependent on the piston velocity. The piston velocity is dependent on the velocity of the cargo. Figure D- 20 clearly shows that the velocities inside the system are governed by the high frequent part of the irregular excitation force. Therefore it can be concluded that the irregular force is dominated by the high-frequent force signal.

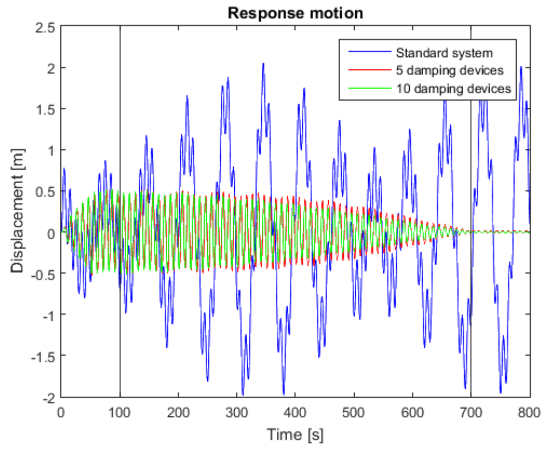

Figure D- 19 Response motion, irregular force case 2

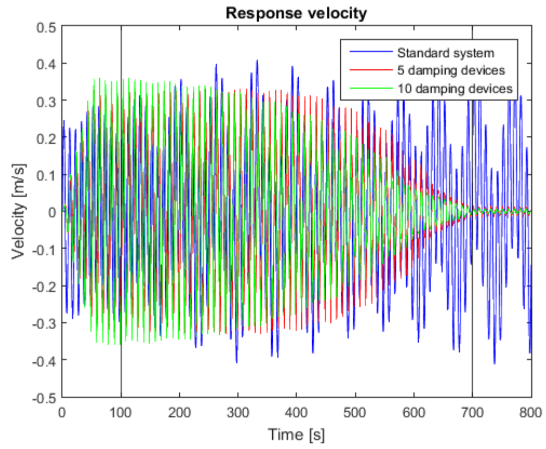

Figure D- 20 Response velocity, irregular force case 2

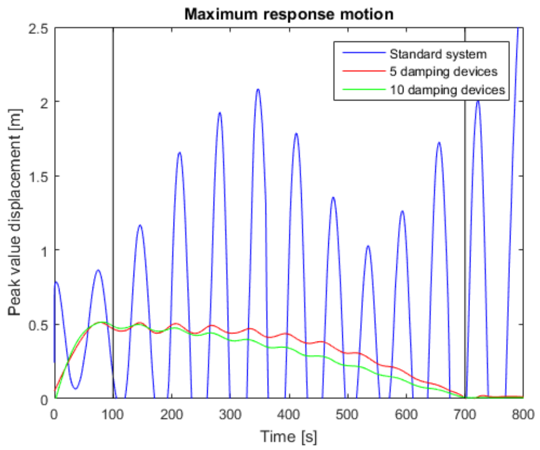

Figure D- 21 Maximum response motion, irregular force case 2

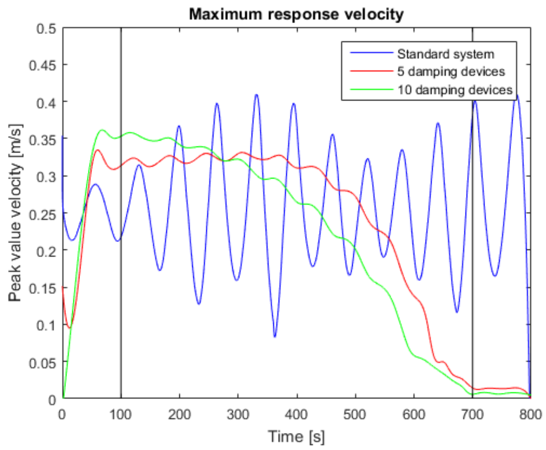

Figure D- 22 Maximum response velocity, irregular force case 2

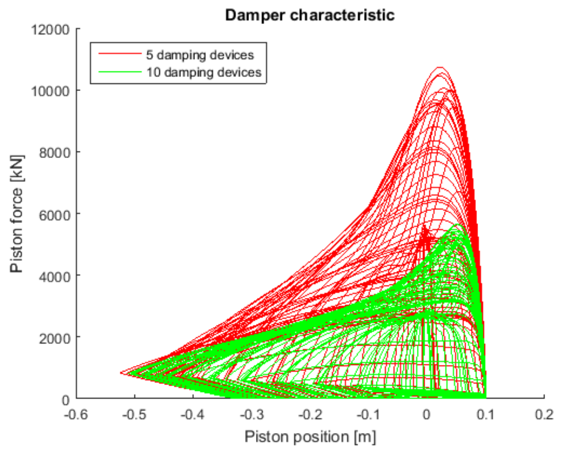

Figure D- 23 Damper characteristic, irregular force case 2

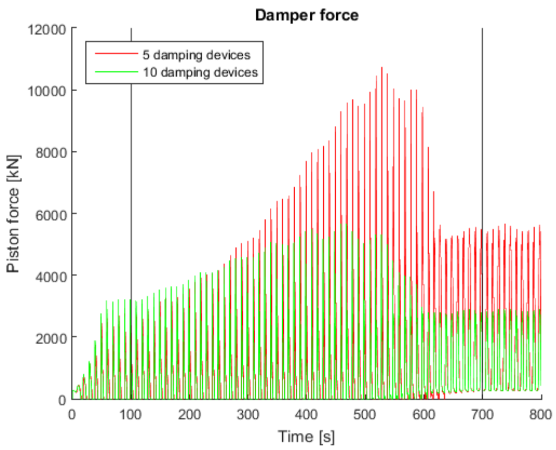

Figure D- 24 Damper force, irregular case 2

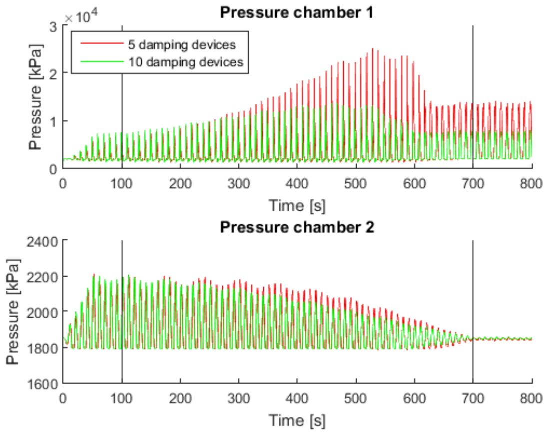

Figure D- 25 Pressure inside the cylinder chambers, irregular force case 2

Case 3: ω_1 = 0.2 [rad/s], ω_2 = 0.63 [rad/s]

In this section a closer look is taken at the response motions of the system when the excitation frequencies are not close to a natural frequency of the cargo handling system. Consequently, the response motions of the system are not driven by resonance. The following values are used in this simulation: ω_1 = 0.2 [rad/s], F_1 = 1000 [kN], ω_2 = 0.63 [rad/s], F_2 = 25,000 [kN].

Both the excitation frequencies can be seen in the response of the present system (see Figure D- 26). The addition of the damping devices increases the stiffness of the cargo handling system. The increase in stiffness has two effects on the motions of the system, at some points in time the motions are larger than for the standard system while at other points the motions are smaller than for the standard system. The motions are smaller because the response of the system changed from motions dominated by mass, to motions dominated by stiffness, for the low frequent part. This caused a reduction in the response amplitude. However the increase in stiffness also induces a shift in natural frequency. This results in larger response motions of the system for the high frequent part of the excitation force. However as the diameter of the control valve within the damping devices is slowly reduced over time, the resulting response amplitude of the motions is also reduced. It is also noticed in the figure that the system fulfils the motions requirement, x ≤ 0.15 [m].

It is noticed that the systems absorb a lot of energy as large force envelopes are visible in Figure D- 30. It is also visible that the velocities of the pistons are governed by the high frequent part of the irregular excitation force (see Figure D- 27). Therefor it is concluded that the damper force is always dominated by the high frequent force signal.

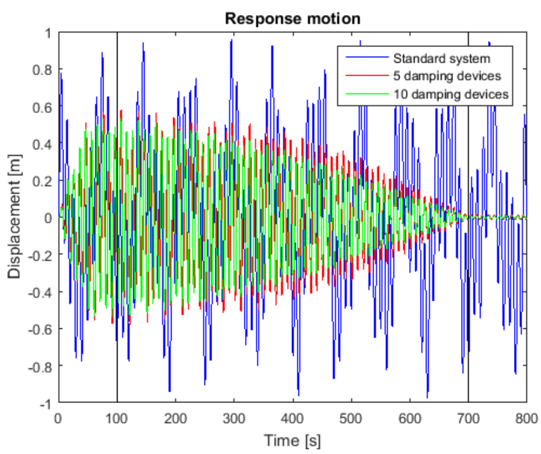

Figure D- 26 Response motion, irregular force case 3

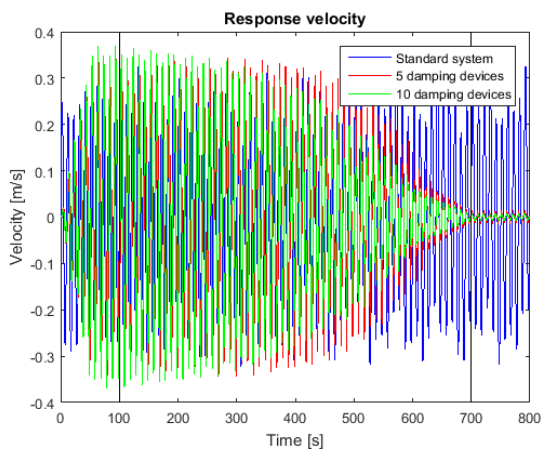

Figure D- 27 Response velocity, irregular force case 3

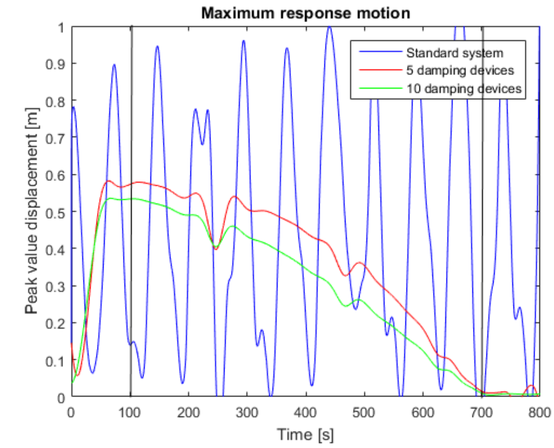

Figure D- 28 Maximum response motion, irregular force case 3

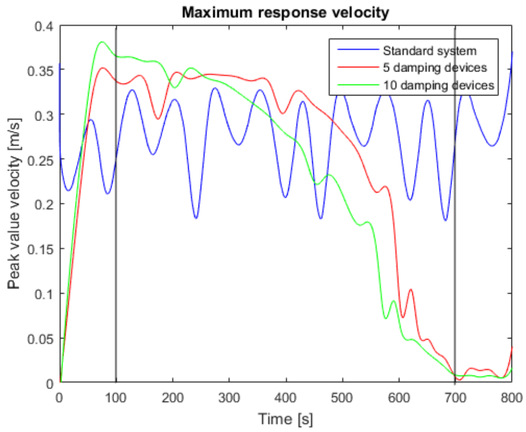

Figure D- 29 Maximum response velocity, irregular force case 3

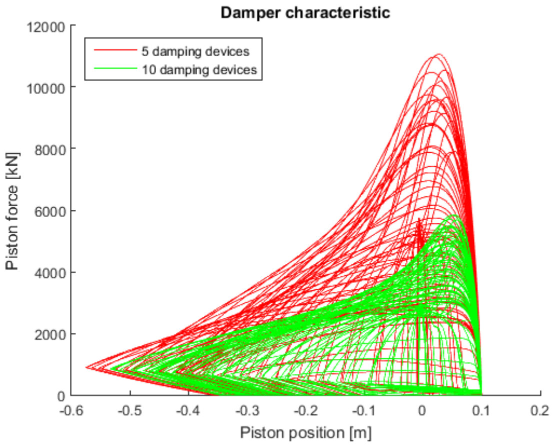

Figure D- 30 Damper characteristic, irregular force case 3

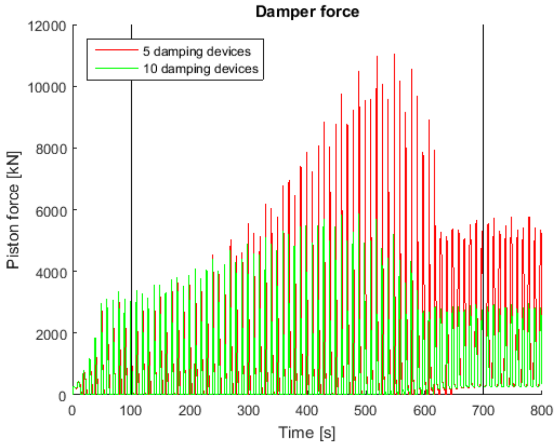

Figure D- 31 Damper force, irregular force case 3

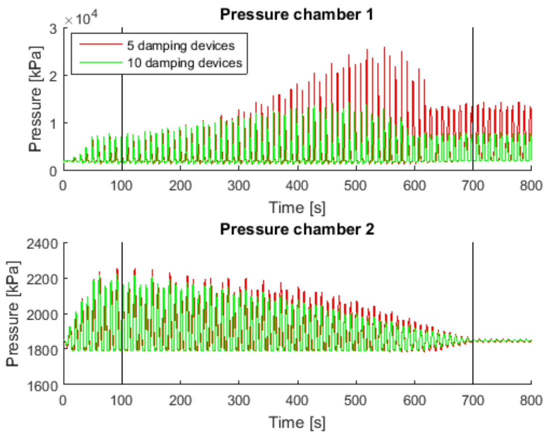

Figure D- 32 Pressure inside the cylinder chambers, irregular force case 3

Appendix E Preliminary design Line tension actuator

This appendix will provide additional information about the design of the *Line tension actuator*. The first section presents the formulas used in the pneumatic system. The second section describes the used Simscape models. The third section presents the dimensions of the system. The fourth section provides additional simulations results for the parametric study. The fifth section presents additional results of the performance of the *Line tension actuator*.

E.1 Formulas pneumatic system

The pneumatic models are based on three equations [21]:

- The ideal gas law,
- The conservation of mass,
- The energy equation.

The following assumptions are made in these equations: the gas is perfect, the pressures and temperatures in each chamber are homogeneous and kinetic and potential terms are negligible.

The ideal gas law is given as [21]:

$$P = \rho R T \tag{E-1}$$

The mass flow rate continuity equation is given as [21]:

$$\frac{dm}{dt} = \dot{m} = \dot{m}_{in} - \dot{m}_{out} = \frac{d}{dt}(\rho V) = \frac{d\rho}{dt}V + \rho\frac{dV}{dt} \tag{E-2}$$

The energy equation is written as [21]:

$$q_{in} - q_{out} + \gamma C_v(\dot{m}_{in}T_{in} - \dot{m}_{out}T) - \dot{W} = \dot{U} \tag{E-3}$$

The used variables are:

C_v	=	Specific heat at constant volume of gas	[J/kg K]
m	=	Mass gas	[kg]
$\dot{m}_{in,out}$	=	Mass flows entering and leaving the chamber	[kg/s]
P	=	Pressure	[Pa]
$q_{in,out}$	=	Heat transfer terms	[W/m^2]
R	=	Specific gas constant used gas	[J/kg K]
T	=	Temperature	[K]
T_{in}	=	Heat incoming gas flow	[K]
$\dot{U}$	=	Change in internal energy	[J]
V	=	Volume	[m^3]
$\dot{W}$	=	Rate of change in performed work	[Nm]
γ	=	Heat capacity ratio	[-]
ρ	=	Density gas	[kg/m^3]

The amount of work which is done by a gas is calculated as:

$$W = \int P dV \tag{E-4}$$

Therefore the rate of change of the amount of work is:

$$\dot{W} = P\frac{dV}{dt} \tag{E-5}$$

The ideal gas relation can be used to define C_v:

$$C_p - C_V = R \tag{E-6}$$

C_p is the specific heat at a constant pressure of the gas. By combining (E-6) and (D-6)the following equation is derived:

$$C_v = \frac{R}{\gamma - 1} \tag{E-7}$$

The rate of change in internal energy is defined as:

$$\dot{U} = \frac{d}{dt}(C_V mT) = \frac{R}{\gamma - 1}\left(\frac{d\rho}{dt}V + \rho\frac{dV}{dt}\right)T \tag{E-8}$$

From equation (E-1) the following relations are derived for the density:

$$\rho = \frac{P}{RT} \tag{E-9}$$

$$\frac{d\rho}{dt} = \frac{dP}{dT}\frac{1}{RT} \tag{E-10}$$

By combining equations (E-2) and (E-7), the following formula for the rate of change in energy is obtained:

$$\dot{U} = \frac{1}{\gamma - 1}(V\frac{dP}{dt} + \frac{dV}{dt}P) \tag{E-11}$$

If equations (E-1), (E-3), (E-4), (E-7) and (E-11) are combined the energy equation can be written as:

$$q_{in} - q_{out} + \frac{\gamma}{\gamma - 1}\frac{P}{\rho T}(\dot{m}_{in}T_{in} - \dot{m}_{out}T) - \frac{\gamma}{\gamma - 1}P\frac{dV}{dt} = \frac{1}{\gamma - 1}V\frac{dP}{dT} \tag{E-12}$$

If it is assumed that the incoming gas flow already has the temperature of the gas in the chamber the energy equation can be written as:

$$\frac{\gamma - 1}{\gamma}(q_{in} - q_{out}) + \frac{1}{\rho}(\dot{m}_{in} - \dot{m}_{out}) - \frac{dV}{dt} = \frac{V}{\gamma P}\frac{dP}{dt} \tag{E-13}$$

The system is used in a very dynamic environment in which the processes occur relatively fast, because the waves have a period of 10 [s]. Therefore the processes within the cylinder also occur relatively fast. For this reason the process is considered to be isentropic. In an isentropic process no heat is transferred, i.e. $q_{in} - q_{out} = 0$. If these conditions are applied to equation (E-13), the following differential equation for the pressure in the chamber is obtained:

$$\frac{dP}{dt} = \gamma\frac{P}{\rho V}(\dot{m}_{in} - \dot{m}_{out}) - \gamma\frac{P}{V}\frac{dV}{dt} \tag{E-14}$$

If equation (E-9) is substituted in equation (E-14), the following differential equation is obtained:

$$\frac{dP}{dt} = \gamma\frac{RT}{V}(\dot{m}_{in} - \dot{m}_{out}) - \gamma\frac{P}{V}\frac{dV}{dt} \tag{E-15}$$

The differential equation for chamber 2 uses the volume and change in volume as input. The amount of volume and its derivative for chamber 2 can be calculated with the following equations:

$$V_2 = (L_2 + x) \times (A_p - A_r) \tag{E-16}$$

$$\frac{dV_2}{dt} = \dot{x} \times (A_p - A_r) \tag{E-17}$$

The used variables are:

A_p	=	Piston head area	[m^2]
A_r	=	Piston rod area	[m^2]
L_2	=	Length chamber 2 at the begin of operation	[m]
x	=	Piston displacement	[m]
$\dot{x}$	=	Piston velocity	[m/s]

The combination of all the above equations leads to the differential equations which are presented in Chapter 0.

E.2 Simscape models

Two different Simscape models are created in order to mimic the system characteristics. One for the hydraulic actuator and one for the pneumatic actuator.

Hydraulic actuator

The Simscape model for the hydraulic actuator is show in Figure E- 1. The model only represents the part of the system which is used during the loading operation; therefore port A of the double acting hydraulic cylinder is coupled to a hydraulic reference.

The operational side of the cylinder is connected to a 3-way directional valve. The low pressure port of the valve is connected to a hydraulic reference, as the fluid will flow to a reservoir under atmospheric pressure. The high pressure port is connected to a hydraulic pressure source. The hydraulic pressure source represents the hydraulic pump which produces a constant fluid pressure.

The piston is connected to a translational spring, through a translational hard stop. The translational spring represents the Dyneema line and is used to pull the piston out of the cylinder. The translational hard stop is used to allow the spring to only experience a tension force. In this way the spring will not produce a pushing force against the piston but only a pulling force.

The piston is also connected to a force sensor. This sensor measures the amount of force the actuator produces. The output signal is fed into the controller. The output signal of the controller is fed into the valve actuator which operates the 3-way directional valve. The valve actuator represents the valve driver.

The hydraulic fluid block specifies the characteristics of the fluid used in the system, like the density, viscosity and bulk modulus.

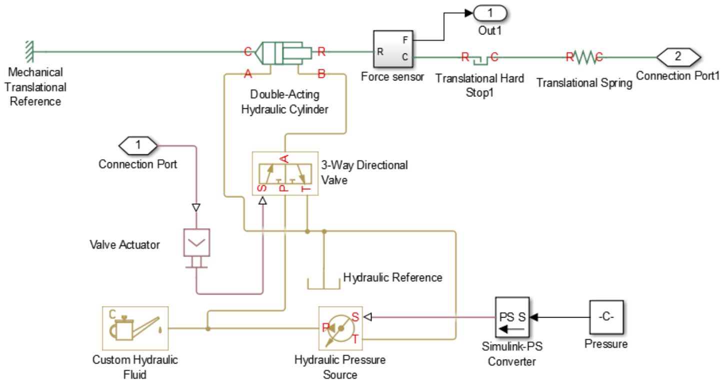

Figure E- 1 SimHydraulics model of the hydraulic actuator

Pneumatic actuator

The Simscape model for the pneumatic actuator is shown in Figure E- 2. The layout of the system is the same as that of the hydraulic actuator. The difference is that the system is filled with gas instead of fluid.

The gas properties block specifies the characteristics of the gas used in the system, like specific heat and viscosity.

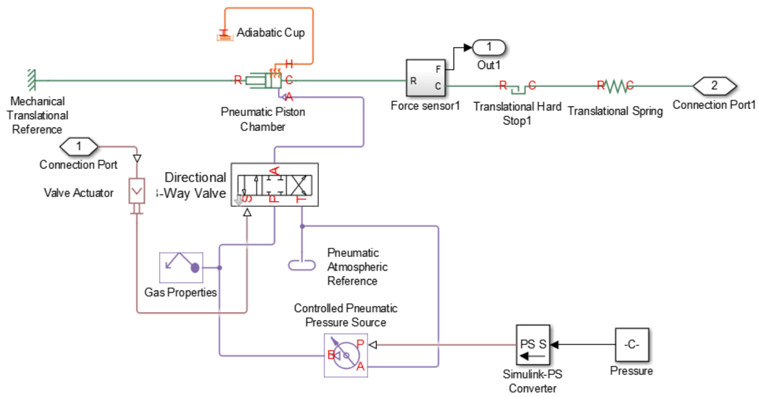

Figure E- 2 SimHydraulics model of the pneumatic actuator

E.3 Dimensions

Before the Simscape simulations can be conducted, the systems need to be initialized. This procedure consists out of defining the dimensions and the initial conditions of the stretch compensation system. The involved variables will be discussed in the following sections.

Fixed variables

The dimensions and conditions of the damping device are considered to be fixed (see Table E- 1). The values of these parameters stay constant for all the loading and discharge operations which the system will perform.

The piston stroke is based on the force/line stiffness relation combined with the resulting motions of the cargo, when the system delivers the correct force. The diameter of the rod is based on the dimensions of the Dyneema line. A characteristic of the Dyneema lines is that they are almost as strong as steel. Furthermore is the piston rod only subjected to tension forces, buckling in the rod is not an issue as no compression force is induced to the rod. The diameter of the piston rod is 0.1 [m]. The diameter of the cylinder is based on the maximum force and pressure in the system. The maximum force which the cylinder needs to produce is 4000 [kN]. It is decided that the maximum pressure of the pump is 300 bar. In order to let the fluid flow from the pump to the cylinder the pressure in the cylinder should be lower than the pressure of the pump. Therefore the diameter of the cylinder is 0.5 [m].

Table E- 1 Fixed properties of the actuators

	Cylinder dimensions		
S	Piston stroke	2	[m]
D	Diameter cylinder	0.5	[m]
$\mathbf{D_r}$	Diameter piston rod	0.1	[m]
	3-way directional valve		
$\mathbf{k_{d\_v}}$	Discharge coefficient valve	0.7	[-]
$\mathbf{D_{v\_max}}$	Maximum diameter valve	0.1	[m]

Hydraulic fluid

The hydraulic fluid used in the hydraulic actuator is Shell Tellus S2 V32, just as for the clamping system. For the properties of this fluid see Appendix D.4.

Pneumatic gas

A pneumatic system is ideally filled with a readily available gas which is nontoxic, chemically stable, non-flammable and causes no corrosion to the system. Two gasses that meet these qualities and are often used in pneumatic systems are compressed air and nitrogen [40].

Compressed air possesses almost all the desired properties and characteristics of a gas used in a pneumatic system. It is nontoxic and non-flammable, however it does contain oxygen which supports combustion [40]. Another big disadvantage is that air contains moisture. The temperature of the gas will change during the operation as the process is isentropic, this will cause condensation of the moisture in the system. Condensed moisture in the system can be very harmful as it increases corrosion. Therefore moisture separators need to be used to filter the moisture out of the air before it enters the system. Also the systems need to be cleaned after each operation, to remove the condensed moisture of the system. The advantage of using compressed air is that an unlimited amount of air is available.

Another gas which also possesses almost all the desired properties and characteristics is nitrogen. Nitrogen is an inert, non-flammable gas. Moreover it does not form explosive mixtures with air or oxygen and does not cause corrosion. However nitrogen is obtained by fractional distillation of air. Therefore the nitrogen needs to be stored in a closed system to be able to reuse it, or a lot of tanks containing nitrogen need to be brought on board of the HTV before the operation starts. These tanks will be delivered from companies which are specialised in the distillation of nitrogen. The nitrogen is saved in tanks in its liquid form. Therefore a liquid nitrogen pump pumps the low-pressure liquid from the storage tank and discharges it as high pressure liquid to a vaporizer which converts it to gas [40].

The advantage of using air is that it can be pumped in the pneumatic system during the operation or, if large quantities are needed, during the travel time of the vessel to the loading or discharge location. It is also an advantage that the system is an open system, the flow which leaves the cylinder can be discharged to the atmosphere, which will save space on the vessel. However the big disadvantage is the amount of moisture in air. As the operations take place in wet environments, at sea or harbours, the amount of moisture in the air is very high. The big advantage of using nitrogen over air is that it will not cause decays to the system and does not include moisture. It is therefore decided to use nitrogen as the gas in the pneumatic system. In order to be able to re-use the nitrogen which flows out of the cylinder it is proposed to create a closed loop pneumatic system. The high pressurized nitrogen will flow from the cylinder to a low pressure chamber. This low pressure chamber can be connected to a high pressure chamber by a pneumatic pump. In this way the nitrogen can be re-used in case the tanks with liquid nitrogen are almost empty during the operation. However the design of this part of the system is outside the scope of this thesis. The properties of nitrogen at 300 [K] are given in Table E- 2.

Table E- 2 Properties of nitrogen [41]

Nitrogen			
C_p	Specific heat at constant pressure	1039	[J/kg K]
C_v	Specific heat at constant volume	743	[J/kg K]
γ	Heat capacity ratio	1.4	[-]
ν	Viscosity	1.874×10^{-5}	[kg/ms]
R	Specific gas constant used gas	296.8	[J/kg K]

E.4 Parametric study

This section will present the simulation results for the different time constants and a varying line tension.

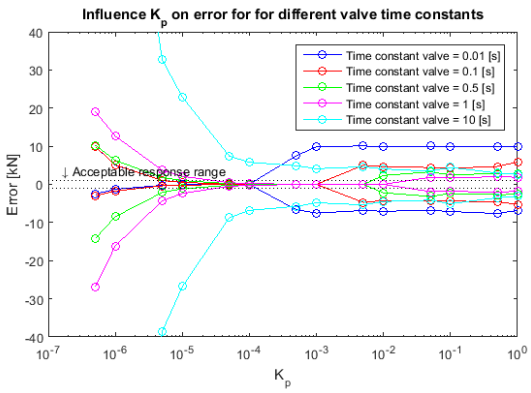

Figure E- 3 Hydraulic system, influence proportional gain, $F_{amplitude}$ = 500 [kN], F_{min} = -10 [kN]

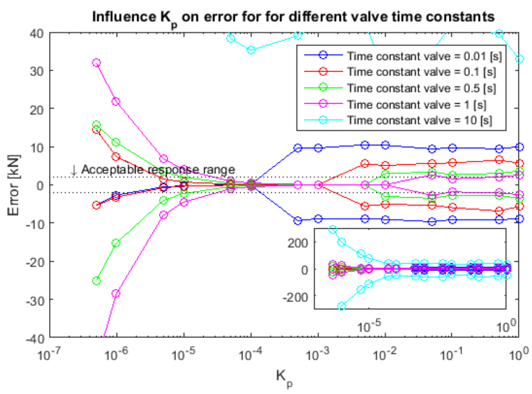

Figure E- 4 Hydraulic system, influence proportional gain, $F_{amplitude}$ = 1000 [kN], F_{min} = -10 [kN]

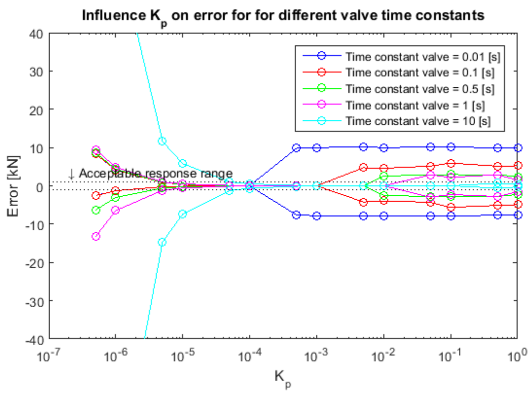

Figure E- 5 Hydraulic system, influence proportional gain, $F_{amplitude}$ = 500 [kN], F_{min} = -100 [kN]

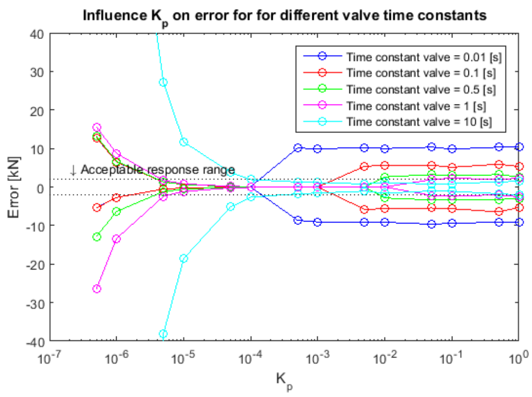

Figure E- 6 Hydraulic system, influence proportional gain, $F_{amplitude}$ = 1000 [kN], F_{min} = -100 [kN]

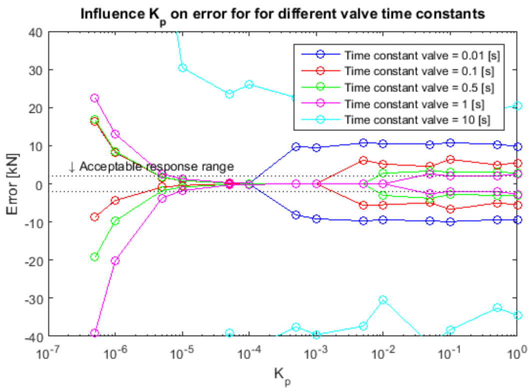

Figure E- 7 Hydraulic system, influence proportional gain, $F_{amplitude}$ = 1500 [kN], F_{min} = -100 [kN]

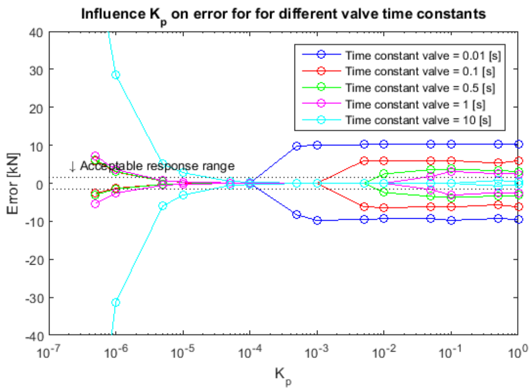

Figure E- 8 Hydraulic system, influence proportional gain, $F_{amplitude}$ = 500 [kN], F_{min} = -500 [kN]

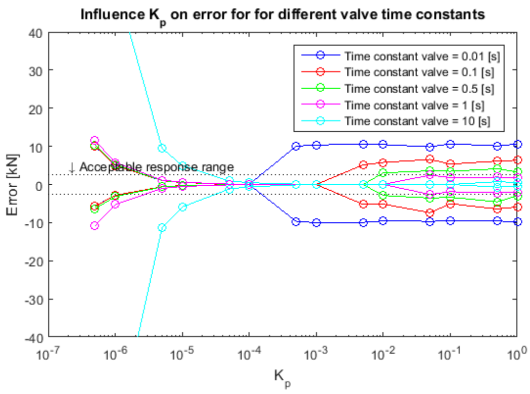

Figure E- 9 Hydraulic system, influence proportional gain, $F_{amplitude}$ = 1000 [kN], F_{min} = -500 [kN]

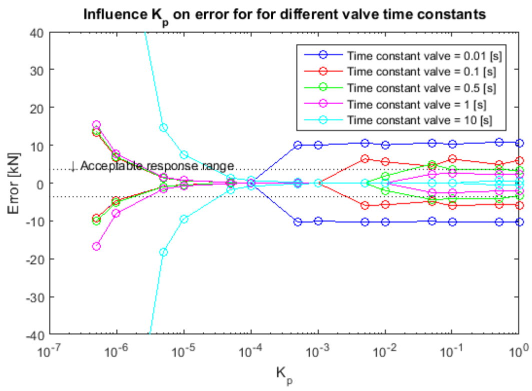

Figure E- 10 Hydraulic system, influence proportional gain, $F_{amplitude}$ = 1500 [kN], F_{min} = -500 [kN]

E.5 Pneumatic actuator 2

In this section the performance of pneumatic actuator 2 is investigated. The actuator needs to deliver a force of approximately 3500 [kN]. The actuator is equipped with 2 Dyneema lines of 35 [m].

It is found that a proportional gain value of 1×10^{-5} produces the smallest error between the required and obtained line tension. The peak flow is 0.21 [m^3/s] and the average flow is 0.007 [m^3/s], the peak mass flow is 71.9 [kg/s] and the average mass flow is 2.5 [kg/s], the peak power is 6.23 [mW] and the average power is 0.22 [mW].

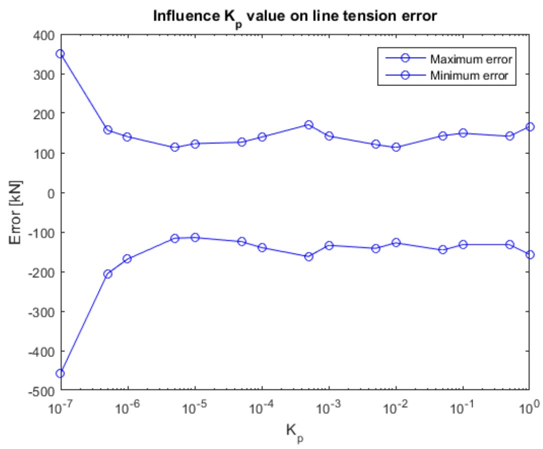

Figure E- 11 Influence proportional gain on line tension error, pneumatic actuator 2

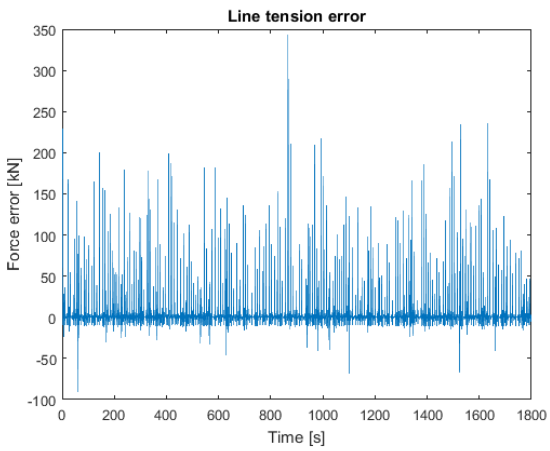

Figure E- 12 Line tension error, pneumatic actuator 2

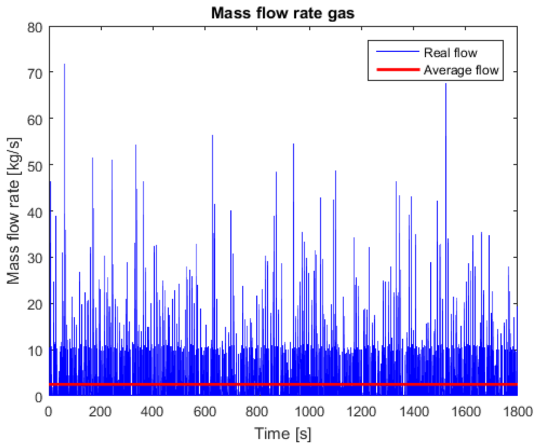

Figure E- 13 Mass flow rate gas, pneumatic actuator 2

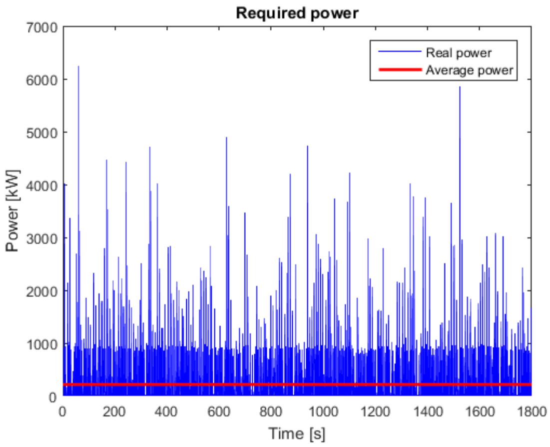

Figure E- 14 Required power, pneumatic actuator 2

Appendix F Software

In this section the global working of the used software packages will be discussed.

F.1 AQWA

The AQWA suite consists of multiple programs of which AQWA Line, AQWA Librium and AQWA Drift have been used in this research study.

AQWA-Line is a program which calculates the linearized hydrodynamic fluid wave loading, composed of radiation and excitation forces, on one or multiple floating bodies. The main analysis technique is the radiation/diffraction theory and the analysis is carried out in a three dimensional direction. The program is based on potential flow, which assumes the fluid to be ideal; incompressible, irrotational and inviscid. On top of this it is assumed that the incident regular wave has a mild slope. The program calculates the first and second order wave forces acting on the bodies in the frequency domain. The output values of AQWA-Line are used as an input for the time domain analyses performed in AQWA-Drift [8].

AWQA-Librium is a program which computes the stability properties (static and dynamic) of a system at its equilibrium position. However only time invariant forces, such as steady wind, current and wave drifting forces, are considered in this program. The equilibrium configuration determined by AQWA-Librium is used as an input for the time domain analyses performed in AQWA-Drift [8].

AQWA-Drift is a program which simulates the motions of a floating structure in irregular waves in the time domain. It uses the hydrodynamic output values from AQWA-Line as input and the static equilibrium position determined in AQWA-Librium, as starting point for the simulation. AQWA-Drift calculates all the forces, including the slow varying wave drift forces, working on the floating structure for each time step in the simulation. From the forces, the position, velocity and acceleration can be calculated [8].

In this research study AQWA-Drift is used to analyse the relative horizontal motions between the HTV and the cargo in time, as the result of the incoming irregular waves and the reaction forces of the Dyneema lines used in the standard cargo handling system.

F.2 MATLAB

MATLAB is short for *matrix laboratory* and is a numerical computing environment and programming language. MATLAB can be used to do matrix manipulations, to analyse and visualise data, to program and develop algorithms and to use and share those applications as a code. MATLAB is used among engineers and scientists around the world [42].

F.3 Simscape

Simscape is a software package within the MATLAB Simulink environment. It contains a set of block libraries and simulation features to model physical systems in the Simulink environment. In contrast to Simulink, Simscape employs the Physical Network approach which is particularly suited to model systems that consist of real physical components. Simsape elements interact with each other by exchanging energy through their ports. Simulink elements represent the basic mathematical operations [39].

The ports on the elements used in Simscape are nondirectional. They mimic physical connections between the elements. The way in which the elements are connected is analogous to connecting real components to each other. Thus Simscape diagrams mimic the physical system layout [39].

Simscape includes multiple physical domains. To model the new cargo handling systems, only SimHydraulics and the Mechanical Translational domains are used.

The Physical Network approach uses two different types of variables to describe the energy flow through an element; the Across Variable (AV) and the Through Variable (TV). The two variables differ from each other in the method used to measure the values of the variables (see Figure F- 1). A Through Variable is measured with a gauge connected in series to an element, while an Across Variable is measured with a gauge connected in parallel to an element. The properties of the variables for the different domains can be found inTable F- 1. In general, the product of each pair of corresponding variables is power (energy flow in Watts) [39].

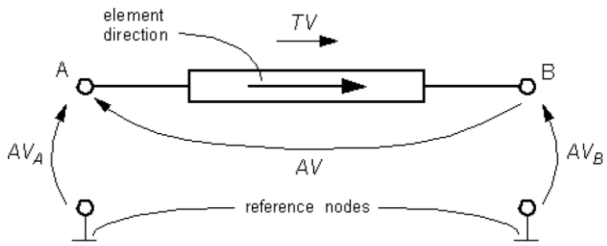

Figure F- 1 Across and Through variables in Simscape [39]

Table F- 1 Across and Through variables Simscape [39]

Physical domain	Across Variable	Through Variable
Hydraulic	Pressure (P)	Flow rate (Q)
Mechanical Translational	Velocity (v)	Force (F)