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# The dynamic mooring force on a wave energy converter moored in a single point

## Calculating the tensile force acting on the mooring structure from scale test measurements

Master of Science Thesis


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Department of Applied Physics
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Cover: A floating device used to keep the WEC hose at the surface.

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# Abstract

This thesis tries to answer the question on how to best moor a wave energy converter that has the shape of a hose. A water wave test tank is used on a model with a scale of approximately 1:2.3 of that of the full size structure and the tensile force acting on the mooring line is then measured with a load cell. By using Froude scaling these measured forces are then used to predict the real force on an energy producing hose out in the ocean.

SWAN simulated forecasted wave data provided by the Swedish Meteorological and Hydrological Institute (SMHI) is used to analyse the wave climate at five different locations along the west coast of Sweden and different potential test sites are listed in table 4.2. This thesis also suggests a mooring design that allowes the moored hose to move with the waves so that dynamic load force on the mooring line can be minimized.

The static load force acting on the mooring of a 48 m hose that has a diameter of 0.5 m should range somewhere around 10-15 kN depending on wave weather and current conditions. A dynamic load force on a hose with the same dimensions should be between 12-37 kN in waves that are 1.6-5 m high. Under extreme storm conditions the dynamic response to the waves could result in forces estimated at up to 5-14 times that of the static drift force value, thus giving a dynamic force acting on the hose mentioned above of up to 50-210 kN. The force on the hose in waves that are around 5 m high should be at the lower end of this force span.

Wave tank tests with 0.7 m waves and a period time of 2.6 s on a 36 m hose with a diameter of 220 mm resulted in a mooring tensile force of maximum 1000 N. Using Froude scaling, the estimated dynamic force on a 83 m long hose of 0.5 m diameter should be around 12 kN in 1.6 m high waves with a period time of 3.9 s and 37 kN in 5 m high waves with a period time of 3.9 s.

# Sammanfattning

Det här examensarbetet försöker att besvara frågan hur en slangformad vågenergi-omvandlare bäst bör förankras. En vågbassäng användes för att genomföra förankrings-tester på en modell i skala 1:2,3. Kraften i förankringslinorna mättes upp med hjälp av en lastcell och Froudeskalning användes sedan för att förutspå den verkliga kraften på förankring-en som en energiproducerande slang ger upphov till.

SWAN-simulerade, prognosticerade vågdata tillhandahållna av Svenska Meterologiska och Hydrologiska Institutet (SMHI) användes för att analysera vågklimatet på fem olika platser längs med den svenska västkusten och resultaten presenteras i tabell 4.2. Den här avhandlingen föreslår också en förankringsdesign som tillåter den förankrade slangen att förflytta sig med vågorna så att den dynamiska vågkraften på förankringen kan minimeras.

Den statiska vågkraften på förankringen hos en 48 m lång slang med en diameter på 0,5 m borde ligga någonstans runt 10-15 kN beroende på våg- och strömförhållanden. Den dynamiska kraften på en slang med samma dimensioner borde vara mellan 12-37 kN stor i vågor som är 1,6-5 m höga. Under extrema stormförhållanden kan den dynamiska responsen på vågorna leda till krafter på förankringen som är upp till 5-14 gånger så stora som den statiska avdriftningskraften. Det skulle i sådana fall ge en dynamisk kraft på nämnda slang på uppåt 50-210 kN. Kraften på slangen i 5 m höga vågor borde ligga i den nedre delen av detta kraftspann.

Vågtester på en 36 m lång slang med en diameter på 220 mm med 0,7 m höga vågor och en periodtid på 2,6 s resulterade i en maximalt uppmätt förankringskraft på 1000 N. Genom att använda Froude-skalning fås att den uppskattade dynamiska kraften på en 83 m lång slang med en diameter på 0,5 m bör vara runt 12 kN i 1.6 m höga vågor med en periodtid av 3,9 s och 37 kN i 5 m höga vågor med en periodtid av 3,9 s.

# Contents

# Preface

I have always been fascinated by the ocean and I began my studies in Engineering Physics at Chalmers University in Gothenburg with alternative energy generation in mind. At the time there were no programs in energy systems in Gothenburg but I figured that a solid math and physics background would provide an excellent basis for future work in the alternative energy field. When the time came to write my Master of Science Thesis and the opportunity presented itself to do so in part at a small water wave energy company I jumped at it.

During the course of this thesis I have had two supervisors: Senior lecturer Magnus Karlsteen at Chalmers University of Technology and Mr Daniel Ehrnberg at Vigor Wave Energy. Both were always available for questions and they always had constructive critique or help to give both when sought for and of their own volition. I owe a debt of gratitude to them both for showing a great interest in my work and keeping me away from dead ends.

I also want to thank Professor Emeritus and scientific advisor at Ocean Energy Centre Lars Bergdahl for providing me with great litterature in the field of Wave Energy and Barry Broman, Senior Oceanographer at the Swedish Meteorological and Hydrological Institute, for letting me use his work on wave data modeling and assisting me in the interpretation thereof.

I also owe a debt of gratitude to Mr Kristian Persson and Mr Roman Madorski at Vigor Wave Energy for assisting me in my experiments and aiding me whenever needed.

# Notations

| | |
|---|---|
| $A$ | *Cross sectional area* |
| $a$ | *Acceleration* |
| $a_i$ | *Amplitude of incident wave* |
| $a_r$ | *Amplitude of reflected wave* |
| $a_t$ | *Amplitude of transmitted wave* |
| $C_D$ | *Coefficient of resistance (depends on Reynolds number)* |
| $C_v$ | *Coefficient of added mass* |
| $F$ | *Fetch, or force* |
| $F'$ | *Force per meter* |
| $F_d$ | *Drifting force* |
| $F_{dyn}$ | *Dynamic force* |
| $F_g$ | *Gravitational force* |
| $F_i$ | *Inertial force* |
| $F_{stat}$ | *Static force* |
| $F_v$ | *Viscous force* |
| $Fr$ | *Froude's number* |
| $f$ | *Scale factor for force* |
| $g$ | *Constant of gravity* |
| $H_{m0}$ | *Significant wave height of the top of the spectrum in fully arisen sea* |
| $H_{max}$ | *Maximum wave height* |
| $h$ | *Wave height* |
| $L$ | *Wave length* |
| $L_0$ | *Deepwater wave length* |
| $l$ | *Characteristic length* |
| $P$ | *Power production* |
| $P'$ | *Power production per meter* |
| $R$ | *Radius of a circular wave motion* |
| $Re$ | *Reynold's number* |
| $s$ | *Scale factor* |
| $T$ | *Wave period* |
| $T_m$ | *Period time of the top of the spectrum in fully arisen sea* |
| $t$ | *Time or scale factor for time* |
| $U$ | *Wind speed or fluid velocity* |
| $U_A$ | $0{,}71*U^{1{,}23}$ |
| $V$ | *Volume* |
| $v$ | *Fluid velocity* |
| $\mu$ | *Dynamic viscosity* |
| $\rho$ | *Density* |
| $\omega$ | *Angular velocity* |

# 1. Introduction

The energy in ocean waves is a huge untapped, renewable and clean source of energy and its potential usage is tremendous. One way of extracting this energy is by using a long hose and the patented Vigor principle. By alternating water and air through an inlet, water batches are trapped inside the hose by air batches. As the water and air batches are pushed forward inside the hose by the sourrounding waves the pressure keeps getting higher inside the hose until it is high enough to power a turbine.
This thesis will examine the magnitude of the mooring force that such a power plant will be subjected to and also suggest a mooring design.

To estimate the forces acting on the moored energy producing hose, theory surrounding floating structures, including a simple form of Morison’s formula, is used together with a whole range of assumptions regarding for example the acceleration of the structure, the flexibility of the structure, what types of waves that are involved, surface smoothness, Reynold and Froude numbers. I have tried to give a short theoretical overview on the subject of water waves and ways to calculate forces acting on a floating structure out in the ocean. I will make good use of Froude scaling. The formulas presented may seem very simple and easy to use at times but they all have severe limitations to their accuracy. I considered trying to simulate forces acting on a flexible hose and I looked into the possibility of solving such a problem with for example COMSOL, MATLAB or SIMULINK. However, a simulation of the fluid dynamics involved when a flexible object moves in an ocean climate proved extremely difficult. Even if you somehow managed to get a hold of the computer power needed to run even the shortest and comparably simple simulations, the results still would likely be off by a large margin.
My work therefore concentrates on actually measuring the forces that arise when the afore mentioned hose is hit by waves of different sizes. This was possible since I had access to a wave testing tank.

Much work was put into designing, ordering or producing the right type of equipment needed to conduct the tests. I was fortunate enough that some of the equipment needed coinsided with the equipment needed to conduct other tests at the wave tank test facility. Mr Kristian Persson in the Vigor Wave Energy team was instrumental in this with his CAD drawing skills and sharp eye for details.

After measuring the forces they are scaled up using Froude scaling and a design suggestion is presented for a hose moored at a single point. The forces attained through these tests and scalings provide the basic input for how sturdy the mooring design of the hose needs to be the day it is placed in a harsh wave climate.

Forces will be stated in Newtons and in tons. The use of tons is commonly used in load cell data sheets and lacks the gravitational constant but will be used for an easier understanding of the forces involved.

# 2. Theory

The energy in a water wave is built up mainly by the speed and the fetch of the wind. The fetch here being the distance which the wind travels over the surface. The magnitude of a wave out in the ocean is thus decided by the size and strength of the weather system and its direction. Maximum fetch is attained when the weather system is moving in the same direction and with a speed equal to half that of the speed of the waves [L. Bergdahl].

In shallow waters, i.e. at depths beneath half the wave length, the topography and the depth both play important roles in determining the size of a wave [L. Claesson et al p.39].
Wind speeds of 1 m/s are needed to create the first capillary waves that are seen as ripples. At slightly stronger winds the capillary waves are then transformed into the gravity waves we call swells. Due to the fact that they are gravitationally driven the energy of such a wave can travel very long distances with negligible energy losses. For a swell to be maintained and not break into white water the velocity of the wind must be below half that of the wave's phase velocity [L. Cleasson et al p.52, p.39].

## 2.1 Fully Arisen Sea, FAS

Given a constant wind speed and enough fetch and durability a state of equilibrium will occur between the losses of energy (for example due to dissipation, wave breaking and turbulence) and the energy added by the wind. This state of equilibrium is called "Fully Arisen Sea" [L.Claesson et al p.78].

According to the "Shore Protection Manual" from 1984 the following formulas for the significant wave height $H_{m0}$ and the period time $T_m$ of the top of the wave spectrum apply in the range from deep waters up to a fully arisen sea:

$$H_{m0} = \frac{U_A^2}{g} * 0{,}0016 \left(\frac{gF}{U_A^2}\right)^{1/2} \qquad (1)$$

$$T_m = \frac{U_A}{g} * 0{,}287 \left(\frac{gF}{U_A^2}\right)^{1/3}, \qquad (2)$$

where $U_A = 0{,}71 * U^{1{,}23}$, U = wind speed [m/s] at an altitude of 10 m, F = a fetch maximized so that the current durability surpasses

$$t = \frac{U_A}{g} * 68{,}8 \left(\frac{gF}{U_A^2}\right)^{2/3} \qquad (3)$$

At the boundary where the sea is fully arisen the "Shore Protection Manual" gives us the following values:

$$H_{m0} = 0{,}2433 \frac{U_A^2}{g}, \quad T_m = 8{,}184 \frac{U_A}{g}, \quad t = 7{,}15\ 10^{-4} * \frac{U_A}{g}$$

In the book "Energi från havets vågor" by Lennart Claesson et al some of the forces acting on bottom fixed structures and floating structures are listed. I'll make a short summary of these forces here and I will later use them in the "Method" chapter to make some sort of estimate of what the forces will be on the studied hose.

## 2.2 Wave forces acting on a bottom fixed structure

The force acting upon a bottom fixed structure with a cross-sectional area A is:

$$F = \frac{1}{2}\rho C_D v^2 A, \qquad (4)$$

where

$\rho$ = the viscosity of the fluid.

$C_D$= a coefficient of resistance that depends on Reynolds number and has a typical value of between 0.1 and 1,5 depending on the shape of the structure.

$v$ = the velocity of the fluid.

If the ratio between the wave length L and the cross-sectional width B is greater than 5 you can use *Morison's formula* to calculate the force F:

$$F = aC_V V\rho + C_D \rho v|v|\frac{1}{2}A, \qquad (5)$$

where

$C_V$ = the coefficient of added mass.

$V$ = the volume of the body.

$a$ = the acceleration of the fluid.

It should be added that the velocity and acceleration of the fluid in the formulae given above are given as the relative movement between the fluid and structure and thus the equations hold even if the structure would be moving. We will make use of this later on in the thesis.

## 2.3 Wave forces acting on a floating structure

A floating structure is affected by both static and dynamic forces. Belonging to the static forces are currents and drifting forces that give rise to a drift that results in a problem of static equilibrium. The dynamic forces depend on the structure's response to each individual wave and the low frequency oscillation of the wave drifting force.

At deep waters the wave drifting force per meter can be written [L. Claesson et al]:

$$F'_d = \frac{\rho g}{4}(a_i^2 + a_r^2 - a_t^2)\ , \qquad (6)$$

where

$a_i$ = the undisturbed amplitude of the incident wave.

$a_r$ = the amplitude of the reflected wave.

$a_t$ = the amplitude of the transmitted wave.

If the reflected wave can be neglected and the wave energy converter absorbs P' kW/m, then the wave drifting force per meter can be written as [Falnes, p.82]:

$$F'_d = \frac{\omega}{g}P' \qquad (7)$$

where

$\omega$ = angular velocity.

*P'* = power production per meter.

We see that the more energy we absorb the bigger the wave drifting power becomes.

## 2.4 Movement pattern of a particle at the surface

Aided by linear Airy wave theory John Fitzgerald and Lars Bergdahl has calculated the movement pattern of a particle at the surface and a particle at the bottom depending on the relation between depth and the amplitude of the waves. The result is shown in the figure below. In shallow waters a particle moves horizontally up to two times the wave's amplitude at the surface and up to one amplitude horizontally at the bottom. In ”Position Mooring of Wave Energy Converters” John Fitzgerald points out that the wave energy mooring problem can be summarized as: *”the need to comply with ever larger horizontal wave frequency excursions despite ever smaller vertical spans in the mooring system available to accomodate such compliance.”* [Fitzgerald, p.11]. Depending on the present sediment type at the bottom the below figure also illustrates the risk of water movements undermining anchors or weights.

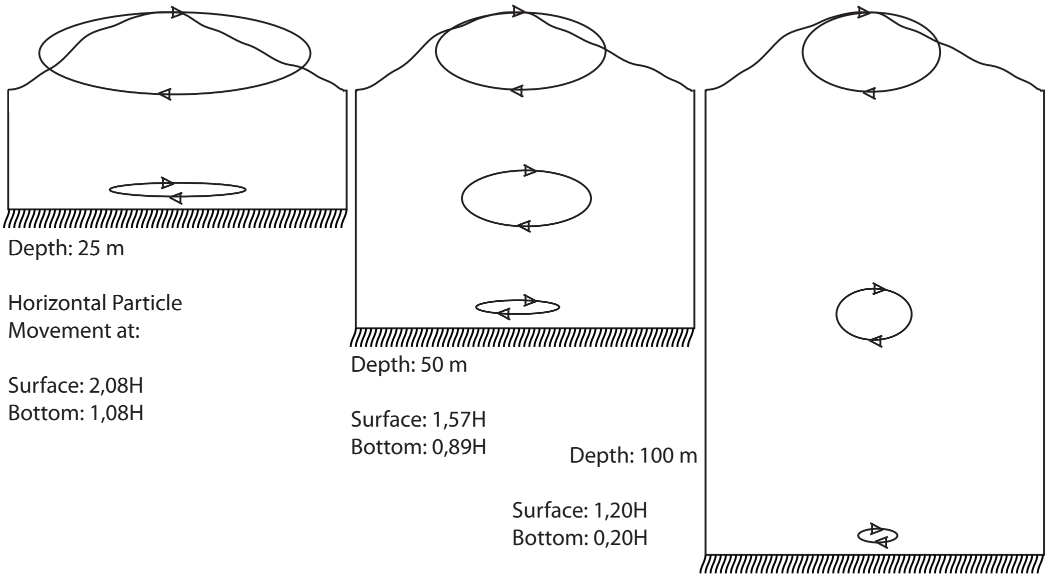


*Figure 2.1: Amplification of the particle movement in the horizontal direction as a function of the wave amplitude H [J. Fitzgerald, p.10].*

## 2.5 Downscaling an experiment in a tank test

This section is manly based on the article ”Guidance for the experimental tank testing of wave energy converters” by Grégory Payne at the University of Edinburgh. In it the reader can find a more thorough presentation on the subject of scaling. Mr Payne lists three kinds of forces that are of comparable importance in the mechanical interactions between fluids and solids: Inertial forces $F_i$, gravitational forces $F_g$ and viscous forces $F_v$. If we let $U$ be the fluid velocity, $g$ the gravitational acceleration, $l$ the length characterising the fluid/solid interaction phenomenon and $\mu$ be the dynamic viscosity we can write:

$$F_i \propto \rho U^2 l^2$$
$$F_g \propto \rho g l^3$$
$$F_v \propto \mu U l$$

From these forces two non-dimensional quantities, namely the Froude number $Fr$ and the Reynolds number $Re$, can be derived:

$$Fr = \frac{U}{\sqrt{gl}} \propto \frac{F_i}{F_g} \propto \frac{inertial\ force}{gravitaional\ force} \qquad (8)$$

$$Re = \frac{\rho U l}{\mu} \propto \frac{F_i}{F_v} \propto \frac{inertial\ force}{viscous\ force} \qquad (9)$$

When conducting scaled model testing it is desirable to maintain the Fr and Re values as constant as possible for the test and the full scale phenomenon. As Mr Payne points out this is not practically possible to achieve for the purpose of tank testing wave energy devices since it would have to involve very large centrifuges and/or fluids with different viscosities. Mr Payne then states the following: ”For many tank-scale WECs [Wave Energy Converters], the net influence of viscous forces on body motion is small and Froude scaling can be assumed to be satisfied”.
This assumtion gives conservative predictions of the full-scale behaviour [Payne, p.6] but it leads the way to what is known as ”Froude scaling”.
If we let s be the geometrical scale between the model and full-scale device and perform dimensional analysis on interesting quantities we end up with Mr Paynes Froude scaling table [”table 2.1”, Payne, p.7]:

| **Quantity** | **Scaling** |
|---|---|
| wave height and length | s |
| wave period | $s^{0.5}$ |
| wave frequency | $s^{-0.5}$ |
| power density | $s^{2.5}$ |
| linear displacement | s |
| angular displacement | 1 |
| linear velocity | $s^{0.5}$ |
| angular velocity | $s^{-0.5}$ |
| linear acceleration | 1 |
| angular acceleration | $s^{-1}$ |
| mass | $s^3$ |
| force | $s^3$ |
| torque | $s^4$ |
| power | $s^{3.5}$ |
| linear stiffness | $s^2$ |
| angular stiffness | $s^4$ |
| linear damping | $s^{2.5}$ |
| angular damping | $s^{4.5}$ |

*Table 2.1: Froude scaling table.*

For example, if we work with a model at scale 1:3, then s=1/3. A wave height of 0.7 m in the tank test then corresponds to a wave height of 2.1 m at full-scale. If A force acting on or caused by the model is measured to be 100 N that corresponds to 2.7 kN at full-scale.

## 2.6 Calculating wave length

Given the water depth and the wave period the following approximation of the wave length can be made [Fenton and McKee, p. 499- 513]:

$$L = L_0 \left( tanh \left( \frac{\omega^2 h}{g} \right)^{\frac{3}{4}} \right)^{\frac{2}{3}}, \qquad (10)$$

where

$L_0 = g\frac{T^2}{2\pi}$ is the deepwater wave length

$\omega = \dfrac{2\pi}{T}$

$h$ = wave height

$T$ = wave period

$g$ = the constant of gravity

## 2.7 Choosing a mooring system

The mooring systems design depends on the following:

- The vertical and horizontal load at the anchor point.
- What is the static load the mooring system should hold for?
- What is the dynamic load the mooring system should hold for?
- If elasticity is desirable or not.
- The type of seabed. Sand, clay or stone?
- Depth.
- How much space the mooring structure is allowed to occupy.
- If there is enough room to, for example drag an anchor deep enough into the sediment.
- How long should the lifespan of the systems or parts of the systems be? 1, 3, 5 or 20 years? (For example inspection every third year and a designed lifespan of 20 years.)
- How easy and fast it should be to inspect and perhaps change parts of the system.
- Survival strategy of the moored structure.
- Cost

## 2.8 Maximum obtainable wave height

There is a theoretical maximum on how high a wave can get out on an open sea before it breaks [J. Fitzgerald, p. 9-10].

The maximum height depends on the water depth, the period of the wave and the inclination of the seabed. The statistical maximum height of a wave with a one hundred years return time is estimated bu Fitzgerald and Bergdahl to 1.83 times the highest wave height in a 1000 waves wave interval.

At different depth the wave with a return time of a hundred years is:

| Water depth | Wave Period | 10 s, Breaking | 15 s, Breaking | 20 s, Breaking | 20 s Sheltered |
|---|---|---|---|---|---|
| 25 m | Wave Height *Horizontal displacement of a particle at the surface.* | 14.3 m *17.2 m* | 17.8 m *28.7 m* | 19.1 m *39.6 m* | 10.0 m *20.7 m* |
| 50 m | | 19.1 m *19.7 m* | 27.8 m *34.5 m* | 27.8 m *42.9 m* | 20.0 m *30.9 m* |
| 100 m | | 20.3 m *20.3 m* | 27.8 m *21.4 m* | 27.8 m *32.7 m* | 20.0 m *23.6 m* |

*Table 2.2: Maximum obtainable wave height at different depth for different wave periods.*

Worth noting is that the highest wave ever measured was 27.8 m high and has been used as a maximum in the above table.

## 2.9 Extreme weather

Not far from Lysekil, close to Väderöarna (The Weather Islands) waves as heigh as 13 m have been confirmed. At a depth of 25 m the highest theoretical height is around 14-19 m depending on the period of the waves and the shape of the seabed.

The highest 10 minutes mean wind velocity that has been measured along a Swedish coast is 40 m/s outside of Gotland. The strongest gust of wind that has been measured in Sweden was 81 m/s at Tarfala near the mountain of Kebnekaise [SMHI].

# 3. Method

The task ahead was to make an estimate of the force brought to bear on a moored hose through a combination of theoretical calculations and measurements on downscaled models in a wave tank. The estimated force was then used to decide on a mooring structure to be tested in an ocean trial within the year of 2012.
Several potential testsites for this trial were investigated but a remarkably few detailed wave data measurments have been carried out on the Swedish west coast (chosen for its close proximity to Gothenburg) throughout history. So I had to look for simulated data instead.
The Swedish Meteorological and Hydrological Institute (SMHI) provided me with SWAN simulated next-day wave data forecasts from 2007-2011 in both grib and MATLAB file format. Senior oceanographer Barry Broman at SMHI's Rossby Centre helped me understand the content of these files and I made good use of his previous work on singling out a representative and manageable amount of data points.
A MATLAB script was then written to fetch interesting testsite data for a particular coordinate. The results can be seen for a few potential testsites in the Results section.

The ocean tests are to be carried out on a 48 m long hose with a diameter of 0.5 m and the following calculations have been carried out on a hose of that size. As a comparison the forces on a hose of 83 m and 200 m with the same diameter has also been calculated.

## 3.1 Force on the hose if it was bottom fixed and stiff

Formula (4) gives us the force on a bottomfixed structure: $F = \frac{1}{2}\rho C_D v^2 A$. We assume that $C_D \approx 1$ [Claesson, p.136] and that we have a current of maximum 3 m/s, consisting of an underlaying current of 1.5 m/s and a wind driven current of about 1.5 m/s i.e a wind speed of 30 m/s [UFC 4-159-03: "Wind driven currents generally attain a mean velocity of approximately 3 to 5 percent of the mean wind speed at 10 m"]. We then get different cross sectional areas depending on the waves' angle of attack on the hose. If the angle is measured towards the normal of the hose we get the following tables:

| Angle of attack | 48 m hose | | 83 m hose | | 200 m hose | |
|---|---|---|---|---|---|---|
| | kN | ton | kN | ton | kN | ton |
| 0° | 113 | 11.5 | 196 | 20 | 473 | 48 |
| 60° | 57 | 5.8 | 98 | 10 | 236 | 24 |
| 80° | 20 | 2.0 | 38 | 3.9 | 82 | 8.4 |
| 90° | 0.9 | 0.1 | 0.9 | 0.1 | 0.9 | 0.1 |

*Table 3.1: Force acting on a bottom fixed, stiff structure. A current of 3 m/s.*

| Angle of attack | 48 m hose | | 83 m hose | | 200 m hose | |
|---|---|---|---|---|---|---|
| | kN | ton | kN | ton | kN | ton |
| 0° | 51 | 5.2 | 87 | 8.9 | 210 | 21 |
| 60° | 25 | 2.6 | 43 | 4.4 | 104 | 11 |
| 80° | 8.9 | 0.9 | 15 | 1.5 | 36 | 3.7 |
| 90° | 0.4 | 0.04 | 0.4 | 0.04 | 0.4 | 0.04 |

*Table 3.2: Force acting on a bottom fixed, stiff structure. A current of 2 m/s.*

## 3.2 Force on the hose depending on produced power

If the reflected wave can be neglected formula (7), gives $F_d = \frac{\omega}{g} P$ saying that we get a larger drifting force the larger the power production is. Since $\omega = \frac{2\pi}{T}$ we get the following table for different values of T and the power production independently of the length of the hose (it is accounted for in $P$ itself):

| T = | P = 100 kW $F_d =$ | P = 250 kW $F_d =$ | P = 500 kW $F_d =$ | P =1 MW $F_d =$ |
|---|---|---|---|---|
| 3 s | 21 kN / 2.1 ton | 53 kN / 5 ton | 107 kN /11 ton | 213 kN / 22 ton |
| 4 s | 16 kN / 1.6 ton | 40 kN / 4 ton | 80 kN / 8 ton | 160 kN / 16 ton |
| 5 s | 13 kN / 1.3 ton | 32 kN / 3 ton | 64 kN / 6.5 ton | 128 kN / 13 ton |
| 6 s | 11 kN / 1.1 ton | 27 kN / 2.7 ton | 53 kN / 5.4 ton | 107 kN / 11 ton |
| 7 s | 9 kN / 930 kg | 23 kN / 2.3 ton | 46 kN / 4.7 ton | 91 kN / 9.3 ton |
| 8 s | 8 kN / 814 kg | 20 kN / 2.0 ton | 40 kN / 4.1 ton | 80 kN / 8.1 ton |
| 9 s | 7 kN / 724 kg | 18 kN / 1.8 ton | 36 kN / 3.6 ton | 71 kN / 7.2 ton |
| 10 s | 6 kN / 651 kg | 16 kN / 1.6 ton | 32 kN / 3.3 ton | 64 kN / 6.5 ton |

*Table 3.3: Drifting force on a power producing unit.*

## 3.3 Force according to Morison's formula

Morison's formula (5) consists of one dynamic part and one static part:

$$F = \underbrace{aC_V V\rho}_{F_{dynamic}} + \underbrace{C_D \rho v|v|\frac{1}{2}A}_{F_{static}}$$

It is applicable if the body be stiff, the acceleration uniform over the body, the diameter of the hose is much less than the wave length and the flow comes from only one direction. We can then calculate the dynamic force by making the following assumption regarding the acceleration:

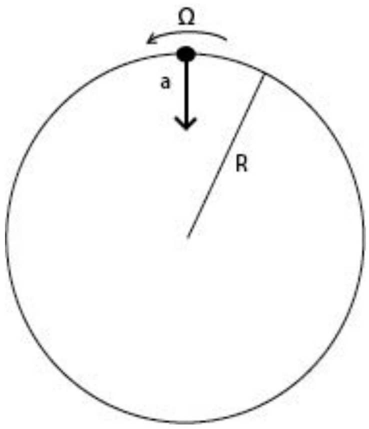


*Figure 3.1: Direction of the acceleration when the hose moves in a circular motion.*

Let the hose move in a circular motion with the same diameter as the wave height and use the following relations:

$$R = \frac{H_{max}}{2}, \quad a = \omega^2 R, \quad \omega = \frac{v}{R}, \quad v = \frac{\pi H_{max}}{T}$$

| *Length of hose* | 36 m | | 48 m | | 83 m | | 188 m | |
|---|---|---|---|---|---|---|---|---|
| $H_{max}$= *2.1, T = 4.2, v = 1.57, a ≈ 1* | 7.4 kN | 0.75 ton | 9.9 kN | 1.0 ton | 17.2 kN | 1.8 ton | 38.8 kN | 3.9 ton |
| $H_{max}$= *5, T = 6.0, v = 2.62, a ≈ 2.7* | 20 kN | 2.1 ton | 26.7 kN | 2.7 ton | 46.2 kN | 4.7 ton | 105 kN | 10.7 ton |

*Table 3.4: Dynamic force acording to Morison's formula for different wave heights and hose lengths. Hose diameter = 0.5 m.*

## 3.4 Dynamic Behaviour

John Fitzgerald and Lars Bergdahl conduct an analysis over the magnitude of the forces acting on a catenary chain when subjected to dynamic motion [article III or p.15-16, J.Fitzgerald]. With a static load force of 200 kN, a scope of 3-8, a depth of 50 m, an attachment point that moves in a circular motion with a radius of 6 m and waves with a periodtime of about 7 seconds we get that the dynamic mooring force ranges from 1000 to 7000 kN. With other words the dynamic force is 5-14 times the static load force. The upper end of this span resulting from extreme waves as high as 27 m. The largest wave height used in this thesis is 5 m so one might assume that the dynamic force factor might lie around 5-6 fr the purpose of this thesis.

A major concern when dealing with dynamic behaviour and mooring lines is to never let the mooring lines slack. If the mooring line slacks it runs the risk of snatching when the full load is introduced again. Anyone who has towed a car using an unelastic cable knows about this effect.

## 3.5 Wave data

There seems to be no good readily available source of historical wave data information at this point in time in Sweden. The following highest observed Swedish wave heights were taken from the Swedish Meteorological and Hydrological Institute's (SMHI) homepage [www.smhi.se].

**Highest observed wave heights**

| Ocean area | Highest wave height | Station | Date |
|---|---|---|---|
| Södra Bottenhavet | 9.8 m | Finngrundet | 2006-10-31 |
| Norra Östersjön | 12.8 m | Almagrundet | 1984-01-14 |
| Sydöstra Östersjön | 11.2 m | Södra Östersjön | 2009-10-14 |
| Kattegatt | 5.9 m | Läsö Ost | 2007-11-09 |
| Skagerrak | 13.0 m | Väderöarna | 2007-01-14 |

*Table 3.5: Highest observed wave heights.*

In the Year 2008 SMHI made a 15 year hindcast calculation of wave heights with the help of the SWAN modelling tool, developed by the University of Delft in the Netherlands. The resolution over Kattegatt and Skagerrak was 6x6 NM and meshed in squares. In 2010 Sina Saremi wrote his Master's thesis on the subject ”Development of a Wave Database in Coastal Areas around Sweden Using the SWAN Wave Model”. In this work he split each square into two triangles and managed to attain a resultion of 3x3 NM over the swedish south eastern waters starting at Stockholm. However I have found no such calculated data with comparable resolution for the western Swedish waters. What I did find out was that SMHI has been saving their one day SWAN model forecasts since 2007. Senior Oceanographer Barry Broman at SMHI provided me with this SWAN data material in MATLAB format and was very helpful in helping me understand the different variables used. A MATLAB program was then written to extract different wave heights, wave periods and wave length for a given coordinate throughout the years. I made good use of Mr Broman's previous work on singling out a managable amount of wave data points. When a coordinate is put into the program the wave data for the closest SWAN data point in the material is selected. The results are presented in section 4.3 in table 4.2. A plot of the SWAN data points used to make table 4.2 can be seen in figure 3.2.

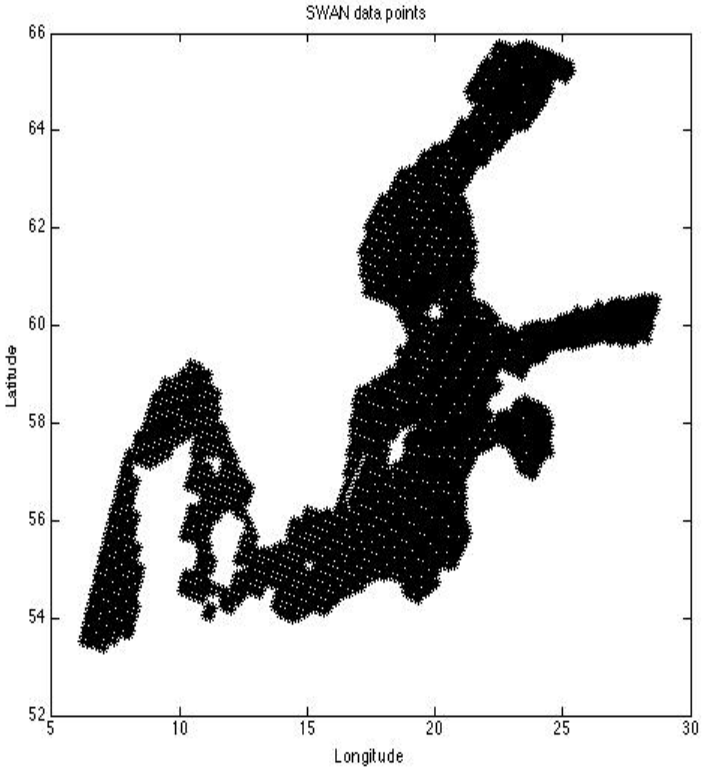


*Figure 3.2: The SWAN data points used in making table 4.2.*

## 3.6 Criterias for choosing a test site

- A spot where the ocean leading up to the test spot is open thus allowing for the waves to build up over large distances.
- A depth of less than half the predominant wave length, but not so shallow that the waves begin to break. This could mean a depth less than 15 meters.
- An inclining seabed resulting in sharper waves.
- Proximity to the power grid.
- Proximity to road access for easier monitoring possibilities
- Proximity to a harbour.

Spots where favorable wave conditions are found can be labelled ”hot spots” and at such hot spots waves can be much higher than the average wave height in the area.

## 3.7 Potential test sites

Through skimming the internet for wave data and good testing locations on Sweden’s west coast the following potential test sistes were listed mainly because there were better available data surounding them or because they had previously been used by others.

1. At Vinga lighthouse outside of Gothenburg, previously the test site for different wave energy converters. The depth is around 20-30 m and the extreme waves are limited in their size. Each wave give around 3-4 kW per meter. But it is considered to be ideal for WEC (Wave Energy Converter) testing [http://www.gp.se/nyheter/goteborg/1.111509-ny-chans-for-vagkraften, 2009-03-29].

2. Outside of Hönö. The location is very similar to that of Vinga with the exception that the bottom conditions might be better suitable for anchoring. The political climate on the island of Hönö might also be favorable and another wave energy project Elskling II has made tests in the vicinity [http://www.gp.se/nyheter/goteborg/1.111509-ny-chans-for-vagkraften, 2009-03-29].

3. North-West of Gullholmen Gullholmen Lysekil, 58.192825, 11.373146. This is the location of the wave energy project ”Seabased”. The following may be read on the web page of The University og Uppsala after a translation into english by the author: ”The project area is situated close to land for availablity and cost efficiency reasons. The area’s average depth, around 25 m, has also played a role when choosing the location as well as the actual bottom substrate, which is a flat sand bottom. The depth makes it possible to carry out relatively uncomplicated divings.” [http://www.el.angstrom.uu.se/forskningsprojekt/WavePower/Lysekilsprojektet.html, 2012-04-10].
   Each wave give around 9-12 kW per meter [http://www.gp.se/nyheter/goteborg/1.111509-ny-chans-for-vagkraften, 2009-03-29].

4. Near the jetty in Halmstad close to Vågbrytaregatan, 56.647757, 12.838039. With a depth of around 7-30 m the area lies protected from Atlantic waves and the extreme waves should therefore be smaller. The area has previously been used as a testsite for smaller ocean-based wind plant prototypes. Close proximity to the power grid makes the site a potentially interesting alternative.

5. Väderöarna, The Weather Islands. This is one of the only spots in Sweden that has available historical wave data. South of Väderöarna we have had the largest waves, over 13 meters high, ever to be measured in Swedish waters by a SMHI

weather buoy. The area is a conservation area, with the exception of the northeastern island of Storö. The island of Storö is supplied with electricity by an under-water power cable from the mainland. The islands constitute a bird protection area and you are not allowed to step onto the islands between the 1st of April and 1st of August. The bottom surrounding Väderöarna is very well mapped due to over 40 years of continuos surveillance of the bio-diversity on the rock bottom. The islands are reached by both the Baltic and the Jutska currents. The harsh wave climate and strong currents leaves only small amounts of sediments on the bottom of of the Islands' West coast [http://hem.passagen.se/vadero/index.htm].

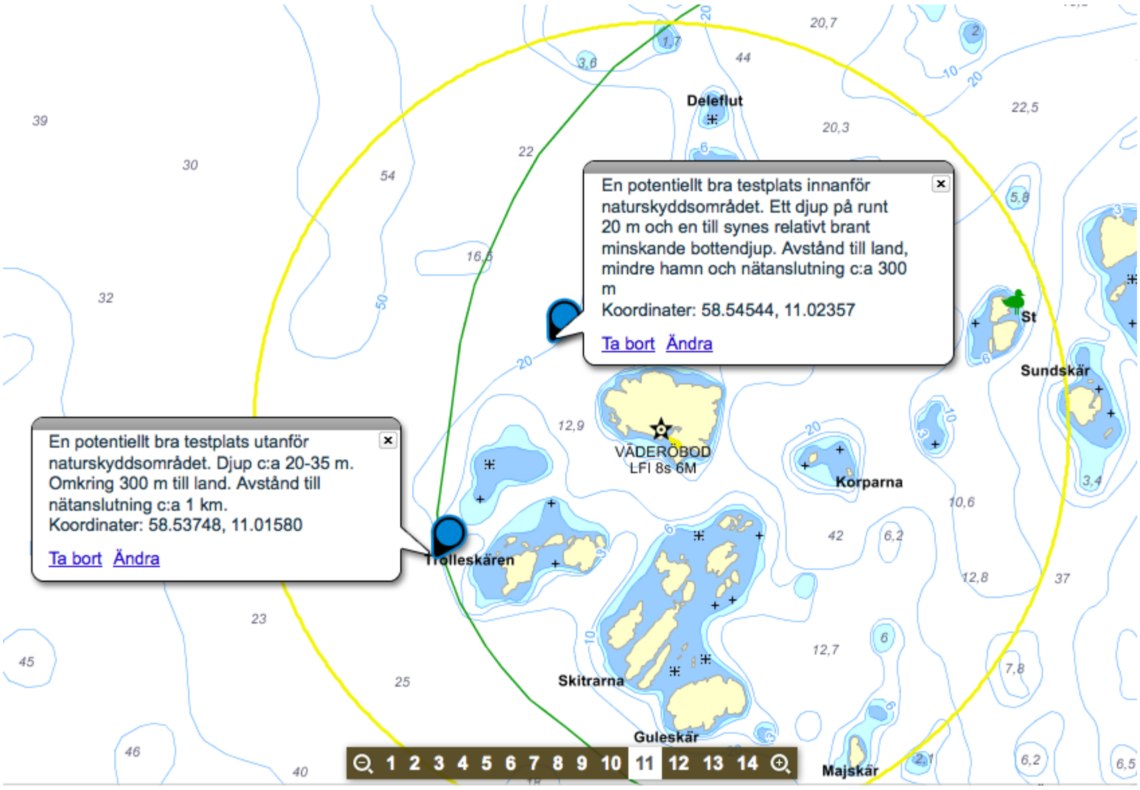


*Figure 3.3: A chart showing part of the Weather Islands.*

## 3.8 The upper mooring point

A hose is to be positioned at the surface so that its inlet is under water at all times. There will be some sort of nozzle that assures that the right amount of air and water is upplied to the hose. For the interest of this thesis it will be assumed that it will be possible to moor the hose in the nozzle and that the nozzle is designed to withstand the forces it will be subjected to. However, how this is done will be of major importance for the life spann of the mooring solution since large forces will be at play around the fastening points.

One could imagine a different fastening point other than at the front end of the nozzle but if this is to be the case one must be careful to design the mooring system so that the mooring line can't collide with the moored structure itself. If for instance an underwater buoy is used the angle of the mooring line connecting the buoy and the hose will be little, especially given that the front end of the hose itself must be kept under water at all times.

The mooring system should be designed so allow the hose to move with the waves as opposed to trying to withstand the waves. If the mooring system can bring the hose to move in a pattern close to that of the particle shown in figure 2.1 then mooring forces should be as small as they get.

Based on a maximum wave height of 5 m the force acting on the mooring structure will be the smallest if the moored structure can float freely with the wave in an elliptical motion. At a depth of 25 m in the ocean the following movement pattern should be desirable:

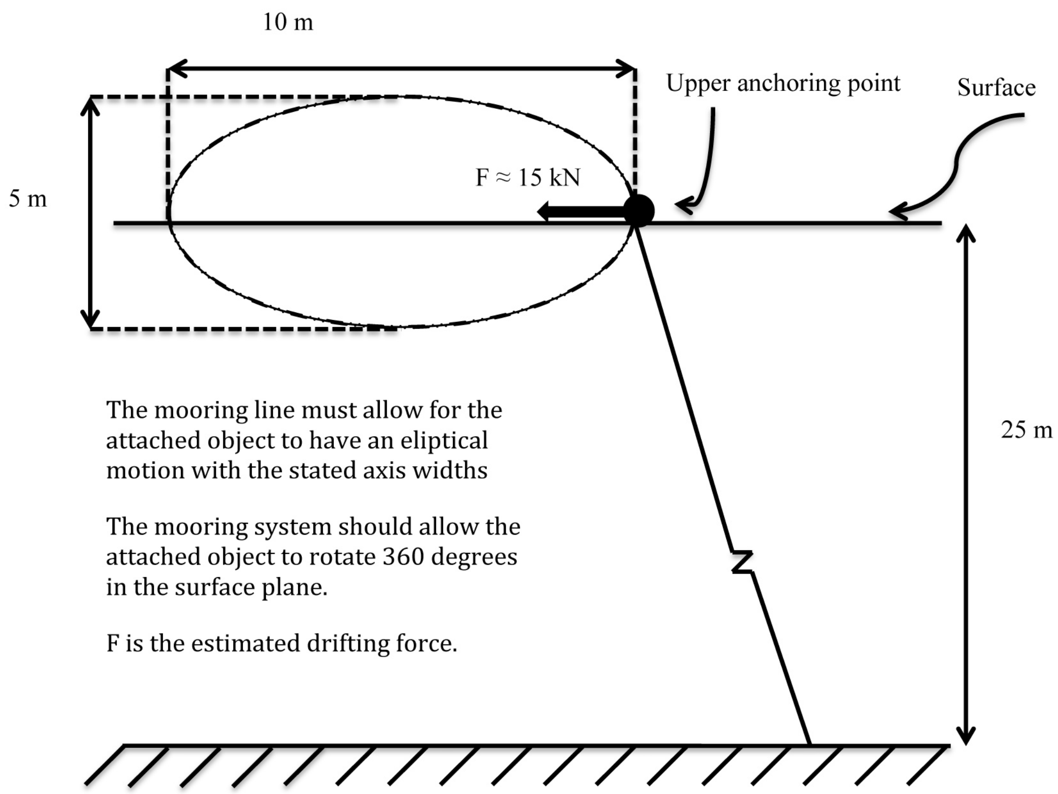


*Figure 3.4: Movement pattern of the upper mooring point.*

## 3.9 Pre-experiment

In order to get a sense of what order of magnitude the tensile forces in the experiments were going to be a pre-experiment was conducted. A smaller hose of 2.4 m with 100 mm in diameter was used. It was cut in sections of 30 cm and every second section was filled with an air filled balloon. The whole hose was then taped together again and the sections that were not balloon filled were filled with water to simulate Vigor's air and water batches to some extent. One end of a rope was then fastened to the hose and the rope was led through two pulleys (one fastened to a structure close to the bottom of the wave tank and one fastened to the ceiling) before the other end of the rope was tied to an empty, graded bucket. When the wave generator was running attempts were made to fill the bucket up just enough so that the wave force acting on the hose could barely lift it up. This required a person holding the bucket on its way down since the waveforce is cyclic in its nature.

Even though the experiment was unsuccessful in determining a specific magnitude of the force, due to the difficulty in balancing the bucket, it still gave a sense of the order of magnitude since a fully filled bucket (5*l*) would cause the hose to drift in a direction against the waves. It was also possible to just hold the rope without the bucket and try to guess how big the force was.

A first estimate put the force on the small hose within the range of 10-100 Newtons.

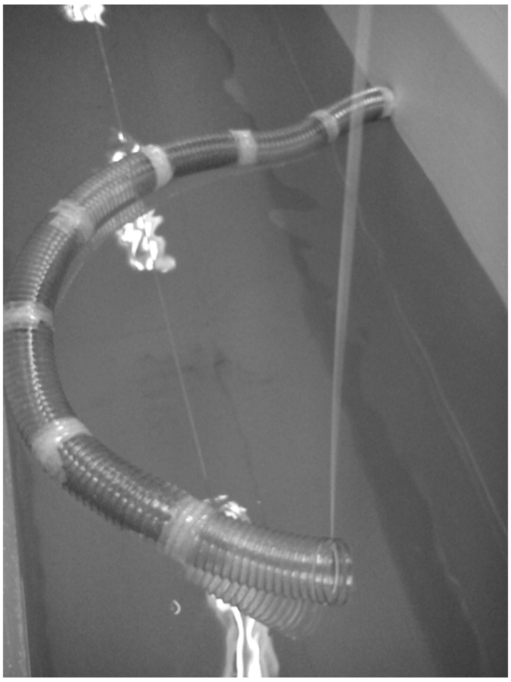

*Figure 3.5: The hose constructed for the pre-experiment.*

*Figure 3.6: Structure build to hold down the mooring line.*

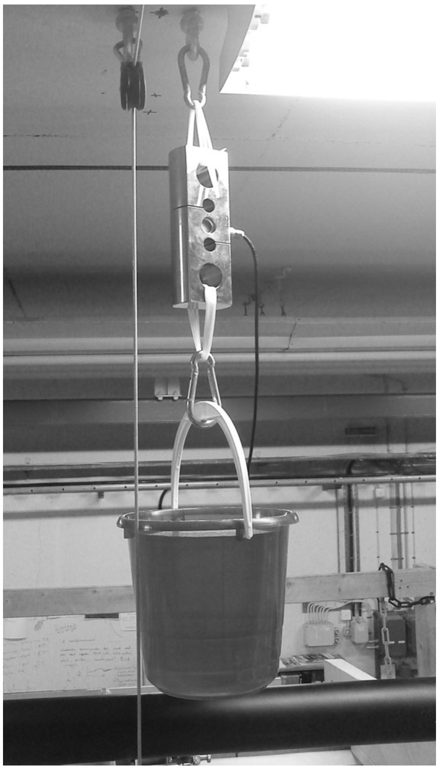

Figure 3.8: A TA4/2 analog transmitter was used to transform the voltage signal from the load cell into a 4-20 mA signal.

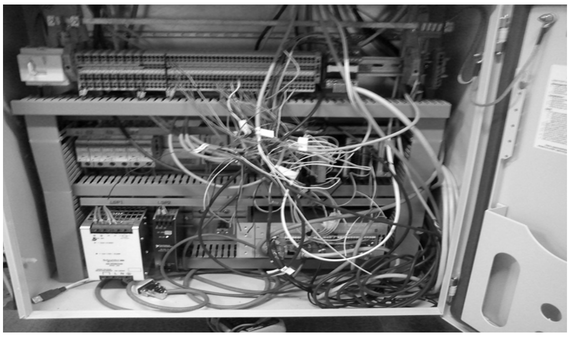

Figure 3.7: A graded bucket filled with water was used to check if the callibration of the load cell was roughly ok.

Figure 3.9: The 4-20 mA signal was connected to a computer card and loaded into a LabView program written with the purpose of logging the measurements.

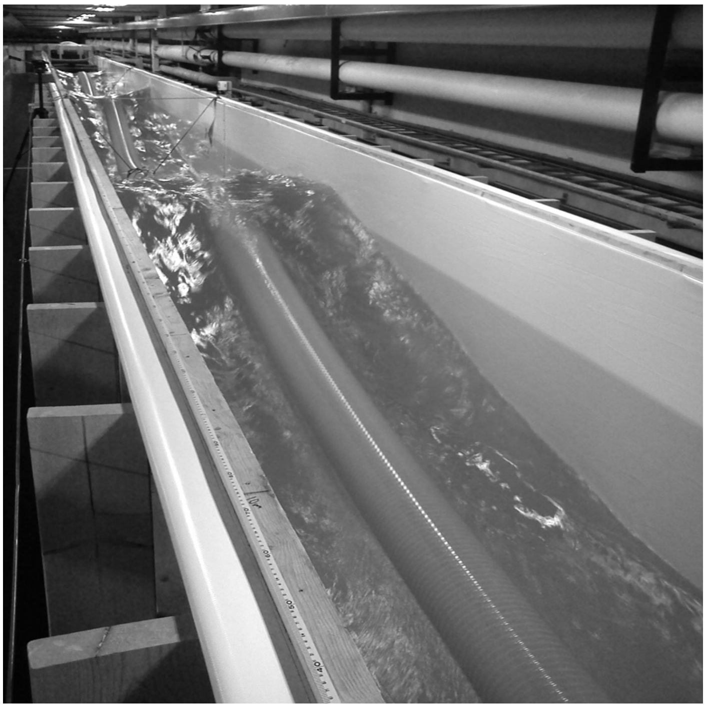

Figure 3.10: The wave tank.

## 3.10 Experimental setup

All tests were conducted in a 51 m long and 1.2 m wide wave tank. Waves were created using a wave creator driven by a hydraulic pump controlled through a LabView program. All test were made on a a 36 m long hose with a diameter of 220 mm. At one end the hose was fastened through a wheel held close to the test tank's bottom and at the other end to a load cell hanging from the ceiling. All tests were made once with 0.3 m waves and once with 0.7 m waves. The period time was 2.6 s in all cases and the water level in the tank was around 1.6 m. The waves were of a sharper tilted sinozoidal form to imitate waves close to breaking.
At the rear end of the wave tank a wave fender killed the remaining waves so that no significant wave reflections were visible. After testing different floating devices, the floating device seen in figure 3.18 was used to keep the hose at the surface about 2.2 m from the hose's front end. Tests conducted with elasticity in the mooring line used the elastic rubbers shown in figure 3.15.

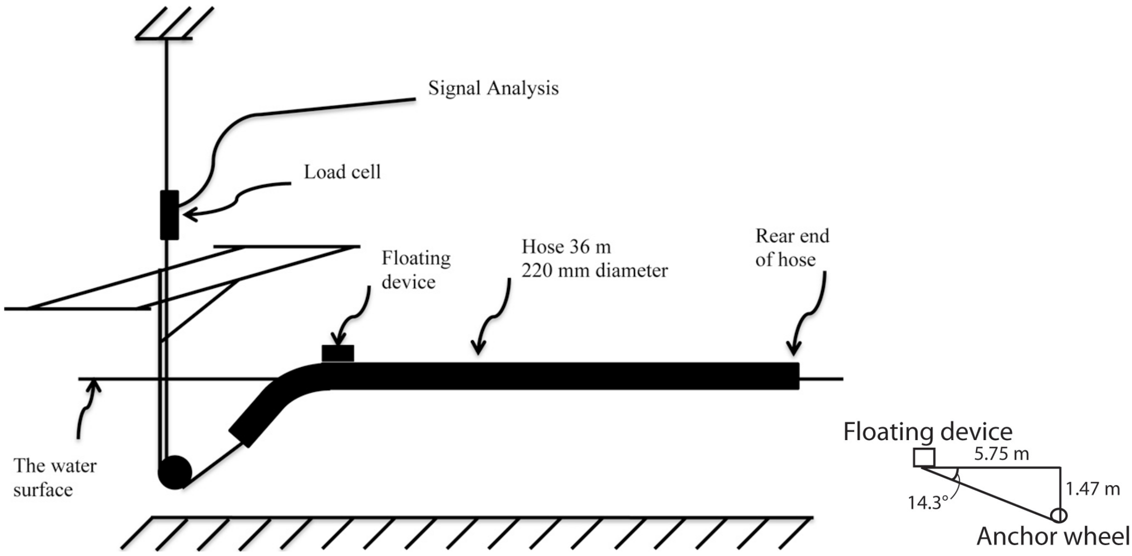


*Figure 3.11: General experimental setup*

The following tests were conducted:

A. The hose is filled with air and fastened to a reservoir at the end. The front end being held up by a floating device.

B. The hose is filled with water and fastened to a reservoir at the end. The front end being held up by a floating device.

C. The hose is filled with water and air batches and fastened to a reservoir at the end. The reservoir together with a special inlet is used to produce the right water and air batches needed to create an increasing pressure within the hose.

D. Experiments A through C were run again but this time with the rear end loose.

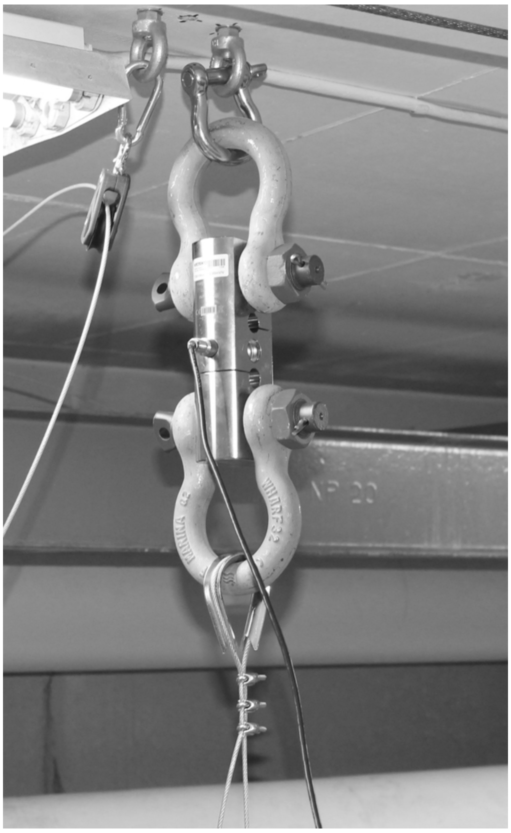

*Figure 3.12: The load cell.*

*Figure 3.13: The bottom mooring wheel.*

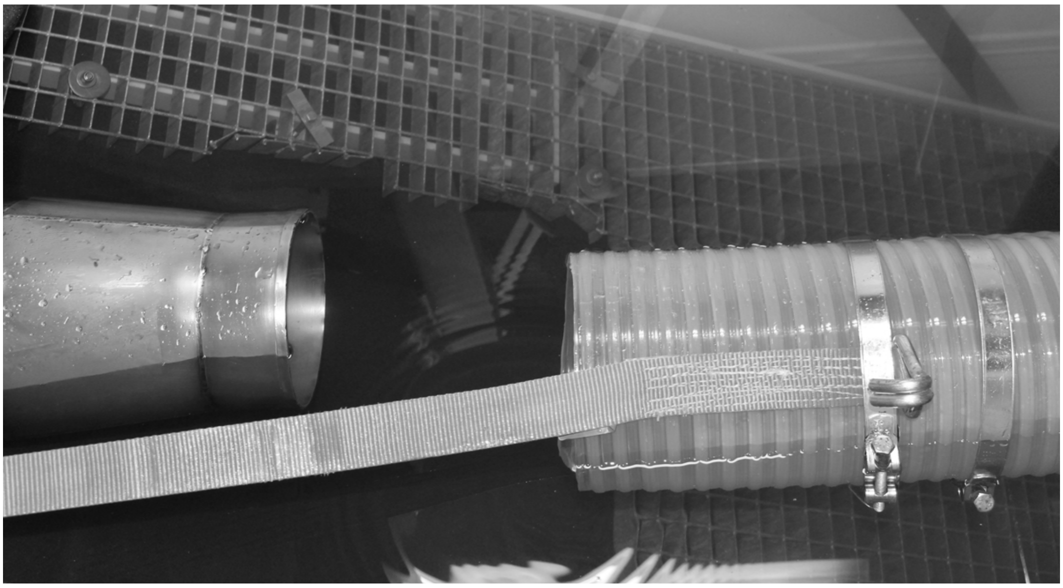

*Figure 3.14: A loose rear end.*

To record the data a VETEK T20-10 load cell capable of measuring up to 10 tons with an error of ±0.023 % was used together with a TA4 voltage to ampere amplifier capable of measuring at 1 kHz. To avoid lagging in the system due to other parallel steering systems used on the same computer the LabView program's measuring speed was set at 10 Hz.

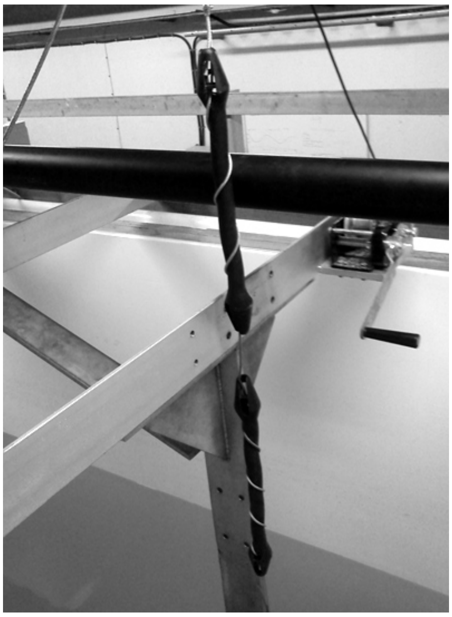

*Figure 3.15: An elastic mooring.*

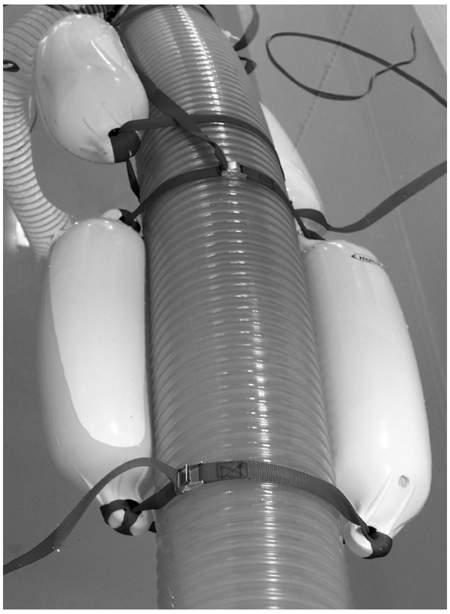

*Figure 3.16: A floating device.*

*Figure 3.17: A floating device.*

*Figure 3.18: A floating device.*

## 3.11 Measuring the current force on the hose

One of the goals of this thesis is to prepare for and submit advice on an experiment on a full scale hose in a sea trial. As the hose is dragged out to sea by boat it will be possible to measure the force it is subjected to by connecting it to a load cell. Different current forces will be simulated by driving the boat at different speeds.

Different experimental setups where the current force was to be measured in the wave tank on a down-scaled hose have been taken under consideration. However, since the force will be measurable on a full sized hose while it is being dragged by boat to its anchoring site these experiments were discarded.

The ideas all involved straightening the wave tank and the usage of a powerful enough pump to pump the water around from one side of the tank to the other.

# 4. Results

## 4.1 The tank test results

The experimental setup described under section 3.10 was used to obtain the following results. All figures give the force on the mooring line. The parameters that were changed were: wave height, elasticity in the mooring line, the content of the hose and if it was fastened at the rear end or not.

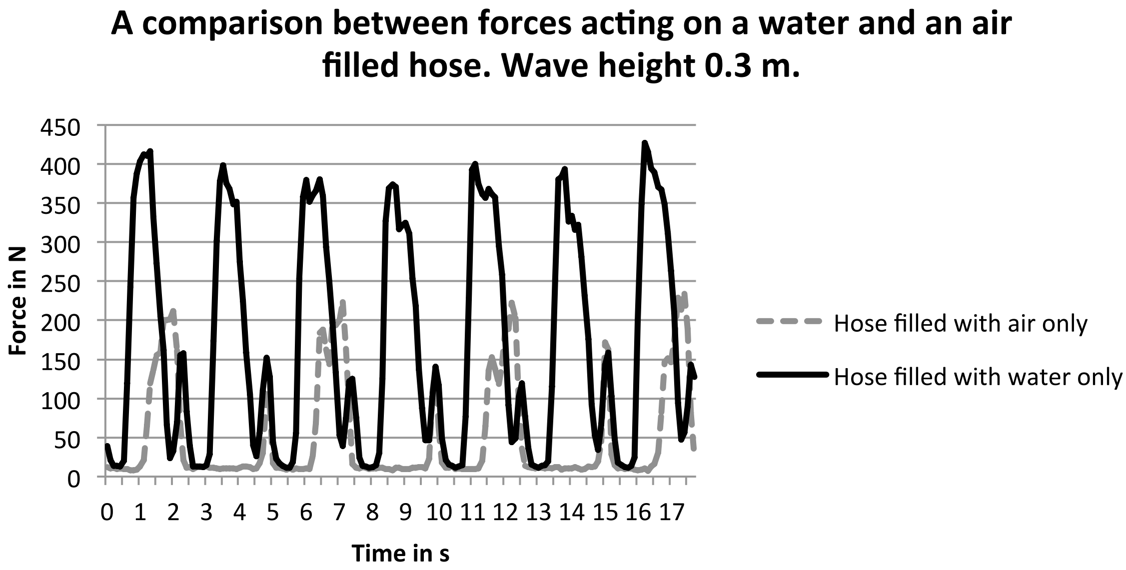


*Figure: 4.1: Air only and water only filled hose. Mooring line without elasticity. Hose fastened to reservoir. Wave height 0.3 m.*

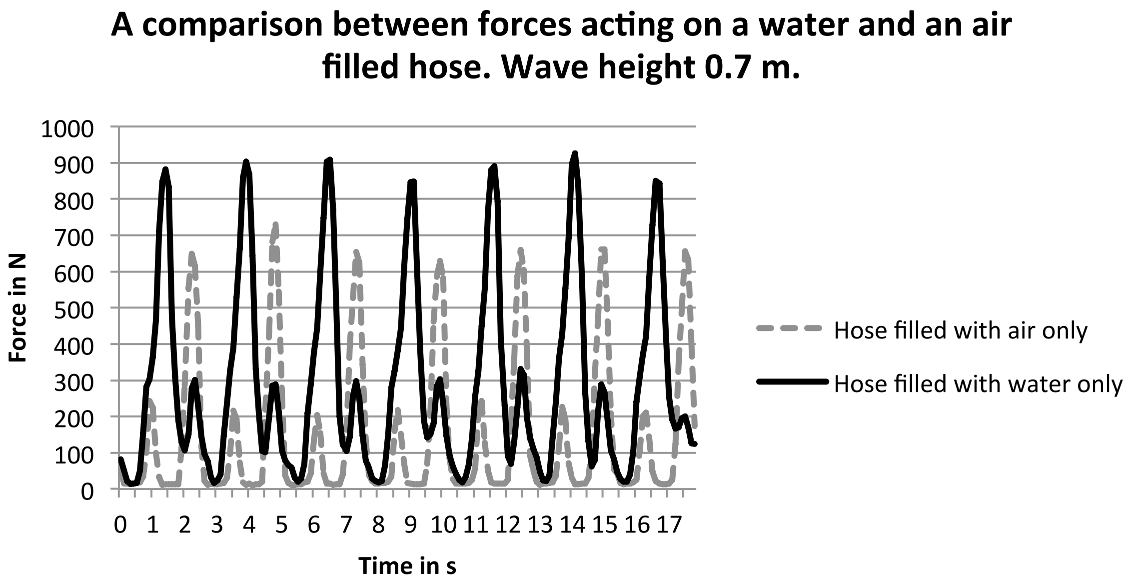


*Figure 4.2: Air only and water only filled hose. Mooring line without elasticity. Hose fastened to reservoir. Wave height 0.7 m.*

We see in figure 4.1 and 4.2 that a water filled hose gives rise to a larger mooring line force than a water filled hose

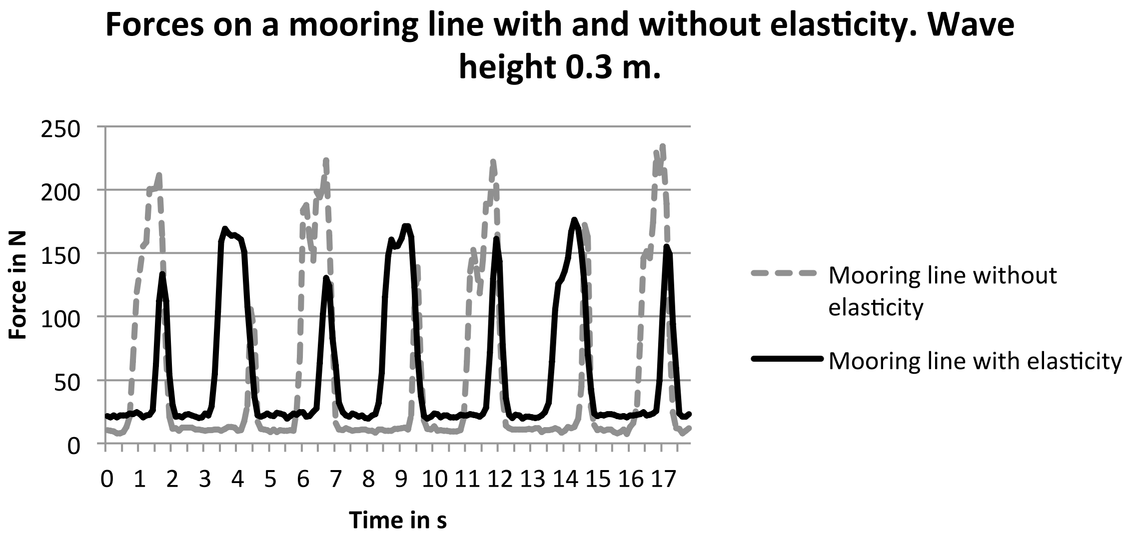


*Figure 4.3: Forces on a mooring line with and without elasticity. Hose filled with air only. Hose fastened to reservoir. Wave height 0.3 m.*

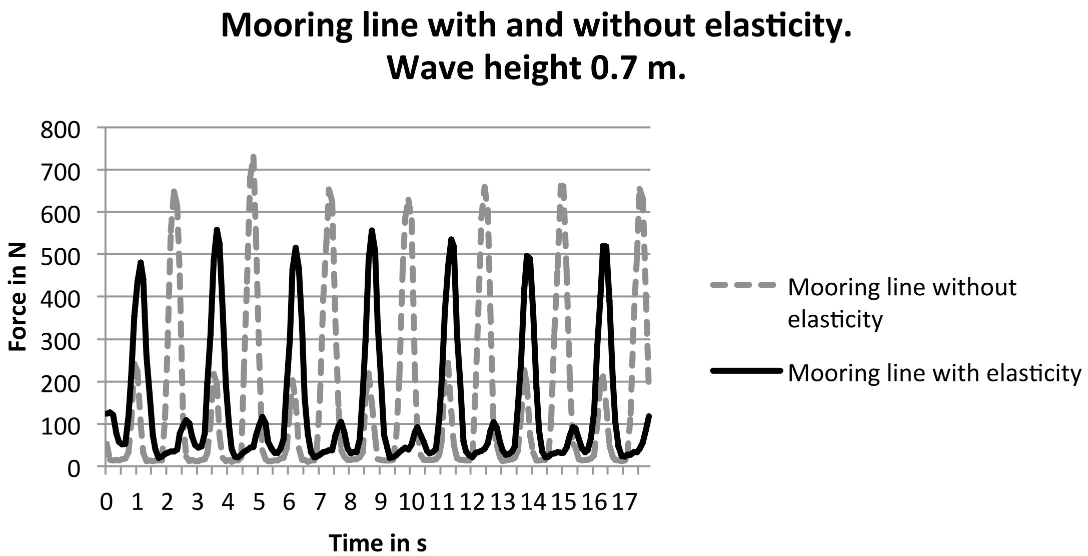


*Figure 4.4: Mooring line with and without elasticity. Hose filled with air only. Hose fastened to reservoir. Wave height 0.7 m.*

Figure 4.3 and 4.4 implies that if the amplitude of the force curve for a mooring line with elasticity is smaller and somewhat wider than the force curve for a mooring line without elasticity.

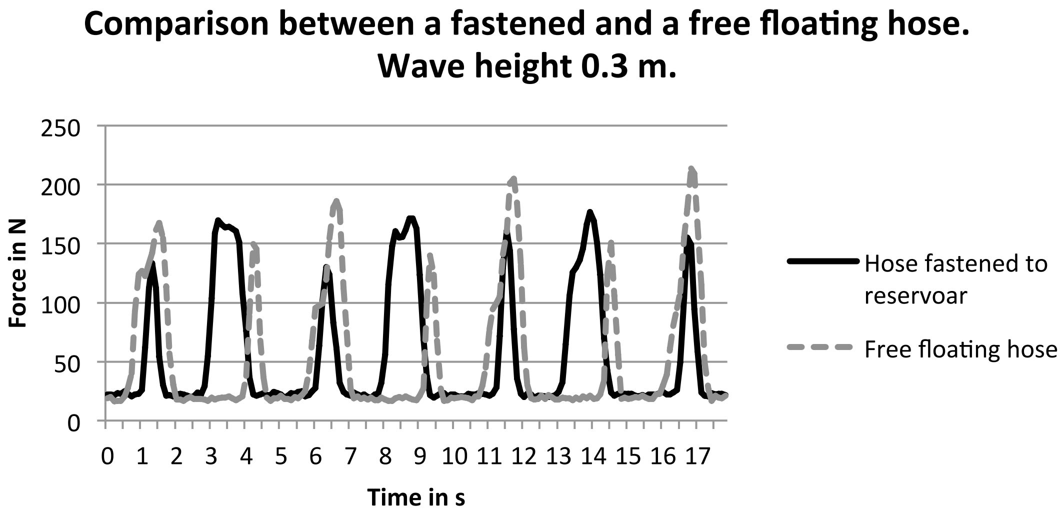


*Figure 4.5: Fastened and free floating hose. Hose filled with air only. Mooring line with elasticity. Wave height 0.3 m.*

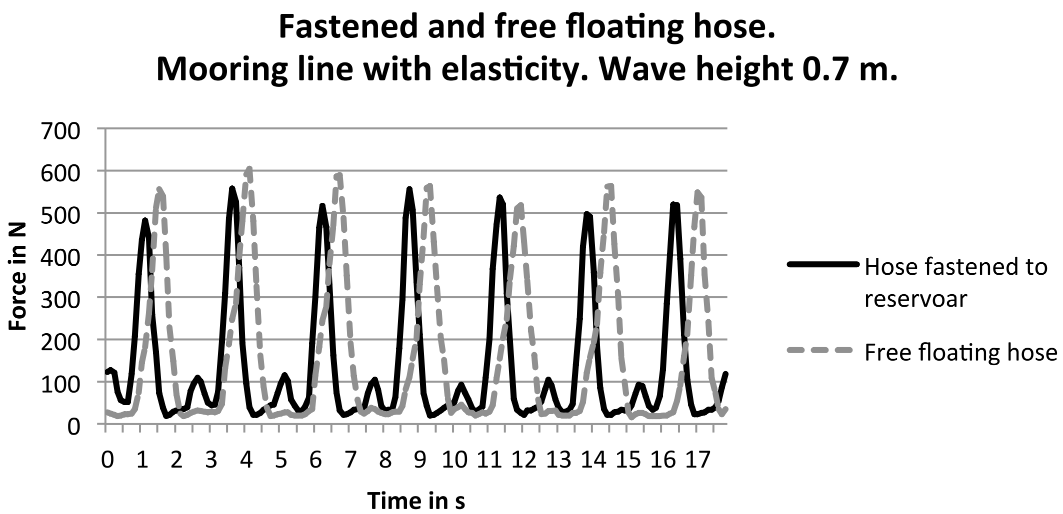


*Figure 4.6: Fastened and free floating hose. Hose filled with air only. Mooring line with elasticity. Wave height 0.7 m.*

Figure 4.5 and 4.6 shows a ”choppier” behaviour for a hose fixed at the end than for a free floating hose and perhaps a more oscillating amplitude for the free floating hose.

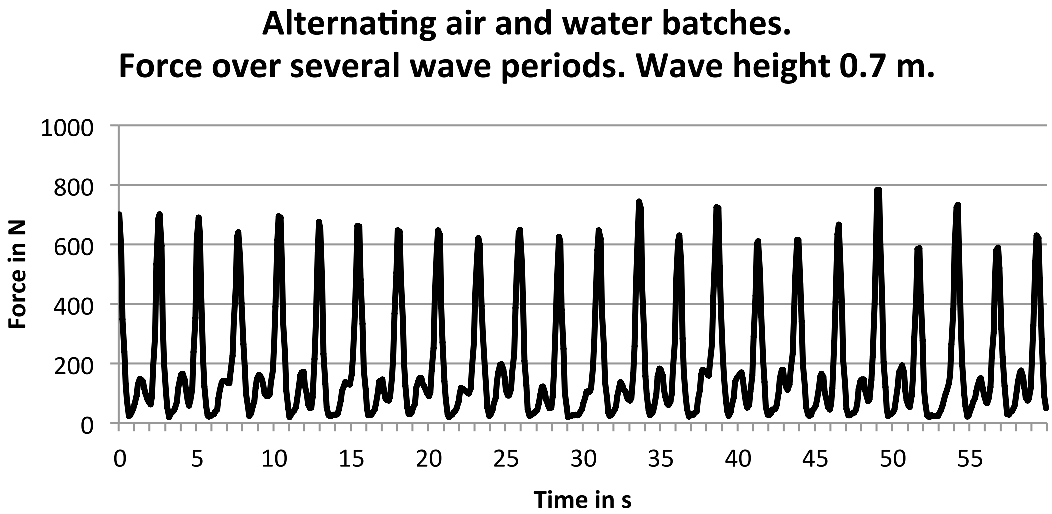


*Figure 4.7: Hose filled with alternating air and water batches. Mooring line with elasticity. Hose fastened to reservoir. Wave height 0.7 m.*

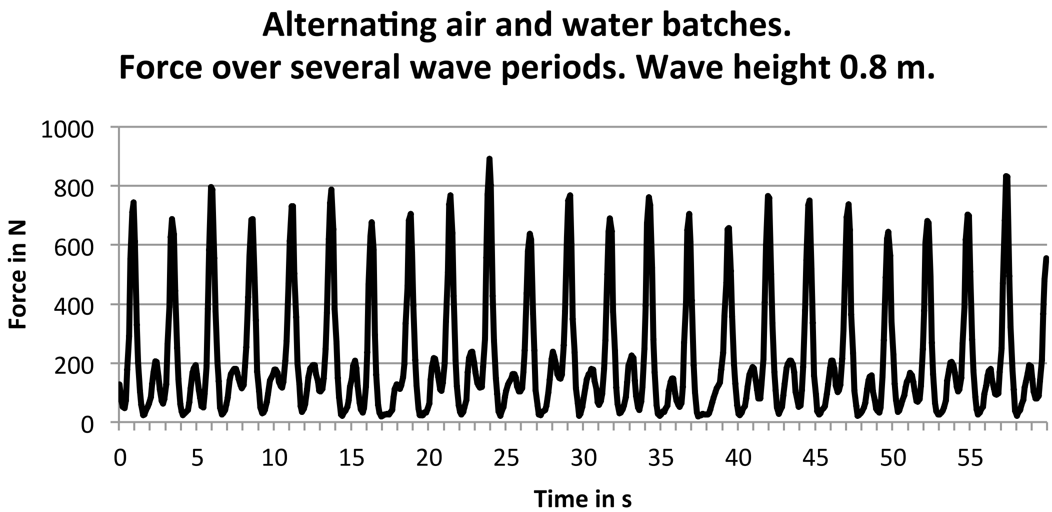


*Figure 4.8: Hose filled with alternating air and water batches. Mooring line with elasticity. Hose fastened to reservoir. Wave height 0.8 m.*

Figure 4.7 and 4.8 shows the variation of the amplitude for a hose with water and air batches over a period of time containing 23 wave periods. Notably is also that the wave height used in figure 4.8 is 0.8 m. The second smaller maxima seen between the large maxima is due to the hose being fastened to a reservoir. We see that the amplitude of the force curve oscillates around 200 N or ¼ of the maximum measured amplitude.

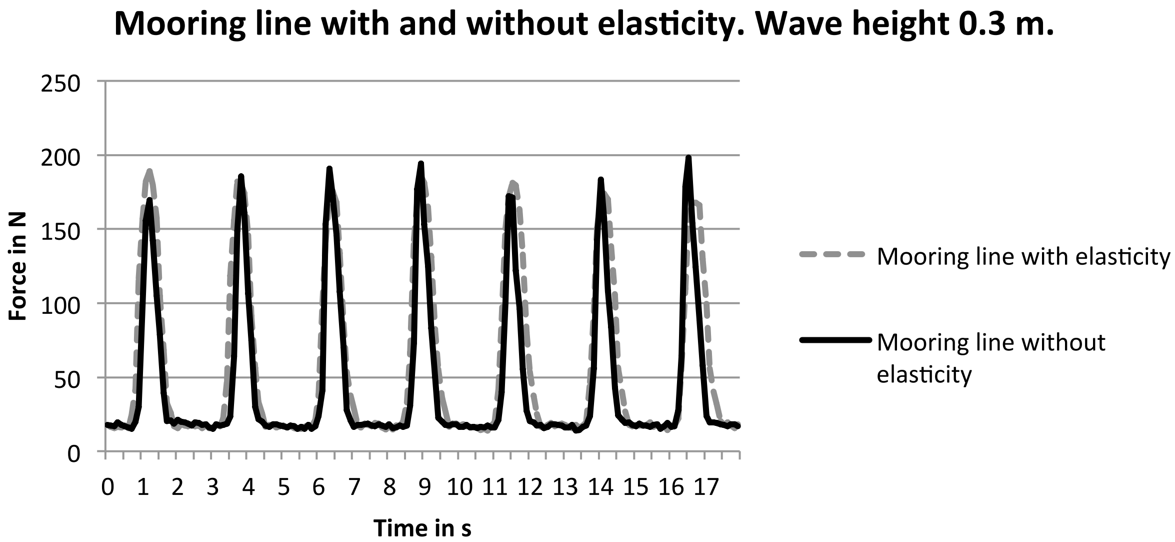


*Figure 4.9: A free floating hose filled with air only. Wave height 0.3 m.*

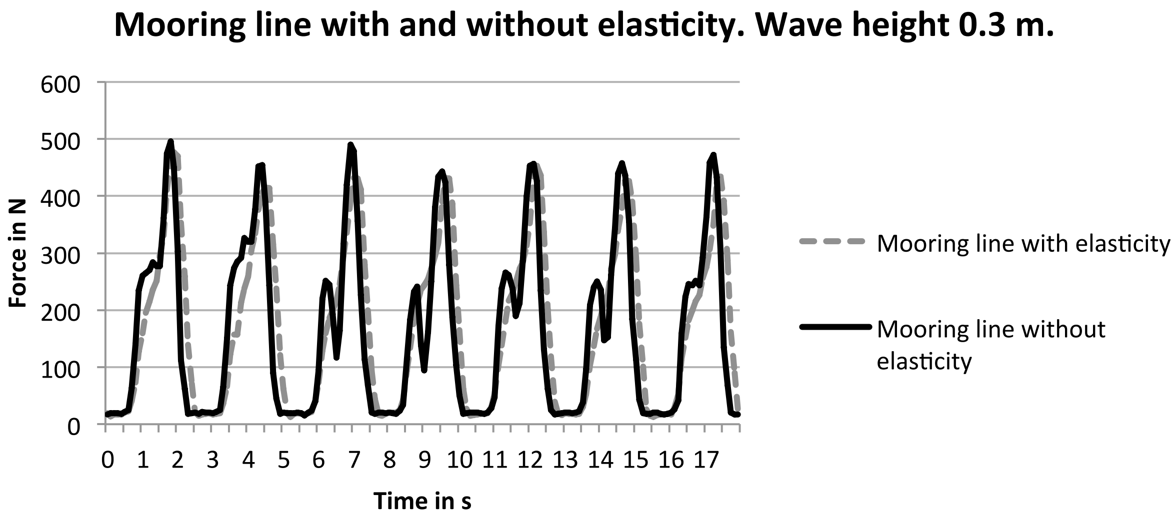


*Figure 4.10: Free floating hose filled with water only. Wave height 0.3 m.*

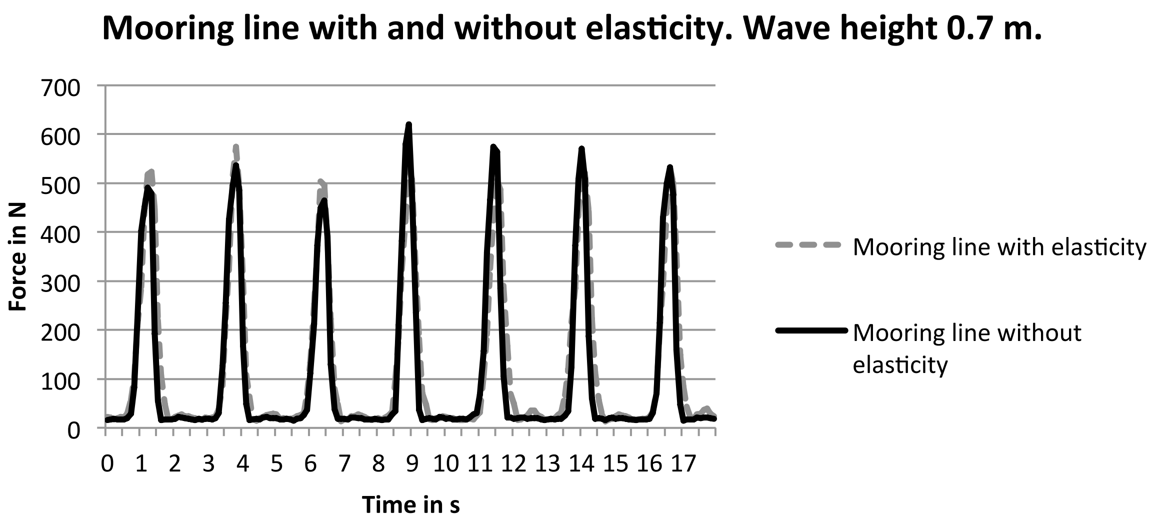


*Figure 4.11: Free floating hose filled with air only. Wave height 0.7 m.*

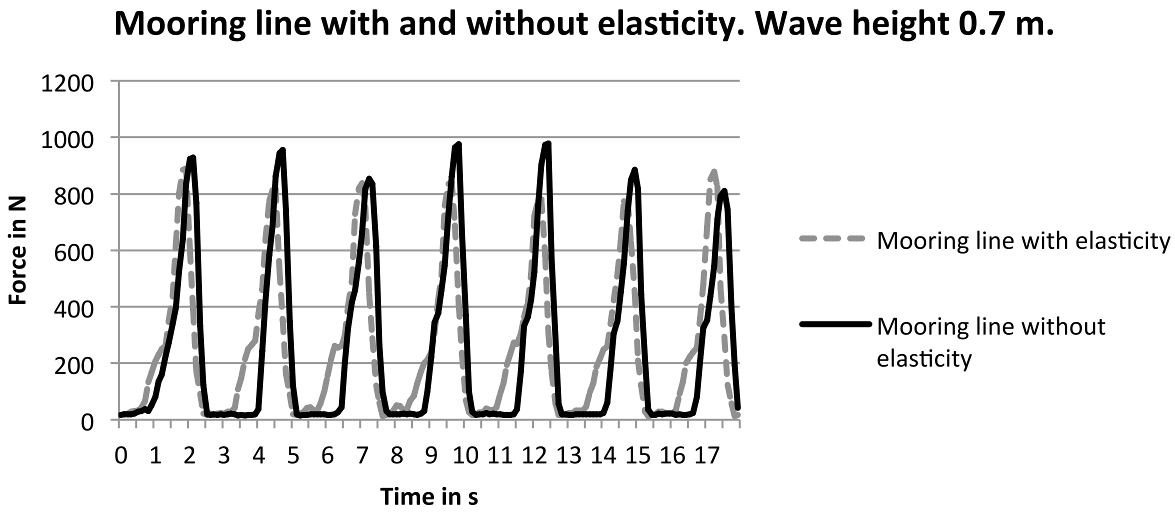


*Figure 4.12: A free floating hose filled with water only. Wave height 0.7 m.*

Figure 4.9 to 4.12 shows us no significant difference between the maximum mooring forces measured for a free floating hose with elasticity in the mooring line and a free floating hose without elasticity in the mooring line. Nevertheless it does seem like the mooring line with elasticity has a wider less steep force curve.

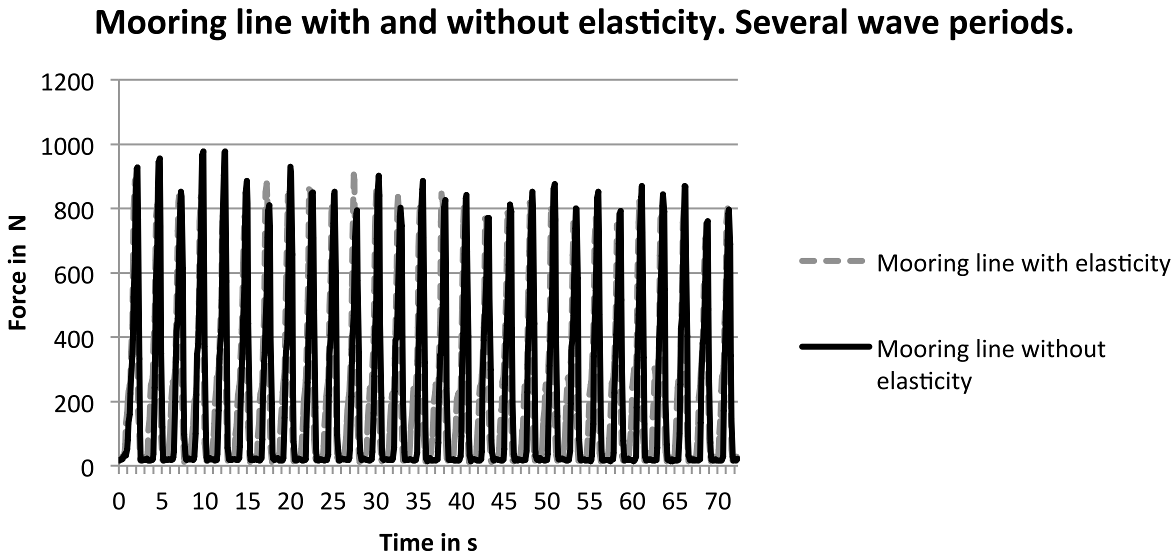


*Figure 4.13: A free floating hose filled with water only. Wave height 0.7 m. 28 waveperiods.*

When the hose is free floating there seems to be no significant difference regarding the amplitude of the mooring force whether an elastic mooring line is used or not.

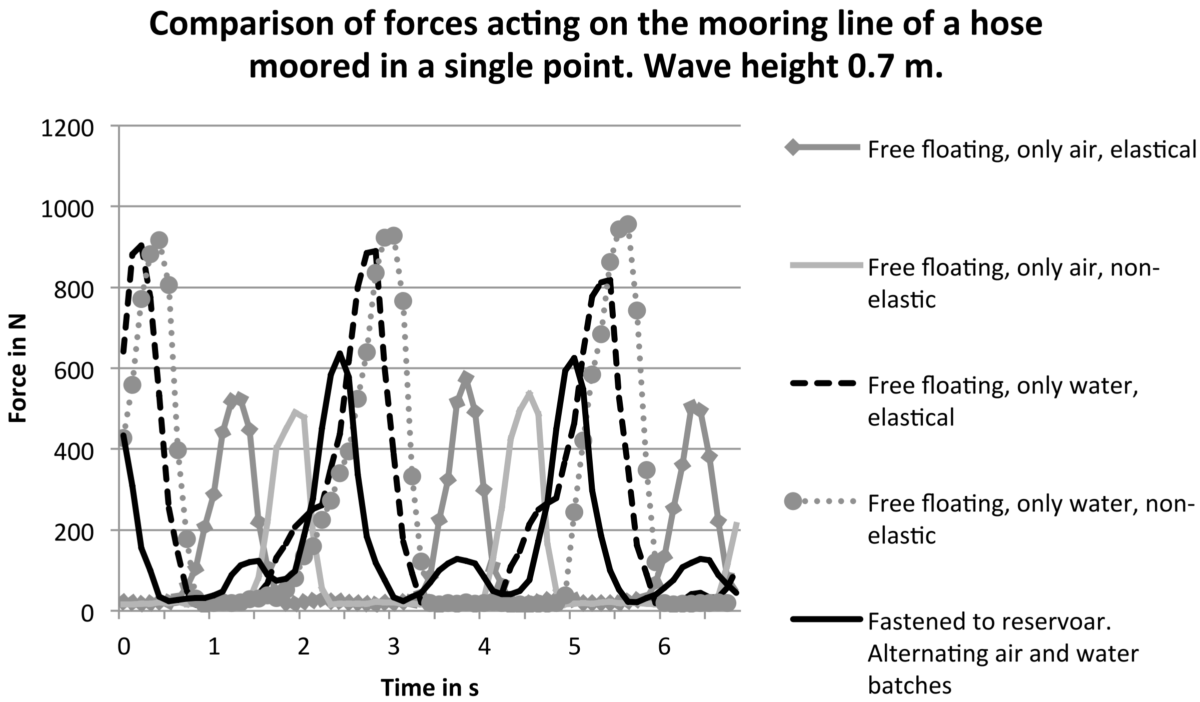


*Figure 4.14: Comparison of forces acting on the mooring line of a hose moored in a single point. Wave height 0.7 m.*

In figure 4.14 we see that the mooring force is the largest for a free floating hose filled with water moored with a non-elastic mooring line. The second largest mooring force is obtained for a free floating mooring line filled with water with an elastical mooring line. We also see the apperance of a second top for the hose that is fastened to the reservoir.

In figure 4.15 below a different test setup was used. At the location of the floting device a weight, a slightly elastical rope and a pulley was used to lift the hose up instead of the floating device.

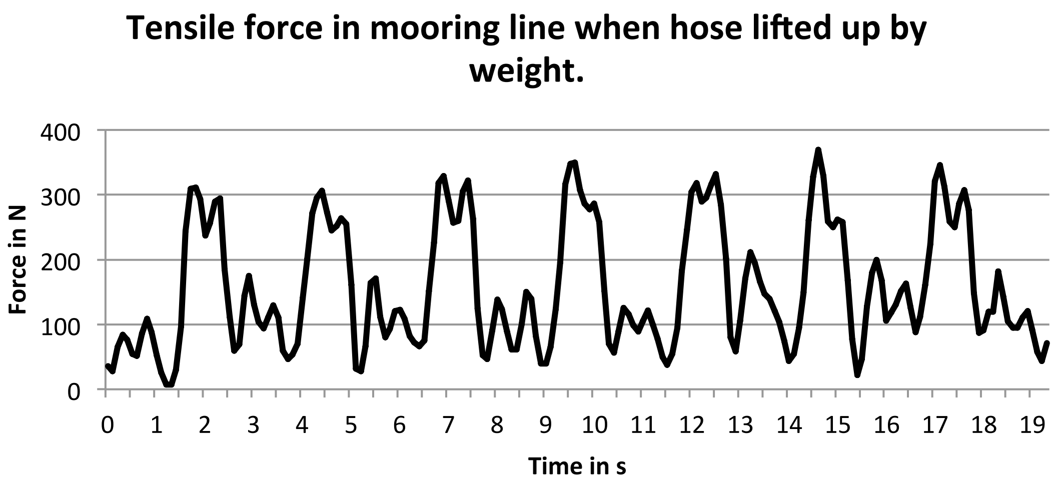


*Figure 4.15: Tensile force in mooring line when air intake lifted up by weight. Hose fastened to reservoir. Wave hight 0.7 m.*

In figure 4.15 we get a figure of more or less where the snatch and snap effects take place on a hose that is fastened and lifted up as stated. These effects happen when the mooring line gets slack which happens more often when the hose is lifted up by the less dynamic weight as opposed to a floating device that regulates the amount of buoyancy it has automatically when it is submerged in or floating on top of a wave.

## 4.2 Scaling up the results

The scale of the model used was 1:2.3 if the full size hose were to be 83 m long and have a diameter of 0.5 m. A scale factor of s = 2.3 means table 2.1 gives us a scale factor for force $f = s^3 = 2.3^3 = 12$ and for time $t = s^{0.5} = 2.3^{0.5} = 1.5$. The maximum force measured in the experiments was around 1000 N meaning that the maximum force on a full sized hose should measure around 12 kN or 1.2 tons in waves that are 1.6 m high and has a period of 3.9 s.

If say the real hose were to have a diameter of around 0.88 m the scale factor used in the tests would be equal to 4. For a hose that is 144 m long and has a diameter of 0.88 m we would get a force of 64 kN or 6.5 tons in waves that are 2.8 m high and has a period of 5.2 s.
A scale factor of 5 would give a hose that is 180 m long and 1.1 m in diameter. In waves that are 3.5 m high and has a period time of 5.8 s the mooring line would be subjected to a tensile force of 125 kN or 12.7 tons.

One might wonder what the force on an 83 m long hose with a diameter of 0.5 m subjected to waves that are 5 meters high would be. If we count backwards using the scale factor 2.3 we see that the waves in the test tank would have to be 2.17 meters for this to be tested. This is not possible with the current size of the wave tank but one might try to predict what the measured value might have been. By looking at figure 23 and 24 it is tempting to draw the conclusion that every 10 cm of wave height adds roughly 100 N to the peak load measured. But this force would be for a hose that is fastened to a reservoar and when we look at say figure 30 we see that we in fact get a force close to 1000 N for 0.7 high waves. If we instead assume that the force doubles every time we double the wave height we would get a measured force of 3.1 kN for 2.17 m high waves in the wave tank. A force of of 3.1 kN would then correspond to a load force of 37 kN on a hose that is 83 m long and has a diameter of 0.5 m subjected to waves that are 5 m high and have a wave period of 3.9 s.

Using the same line of reasoning the results are summarized in the following table:

| Length of hose | Diameter of hose | Wave height | Wave period | Dynamic load force |
|---|---|---|---|---|
| 83 m | 0.5 m | 1.6 m | 3.9 s | 12 kN or 1.2 tons |
| 83 m | 0.5 m | 5 m | 3.9 s | 37 kN or 3.8 tons |
| 144 m | 0.88 m | 2.8 m | 5.2 s | 64 kN or 6.5 tons |
| 144 m | 0.88 m | 5 m | 5.2 s | 114 kN or 11.6 tons |
| 180 m | 1.1 m | 3.5 m | 5.8 s | 125 kN or 12.7 tons |
| 180 m | 1.1 m | 5 m | 5.8 s | 179 kN or 18.2 tons |

*Table 4.1: The dynamic load force on a hose moored in a single point.*

## 4.3 Test site data

| Test sites | Vinga | Hönö | Lysekil | Halmstad | Väderöarna |
|---|---|---|---|---|---|
| Coordinates | 57.62709, 11.60226 (SWAN point 57.5284, 11.6008) | 57.68815, 11.59683 (SWAN point 57.5284, 11.6008) | 58.192825, 11.373146 (SWAN point 58.2241, 11.2441) | 56.647757, 12.838039 (SWAN point 56.606, 12.5726) | 58.54544, 11.02357 (SWAN point 58.4765, 11.001) |
| Depth | 20-30 m | 25 m | 25 m | 10-20 m | 20-25 m |
| Bottom | Rock | Clay/Sand/Roc k | Clay | Rock/Clay | Rock |
| Minimum Average Wave Height | 0.6 m in July 2011 | 0.6 m in July 2011 | 0.4 m in May 2008 | 0.2 m in May 2008 | 0.4 m in May 2008 |
| Maximum Average Wave Height | 1.8 m in January 2007 | 1.8 m in January 2007 | 2.1 m in December 2011 and in January 2007 | 1.7 m in January 2007 | 2.1 m in December 2011 |
| Maximum Wave Height during 2007-2011 | 5.1 m in November 2011 | 5.1 m in November 2011 | 5.1 m in January 2007 | 4.8 m in November 2011 | 5.1 m in January 2007 (>13 m measured in 2007) |
| Five Year Average Wave Height 2007-2011 | 0.9 m | 0.9 m | 1 m | 0.7 m | 1 m |
| Portion of waves >1m Five-Year Average | 33% | 33% | 36% | 24% | 37% |
| Portion of waves >2m Five-Year Average | 9% | 9% | 13% | 6% | 14% |
| Minimum wave period | 2.3 s in May 2008 | 2.3 s in May 2008 | 2.5 s in May 2008 | 2.0 s in April 2008 | 2.4s in May 2008 |
| Maximum wave period | 4.5 s in January 2007 | 4.5 s in January 2007 | 4.9 s in December 2011 | 4.4 s in January 2007 | 4.8 s in December 2011 |
| Span of yearly average wave periods 2007-2011 | 3.3- 3.4 s | 3.3- 3.4 s | 3.4-3.7 s | 2.7-3.1 s | 3.4-3-6 s |
| Wave length | 17 | 17 | 20 | 13 | 19 |
| Speed of currents at location | <1,5 m/s | <1,5 m/s | <1,5 m/s | <1,5 m/s | Up to 1.5 m/s (Jutska strömmen) |
| Distance to land | 520 m | 240 m | 300 m | 200 m | 300 m |
| Other | | | Close to the company ”Seabased”'s testsite | | |

*Table 4.2: Test site data.*

## 4.4 Mooring models

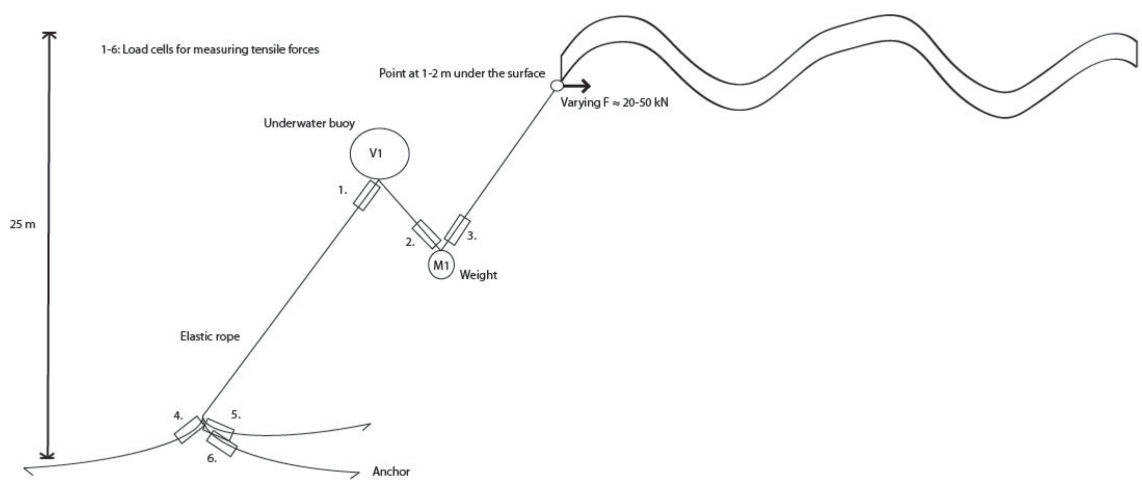


*Figure 4.16: A mooring model using an underwater buoy and a weight. Three horizontally loaded anchors.*

The three anchors are used to secure a horizontal load on the anchors independently of the direction in which the hose at the surface is pointing. The solution presented in figure 4.16 could be a mooring option where the seabed permits the use of anchors. If the underwater buoy and the weight position themselves too close to being vertically positioned to one another the mooring line runs the risk of getting entangled or worn.

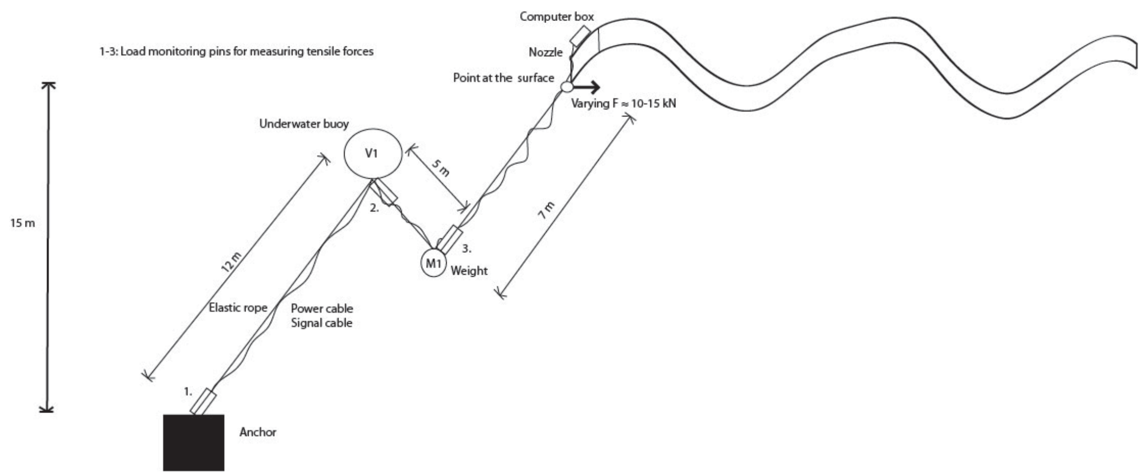


*Figure 4.17: A mooring model using an underwater buoy and a weight. The anchor is a gravity based anchor which in most cases would mean a block of concrete.*

In harsh wave environments at shallow depths the seabed often consists of rock due to the sediment being washed away by the waves. A large enough concrete block would be a secure option. The tilted Z mooring solution offers a larger horizontal compliance in the mooring lines than the solution presented in figure 4.18.

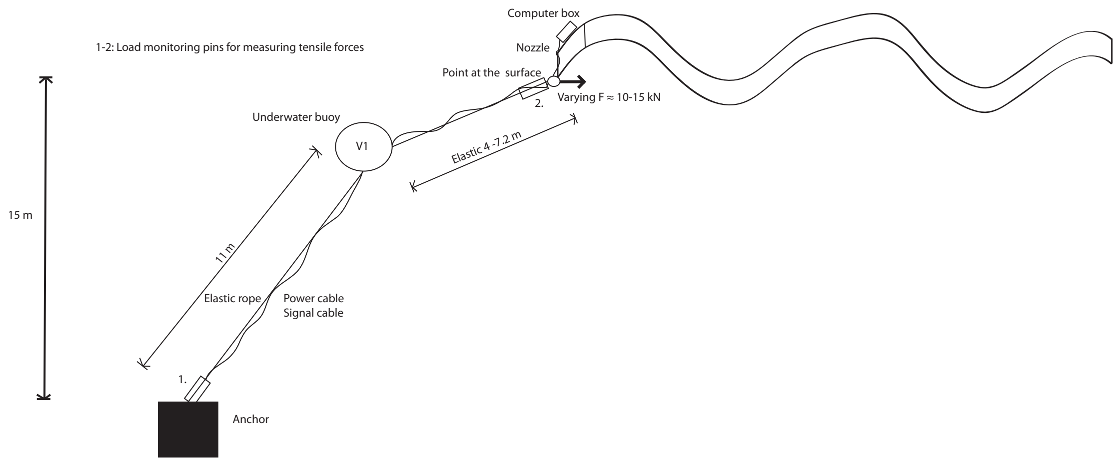


*Figure 4.18: A mooring model using only an underwater buoy. The anchor could consist of either a concrete block or one or several anchors.*

The vertical mooring force on the achor is greater in figure 4.18 than in figure 4.16 and 4.17 if the buoy is of the same size due to the fact that there is no weight present to help bring it down. In the solution shown in figure 4.18 there is no risk of entanglement and the hose can rotate freely.

# 5. Discussion

## 5.1 Mooring line forces

Throughout this work Froude scaling has been assumed to be permissible. Should viscous forces' net influence on the body's motion be found to be other than small then Froude scaling does not apply and no conclusions can be made by using Froude scaling. The studied hose was for instance not completely smooth and had in fact a slightly rough surface in the form of a thicker rubber that spiraled around the hose to give it more stability. The rubber spiral around the hose did not create any visible turbulence or eddies around the surface of the hose and therefore the assumption of small viscous influence on the hose was assumed to be satisfied. This leads to conservative force predictions as Mr Payne states in his ”Guidance for the experimental tank testing of wave energy converters”.

Another built-in error in the measurements is that the wave tank is perhaps too straight in comparision to the size of the hose itself and as the wave moves along the hose it will be absorbed or hindered by it to a significant extent. In the ocean, where no limits apply to the width of the wave front, the wave will diffract towards where it is being absorbed and thus the wave force acting on the hose will be greater in reality than in the wave tank.

When looking at the figures in section 4.1 depicting the mooring force for various conditions the following conclutions can be made:

- A heavier hose (filled with water) results in a larger mooring force.
- Elasticity dampens the amplitude somewhat and evens the force out over a larger time interval.
- A hose that is free floating gives rise to a mooring force that has a greater variation in amplitude than a hose that is fixated in both ends.
- When the hose is moored in both ends both the top and the bottom of the wave give rise to spikes in force amplitude.

The reason for the greater load forces when the hose is heavier could be accredited to the fact that it then lies more submerged in the water than a lighter air-filled hose, that will stay more on top of the wave, and thus the area hit by the energy of the wave. On the other hand the buoyancy of an air-filled object is greater than if it is filled with water so one might expect that the tensile force in the mooring line be greater the better the hose follows the wave's crest. Another explanation might be that once the hose is put in motion the momentum of the heavier body is greater than that of the lighter body and that that is what is seen in the measurements.

Had the mooring line been place at the surface it would be expected that there'd be two spikes in the measurments per wave period, since the mooring line would be stretched out at the top and at the bottom of the passing wave. Now, since the mooring wheel through which the mooring line is connected is held below the surface the mooring line forms an angle up to the hose and therefore this angle affects the amplitude of the two measured spikes. Also, when the hose is being lifted up or pushed away it is being so by the full force of the wave since it is so to speak standing in the way. On the other hand when the hose is moving towards a wave minima it is being mainly gravity driven and the direction of the motion is towards the mooring line, thus the force should be smaller.

A faster sampling rate than 10 Hz might have revealed higher spikes for the non-elastic mooring line. Then again, the hose itself was rather flexible and elastic so even when the mooring line had no elasticity there was some elasticity built into the system. A sampling rate of 25 Hz could be obtained by a mere change of computer settings, but when used the overall steering system for some reason started lagging and 10 Hz was the highest stable rate that could be used. However the water's viscous influence on the speed of objects emerged in water should help in reducing the highest load spikes. A measurement rate of 10 Hz could be sufficient to spot the highest peaks. The tests that were run were also carried out during a much larger number of wave periods than those 23 wave periods used in most figure plots and had there been larger load peaks, chances are that they would have been spotted at least once or twice as a highly deviating measurement. No such deviating peak could be found in the data material.

The up-scaling of the wave tank tests resulted in a full sized hose that is 83 m long and has a diameter of 0.5. If we compare a hose with these dimensions with the other results for a hose with those dimensions we get the following table:

| Method | Wave height | Wave period T | Force (tons) | *Reference* |
|---|---|---|---|---|
| Tank tests and subsequent scaling | 1.6 m | T = 3.9 s | 12 kN (1.2) | *Table 4.1* |
| | 5 m | T = 3.9 s | 37 kN (3.8) | |
| Dynamic part of Morison's equation | 2 m | T = 4.2 s | 17.2 kN (1.8) | *Table 3.4* |
| | 5 m | T = 6.0 s | 46.2 kN (4.7) | |
| Static part of Morison's equation Angle of attack 80° | 3 m/s current (30 m/s wind speed) | | 38 kN (3.9) | *Table 3.1* |
| | 2 m/s | | 15 kN (1.5) | *Table 3.2* |
| Power producing unit | 100 kW | T = 4 s | 16 kN (1.6) | *Table 3.3* |
| | | T = 5 s | 13 kN (1.3) | |
| | 250 kW | T = 4 s | 40 kN (4.1) | |
| | | T = 5 s | 32 kN (3.3) | |

*Table 5.1: Comparison of forces on mooring lines due to different methods of calculation.*

If we disregard the fact that the wave periods and wave heights are not in absolute compliance with each other between the different methods listed in table 5.1, we are struck by how similiar the forces' order of magnitude are for simliar wave heights.
In the wave tank there is no underlying current and no wind so the closest comparable figures should be those from the dynamic part of the Morison's equation. The tank tests and subsequent scaling gave a tensile force of 12 kN for 1.6 m waves and Morison's equation gave 17.2 kN for 2.0 m waves. This can be seen as a very good correlation between theory and experiment. For waves that are 5 m high the numbers for the tensile force are 37 kN from the tank test and subsequent scaling, and 46.2 kN from Morison's equation. The correlation continues to be good for higher waves, although more assumptions were made regarding the scaling of the measured force to bring the wave height up from 1.6 m to 5 m.
The drift force formula (7) for a power producing unit includes dynamic effect and does not consider the shape of the power producing unit. It does however take into account the waves period and if we assume that a hose of the aforementioned dimensions, or the order of dimension, can produce around 100 kW we can compare them with the static part of the Morison's equation. We see that the figures for a current of 2 m/s (that

includes an underlying current of 1.5 m/s and a wind driven current of 0.5 m/s due to an average wind speed of 10 m/s), which might be considered a normal to higher than normal sized current, corresponds surprisingly well with the relatively simple drift force formula.

One should be very careful not to draw far reaching conclusions from the results presented in table 5.1 since a great deal of assumptions have been made to reach these seemingly neat numbers.

The order of magnitude of the dynamic force on an 83 m long hose with a diameter of 0.5 m seems to be 10-50 kN depending on the waves. The force on a 48 m long hose with the same diameter should therefore be smaller than that. Out in the ocean we must add a static drift force to the dynamic force and the final number of magnitude acting on a 48 m long hose with a diameter of 0.5 m should therefore still be around 10-50 kN in waves that are up to 5 m high.

A more elastical system than the one tested in the wave tank should reduce the dynamic peak load.

## 5.2 The drift force

Attempts to calculate the drift force were made in section 3.1-3.3. They were all, with the exception of formula (7), calculated under the assumption that the hose be a stiff, inflexible structure. This is off course a simplification, but it might still be worthwhile to check the results and speculate about wether or not the drift force would be larger or smaller in reality. Personally I think the drift force will be smaller for no other reason than that when the hose is subjected to a static drift force it should be able to bend and adjust itself so that the drift force is minimized. However, there might be a dynamic response to the waves in the flexible hose that complicates the analysis.
In section 3.4 I state that under extreme storm conditions the dynamic response to the waves could result in forces estimated at up to 5-14 times that of the static drift force value, thus giving a dynamic force acting on a hose subjected to a drift force of 10-15 kN [table 3.2] of 50-210 kN.
These values should be considered maximum estimates of the force and were calculated under the assumption that the hose be a stiff, inflexible structure. Such large strain might suggest that the hose itself might break before the mooring structure is subjected to such large forces.
The upper end of the 50-210 kN interval is for very high waves. In this context waves that are ”only” 5 m high may be considered to be part of the lower end of this interval.

Since the drift force is hard to calculate on a flexible object the suggestion is that this force be measured by dragging the hose after or alongside a boat at different speeds.

## 5.3 Test site data

The wave data given in table 4.1 is based solely upon one-day forecasts and is therefore as reliable as the SWAN data itself. This really isn't too bad since the SWAN forecasts are known to be fairly accurate when trying to calculate next day's wave climate. The calculated wave value is for a specific SWAN coordinate point and not necessarily for the exact desired coordinate. However, this should be a minor problem since the wave climate can be assumed to be overall more or less the same for two adjacent coordinates. Hot spots have not been taken into consideration since they require a more thorough knowledge about a coordinate's specific suroundings but they are of course almost by definition interesting spots when found.

## 5.4 Mooring models

The mooring models presented do not present any new findings when compared to, for instance, Fitzgerald's "*Position Mooring of Wave Energy Converters*". It does however make good use of this publication's advice on how to minimize the anchor forces. The designs in figures 4.16-4.18 all comply with the movement pattern in figure 3.4.
The big question when choosing one of these designs is what the size of the buoy and the weight should be, as well as whether or not the mooring lines should have elasticity, and if so how much elasticity is optimal? This poses a very difficult fluid dynamics problem and simulating such a problem accurately in different wave climates continues to be very time consuming, if not near impossible.
The best way forward in this field would therefore probably be a trial and error approach. The dynamics of the tilted Z mooring solution shown in figures 4.16 and 4.17 needs to be studied to decide on how entanglement of the mooring lines can be avoided.

If one is to choose between the three mooring suggestions, without further studying the effects of the tilted Z solution in figure 4.16 and 4.17, one should pick a solution like the one presented in figure 4.18 since there is no risk of entanglement in this solution.

# 6. Conclusions

A first ocean test with measurements of the forces acting on a 48 m long hose with a diameter of 0.5 m should be made using mooring model 4.18 since its mooring lines run no risk of entangling themselves. The mooring structure should be designed to withstand a dynamic force of 50 kN and a static current force of 10 kN. If the system is elastically compliant enough to move with the waves, the dynamic force should prove smaller than this in reality.

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