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WAVE ENERGY CONVERTER

Master Thesis Spring 2012

Written by Jose Vazquez Taboada

University of Stavanger

To my parents Josefa Taboada and Nicolas Vazquez

To my rest of family, and specially my Uncle David Taboada decease.

PROBLEM DESCRIPTION

The work along this Thesis will carry out the installation of wave energy devices for power extraction, from the kinematic and potential energy of the waves to lineal movement, through a WEC (wave energy convertors), formed by different types and energy transmission systems from point absorber to drive generators.

The whole power take off system for each device, will be different for each emplacement, the first one will be stay onshore and the second one offshore, due to different ways to carry out and eject each one.

Analytical calculations, CAD models and FEM analysis, will be the basic aim for carrying out the report. Such as, the conceptual study of the present technology to determine behavior and efficiency for each wave energy device.

ABSTRACT

The principal objective of this thesis has been to study and research on Water Energy Convertors (WEC) devices, carrying out theoretical and analytical approximations.

Nowadays, the demand of renewable energy resources is increasing and tent to increase deeply in the next couple of years. This has been forced mainly by oil crisis and by the need of creating energy by clean resources.

The main proposal has been to find out the maximum efficiency level for two different types of convertors and energy transmission process from point absorber to drive generators. Carrying out the dimensioning process of each device separately, according to exigencies demanded for each one.

The entirely thesis is oriented completely from an application point of view, using the engineering background of maritime technology, mechanical and offshore industry.

An interesting arriving point was obtained from Scatter diagram, an optimal potential energy range was captured by wave's motion with wavelength from 7.5sec to 10.5sec period. This range will be the most appropriate range of work for WEC.

The main motivation of this report is to provide two different WEC application approach from academicals perspective to industry.

PREFACE

This thesis report is written in partial fulfilment of the study of Master of Science (MSc) in Mechanical Engineering. The thesis part of the study was carried out at the Faculty of Science and Technology, at University of Stavanger, in the period February to first July of 2012.

Firstly, I would like to thank my supervisor Assoc. Professor Hirpa G. Lemu for his whole process of guidance and teaching, as well as, his constant trust in my dairy work and his perfect and huge orientation constantly, in his small office.

Secondly, this entire work could not have been a reality without the practical help provided from Olaf Akevoll, who allowed me to work on this interesting project idea and technology. The entire work of this thesis is based on his patented ideas. I appreciate the way he fixes meetings.

Thirdly, this thesis work could not have been possible without the fidelity of all the members concerning this faculty, such as the help provided by the library, who provided me the necessary literature for this work.

This work would not have been possible without the motivation from entirely climb group of SiS sport, especially from my friends Christian and Jakeb. As well as the productive and nice talks and discussions from Robert and Whida (Offshore Engineering Students), in the laboratory computer room, where I carried out almost all the time of this entire thesis.

Also, I would like to thank my home university (University of Vigo), where I started this study and got the background knowledge for this thesis work, as well as, all teachers integrates, especially the Professor Jose Antonio Perez.

Last but not least, I would like to thank my close friends at University of Vigo who have always surrounded me with nice relationship at all moments, as well as, many people who have shared their own time along my life until now. And finally, my most big and deep thanks goes to my parents, Josefa and Nicolas, who gave me the chance to study and supported me along my studies and my entire life so far, suffering and working hard every day.

Jose V. Taboada

Stavanger, July 2, 2012

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CHAPTER 1 INTRODUCTION

1.1 General Background

1.1.1 Wave Energy

There are six major renewable energy resources in the oceans: waves, tides, ocean currents, thermal, salinity gradients and the biomass. In reality, wave energy can be considered as an undiffused form of solar energy. The water of the oceans is in constant movement, the gravitational pull of the sun and moon oscillates the surface of the oceans twice a day while the wind disturbs it into waves (Figure 1.1). Winds are created by the differential warming of the earth and, like they pass over open bodies of water, they transfer some of their energy to a form of waves. The wave energy flux is stored in waves as both kinetic energy (in the movement of the sea water particles) and potential energy (in the amount of mass of sea water displaced from the mean sea level).

The amount of energy transferred, and therefore the size of the resulting waves, depends mainly on the wind speed, the length of time for which the blows and the distance over the surface which it blows.

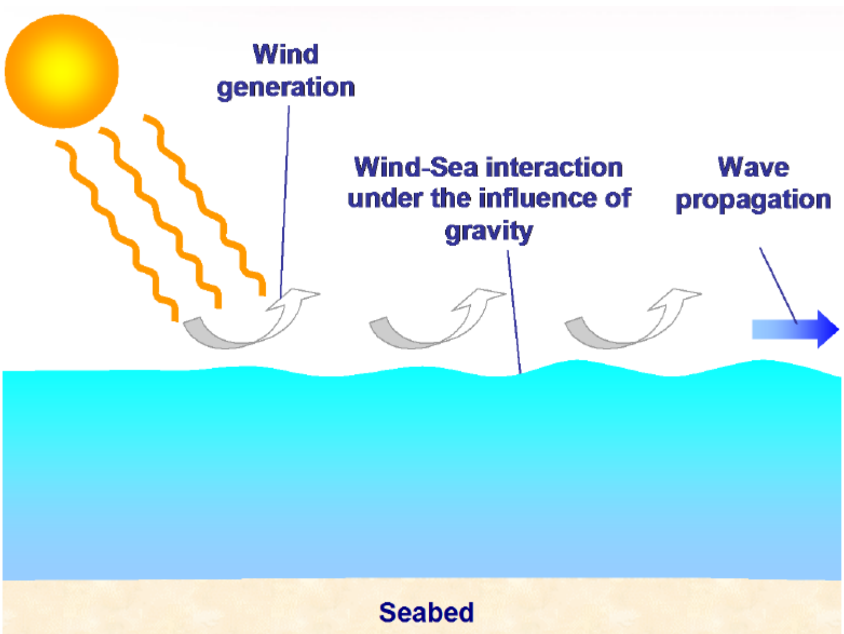


Figure 1.1. Concept of Wave Energy [1]

- **World Global situation:**

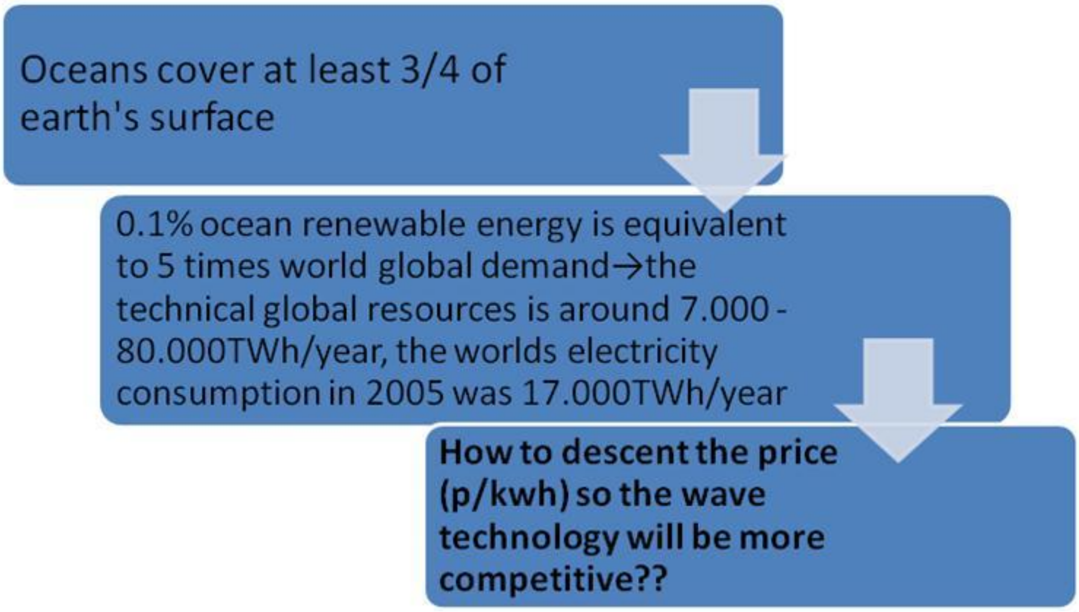


Figure 1.2. World Global situation

1.1.2 Characteristics and types of waves

The main important parameters used to describe waves are their length and height, and the water depth over which they are propagating. The propagation environment (shallow water, transitional and deep) depends mainly on the level of the depth. All other parameters, such as water velocities and accelerations, kinematics (motions) under waves can be determined theoretically from these quantities.

The principal types of sea waves are deep water waves, inshore waves, and destructive, constructive, refracted waves. In addition, there exist two main grouping of waves based on regular and irregular waves.

The linear waves, known as sinusoidal waves, describe a regular movement, in contrast to nonlinear waves. The problem is that real ocean waves are normally not regular. As a result, the wave height and wave period change in a stochastic manner, depending of the value of each parameter. In this report, we assume that the amplitudes of the waves and oscillations are sufficiently smaller for linear theory to be a good approximation, according to specific the date to implant this technology, as I well showed in Appendix A. But to get better approximations and results, I will prove in the next couple sections, how it will be much better to use other recently theories of waves.

1.1.3 Representation of the waves

Sea waves can be described as regular waves (sinusoidal), shown or irregular waves. Though ocean waves are actually never sinusoidal, low amplitude swells may come close to sinusoidal, as shown;

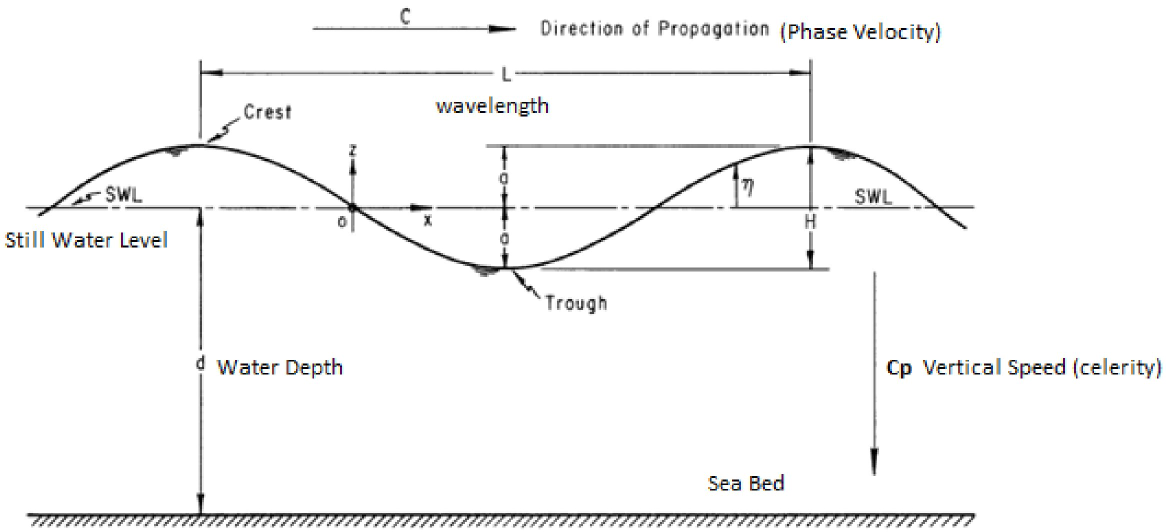


Figure 1.3. Representation of a regular wave [2]

1.1.4 Sea water depth propagating

The propagation of the waves depends directly on the depth and length of waves. As depicted in Figures 1.4 and 1.5, both represent the main parameters for a global distinction of wave characteristics with respect to each situation in the ocean. As given in Appendix B, the relationships between these parameters change considerably and designated as the Ursell parameter (Ur) as expressed below:

$$\boldsymbol{Ur} = \frac{\boldsymbol{\lambda^2 \cdot H}}{\mathbf{d}} \begin{cases} \geq 100 \rightarrow \quad \text{Linear (Airy) Wave Theory.} \\ \\ \leq 100 \rightarrow \quad \text{Non-linear Theory.} \end{cases}$$

Another possible classification, as depicted in Figure 1.4 and Table 1, is as deep water, transitional and shallow water waves. This classification is basically based on the ratio between the depth (d) and length (L) of the wave. Directly, both parameters are related firstly with current forces (depth) and wave forces (length). As well we can observe on figure 1.4.

By another side, it is clearly reflected on figure 1.5, that the amount on energy transported a long currents speed varies, depend the situation (deep, transitional or shallow water), for instance, in shallow water the currents speed tend to be heights than depth water, insomuch as the amount of energy disperse will be bigger on deep water, that means that the speed currents are lower. As well as, the way described by currents in each situation. As a consequence, the trajectory described by a particle, will be change considerably.

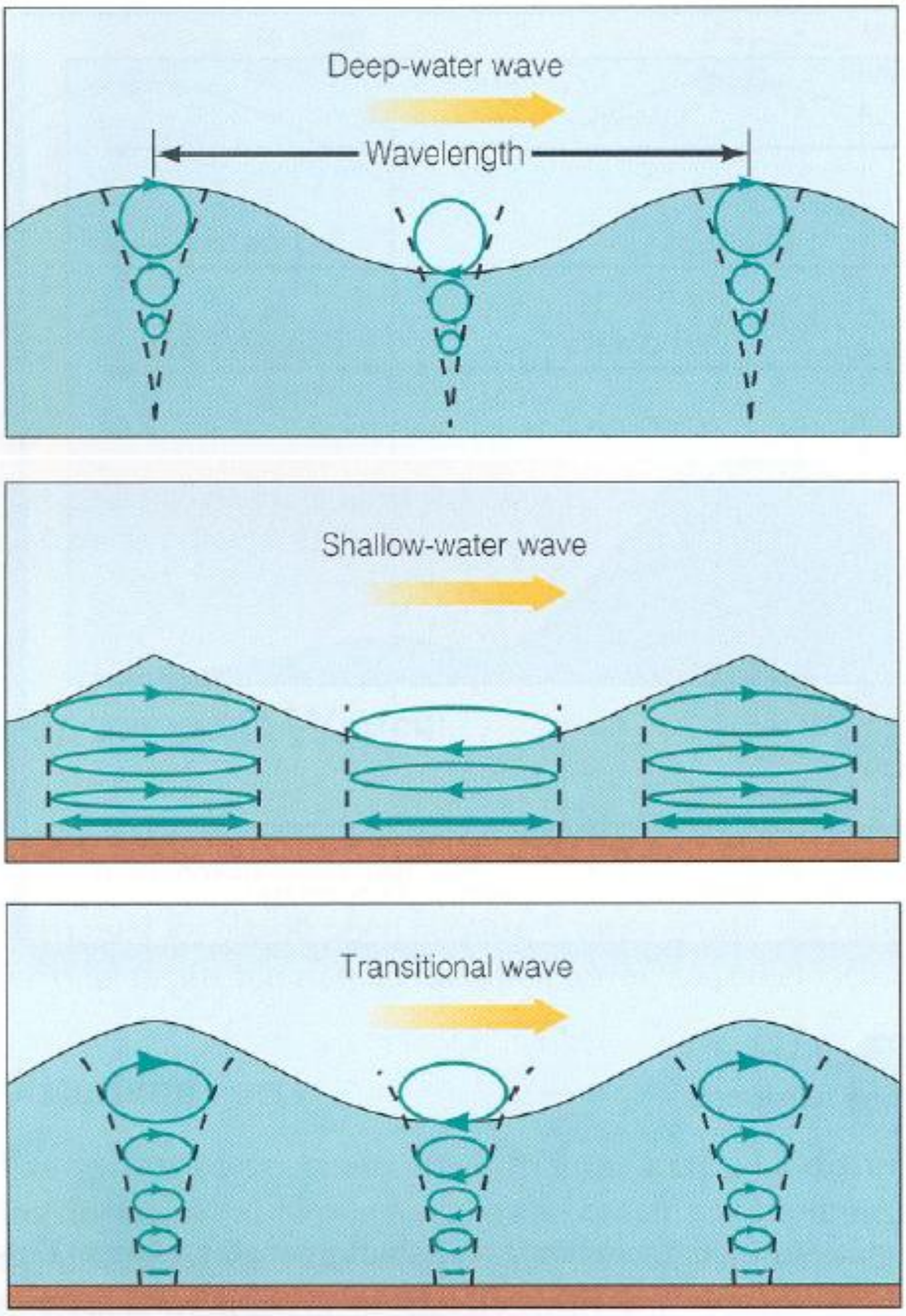


Figure 1.4. Wave depth propagation [3]

Table 1.1. General classification

Classification	d/L	2πd/λ	tanh(2πd/λ)
Deep water	>1/2	>π	≈1
Transitional	1/25 to 1/2	¼ to π	tanh(2πd/λ)
Shallow water	<1/25	<1/4	≈2πd/λ

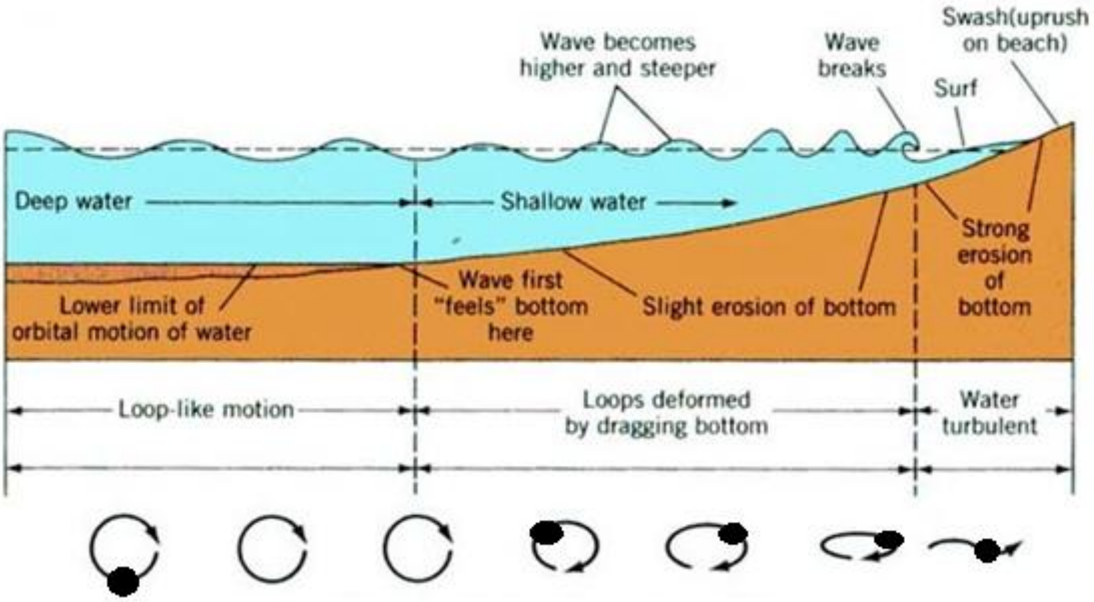


Figure 1.5. Change in orbital motion of water particles

My initial interactions to focus and calculated all parameters of waves, indicates on Appendix B, was from shore protection manual [2] with the date from North Sea and assuming constant seabed of 75m depth. The only different a long development of all theories from linear to fifth order is grade of approximation on waves in the ocean.

All I want to illustrate on Appendix B is how we can get a much better approximation, for instance, from linear to third stoke order.

In the next table and figure, we can examined how much can influence the relationship between depth(d), height(H), period(T) and length(λ), depending the situation (shallow, transitional or deep water) to give a response on surface behavior.

Table 1.2. Max and Min values of wave parameters.

	Ur	H/(g*T²)	d/(g*T²)	d/λ...
Max.	18,531	0,008	0,849	5,337
Min.	4,113	0,004	0,045	0,297

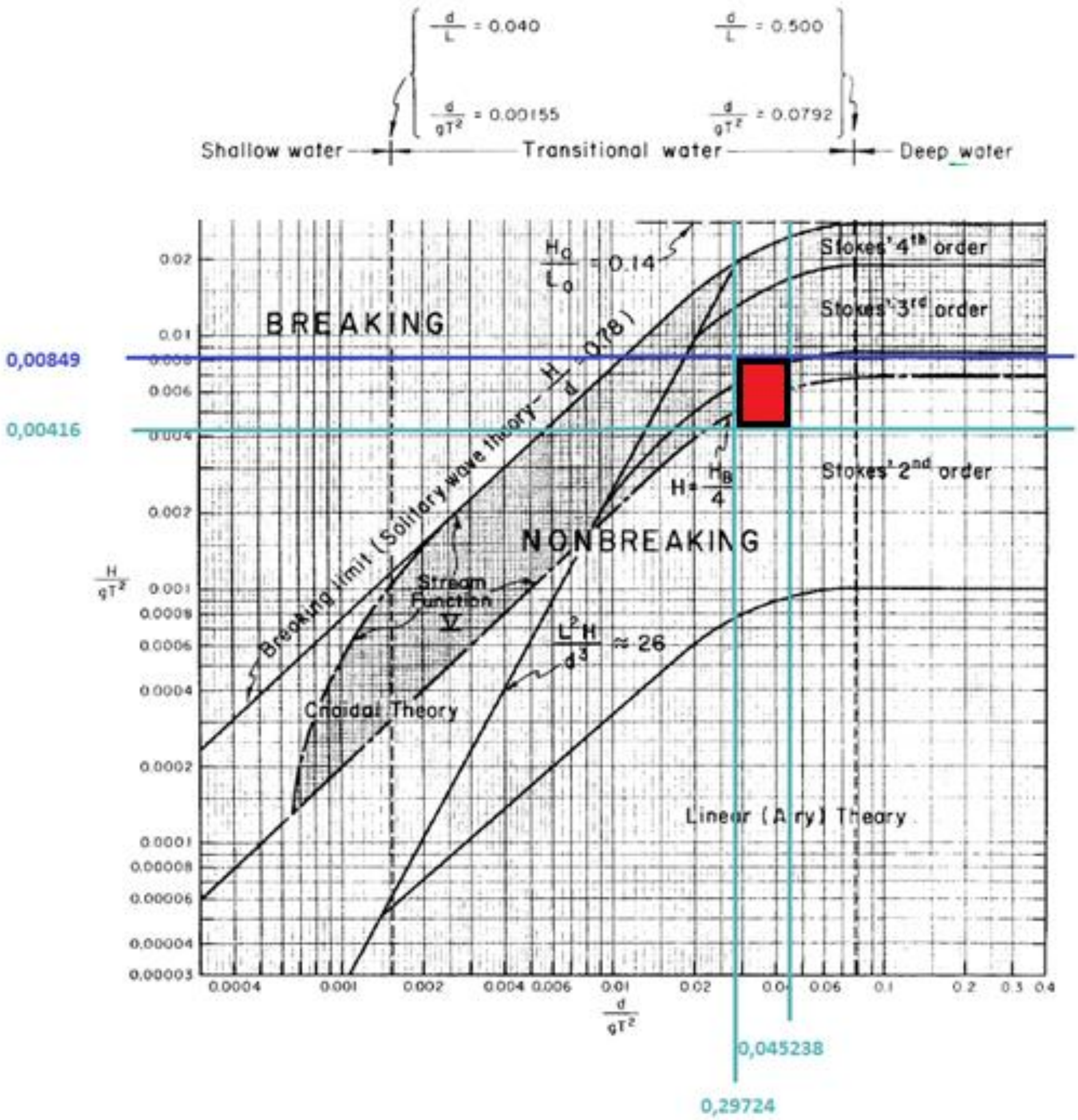


Figure 1.6. Regions of validity for various wave theories [2]

To reach a main theoretical conclusion was, that all calculations which ones I have reflected on Appendix B, will be more precise to use third stokes order (theoretical) than linear theory, as well we can observed on the last figure 1.8, but to be more precise, as well recommend DNV, we must work with fifth order stokes and dean stream, to get best approximations. Hence, all the future calculations and results of environmental behavior of North Sea, which one I am carrying out by Orcaflex, will be calculated on fifth order.

1.1.5 Ocean wave converter (OWEC)

Many devices of ocean wave convertors are also known as offshore wave energy devices. The first known patent to take energy from ocean waves was given out in 1799 and was filed in Paris by Girard and his son [4]. Real research on wave energy didn't begin development until 1973 (oil crisis). Then in 1980 when gas was cheap, people stopped funding wave energy. Actually with all the focus on renewable energy, people are once again looking to the ocean for cheap, known as clean power. All the projects done around the world, especially in Scotland (University of Edinburgh) by Stephen Salter [5], and in Norway (NTNU) by Kjell Budal[6] and Johannes Falnes[7] have demonstrated, within the last decade, that it is possible to get energy from the waves efficiently according to each device over the specific fetch area on sea conditions. Almost all these devices have been installed with respect to the conditions of waves, but others still prefer simple patent idea.

The basic idea of wave energy conversion process can be stated in very common terms as follows: the torque (or force) produced in a system by an incident wave conditions relative motion between and absorber and resistance point, which acts directly on, or drives a working fluid through, a generator principal mover.

This kind of technology demands specific conditions to satisfy the requirements to efficiently capture the energy from the waves, because as we can see over the map (Figure 1.7), the wave energy flux (Kw/m) around the world is not uniformly distributed, the better conditions to implantation of this technology are distributed between 30^{0}latitude to 60^{0}North and South hemisphere, inside this range we have variation of different values of wave surface power, depending on depth, period and height of the wave, from these last three parameters, we can derived almost the rest of parameters of the theories of waves.

From a technological point of view, there are good causes for deploying wave energy convertors at exposed within this range, not because the wave power level is sufficiently high, but because of the fact of avoiding problems associated with extreme waves. The assessment of the wave energy resource is a basic essential for the strategic planning of its utilization and for the design of wave energy devices. Each device is efficiency for specific location, as well as their cost. The main method to approach a measurement of capture efficiency is thankful the wind wave, respect to the angle of wave direction, as we can observe in the graph (Figure 1.4) below. As we can see over theses wave roses, the average angle of wave direction (θ) varies depending on the sea location, given that the way of wind change. Along this research, I will assume along this research that the wave direction angle will be 180^0. According to the last cited, the marine currents also tend to follow the same angle direction.

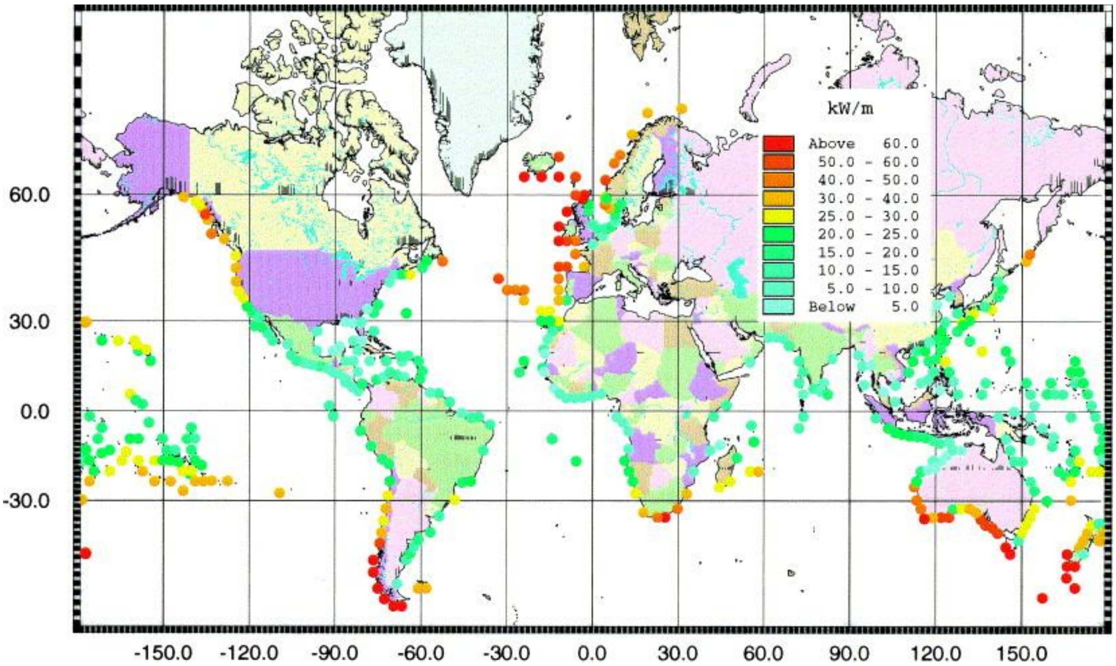


Figure 1.7. Global distribution of time-average wave power [8]

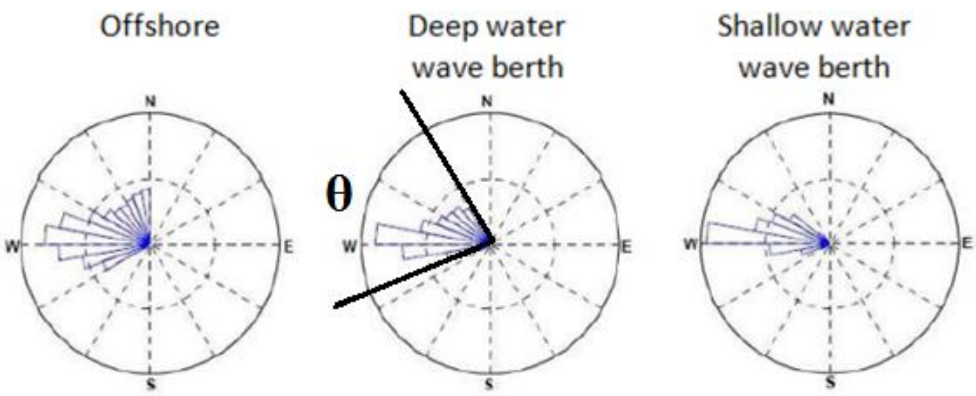


Figure 1.8. Annual wave power roses(North Scotland)[9]

1.1.6 WEC Devices

Many of the different types of existing wave energy converters may be grouped in various ways, for example, with respect to their horizontal size and orientation. If the dimension is very small compared to the typical wavelength, the convertor is named a “point absorber”(Falnes and Budal 1975), while if the extension is comparable to or bigger than the typical wavelength, the convertor is called as “line absorber”(for instance, pelamis) [10], but others author prefer to use “terminator” and “attenuator” [11]. We can observe these differences clearly from the figures given below (Figure 1.5. and 1.6.). The main different between both is the wave-power absorber whose horizontal extension is rather small compared to the wavelength. This is the reason to name these devices as a point absorber. The convertors are also classified according to their different locations.

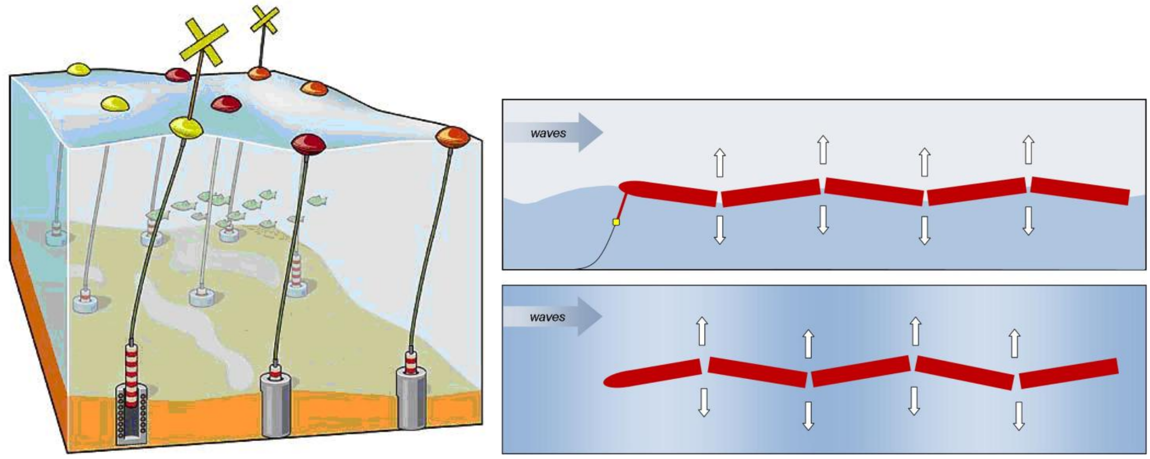


Figure 1.9. Point and Linear absorber.

The particular specification for the point absorbers should be installed in locations (shallow water and transitional) where they are of least inconvenience to navigation and fishing. Due to the cost of electric power transmission by subsea cables the distance from on land should preferably be less than a few kilometers, also the losses could be considerably lowers.

1.1.7 Cape Verde principle by Euro Wave Energy (EWE)

This research focuses on the point absorber, the device from EWE that can be classified as hydrodynamic of offshore device. As illustrated in Fig. 1.7, the principle of this device consists of a floating buoy, placed on shallow water and near shore, coupled via a wire cable directly to connect with driver wheel that is placed the seabed, and then the wire cable extends to outside of sea (on land). It is then connected underneath gearbox system through a transmission belt, and then joined by another cable until top of the tower and fixed with weight moving up and down. The weight maintains the whole tension of the wire cable. The stroke length of this weight will correspond to the maximum possible height value of the waves (that is around 14 m). The principle of this mechanism can clearly be observed from the descriptions on the global system below (Fig. 1.10).

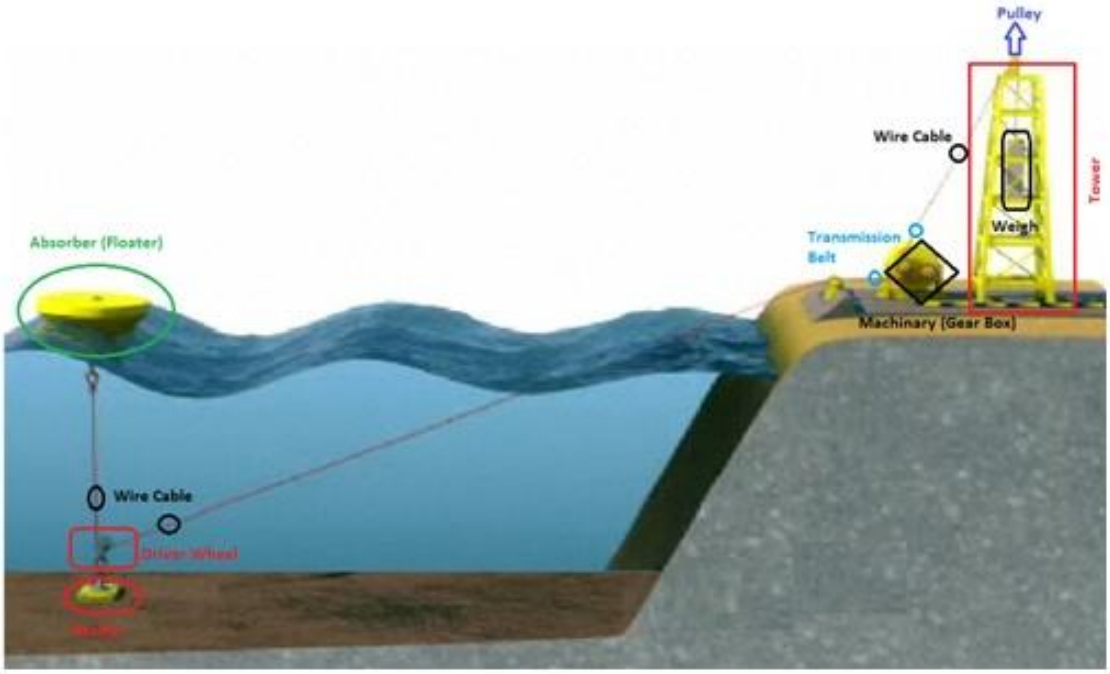


Figure 1.10. Cape Verde device by EWE, adapted from [12].

The main purpose of this mathematical model is to find out the wave loading on a fixed or floating floater of agreed design; in absorber hydrodynamics it is the response of a specific floater design to certain fetch area conditions that is of interest. The approach is that the device must perform sufficiently well to absorb an acceptable amount of power for moderate and small seas, that means shallow water and near shore.

According to EWE [12][1], the difference between this version of the technology and the other models is that only the absorber and anchoring system is placed subsea. This has many advantages including

- Inexpensive to produce, transport and install locally.
- All components are over the counter products.
- Simple installation.

[1] [12] http://www.eurowaveenergy.com/cape-verde-version/

Conversion Global System Outline:

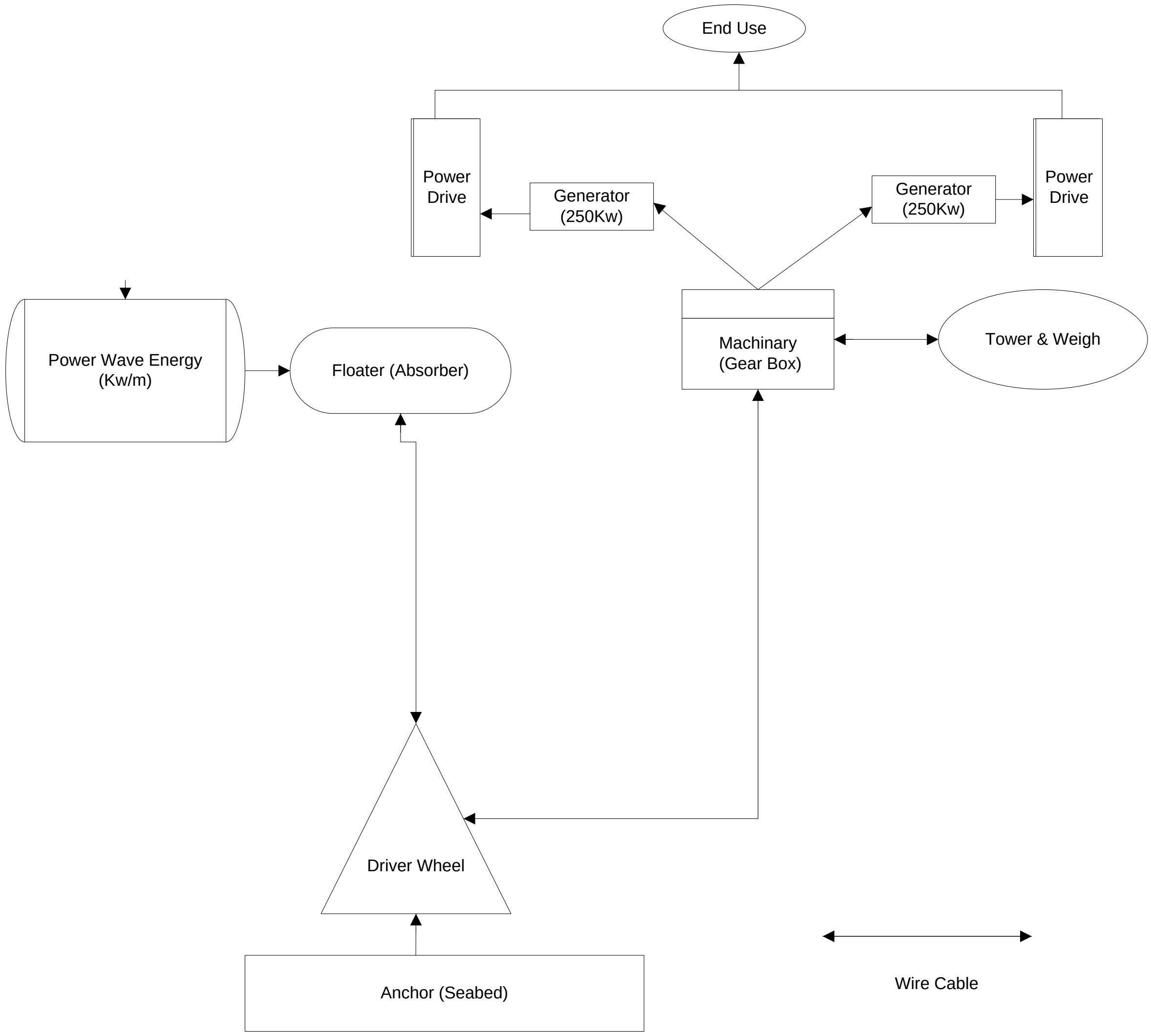


Figure 1.11. Conversion System of Cape Verde.

1.1.8 Flexible Drive Line by Euro Wave Energy (EWE).

The initial patent well showed on web page [8][2], consisted in three main parts:

- Buoyancy.
- Main module.
- Tension legs.
- Anchor seabed.

We can consider the main part of whole system, as main module, which one formed by:

- Drive line (wire cable or chain).
- Generator.

This is the second model patent by EWE, the main difference with the last model, explained on last section, are;

- Whole system placed between shallow water and transitional water.
- Generators are submersed inside main module.
- Complex installation.

From the economical point of view, the installation work and equipment, as well as the whole maintenance process, will be more expensive than Cape Verde device. Also, from a perspective of Offshore Engineering, the main module has a lot freedom degree and highly flexible, as well we can see, it is suspended from a single point from floater and anchored on seabed bottom. The whole system must be resist all currents, which ones are making drag forces and moments, as well as give good response, according to the motion of all currents.

The whole principle of this convertor energy device can be observed from the next representation below (Fig. 1.12).

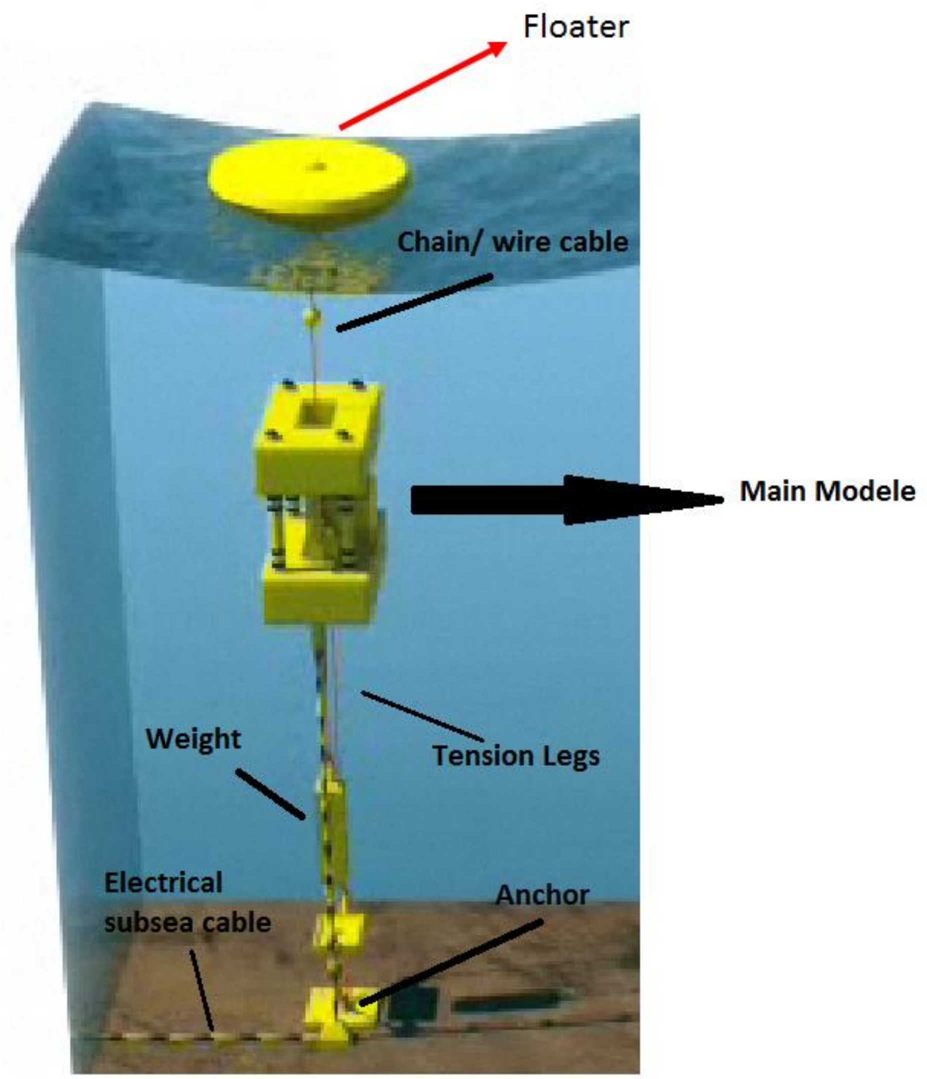


Figure 1.12. Flexible Drive Line by EWE, adapted from [9].

2 [13]http://www.eurowaveenergy.com/flexible-drive-line/

Initial Conversion System Outline:

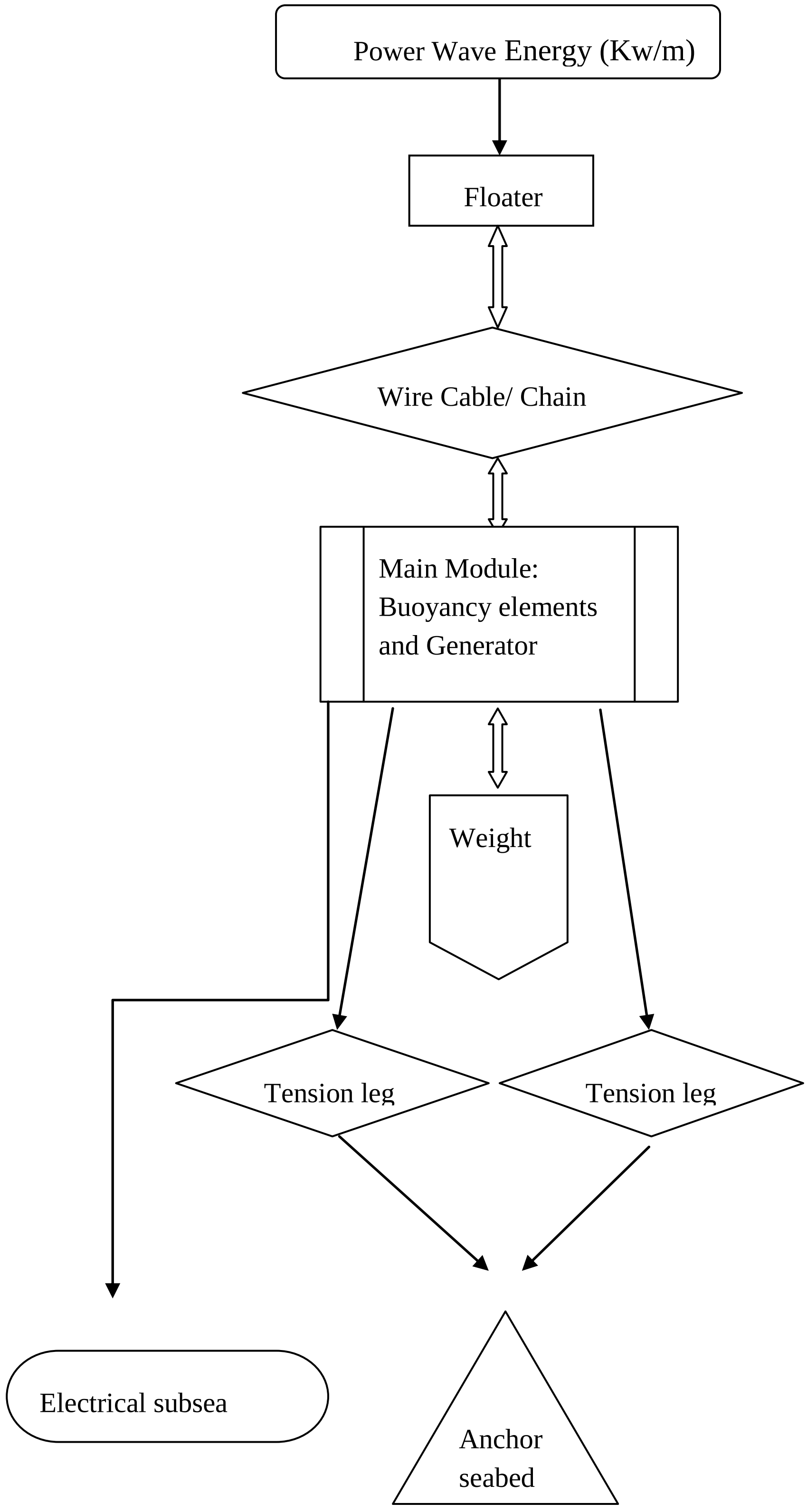


Figure 1.13. Initial Conversion System.

1.2 Outline of the Thesis

The present report is distributed into four chapters. Chapter 2 begins with a general classification of WEC, where I am describing the different types, such as the selection process and analyses of type floater according with the global distribution of wave Power in North Sea.

Hence, this chapter focuses on determination of the most appropriate floater for each device.

Chapter 3 takes an intensive explanation of assemble and lifting process for each device, as well as the description how each WEC device works in each emplacement. Also, described the analytical graphical results of hydrodynamic load, and the corresponding selection and analyses of Anchor System used for each case in particular.

Finally, Chapter 4 are explained the integrity of dimensioning process, for each Machinery, as well another bodies, such as an design and analysis of a Tower model for Cape Verde device.

CHAPTER 2 ANALYSIS AND DESIGN OF FLOATING CONVERTER

2.1 Classification of wave energy convertors (WEC)

Many different types of possible wave energy convertors do exist. The global classification given to the device as being a *Point Absorber*, a *Terminator* or an *Attenuator*, according to their horizontal size and orientation (Fig. 2.1). Another classification is according to their different locations (Onshore, Near shore or Offshore).

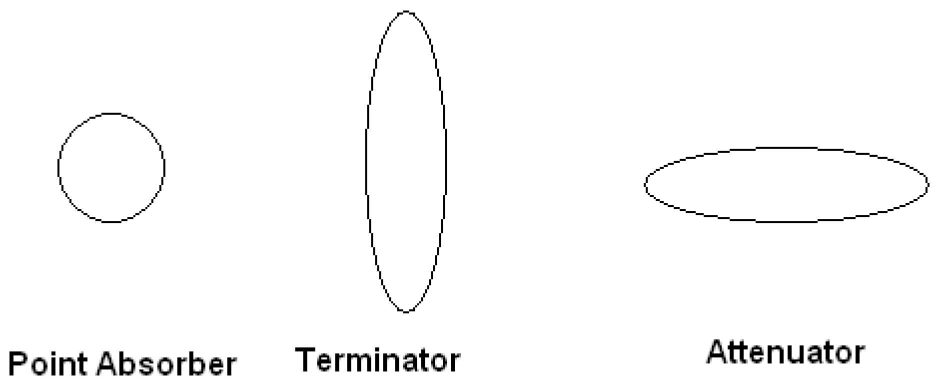


Fig. 2.1. *Point Absorber, Terminator, Attenuator*

The main differences between all of them are:

- *Point Absorber:* is quite small to the wave length and it can capture energy from a wave front greater than the dimension of the absorber.
- *Terminator:* the horizontal axis is parallel to the incident wave crest. The reflected transported waves determine the real efficiency of the device to use.
- *Attenuator:* also called linear absorber (usually are placed in a line) the principle axes are situated parallel to the direction of the incoming wave due to orientation close to parallel to the direction of wave propagation.

The levels of efficiency varies in all, but the *Point Absorber*, which is the one used in this research, does not have a principal wave direction and is able to catch up energy from waves incoming from any direction.

We can make up many combinations of absorber energy converter and structure type is possible, but the most known classification for wave energy conversion process is shown in Fig.2.2.

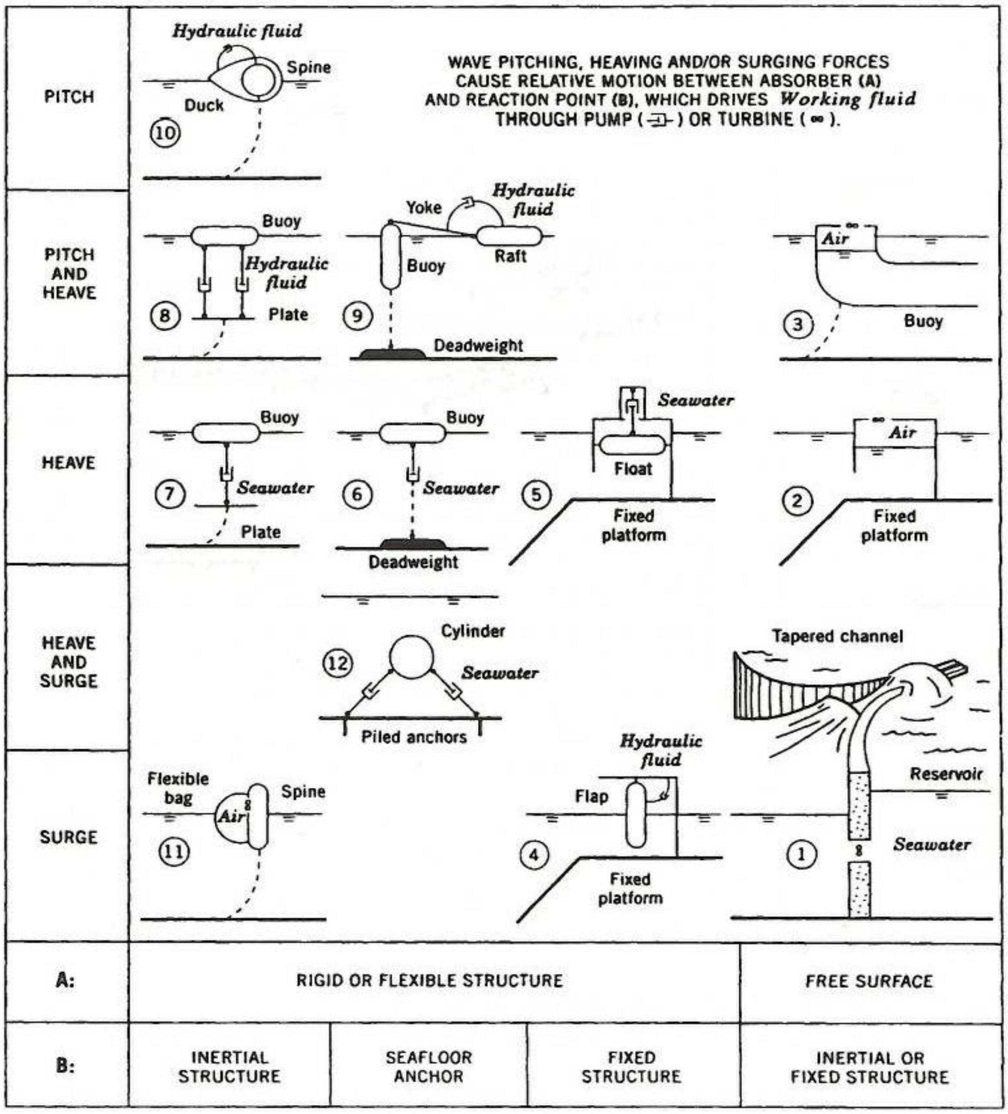


Figure 3.1 Classification of wave energy conversion processes based on mode of energy absorption (pitch, heave, surge or combined modes), type of absorber (A), and type of reaction point (B)
Source: Hagerman, G. 1995a. (Reprinted by permission.)

Figure 2.2. Classification of WEC processes [14].

2.2 Sizing of WEC.

With respect to wave parameters of a North Sea that will be used as the base for the design of this wave energy converter in this research, and the accompanying electrical system (generator AC), will be key to take off the most optimal energy in WEC.

On one hand, the global system will be designed to resist extreme waves, which can be found through the "worldwaves data" (www.globwave.org) or from Appendix B (North Sea date), but all this data are more specific with Scatter diagrams (Appendix A), as well named in chapter 1. The global system is optimized for these extreme waves then it will operate at power levels more than an order of lower magnitude than it's happened almost all the time. As well as a WEC system is hardly be viable from an economic perspective.

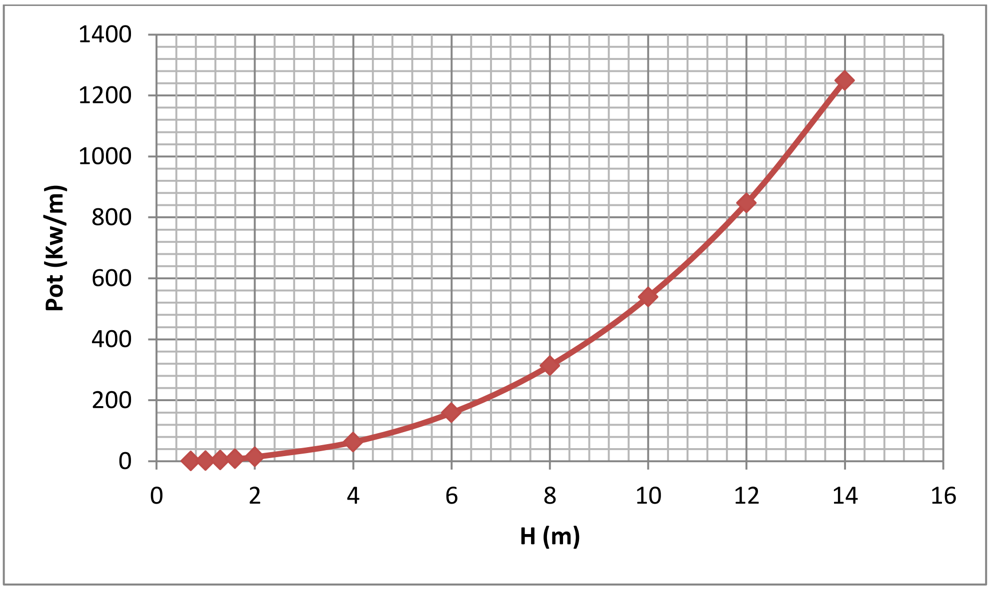


Figure 2.3. Global distribution of wave Power on North Sea

The plot in Figure 2.3 shows the global distribution of Wave Power (Kw/m) in North Sea for harmonic waves, as function of period (T) and height (H). On this graph we can see the most extreme power including the levels of lower power. But the most important data, which is the one, being discussed in this report, it's the levels of average of Wave Power (around 291Kw/m) and for regular wave (around 63Kw/m) with 4 meters height and 8 seconds period, the developed solution in this report, is to build the global system smart enough so that it will support extreme conditions on sea, as well at the same time limit the optimal power production (output power) of the generators because the output from the generator will change with the frequency speed of the point absorber. The absorber has huge fluctuations of power and power peaks that will rise levels couples times bigger than the average power production (around 250kw by generator). Also we can classify the power on the sea by Energy Flux or Density.

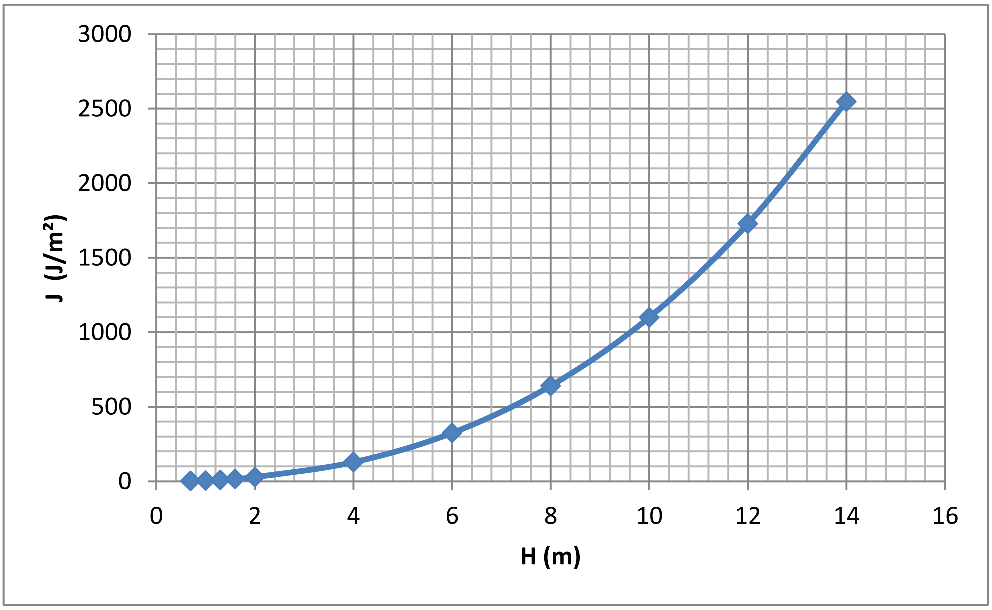


Figure 2.4. Energy Flux on North Sea

As we can see on the graph in Fig. 2.4, the average Energy Flux (J) transported in waves is around 593 J/m².It is also possible to assume that this represents the Energy Density (E).

By another side, to get a more deeply understanding about scatter diagrams (Appendix A), as well as their usefulness on energy captured from waves, we must take a view on next graph, extracted;

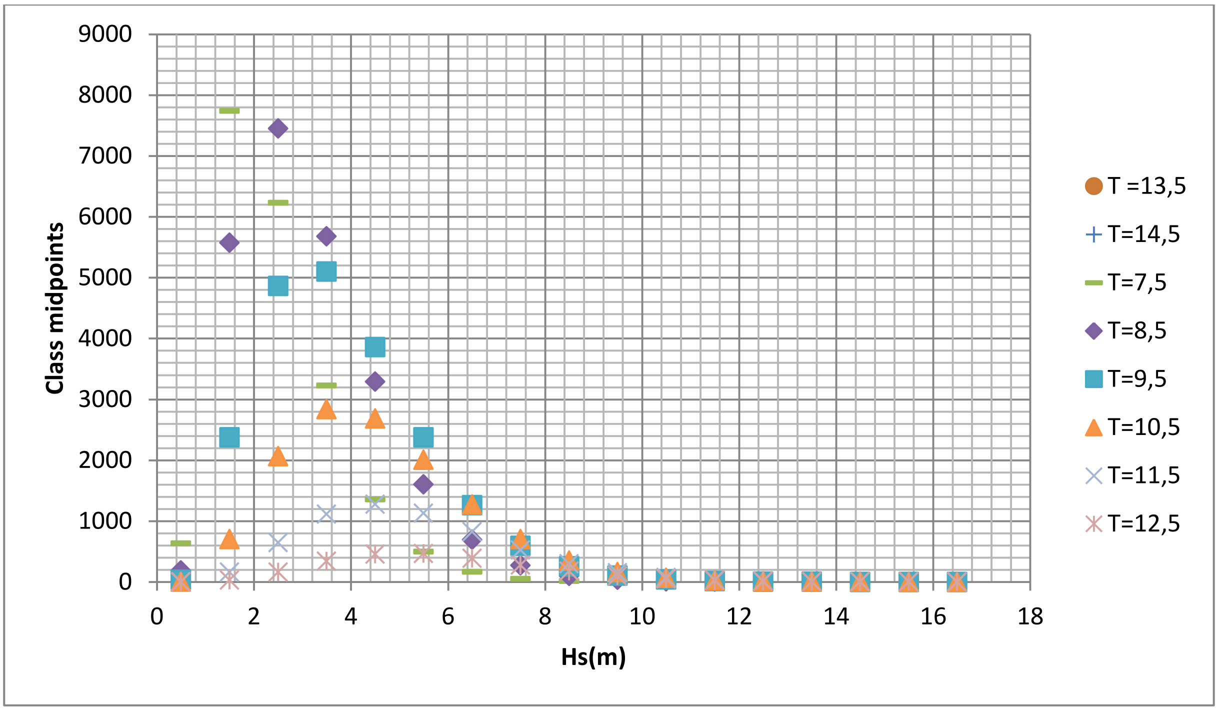


Figure 2.5.Curves of Scatter Diagram.

We can see clearly, that the maximum concentration of class midpoints is situated, for periods between 7,5sec to 10,5sec. Also if we make a comparison with the power captured (Kw/m) in function of all periods, see graph below;

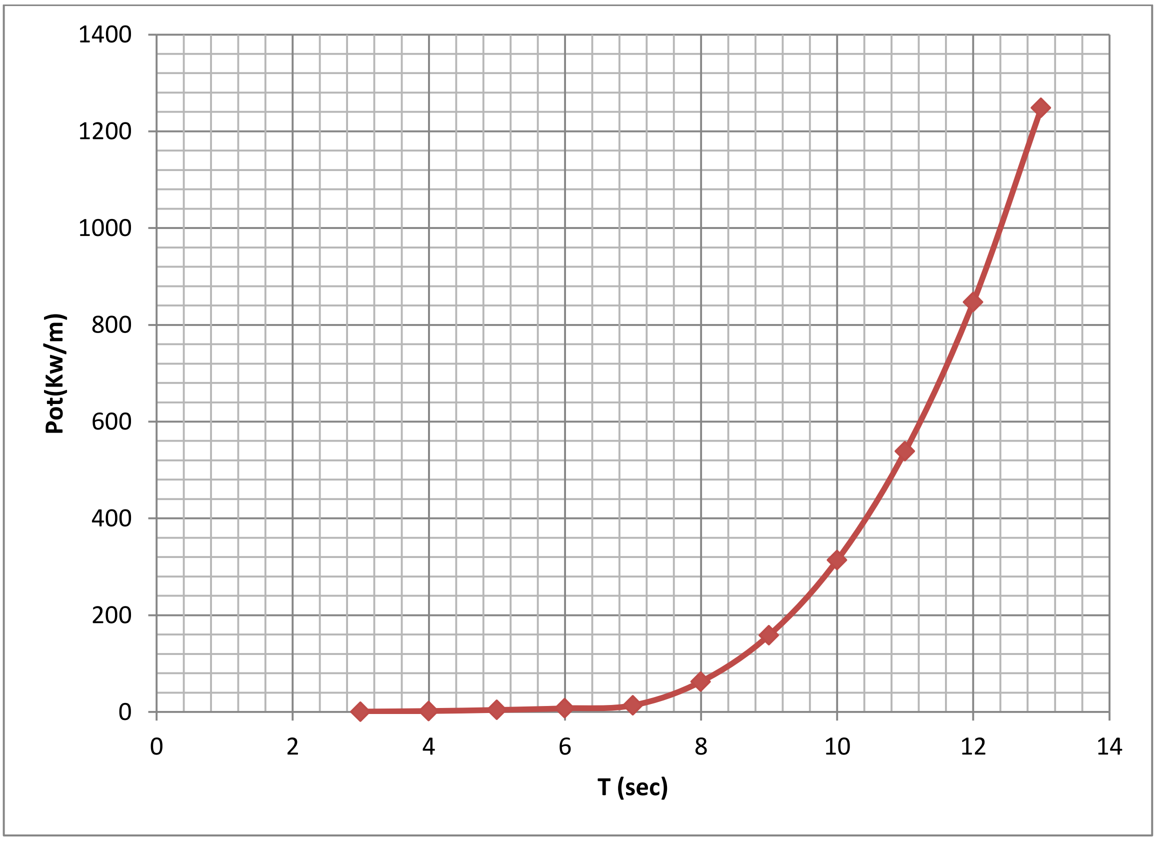


Figure 2.6.Power waves vs. Periods on North Sea

From my point of view, the optimal potential energy captured from the waves, will be after a value of 7,5sec to up, after this values, we can observe sharp increase of Power Wave Energy.

According to this last data and these short conclusions, the aim of this study is to be able to make this kind of devices for WEC, to take off the maximum amount of energy from the waves.

2.3 Selection of type Absorber model (Floater).

From point of view of getting optimal control and design with the selected floater, from commercial industry, although to get optimal results from a hydrodynamic analysis for this specific research, was necessary to select the most appropriate floater. The criteria to select a correct floater for this application must satisfy;

- **Optimal Control** between the floater and motion of waves.
- **Maximum power absorption** based upon hydrodynamic considerations alone corresponds to optimal control, as well as it determines both the amplitude and phase of the proper motion. Also the maximum power absorption are determined by **wave breaking limit** between the diameter(D) of the floater selected and the parameters (H, λ) from motion waves.
- The appropriate **Buoyancy** and **Width/Diameter** capacity to absorber the mean power intercepted close to surface waves.
- Efficiency **Drag Area**, to satisfice a correct pressure up-down under the floater, as a result to take on the maximum power absorption.

2.3.1 Axisymetric Floater

For a general rigid floater motion is composed of three rotational modes of motion (roll, pitch and yaw) in the directions of and about the (x,y,z) coordinate axes, and three rotational modes of motion (roll, pitch and yaw). From commercial industry, I had selected two initial models of floater, as we can see below (Figures 2.5)

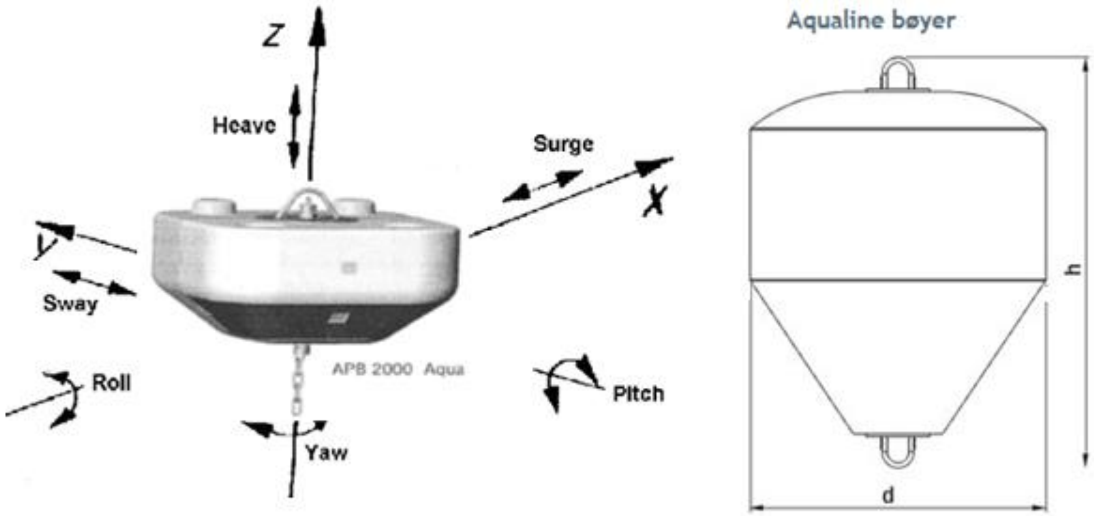


Figure 2.7. Buoy models [15].

At the initial analysis, it is necessary to have clear understanding of the difference between both Floaters in terms of the wave breaking limit of the hydrodynamic forces over these floaters. This can be observed from the graphs in Fig. 2.6 and 2.7

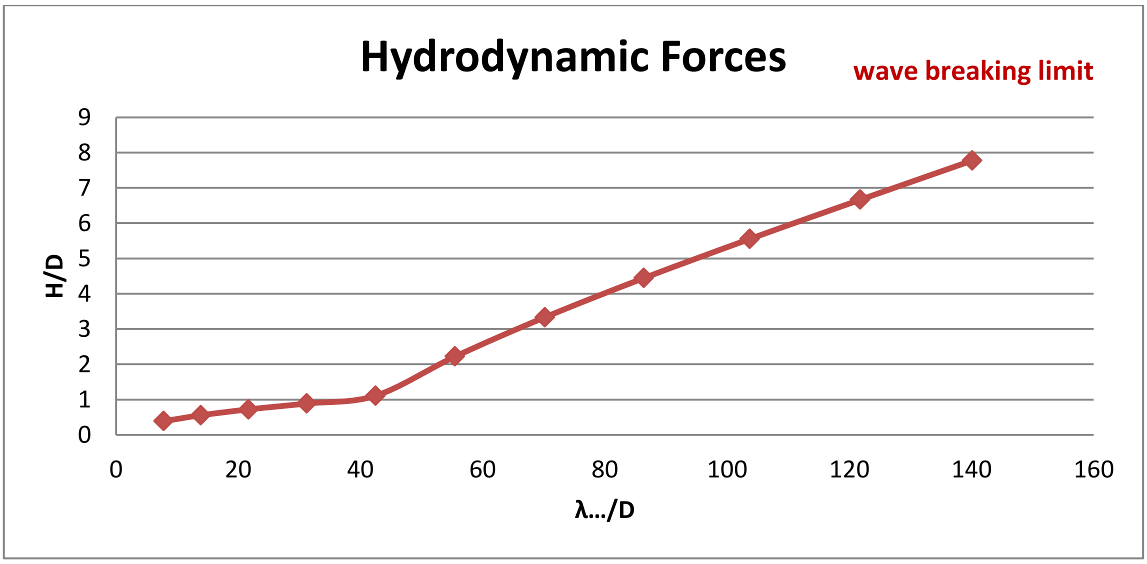


Figure 2.8. Wave breaking limit of "2000 Aqua buoy".

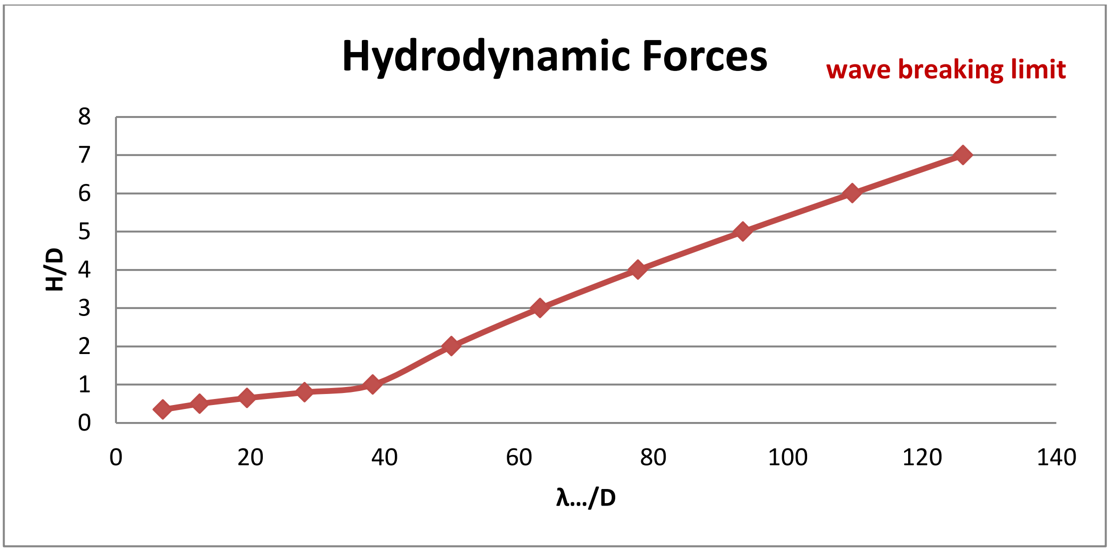


Figure 2.9. Wave breaking limit of "Aqualine buoy".

The result of both graphs, are that with "2000 Aqua buoy" we have more high (H/D) than the second one, which means, the first one provides more energy. Also the buoyancy and drag area on the first one are more efficient than the second one.

To reach a main conclusion between both floaters, the type of floater chosen, is the "APB 2000 Aqua" buoy model, I must make clear that this model is the most appropriate for WEC devices. According to all declared before, I will run both WEC devices with APB models.

For the first the device ("Cape Verde") only will be necessary one floater. In contrast, the second device ("Flexible Drive Line"), I will carry out another completely different design and assembly way, to eject the whole project. This second WEC must have two structures (up and down) to join in totally of five buoys which ones will maintain the proper buoyancy and stability of the whole system. We can observe in the next figure, bellow;

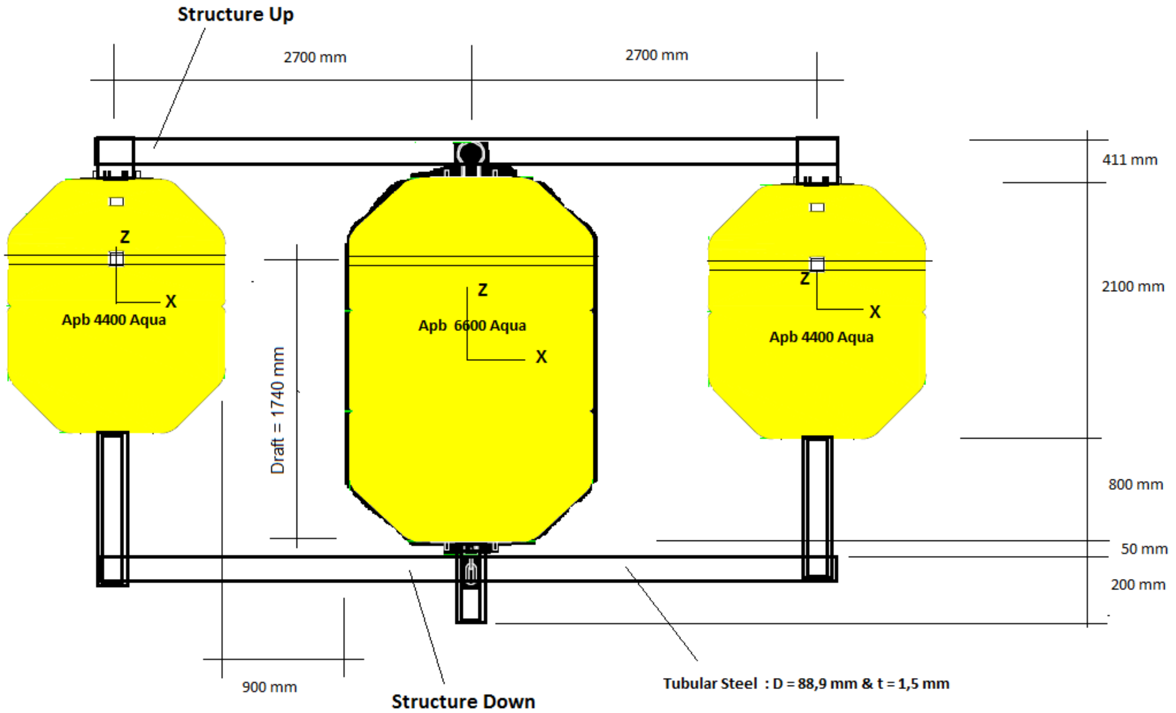


Figure 3.0. Frontal view of Buoys and structure assembled.

Our wave breaking limit, will be the same as "2000 Aqua buoy", due to the diameter in all buoys, will remain constant (1,8m). Hence, the only differences a long this section, will be type of floater selected for each device, due to the functionality of each WEC and objectives, are different. We could evaluate it, in the next Chapters.

2.4 Analysis of type Absorber

2.4.1 Main Parameters and Dimensions

Part of the values and data as such the dimensions described below are from catalogues. A other parameters such as drag area or even draft, were calculate by hand, while the rest of the parameters, are taken from Orcaflex program which has been used to carry out simulations and interactions. These simulations were done with input by one direction of motion waves that means 180^0 over plan XZ, for wind and currents and of course with all data from waves of North Sea. As we can observe, on next table, the most notable difference between values is **draft** value and **buoyancy**, which one varies depending of Wave Energy Converter, also these parameters must be consider as key values for each device.

Table 2.1. Parameters and Dimensions.

Table 1

Axi-symmetric Floater

Parameter	Value			S.U.
WEC	Cape Verde	Flexible Drive Line		
Type Model	APB 2000 Aqua	Apb 4400 aqua	Apb 6600 aqua	
Diameter/Length	1,8	1,8	1,8	m
Height	1,6	2,1	2,9	m
Volume	2,2	4,4	6,6	m^3
Mass	350	590	890	kg
Draft	1,18	0,94	1,74	m
Drag Area	0,5904	0,5904	0,5904	m^2
Buoyancy	2060	4000	6000	kg
Centre of gravity	(0,0,0)	(0,0,0)	(0,0,0)	m
Moment of inertia(Ixx)	232166,6667	482325	1024241,667	t/m^2
Moment of inertia(Iyy)	232166,6667	482325	1024241,667	t/m^2
Moment of inertia(Izz)	315000	531000	801000	t/m^2

2.4.2 Hydrostatic Stiffness on Floaters

The Hydrostatic Stiffness matrix (K) is only specified for roll, pitch and heave directions, it is applied in dynamic only if the Floater “primer motion” is set to calculated six degrees of freedom.

According to theoretical and practical method as described earlier, are using this program (Orcaflex) to calculate the Hydrostatic Stiffness at floater, the results obtained are showed below:**[Fheave, Mroll, Mpitch] = K [Oheave, Oroll, Opitch]**

These last components are;

Force (KN), Moment (KN·m) and Rotation (Radians)

$$
\mathbf{K} = \begin{array}{ccc} \text{Roll} & \text{Pitch} & \text{Heave} \\ k_{11} & k_{12} & k_{13} \\ k_{21} & k_{22} & k_{23} \\ k_{13} & k_{23} & k_{33} \end{array}
$$

$$
\mathbf{K}\ \text{(kN/rad)} = \begin{array}{ccc} 0 & 29e^3 & 14e^3 \\ 270e^3 & 0 & 0 \\ 0 & 10e^6 & 0 \end{array}
$$

As well we can see in this matrix, the hydrostatic stiffness components for the surge, sway, and yaw directions are all zero, but not for roll, pitch and heave. All the values zero, means that some components zero.

2.5 Chapter Summary

The methodology development along this chapter was divided into four concepts (classification, sizing, selection and analysis). All I wanted, to express with these last concepts, is how this chapter has addressed in the way to get the most optimal floater from commercial industry, as result take off the maximum amount of energy from the sinusoidal motion from the waves.

To reach a conclusion, the optimal movement of the floater on surface waves has directly optimal captured efficiency of the output power production into generator.

CHAPTER 3 DIMENSIONING AND ANALYSIS OF LIFTING SYSTEMS ON WEC

3.1 Introduction

As expressed chapters one and two, this devices ("Cape Verde" and "Flexible Drive Line"), must be designed for extracting the maximum amount of energy required, although the whole lifting should be designed in such a way that it can carry the resonant of the floater.

The relative movement between the floater and waves must be depending directly from the global lifting and main supposition of one direction motion waves with 180º.

The lifting process, which is going to be discussed through the dimensioning process, was initially viewed as mooring lines design for WEC. This is because the functional aspects of mooring system will have a dynamic response to the waves, but the application of this dynamic wire cable is another, due to the fact that it must satisfy the whole movement of each device. For instance, taking the idea of a normal mooring system, the nominal diameter upper is more than 30mm, because the functionality is absolutely another, as we well know, it must resist high levels of strength in tension, that's why the section of cable must be thick.

For better understanding of these two WEC devices, it is important to clarify and distinguish, the two main concepts of Global lifting Systems undersea for each device, given that "Cape Verde" require different way to lift than "Flexible Drive Line".

For instance, the first device only demand a simple task to take off the wire cable from sea though hydraulic lifting and pulling device from onshore. While, the second one, will carry out absolute on offshore.

Also, the whole hydrodynamic load caused by the currents will be constraint for the second device, not so much at first one, due to the important will be height of waves.

3.2 Global Lifting System undersea

3.2.1 Lifting of Cape Verde Device

The lifting process along whole system of this device was rather clear showed on Chapter 1(figure 1.7.), the main **specifications** must have the wire cable selected, are described below:

-High flexibility.

-Strong against abrasion.

-High breaking load.

-Rotation and resistance.

The main reasons why the type of cable should have these last specifications are related mainly with a dynamic and resistible response along with the whole cable and post sizing of snatch block on sea bottom, the one that will be described in the next sections, so such the environmental conditions of the sea.

With respect to the whole static and dynamic analysis, the iterative calculation process is divided into three main blocks;

1º. Tension leg.

2º. Catenary.

3º. Snatch block and Anchor.

All these blocks are connected together through other lifting items (such as wire slings, link offshore...etc) which ones helped to connect the whole cable from surface of waves to machinery on land, it will explained on section 3.2.1.1.

Firstly it is assumed that the tension leg from the surface to bottom is close to the vertical component (yaw axle), but the reality is completely different. So there exist so many degrees of freedom along the three components. Therefore the results obtained from the vertical movement are increased by at least 200% more. This also protects the whole system from environmental extreme situations that are known as safety coefficient from engineering perspective.

The losses around pulley (snatch block) are also accounted for and assumed to be 20%.

The method used to get all analytical calculations a long whole cable, could be named as “quasi dynamic and static method”. This quasi dynamic and static method, implies both situation at the same time (static and dynamic), both should carry out analyses for each situation.

All calculations are carried out according to Static Equilibrium, firstly directly over snatch block, and then directly transmitted over the anchor over seabed soil. Such as, follow the recommendations from DNV-RP/OS-E301 and E304 by Det Norske Veritas, for this kind of anchor system, especially for general conditions of marine soil. For more details, the main dimensions of the Anchor and Snatch block are given in Appendix C.

And the last one (catenary), calculations on both static and dynamic analysis are done assuming a slope of around θ=66º, with regard to the initial dimensions as depicted in Figure 1.10.

The global design fatigue life for the stainless steel and other structural parts was assumed with a specific service life of minimum 20 years.

In this section, the main components that constitute all the whole system undersea are described. This is also shown in the flowchart depicted in next Fig. below;

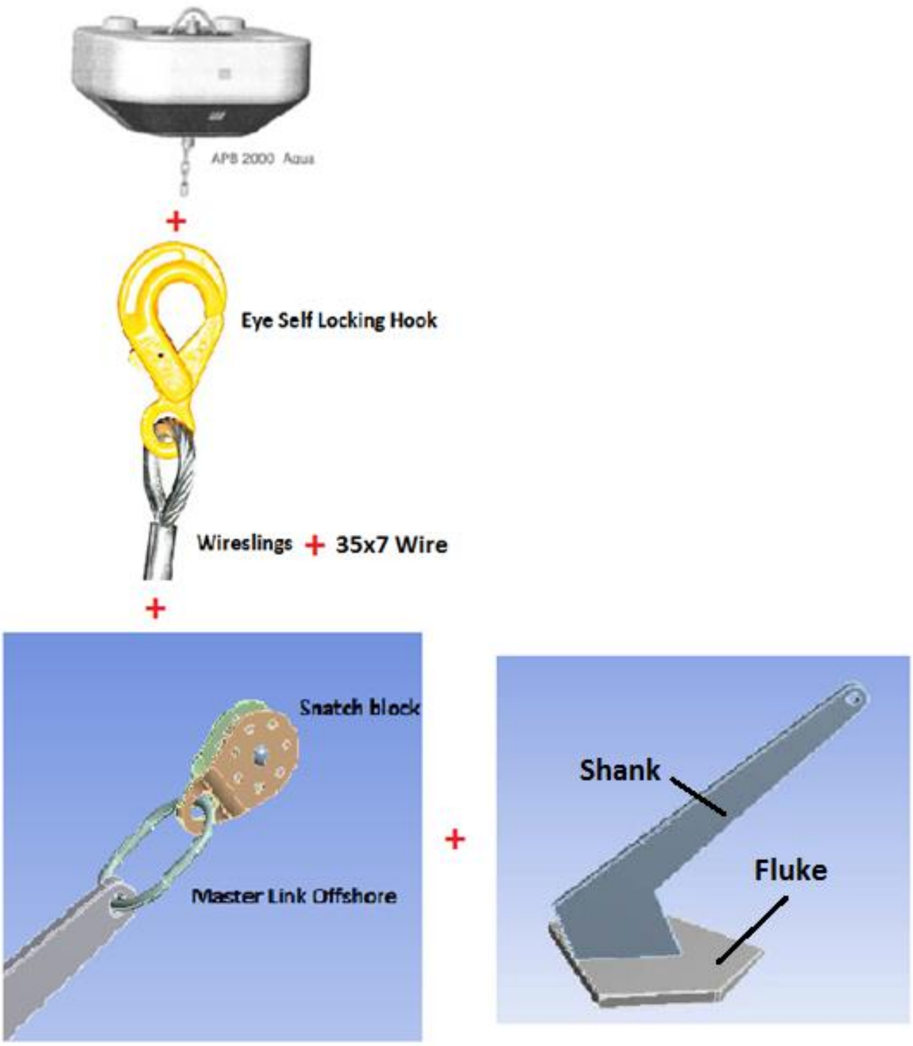


Figure 3.1. Flow chart of the global system undersea.

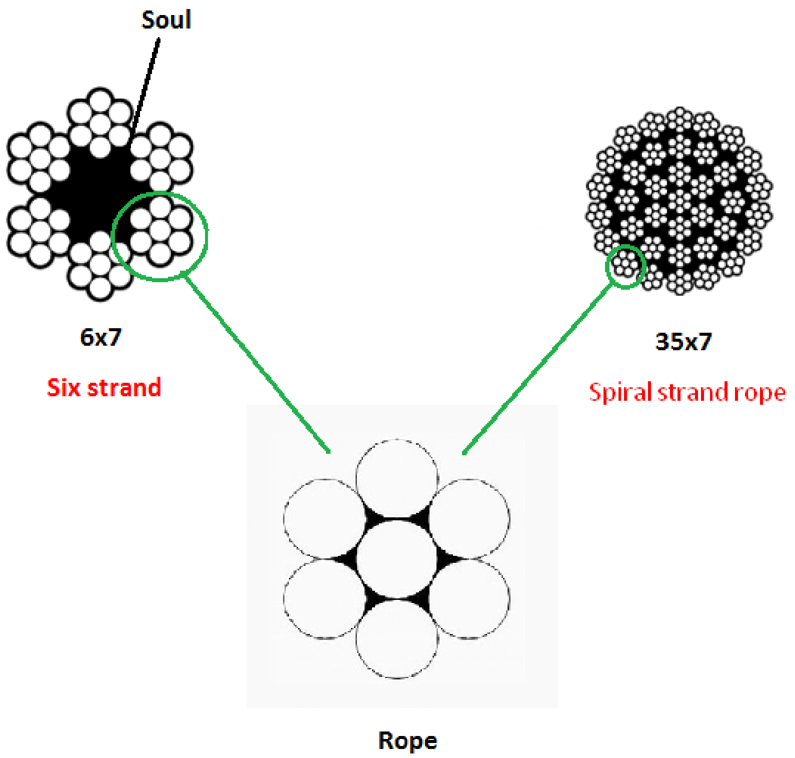
Starting with the first component (Eye Self Locking Hook), as well shown in the flow chart, the component is made from stainless steel and has a maximum capacity around 21ton. All of the mechanical properties and dimensions of the component are obtained directly from manufacturer's catalogue.

The second main component is a wire slings component. This component is well used for almost all applications to keep relation with lifting components, and it has two main small parts (thimbles and sling), both depend on sizes of fitting wire ropes from the nominal diameter of wire cable.

3.2.1.1. Selection of suitable wire cable

To develop a correct method of selection of the wire cable, step by step, firstly, we must respect the guidelines provided by DNV-OS-304 by Det Norske Verites, and follow this whole recommendations from a practical perspective. In this research, the whole wire cable will be exposed to two main physical forces: tension and fatigue. Thus, the **tensile strength** and **fatigue life**, especially around snatch block (pulley), are studied.

According to the mentioned physical forces, the study in this research can focus on and specify more throughout the whole sizing process. This involves mainly classification of wire cables as types of **section** and **winding** (*Regular* and *Lang* lay). The type of section, as well described in DNV-OS-304, there are four huge gropes of sections: six strands, spiral strand rope, half locked coil and full locked coil. The two main ones are focused on in the selection for this study: **six strand** and **spiral strand ropes**, as well as are the most common types from manufacturer's catalogues. These are shown in the following simple representation (Fig. 3.2).



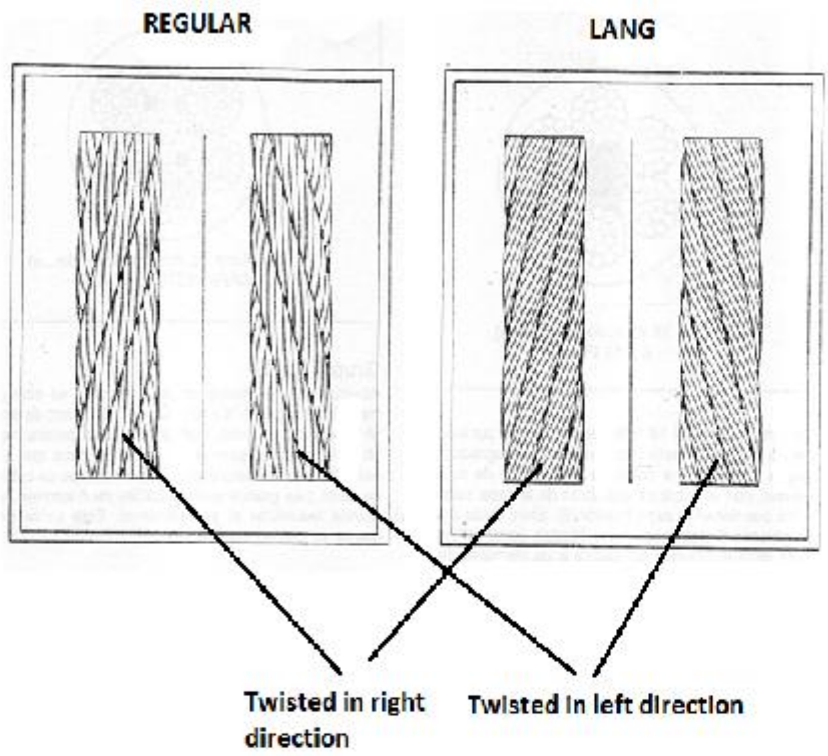


Figure 3.2. Straight section and Winding representation [16]

As the name implies, the wire rope can be described as the part that integrates the main parts. It is a rope that integrates wires and also with soul in the middle. The meaning of the nomenclature of wires ropes has two main parts, the first number indicates that the ropes are integrated wires, and the second number is the number of small wires. Even it could make a third classification, with respect to the type of tension that is being transferred (traction or flexion). This last parameter is the key to take a correct selection of wire rope, depending directly on the functionality to be used. Wire cables have a high capacity of traction that are for instance the model with less number of wires (6x**7**), is most appropriate for mooring systems. Wire cables could satisfy high levels of flexion (6x**37**), this last ones are useful for pulleys.

These three main classifications are clearly shown in the next graph (Figure 3.3).

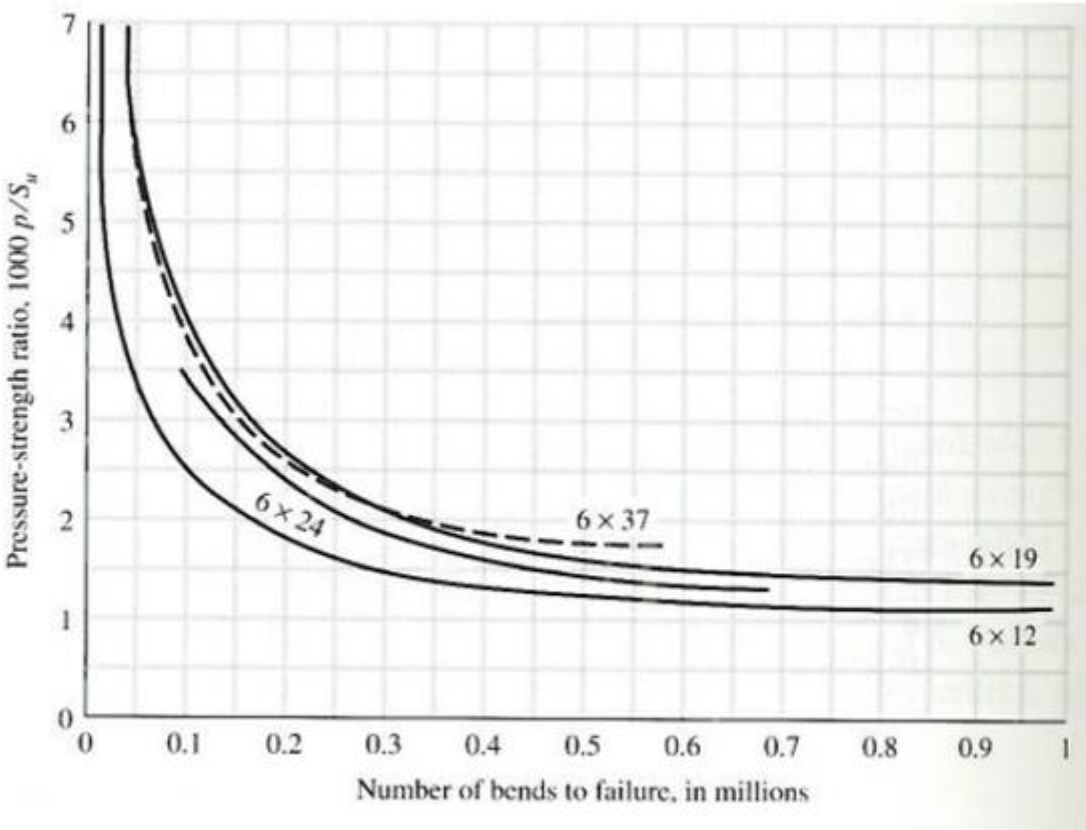


Figure 3.3. Experimentally fatigue life and sheave pressure [16].

The graphs in the above figure represent the experimentally obtained relation between the sheave pressure (over section) and the fatigue life of wire rope, for the most common types of wire cables.

As can be observed in the graphs, the model 6x37 has short life compared with 6x19, because as mentioned before, wire cables with more numbers of wires allow higher range of flexion.

In any cases, the wire cable that is more appropriate for our project could be **6x37** model, but finally, it has been decided to choose **35x7** model, because the technical characteristics are the most closer to the initial specifications, described in Section 3.1. , that means, 35x7 has high flexibility than 6x37.

According to this last conclusion, we need to focus on **spiral strand** wire cables, from manufacture's catalogues, as can be observed from the graph in the next figure (Fig. 3.4).

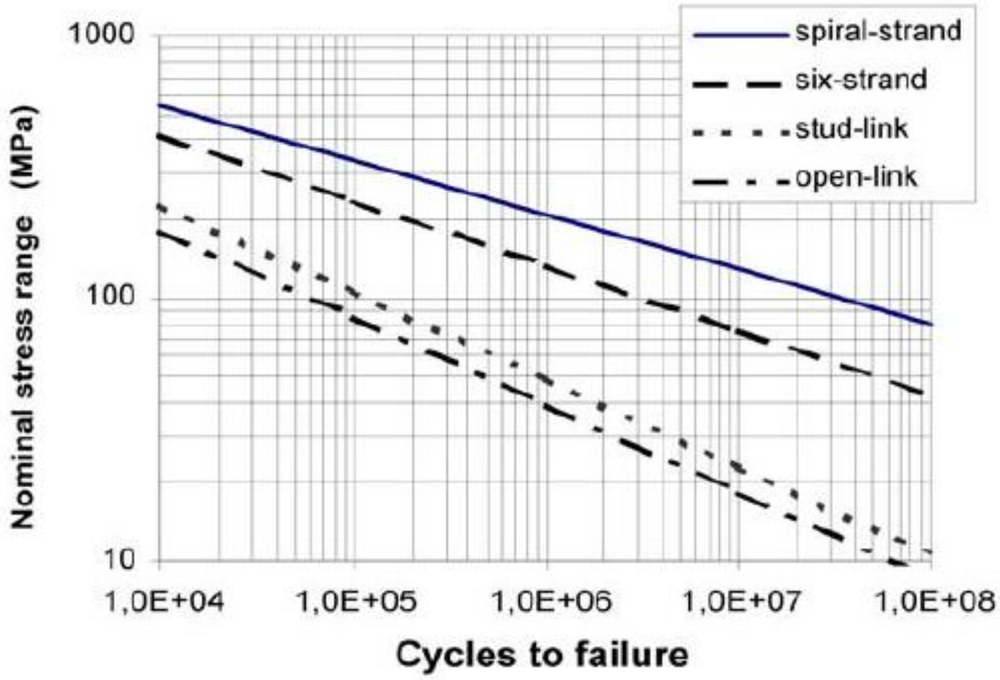


Figure 3.4. Design S-N Curves [17].

As documented in the DNV standard (DNV-OS-E301, section 2.203), the S-N curves for steel wire ropes assume that the rope is protected from the corrosion effect of sea water.

3.2.1.2. Determination of nominal diameter for 35x7 Wire.

To select the most suitable wire cable, study of all principal tensions concentrated over the section of wire cable along three principal axes, including **normal/nominal (σ)** and **shear stress(Su,d) should be done**. But firstly, it should be clear that our selection must be lower than minimum breaking load or allowable tension, the one that the cable is subjected to, that means, the first thing one needs to determine will be finding the maximum tensile force in the wire cable. And secondly, the normal and shear stresses should be checked as they are lower than the permitted load for the wire cable.

- **Maximum tensile load on whole wire cable:**

To calculate the maximum traction tension in the cable, calculation methods using the quasi dynamic and static method are developed, where both are determined for differential unit of one meter, and then extrapolated to the whole cable.

A long this wire cable, we have three main parts:

Tension leg (or vertical cable), **snatch block** and **catenary** (with slope of 66^0). According to this distribution, I have ejected all analytical calculations. As it is well known, the stress is directly transmitted through the whole cable. This can be observed from the corresponding representation on physical forces for both cases (static and dynamic);

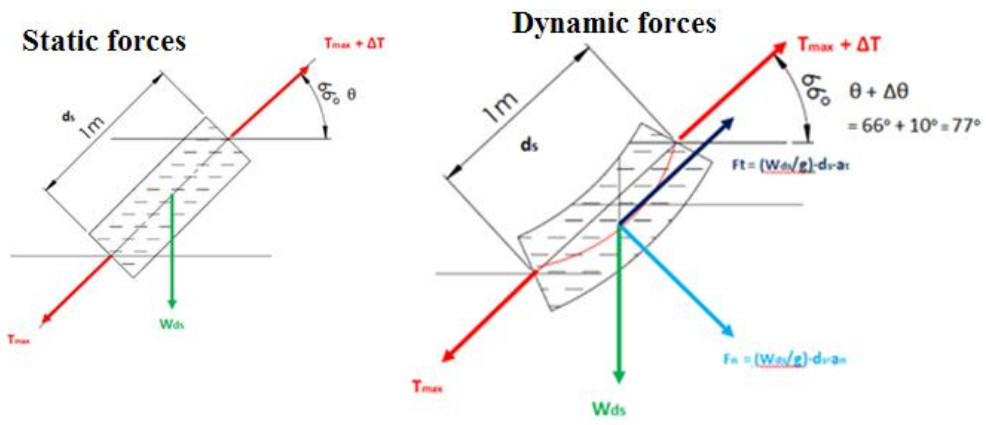


Figure 3.5. Static and Dynamic forces.

Also I have assumed a few more details, especially from the Dynamic Forces. I have considered that the normal acceleration must be cero, because the radio is really huge, therefore the normal acceleration is value insignificant, in addiction, was fixed an incremental value (Δθ) of ten degrees more or less a long whole slope, for another details (see Table 3).

Calculations for one unit of metro (ds =1 m):

-Static Forces;

$\Sigma Fh = 0$;

[T+ΔT]·cos(66+10) - T·cos(66) = 0

$\Sigma Fv = 0$;

[T+ΔT]·sin(66+10) - T·sin(66) - Wds = 0

ΔTs = 4,12 New

Rs = 10 New

Ts = 5,92 New

-Dynamic Forces;

$\Sigma Fh = 0$;

[T+ΔT]·cos(66+10) - T·cos(66) + Ft·cos(66) = 0

$\Sigma Fv = 0$;

[T+ΔT]·sin(66+10) - T·sin(66) + Wds - Fn·cos(66) = 0

ΔTd = 3,30 New

Rd = 18,83 New

Td = 15,53 New

RT =10 + 18,83 = 28,83 New

The main results which ones, have gotten, are described on the next table;

Table 3.1. Max Tension

	Value	**S.U.**
Tension Leg (Tv)	19877,51	N
Catenary (Tc) & slope (θ=66⁰)	8537,47	N
Losses on Snatch block ↓	20	%
Safety coeff.↑↑	200	%
Max. Tension (Tmax) →	45463,97	N
Max. Tension (Tmax) →	**4,634**	**ton**

We can assume that the real value is around five tones, which is only an indicative value and it should be checked after the value of the maximum tension of our wire cable (35x7) is determined with some lower nominal diameters as shown below.

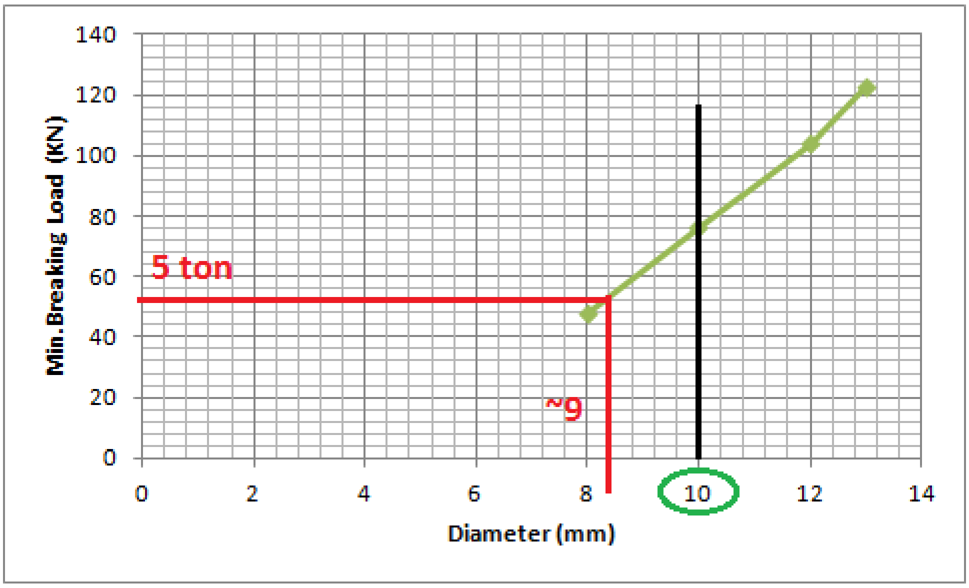


Figure 3.6. Min Breaking Load vs. diameter.

All the values of minimum breaking load, was calculated with respect to a tensile break between (1960 to 2160 N/mm²), specification from catalogue[2].

[2] [14] http://www.cosaltoffshore.com/

As can be observed from the above graph, it has been decided to choose a cable size even higher than what the theoretical calculations given therefore our wire cable that can carry the next steps of sizing will be **35x7** with **10mm** (3/8").

3.2.1.3. Stress Analysis of Wire cable and Snatch block friction.

Taking our last value of traction tension (5ton≈11027lb), this one is the tensile force rope (Tmax). The analysis is carried in two ways;

1º. Nominal **(σ)** and Shear stress **(Su,d)** of the wire cable.

2º. Wire cable and Snatch block;

- Bearing pressure **(p)**.

- Ultimate Strength **(Su)** of wire.

We need to use the formula of breaking load **(Smbs)** from DNV-OS-E304 Ch.2.Sec7 and also tensile break (f), given as.

$$Smbs\ (KN) = f \cdot t \cdot K1$$

The total number of wires (t) is seven and over the lay factor, it is assumed 0,78. Also with the value obtained, we can calculate the direct simple shear stress **(Su,d)**, and this, with the corresponding nominal tension **(σ)** is represented in the figure below (Fig. 3.7);

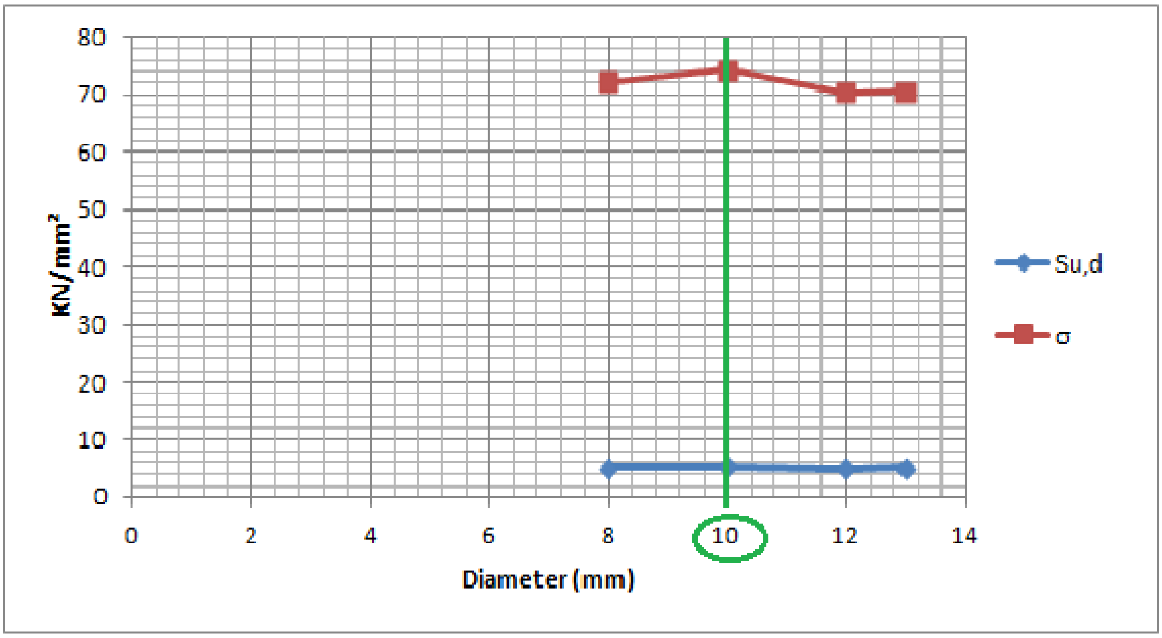


Figure 3.7. Nominal tension & Shear stress vs. Diameters.

As can be seen in the above graph, the corresponding obtained values (5,30 and 74,28KN/mm²) are the maximum values that could be admitted for our wire cable. Coming back to maximum tensile tension, we can obtain the normal and shear stress, we find the corresponding values to be (0,58 and 5,78KN)

The second approach of the calculation is done according to Figure 3.3, as show below;

Data:

Tmax = 5 ton =5000kg =11027lb

dn = 10mm

D(sheave pitch diameter) = 276mm (see Appendix 3)

This must be checked if this sheave, compared with the limitations from (DNV-OS-304.Ch.2.Sec.4) for general design of fairleads;

$$\frac{\mathrm{D}}{\mathrm{dn}} \geq 16$$

Clearly our sheave, satisfy this limitation.

- **Bearing pressure** (p): $p = \frac{2 \cdot \mathrm{Tmax}}{\mathrm{d} \cdot \mathrm{D}}$ (kpsi)
- **Ultimate Strength** (Su): $\mathrm{Su} = \frac{2000 \cdot \mathrm{F}}{\mathrm{d} \cdot \mathrm{D}}$ (kpsi)

The calculations obtained from these last two formulas are;

p = 800Kpsi; Su = 800,510Kpsi

As well as, the corresponding allowable fatigue between wire cable dimensioning and pulley selected, are calculated below;

$$Ff\ \ Kpsi \cdot mm^2 = \frac{\frac{p}{Su} * Su * d * D}{2} = 110400$$

With both values and referring to the graph on Figure 3.3., we can conclude how the relation of pressure/strength ratio for the cable 35x7 could be, but unfortunately we don't have the experimental curve of this cable from the manufacturer. The main objective is to prove, if this wire cable is completely protected from fatigue life.

Hence, upon arriving at the results of the above calculations, even it is possible to dimension the wire cable, taking account of appropriate calculations with axial and bending stiffness for each cable. Some more properties are shown on next table.

Table 3.2. Main Parameters and Values.

Wire rope: spiral strand rope	35x7	
Parameter	**Value**	**S.U.**
Nominal Diameter (Øn)	10	mm
Total length (undersea)	300	m
Weight by length (in water)	0,3915	Kg/m
Breaking load (Smbs)	416,717	KN
Shear strength max (Su,d)	5,305	KN/mm²
Normal Stress(σ)	74,281	KN/mm²
Young Modules	82,737	KN/mm²
Axial Stiffness (Ka)	3051,694	KN
Bending Stiffness (Kb)	98776,831	N/mm²
Seabed inclination	0	cte
Water depth	80	m
Speed in transver. direction (on bottom)	0,061	m/s
Drag Force	0,019	N/m
Inertia Force	0,019	N/m

3.2.1.4. Design and Analyses of Snatch Block.

This section represents the assemble model of the snatch block. The block is made of stainless steel and is composed of the following main components: axle, pulley and cover plates (Fig. 3.8).

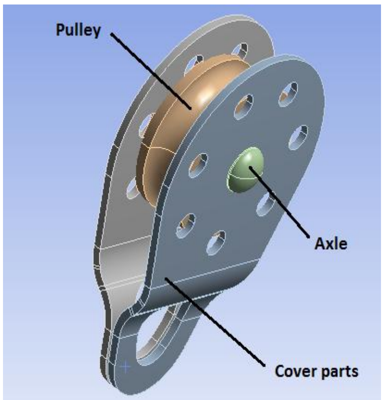


Figure 3.8. Snatch Block model

This component can be consider as key component at the global system level, because they establish the link between wire cable and anchor, as well as link offshore. In addition, the link situation undersea is about 80m deep from the anchor point.

Since the whole load and fatigue on all components is quite high, as it was calculated before, I decided to carry out suitable stress analysis between the pulley and axle in this calculation, and it has been found that the moment generated from wire cable around the pulley is around 6770 N·m with a nominal radius of 138mm.

On the other side, taking into account the fact that the element is quite small, I have considered zero drag and lift force.

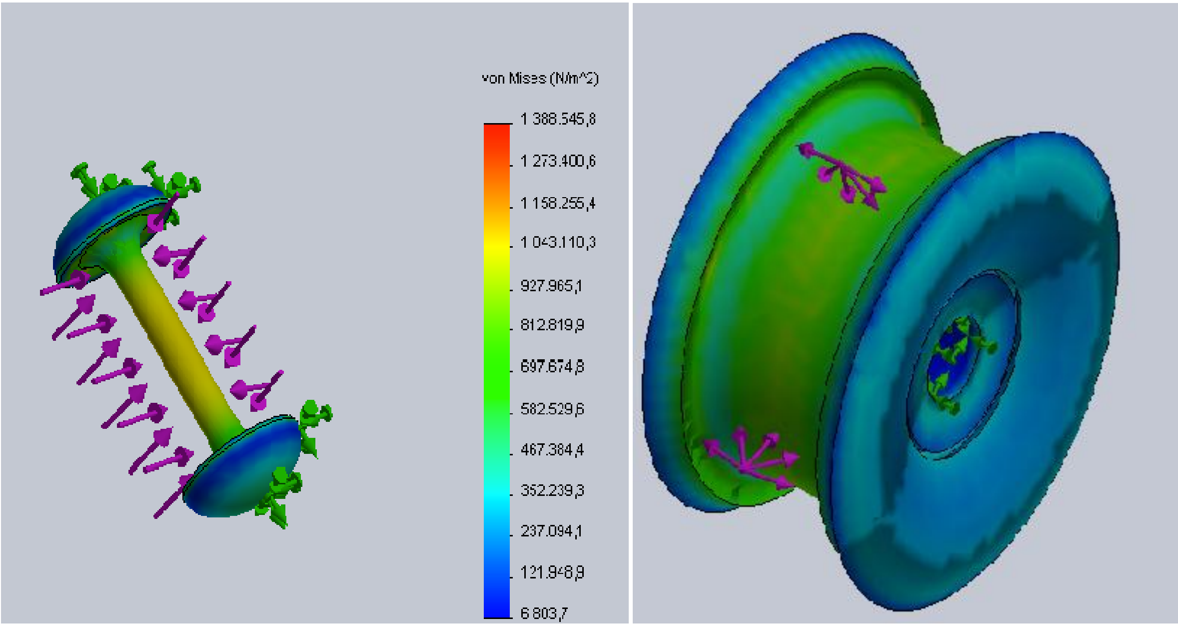


Figure 3.9. Stress analysis on axle and pulley.

Figure 3.8 shows the von Mises stress plot of the axle and the pulley. As shown, the maximum von Mises stress value is 13,25MPa, and if we take a look at the table of properties of this component (Table 3.3), we can conclude that the material selected is quite acceptable.

Table 3.3. Properties and Parameters of Snatch Block.

Parameter	Value	Unit
Stainless Steel	Sy = 207 ; Sc= 207 ; Su= 586	MPa
Weight (in water) = Wa'	46,11	Kg
Volume	0,006785	m^3

Centre of Gravity pulley	x=2,0136e-011;
	y=2,0926e-011;
	z=-5,25e-002
		m

Moment of inertial(Ixx)	6,84E-02	$Kg{\cdot}m^2$
Moment of inertial(Iyy)	6,84E+02	$Kg{\cdot}m^2$
Moment of inertial(Izz)	0,10158	$Kg{\cdot}m^2$

3.2.1.5. Ideal Model and Analyses of Plate Anchor.

The main function of the plate anchors is to give support against the high vertical loads from the anchor line.

From anchor manufacture companies we have different kinds of models, for instance, SEPLA, PADER, ANCHOR LINE…etc, but all of them have different areas of application.

The intended application of these kinds of anchors is for mooring products in aquaculture industry or mooring lines for oil industry, on the other hand our application will focus on Anchor line or known as Plate anchors.

We can make a shortly explanation between different of SEPLA and PADER anchor, if we take a look to next figure, showed below;

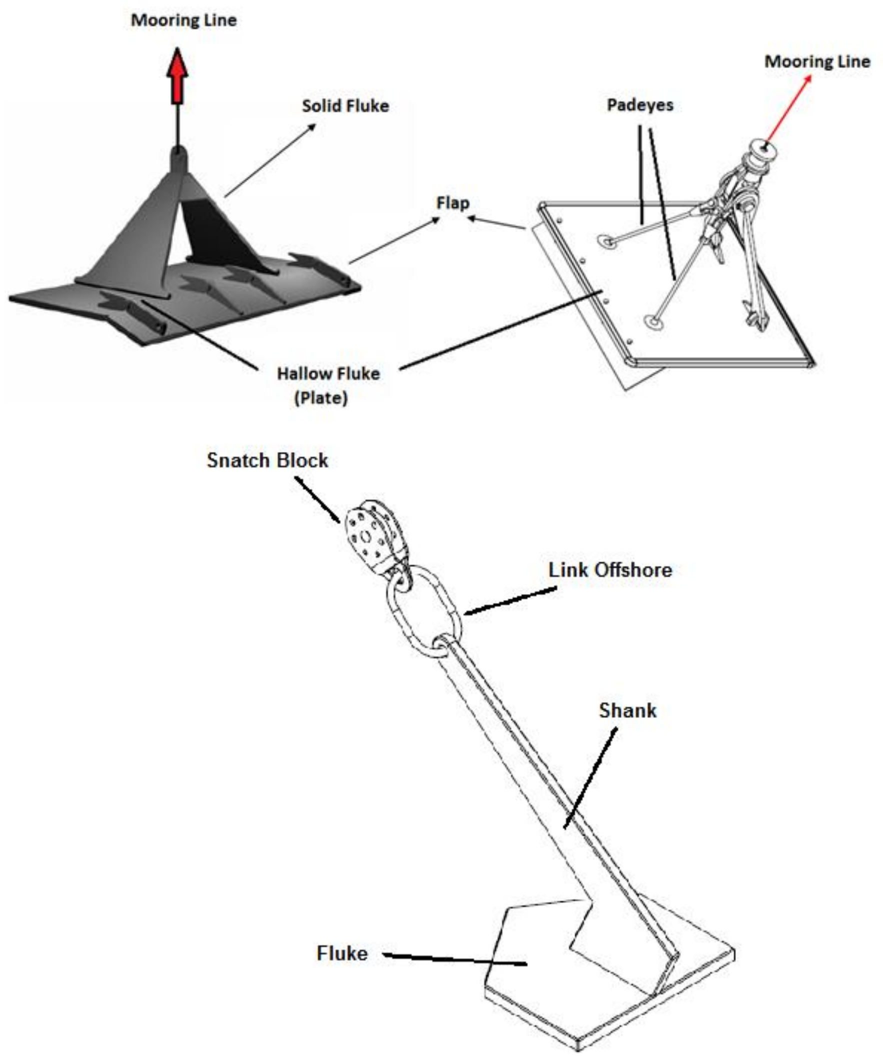


Figure 3.10. SEPLA, PADER and ANCHOR LINE models [17].

SEPLA (Suction Embedded Plate Anchor):

- Vertical Orientation.
- Two bodies welding (Solid and Hallow).
- Two types;
  - ·Small sizes (temporal installation)
  - ·Biggest sizes (permanent installation).
- Big Flap area.

PADER:

- Two bodies (mooring connection and plate) connected by Padeyes.
- Triangular Stiff lifting bridle.
- Small Flap area.

ANCHOR LINE:

- Two bodies welding (Shank and Fluke)
- Triangular Plate
- Mayor penetration on soil seabed.

The main differences or advantages between these last model of anchors, are that the first one (SEPLA) is absolutely welded and has more plate area, that means more flap, but the second one, although triangular stiff lifting bridle allows for a controlled gyration of the plate, as well as avoid rotation on the whole plate, in contrast, has less flap area in compare with SEPLA model. Has more flap area involve to resist highest mooring loads, due to is transferred directly the soil seabed, partly by the whole plate and mainly by the flap. I will prove in next couple sections, how SEPLA model will be suitable for Flexible Drive Line model.

Anyway, any anchor model from manufacturer industry must be satisfying the application for which one will be used.

By another side, to design of plate anchors, one should provide information from three main fields: Soil parameters, Geotechnical data and Operation field. All the above-mentioned main fields should be based on dependable information concerning specific location and soil properties.

It is hypothetically assumed, as shown in Fig. 1.10 & 1.12 in Chap. 1, that the foundation is with soft clay and sand, with constant level surface over soil marine (75m). These last concepts will be the base of all theoretical calculations carried out in this section.

The model of plate anchor, the one modeled was according to the way of slope and anchor line catenary tension, that’s why this model has this geometry, and also it will provide much better depth of penetration for the anchor over soil marine. This will help considerably to reduce the overall burden of resistance along the wire cable.

The two main parameters to start with the design work must be the depth of penetration of the plate (Zplate) and the plate width (Wplate), as well as the angle of penetration (β). These are the main parameters that satisfy the requirement that the anchor penetrates deeper where the soil normal and strength components on fluke tend to be higher and give rise to resistance over marine surface. This is due to the fact that it will have a good tensile resistance along the whole wire cable. The material selected was structural steel whose technical properties are given in the next table;

Table 3.4. Properties and Parameters of Anchor model.

Parameter	Value	Unit
Structural Steel	Sy = 250 ; Sc= 250 ; Su= 460	MPa
Weight (in water) = Wa'	1715,379	Kg
Volume	0,251167	m^3
Centre of Gravity	x=-0,67 ; y=0,36 ; z=0,08	m
Moment of inertial (Ixx)	242,76	$Kg \cdot m^2$
Moment of inertial (Iyy)	748,66	$Kg \cdot m^2$
Moment of inertial (Izz)	543,36	$Kg \cdot m^2$
Drag Force	2,892	N/m
Lift Force	2,313	N/m

On the other side, it must be mentioned that the installed penetration depth and its verification is not the goal of this research and this must be carried out based on acceptance criteria from installation contractor companies in situ.

The first verification criterion that we must clearly know is if our plate anchor can resist the whole resistant of interlock is showed below;

$$\mathrm{Zplate} \leq 4{,}5 \cdot \mathrm{Wplate}$$

If the corresponding values are substituted, the result will be about 67,5 times the restriction of penetration, so such an anchor resistance will be quite safe with the dimension inputs on the model design.

➢ Equilibrium Calculations of whole Anchor and Snatch block:

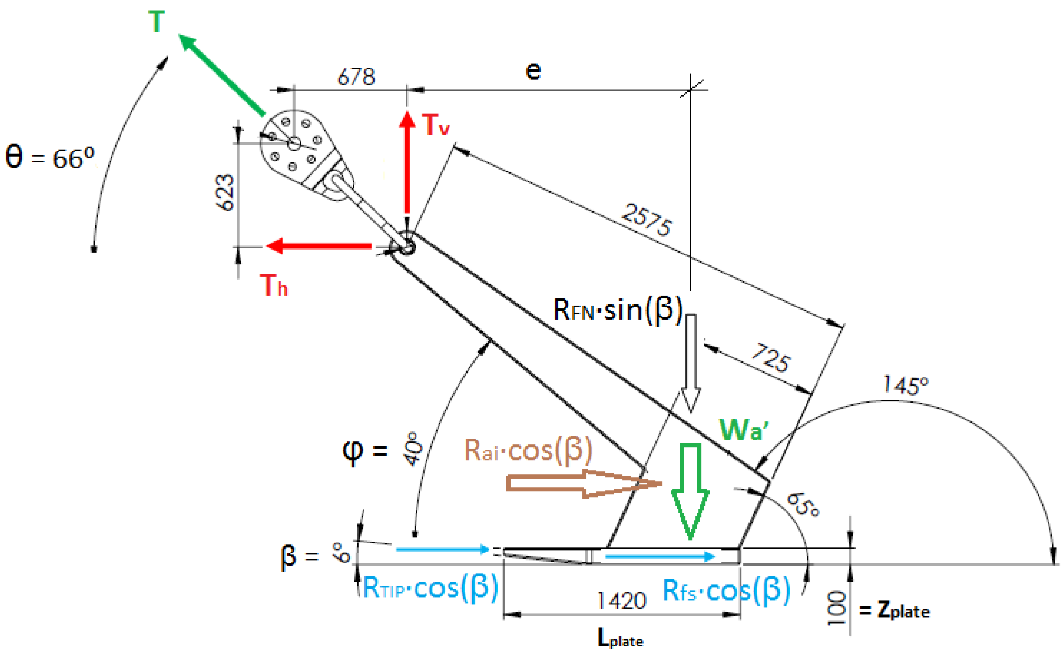


Figure 3.11. Principal reaction forces on plate anchor.

Figure 3.9 shows the free body diagram of the anchor under the tension load T.

The equilibrium calculations are done as follows:

Horizontal Equilibrium;

$$T \cdot \cos(\theta) = \sum_{i=1}^{n} Rai \cdot \cos \beta + Rtip \cdot \cos \beta + Rfn \cdot \sin \beta + Rfs \cdot \cos \beta$$

Vertical Equilibrium;

$$T \cdot \sin \theta = \left[\sum_{i=1}^{n} Rai \cdot \sin \beta + Rtip \cdot \sin \beta + Rfs \cdot \sin \beta \right] + Rfn \cdot \cos \beta + Wa'$$

The definition of the load components and the obtained results are shown in Table 6 below.

Table 3.5. Main Parameters and results.

Equilibrium Calculations	value	units
θ (slope angle or orentation of anchor line)	66	o
β (penetration angle direction)	6	o
φ (fluke angle)	40	o
Rfn(soil normal resistance at the fluke)	12	KN
Rfs(tip resistance at the fluke)	0,4	KN
Rtip(tip resistance at the fluke)	12	KN
Rai(soil resistance at the anchor components)	12	KN
Wa'(submerged anchor weight)	16,83	KN
e(dist. Center gravity)	2	m
Width plate	1,5	m
T_v (θ=66)	28,70	KN
T_h (θ=66)	62,74	KN
T(max)	69	KN
T(max)	**7,03**	**ton**

The obtained maximum tension is the maximum admissible value for this anchor, while the rest of the values (Rfn, Rfs, Rtip, Rai) are taken from DNV-RP-E301/302. They are soil parameters, and further details are given in Table 6 and Appendix 3.

As already discussed in Section 3.2.1.2, the maximum tension obtained was **5 ton**. Thus, it can be concluded that this anchor is at least by170% safe with respect to the transmitted tension from wire cable.

From the above analysis, it is possible to guess one of the main aspects as follows:

$$\uparrow \text{Zplate} \ \& \uparrow\uparrow Wplate \ \& \ \uparrow\uparrow Lplate \leftrightarrow \uparrow \text{Area plate} \rightarrow \uparrow\uparrow \text{Tmaxadm}$$

➢ Stress Analysis:

Fig. 3.10 shows the global stress analysis that was carried out with all main physical forces (Drag, Lift, Weight and Tension).

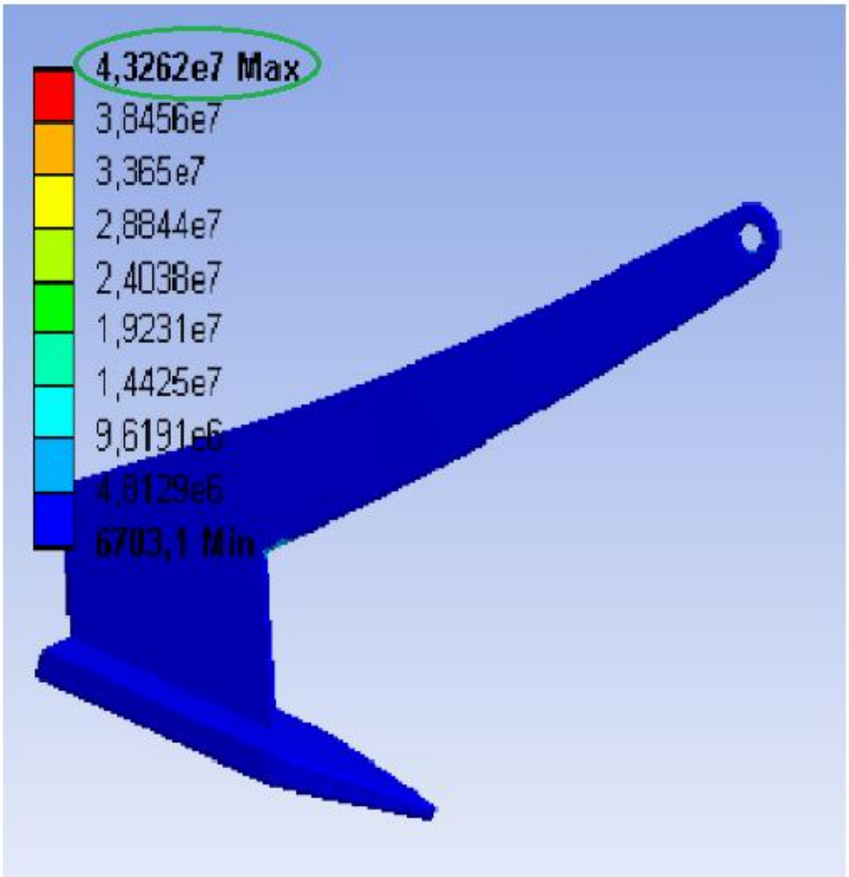


Figure 3.12. Analyses from Von Misses Principle.

The result obtained from this analyses shows that the maximum stress is around 43,2 MPa and the total de deformation is insignificant. It is important to mention that all these results are valid based on the hypothetical case, i.e., it is sufficiently anchored on seabed surface. Also we can determine the maximum number of cycle’s life, showed on next graph below;

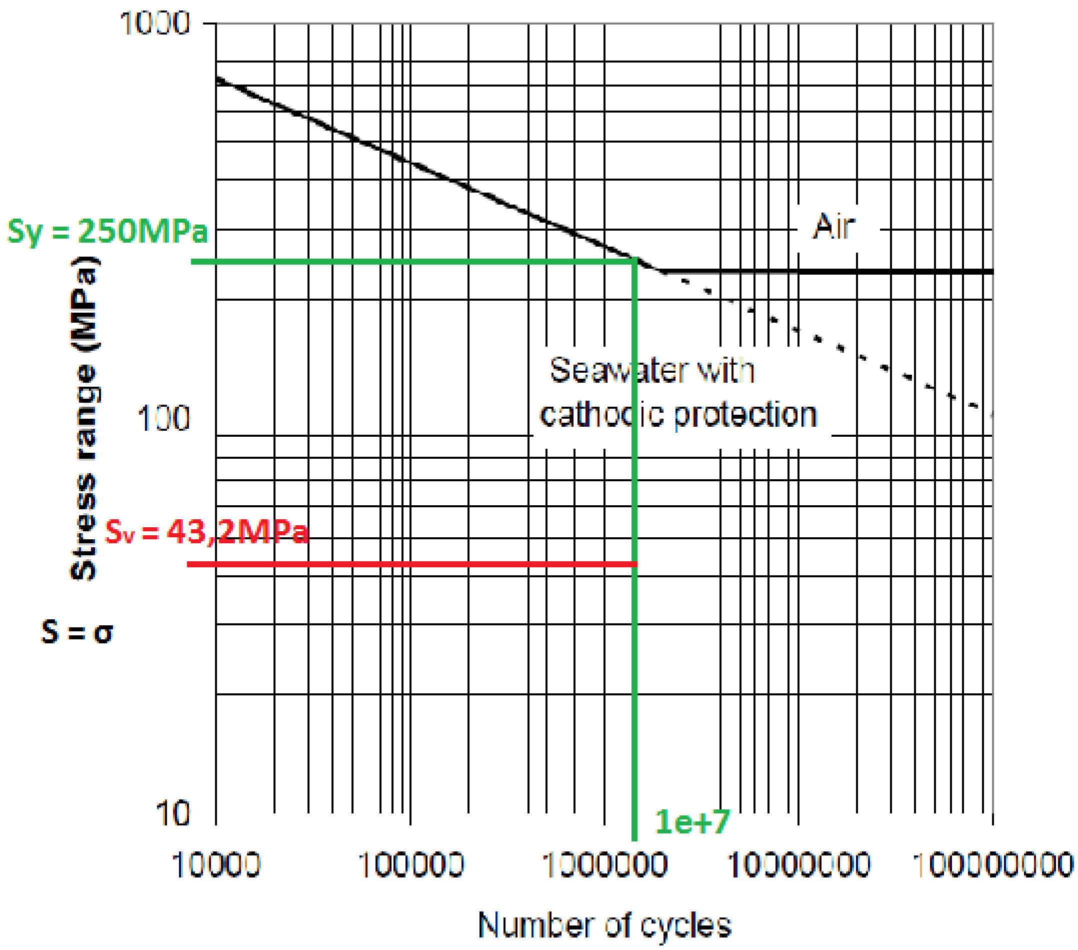


Figure 3.13. S-N for High Strength Steel [19] .

The S-N diagram for Strength Steel is only a representation of "Stress front Life Cycle" on yield tensile of material and Von-Misses approximation, we can observe clearly that this model could support plenty Stress, for long life cycles.

3.2.2. Lifting and Hydrodynamic Load on Flexible Drive Line.

Taking as starting point, the initial idea patent by EWE, showed on figure 1.12. (Chapter 1), this whole device requests a new and different research to lift it, in its entirety.

Firstly, the whole assembly process for all the components and bodies, as well as if all of them will give a correct full movement as WEC.Secondly, the whole system should have appropriate design and dimensioning to facilitate maintenance process undersea, throughout their lifetime (around 20years). And thirdly, the whole Hydrodynamic Load incident must be keep enough though a mooring control (tension legs).

After, to have clear these three main factors, the whole lifting and assembly on this device, will be divided in four ways ;

1. Lifting and Assembly of Body2.
2. Hydrodynamic Load on Body 1(Main Body).
3. Buoyancy of main Body.
4. Sea-bed Anchoring System.

Both ways to eject the lifting process are showed, in the next representation**;**

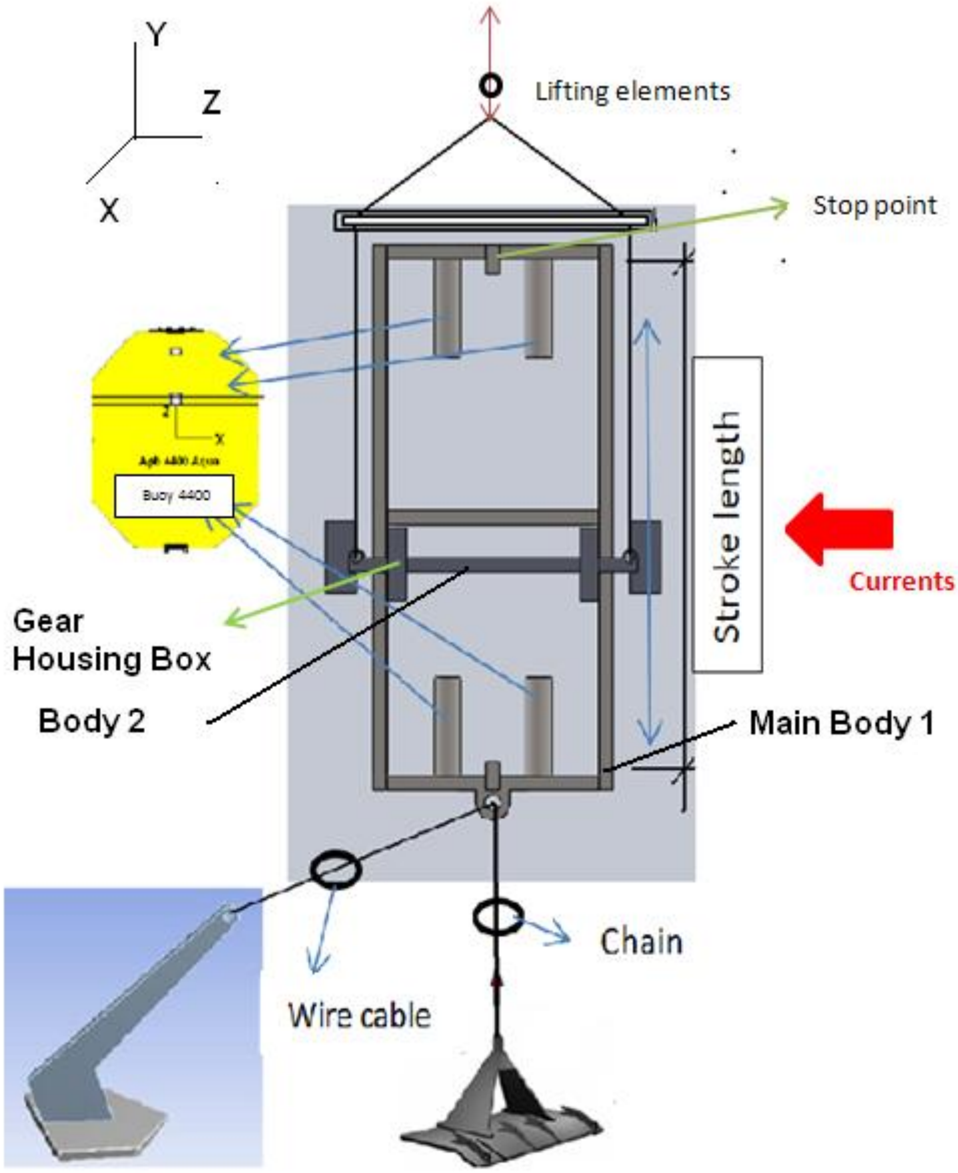


Figure 3.14. Representation of Flexible Drive line Model

3.2.2.1. Lifting and Assembly of Body 2.

The general method to lift and assembly all the components until Body 2, must be according to with the way described by the linear movement (up-down), starting from the surface of the waves to the maximum stroke length under water, described for the mechanism in moving. As well as, the whole proof tension along all chains connects, must be at least three times higher than restoring force, well known as safety coefficient, just to protect against extreme environmental conditions.

As well was showed in the last figure, the whole system must described a linear movement over each helicoidally drive gear track (eight), which ones are connecting both bodies (2 and 1), that means, the four chains are joined to spreader bar must be have almost the same tension in moving (up-down) to satisfy the most optimal movement of the whole device. Just as, will be explained on the next Chapter 4.

Anybody in linear movement, depends directly of the force (moving mass) and linear speed. Therefore, the whole lifting and assembly should be balanced in its entirety. To carry out it, will be necessary to have under control two main factors (total **Mass** and **Buoyancy**);

- **Total Mass :**

Based on vertical equilibrium of this body in movement, which one as main parameter has the total mass underwater (**m_w**), assuming that the opposite reaction along the vertical direction is the whole restoring force (**Fs**), which one I consider, are close of said direction.

To carry on and prove all the theoretical calculations of the global restoring force, firstly it is necessary to know, how much will be this total weigh, see table below;

Table 3.6. Parts and Total Mass.

PARTS	MASS (Kg)
Struct.join Bouys	530,170
Chain (35m)	42,63
Chain Slings grade 8	26,521
Spraider bar struct.(7x7)	400
Body 2	13050
TOTAL (Mt) →	**14049,321**

Also, I must said that for Body 2, are include the total mass, that means, the approximation from catalogues with all parts (beams, gear racks…etc), are integrated it. For it, even, I have assumed a 5% more.

By another side, I must said that the obtained mass, it will be elemental value to get approach of lineal speed on main body in next chapter, because it will stay uniformly distributed along each helicoidally drive gear track.

➢ Buoyancy, Restoring Force and Draft distance :

All theoretical calculations carry out concerning with the Buoyancy of Body 2, was basically, take the starting view from table 1 and fig 2.8 on the last chapter, to get approximation how much will be the entire Buoyancy necessary, will be necessary to have under control the main parameters, which one related it so such the steps to follow;

$$Fs = B - \boxed{Mt}$$

$$Fs = B \cdot y$$

$$Draft = Hb + y$$

As we can see in the last formulas, the whole restoring force (**Fs**) and total mass (**Mt**), must be appropriates to have a good Buoyancy (**B**). To determine it, which would be a good one, I have effort it though operational technique, in which one, I have chosen Buoyancy at least 35% bigger than total mass.

As well I cited on last Chapter 2 in section 2.3.1, the total Buoyancy of five buoys, are around 22000 Kg.

Substituting these values obtained previously, the **total restoring force** (Fs) is **7950.67 Kg**, as result a **draft** of **1.74m** (showed on fig. 2.8), which one depends of height of buoy(**Hb**) and the vertical displacement(**y**).

3.2.2.2. Hydrodynamic Load on Body 1(Main Body).

I have named this Body 1 as main body, due to is mainly element on the whole system, it must stabilized and attune the more rigid vertical position along of 15m length. As well as, the working range to set this main body must be between 40 m and 55m from surface and 25m to bottom, this entails a tough task.

Not only, from the way how it will be located and assembly in situ, if not, how it must be resist and support all currents, which ones are making a strongest drag forces perpendicular along external face of said body (plane X-Y), as result causing a moment along pitch direction(x). To avoid all these issues, the lifting adopted along the next sections should be safe. As well I have noted and assumed on Chapter 1, the currents will move in the same way as such the waves 180º along plane z-x, as well showed on (figure 3.14).

Before to proceed with the lifting process, firstly we must have clarified how much will be the displacement of this body, induced by currents. And secondly, after have calculated the range displaced, will be ejected the process of lifting, though tension legs, anchored to marine bottom. Theses tension legs will help to reduce this displacement.

To have under control more or less, how will be the behavior of currents and his consequences over the main body, will be explained step by step along this section.

All cited before, entails a deeply analyses, which on will be proved and showed graphically as briefly way.

This graphical demonstration will carry on in the next order;

1. **Speed of Currents (Uc).**
2. **Potential flow curve.**
3. **Drag Forces (Fd).**

➢ Speed Currents (Uc) :

From an offshore engineering perspective, we could guess that the currents speeds close to the surface should be close to 1 m/s and almost zero on the seabed, of course, for a normal situation.

But to be more precise, all the date of these speed currents was extracted directly imported from DNV-RP-C205, and exported to “Orcaflex”, to get a general orientation how will be behavior of the currents for normal conditions of sea, as shown bellow ;

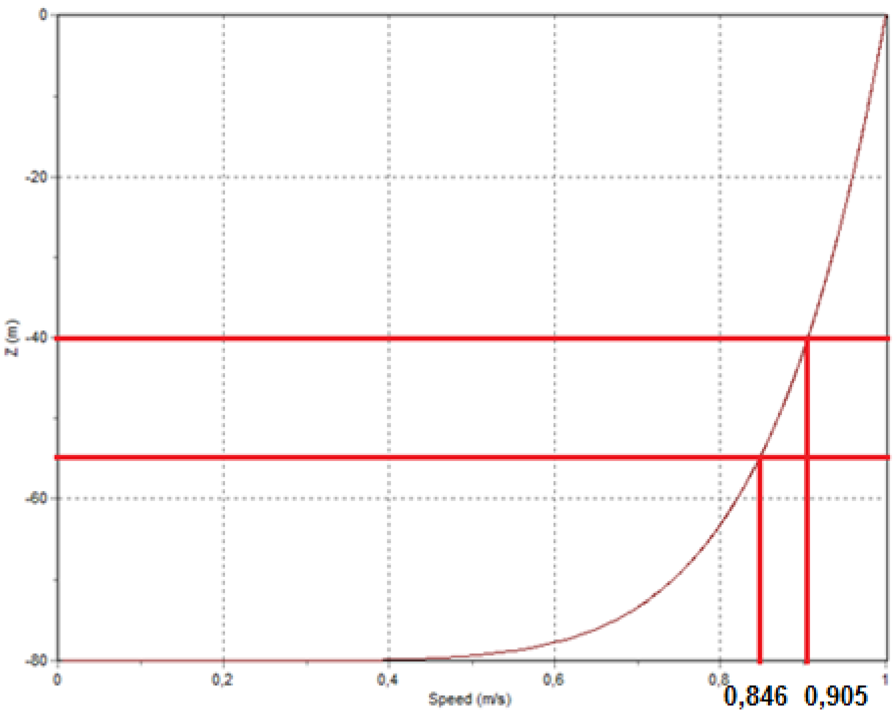


Figure 3.15. Depth vs. Currents Speed

As well we have observed on the last figure, the maximum current speed will induce to originate the maximum displacement in that point (at -40m).

➢ **Potential Flow curve** :

To understand how the pressure variation (ΔP) anybody submerse around his wall, respect to all currents and his coefficient of pressure(Cp), can push notable his trajectory respect to vertical direction (gravity), primarily due to the angle (θ) of the currents, as described in the next equation and graph below;

$$\Delta P = \frac{1}{2} \cdot \rho \cdot Uc^2 \cdot Cp \ ; \qquad Cp = 1 - 4 \cdot sin^2\theta$$

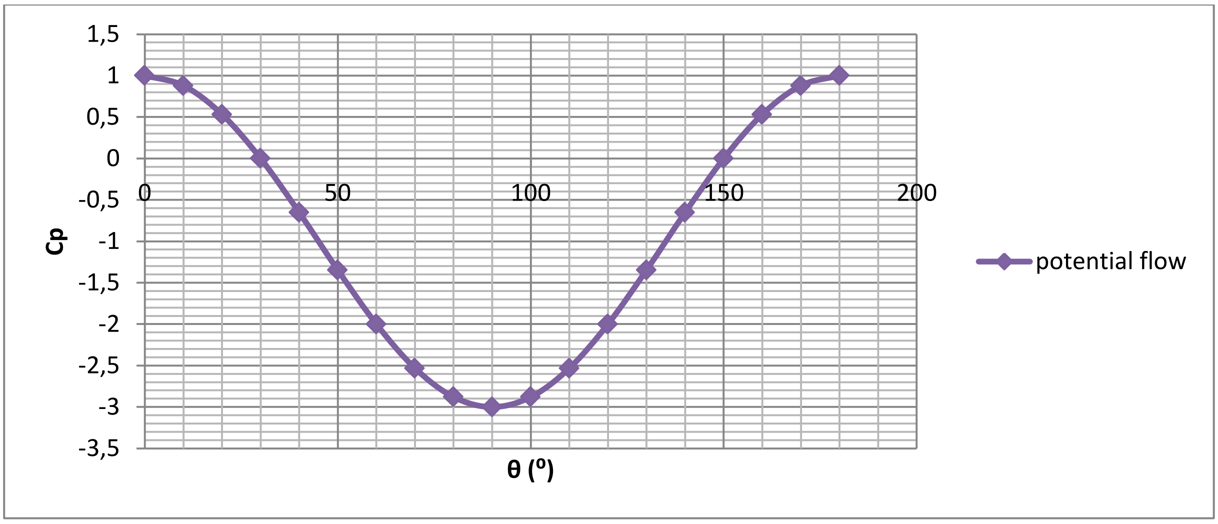


Figure 3.16. Cp vs. Angle described (θ)

All I want to illustrate with the last graph is only that maximum value of pressure on one body submerse, is situated around 90º perpendiculars to plane x-y, that means, the maximum concentration of stress are located around that value. Also, it will stay clearly indicated on next section, as well.

- **Drag Forces** :

As well I have citied, on initial introduction of this section, all drag forces should analysis in a criteria, from the biggest ones to small ones, that means, the range of displacement will be varies along the heave direction (axle z). But to be more specific, how I have decided and implant this geometry on the model, was basically, respect three main parameters;

- Roughness of material (ε).
- Maximum diameter of the body (D).
- Reynolds number(Rn)

After, to carry out optimal calculations between values of diameter, due to roughness of structural steel (ε = 4, 4exp-5) remains constant, the most optimal theoretical value was a diameter with 6,8m. I would like to explain how understand it as optimal value;

$$\frac{\varepsilon}{D} = \downarrow\downarrow$$

$$\boxed{Rn = \frac{Uc \cdot D}{\nu}} \qquad \boxed{Fd = \frac{1}{2} \cdot \rho \cdot (Uc)^2 \cdot Cd \cdot D \cdot \sin(\theta)}$$

$\Downarrow$ **Fluid dynamic Resistance** $\Downarrow$
(FDR)

As we can see on (figure 3.14), these global FDR is considerably reduce, due to the model designed is absolutely empty. Even more, this full resistance could be seen a little decreased if for this main body, have chosen rounded corners, which ones will decreased all torsion moments, caused by the circular currents along sway direction (axle Y). But finally I opted, commercial and standard profiles, as we have seen on (figure 3.14 or Appendix D). Also, I must said, the constant values (ν, ρ, and Cd), have adopted from the specifications and guidelines (DNV-RP-C205).

All the graphical calculations obtained, are showed below;

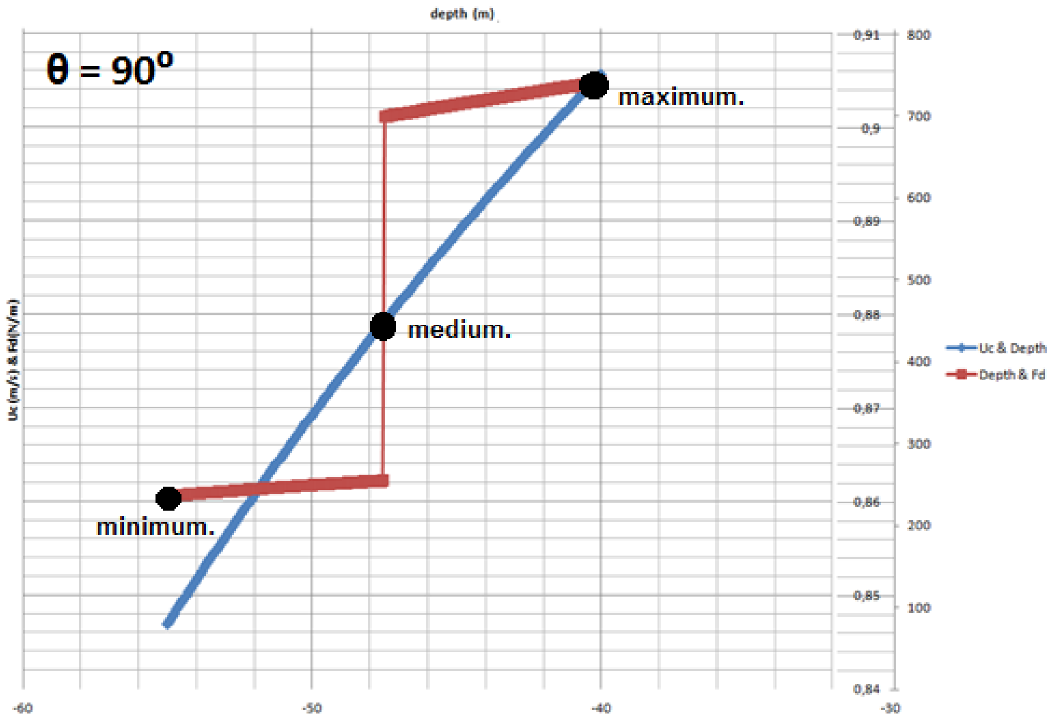


Figure 3.17. Depth vs. Uc & Fd

The last three points, represented on the last graph means drag forces (N/m) applied over the main model fixed in a normal equilibrium position thought the tensions legs, which ones will be explained in the next couples sections.

Also, could be a representation of this drag force (N) along the sway direction (axle Y), figure below;

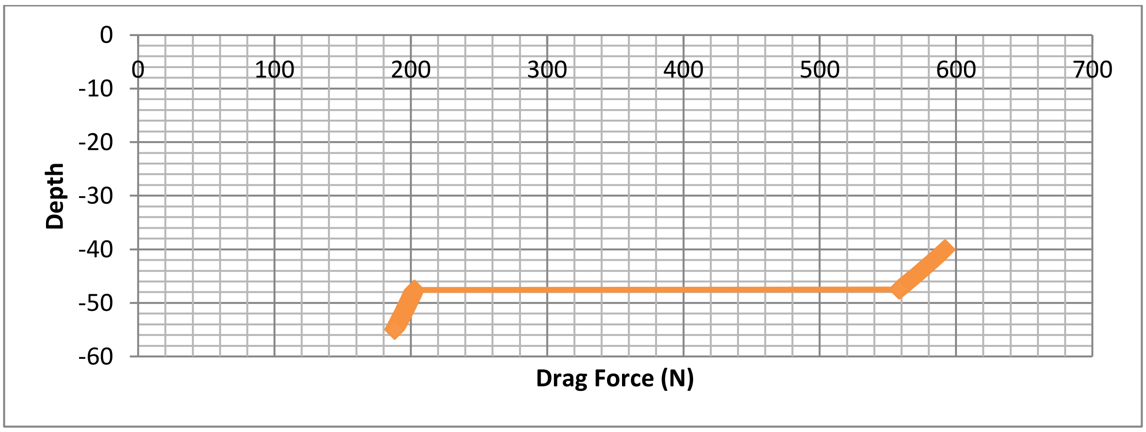


Figure 3.18. Depth vs. Fd (N)

As well we have observed, on the last two graphs, we have three main points, which ones will stay clearly affected this full drag force. Hence, all the values vary of a normal downward way, exception a range of 0,2m that variation is huge, this is because in that range, is situated between two empty spaces (see figure 3.10), which ones, are causing a considerable variation of FDR.

To reach a conclusion that I can extracted from these last graphs, are showed in the next table, below;

Table 3.7. Characteristic Values.

Direction / Turn	Sway(Y)	Heave(Z)	Heave(-Z)	Roll (-X)	Heave(-Z)	
Point	**Depth(m)**	**Uc(m/s)**	**Fd(N)**	**Md(Nm)**	**δh (m)**	**αmax(º)**
Maximum	-40	0,905	592,304	8884,563	3	21,801
Medium	-47,5	0,878	202,914	1521,859	1,02	7,744
Minimum	-55	0,846	188,339	~0	~0	~0

Values obtained, is assume from hypothetical case that the two principal tension legs (chains) are close to the vertical position.

But as well I have showed on the last table, the horizontal displacement (**δh**) and moment (**Md)**, is considerable high, that's why the main model requires two more legs, to reduce it. All of them will prove it in the next couple sections.

3.2.2.3. Buoyancy on Main Body.

Requirements, to fix and keep the main body at 25m from marine bottom, demands correct Buoyancy as well as Tension legs to stabilize it.

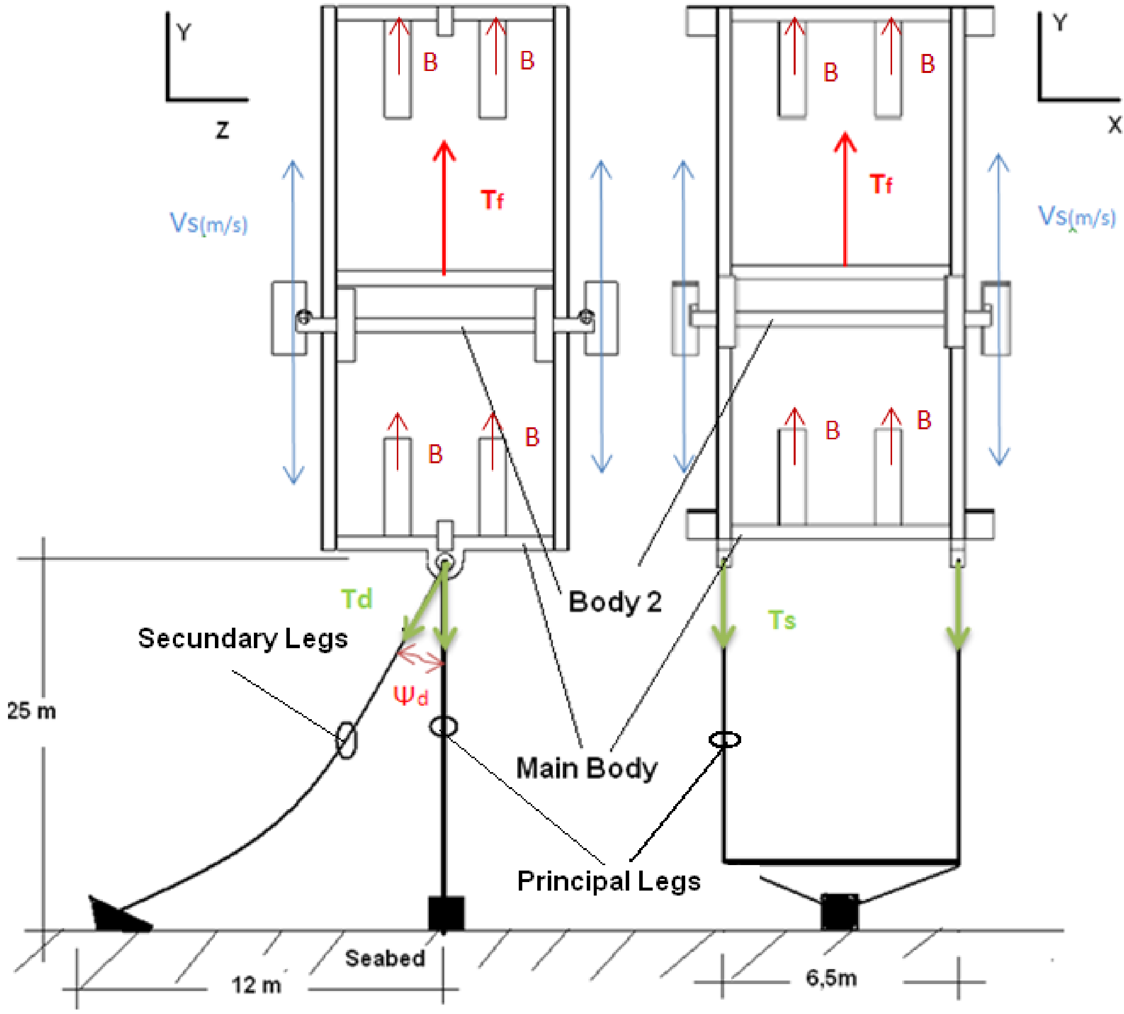


Figure 3.19. Bodies and Tension Legs

The whole Buoyancy integrated with eight buoys (“Apb 4400aqua”) each one in each cone, inside of the main body, must be notable higher than the maximum tension legs, which one it will be submitted along its lifetime service (20years).

Just as I have explained in previous sections of this chapter, the buoyancy is related with total mass underwater and restoring force of legs. There are the results obtained;

Table 3.8. Parameters of Main Body.

Bouyancy(kg)	Masa(Kg)	Trestoring(Kg)
Apb 4400 aqua (8)	Main Body.Module1	
32000	8700	**23300**

3.2.2.4. Sea-bed Anchoring System.

To carry out all this process, it will be necessary to take into account mainly the hydrodynamic load explained previously, due to the legs anchored, must try to keep the main body in a most vertical position, also as safety coefficient(recommended by “CarlStahl”) the tension legs need to have at least three times more tension than the total Buoyancy.

The corresponding Sea-bed Anchoring System will be dived in three main categories;

1. **Two Principal Legs.**
2. **Two Secondary Legs.**
3. **Anchors.**

- **Two Principal Legs:**

These two tension legs situated at 25m from the bottom, I will consider them as principal ones, due to their vertical position and tension along each one, must be keep the main body against currents, and as well the type selected was chain (28 x 150), both legs, must have in constant tension.

The corresponding theoretical calculations have been calculated as two main situations (Static +Dynamic). For a static state, I assumed that the vertical angle (α) was close to zero, and for a dynamic I have applied the exactly angle caused by the maximum displacement (see table 7). Also, I must said that this kind of calculations was shown in sections before, that means, I consider enough defined as a table results, see table below;

Table 3.9. Restoring Force of Chain Legs.

STATIC ANALYSIS		$\alpha = 0^{o}$ (slope)
dS (m)	δh =0 m	25
Ts (Kg)	static tension	**23300**
n^{o}legs	Chain 28 x 150	2
ΔTs (Kg)	10% of Ts	2330
Rs = T + ΔT	each leg	**25630**
Rs = T + ΔT	both legs	51260

DYNAMIC ANALYSIS		αmax = 21,801^{o}
dS (m)	δh =3 m	25
Td (Kg)	dynamic tension	**21633,568**
n^{o}legs	Chain 28 x 150	2
ΔTd (Kg)	10% of Td	2163,356
Rd = T + ΔT	each leg	**23796,925**
Rd = T + ΔT	both legs	47593,851

Trestoring(Kg) = (Fs)	**Min.Breaking Load(kg)**	**Safety (%)↑**
49426,925	80326,197	38,46

This last values obtained, are for each leg.

To reach a conclusion between the restoring forces obtained on Table 8 and total restoring force with two chain legs, these two legs could resist at least 4 times more than whole resistance, due to the chain selected are appropriated.

- **Two Secondary Legs:**

The main functionally and purpose, of these two legs emplaced behind of the main body (see fig.3.14), is to reduce the moment generated around X direction (see table 7). To avoid it, I have decided to connect these two legs from the same point connect, as well principle legs until the corresponding anchors situated around 12m from the principal anchor. As shorter the distance between secondary anchors to principal one, as much we can reduce the drag moment generated, as well the displacement. Also, these legs don't must be in constant tension, rather as catenaries legs.

The type of legs selected, was wire cable (6x36), by a simple reason stresses supported, due to the secondary legs are not in constant tension, as principal ones. As own criterion for sizing, of these legs, I have decided to situated their anchors at 12m, cited before, and with a range of slope varying from 22º to 15 º, depending if it is in static tension or dynamic, so such an incremental of 5º ($\Delta\psi$) for both state. Also, as well safety coefficient I considered an incremental of 20% more in static analysis than dynamic. Therefore, these tension legs are in maximum tension (static) when the maximum hydrodynamic load is pushing and minimum tension (dynamic) when the main body is in a resting situation. See table results, below;

Table3.10. Restoring Force of Wire Cables Legs.

STATIC ANALYSIS	**(max.tension)**	ψs=21,801º ; Δψs=5º
dS (m)	δh =3 m	29
Ts (Kg)	static tension	9252,380
ΔTs (Kg)	30% of Ts	2775,714
Rs = T + ΔT	each leg	**12028,095**
nºlegs	Wire cable 6x36	2
Rs = T + ΔT	both legs	24056,19048

DYNAMIC ANALYSIS	**(untension, only weight cables)**	ψd = 15º ; Δψd=5º
dS (m)	δh =0 m	29
Td (Kg)	dynamic tension	4350
ΔTd (Kg)	10% of Ts	435
Rd = T + ΔT	each leg	**4785**
nºlegs	Wire cable 6x36	2
Rs = T + ΔT	both legs	9570

$T_{restoring}$(Kg) = (Fs)	**Min.Breaking Load(kg)**	**Safety (%)↑**
16813,095	26197,757	35,822

The main objective, of these restoring force achieved, is how these two tension legs could reduce significantly the displacement induced by the currents.

- **Anchors:**

As well I have explained and shown on section 3.2.1.5 from commercial industry we have plenty model of anchors, but main challenge will be choose the most appropriate one for each application, the sea-bed anchoring system dimensioned, demands two kind of anchors;

- **SEPLA ANCHOR (principal legs).**
- **ANCHOR LINE (secondary legs).**

SEPLA ANCHOR, will be used for the two principal legs with ones have the maximum vertical restoring force, this model of anchor, is the most suitable for these tension legs, as well I have explained in said section, according to the high restoring force(tension) which one is subjected.

The specific model, must be the highest one (15ftx30ft&87tons) for permanent installations.

By another side, I have contemplate don’t carry out any analysis of stress, insomuch as the total restoring force admitted for this anchor is almost a 44% higher than supported by the chain tension legs(49,42ton).

ANCHORE LINE, will be installed for each wire cable leg explained previously, the main reason because I opted to use this model, will be explained below. Firstly, the restoring force is considerably smaller than the principle legs, and secondly, the geometry of this model, is corresponding to the slope of each leg as shown in (figure 3.10), that means, the interlocking of these model with this specific position, will advantage to fix notable the whole tension, and as a result, reduce the displacement of the main body. Also, I leave aside to carry out any stress calculations, due to will be enough to choose a model with a restoring force with 24 ton.

3.3. Chapter Summary.

Along this chapter, the appropriate assembly process in fully, as well as the elements suitable for each device, was main challenge, due to each device demand different conditions as WEC. If I must do an evaluation of difficulties for each device, obviously the Cape Verde has simple installation as well as the hydrodynamic load don't have no major complications compare with Flexible Drive Line, given that the first one the main module is located onshore and second offshore, for these last reasons the mooring conditions are completely different.

From these points of view, the whole study of this chapter concludes how each WEC have different constraints, as result each ones requires calculations in different ways to carry out.

Have reached a better approximation of all these calculations shown, as well as graphical results, will be though CFD programs (SIMO HydroD or AquaSim).

CHAPTER 4 MACHINARY

4.1. Introduction

As well was distingue in the last chapters, this chapter is devoted primarily to dimension and design and transmission on WEC. I will carry out it in two ways, the first one for Cape Verde Device and second one for Flexible Drive Line, both have machinery equipment emplaced in two different places(on land and offshore), as well was described in previous chapters. Basically, entails two studies absolutely different, although sharing some aspects from Mechanical Engineering Design.

To make a brief orientation, both share a linear transmission system but in two different ways, in the first device this linear movement is transmitted by a belts transmission moored by cables on shore (represented on Appendix E), and for the second one, thought Helicoidally drive Gear track are connect and join both bodies (explained in the last Chapter 3) though a Gear Rack transmission. Also, the entirely dimensioning of components will carry out according to real components from manufacturer's catalogues.

4.2. Global Transmission System on WEC

4.2.1. Cape Verde Device

As well was showed on fig. 1.10, the whole integrate transmission system used was wire cable joined to belt in both side, which one is the most appropriate for power transmission system, with heavy loads in movement, so such their use reduces the cost and the design of a machine.

So such, in terms the efficiency, of losses and safety over other elements (driver wheels, bearings…etc) tend to be much better than chains for instance. Among all types of Belts (flat, timing,…etc) the model choice was Timing Belt with double teeth on each side, due to his geometry is the most appropriate one for the model designed, as well the high force in movement forward and backward, which one is transmitted directly on each driver wheel(or known sprockets too). As other good properties of transmission with belts are the elimination of the restriction on lineal speed, their teeth makes it possible to run any speed. Disadvantages, the limitations of breaking tension, as well as the high dynamic fluctuations caused at the belt-tooth or even the cost as well.

Dimension a transmission system though timing belts, capable of transmitting an input power with a constant angular speed on driver wheels, the driver shaft activated directly the movement a helicoidally gear box, whereby transmitted the appropriate input power to a generator AC.

The main details and input date are on the **Appendix E**, linear power energy in movement, has two main elements to eject it, weigh from tower and the total resultant force from subsea, which one was obtained in previous chapters (around 5ton). These force from the wire cable subsea, have a small losses after passing over the pulley on land, which ones are around 9% less, get a final value of **4567,73Kg** to Machinery. As well I have showed on Appendix E, this device has two main global components;

-Tower.

-Machinery.

4.2.1.1. Design and Analysis of Tower Model

The design and Analysis of the Tower model, must be focus from a point of view of Civil Engineering, was according to main external requirements due to it is summated;

- Moment (Mz) of 24525Nm.
- Horizontal Pressure of wind onshore 8000N/m².
- Principal Legs on ground recess.

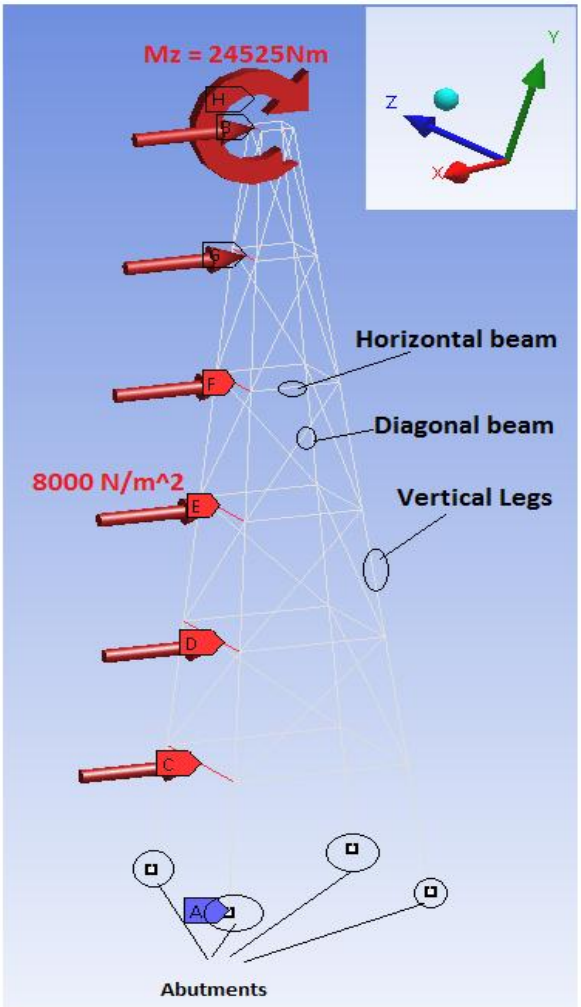


Figure 4.1. Representation of Pressure and Moment on Tower

The profiles section, elected for each beam, has been three types:

- Vertical ones (solid Square section) for Principal Legs.
- Horizontal ones (Tubular Square section).
- Diagonal and Horizontal ones (solid Triangular section).

The whole structure, is absolutely welded, as well as the material of each one, is Structural Steel (**Sy=250MPa**).

The way carry out the selection from commercial profiles, was based physical efforts, which ones are acting on whole tower, due to each Beam has different section areas, as well as from the mayor section area to less, will be reduce the tensile stress transmitted between all beams.

$$\downarrow \sigma = \frac{Faxial}{A\uparrow}$$

So such, the main physical properties, are described on **Appendix G**.

The main solicitations to which ones, the whole tower is subjected, are;

- Stress (Pa).
- Bending (Pa).

The results obtained, from the Analysis made, are showed below;

Table 4.1. Stress and Bending results.

Minimum Combined Stress	Maximum Combined Stress	Minimum Bending Stress	Maximum Bending Stress
-6,086e+006 Pa	-1,0048e+006 Pa	-5,0836e+006 Pa	2,9342e-010 Pa
4,8097e+005 Pa	6,0938e+006 Pa	-2,9342e-010 Pa	5,0836e+006 Pa

If we compare the stress range, due to Structural Steel can admitted, we can see clearly that integrity of the tower, will have long life cycles along service life.

4.2.1.2. Dimensioning process of Machinery

These linear force transmitted from outside to inside machinery for a normal conditions of sea behavior, tend to be constant, due to it is compensated and balanced by a weigh in movement inside the tower, describing a constant stroke length as well. Also, along the whole process of theoretical results obtained, I must said that the entire development will be according a range of input values, insomuch all results obtains are from theoretical approximations.

The dimensioning starts directly from the lineal force in movement forward and backward, which one carries a constant linear velocity along the whole timing belt (7m), to the rotational speed of each drivel wheel, and as a result an input Power. These input Power, is transmitted directly to each shaft, which one is supported though a Bearing fixed on each hole of the structure support. The average Power obtained, from the lineal movement to rotational, has his corresponding losses, because of two tensor wheels situated before, as was showed on Appendix E, cause losses. The functionally of this two tension wheels, are to guide the belt in the position whished, according to the initial design established by EWE, as well as requirements demanded by the whole device.

Take as input value the tangential force (**4567,73Kg**) from the timing belt, the rest of elements, which one are integrate the whole design must resist all stress transmitted, as well provides the mechanical energy to move and rotate the generator.

The criteria established to carry out it, the whole dimensioning process, was basically, though two main inputs:

-Maximum Input Power (Kw).

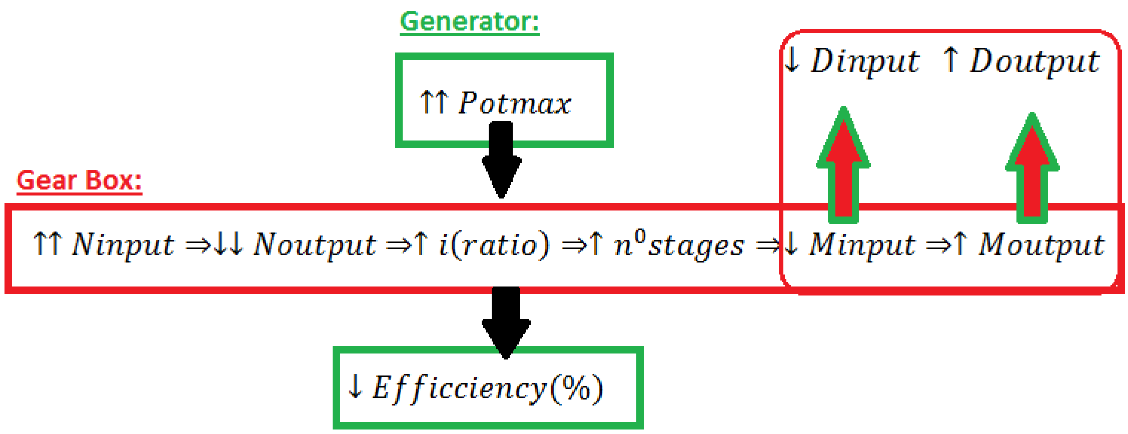
-Maximum Efficiency (%).

To eject these two last inputs, carries a whole compensation between all elements, which ones, are integrated the whole system (from Outside to Inside).

So such the power transmitted, is going from outside to inside. For these, last elementary reason the dimensioning process must be ejected in the same way (from Out to In). The main elements which ones, are transmitted the input power, are;

1. Drivers Chain Wheels (a&b).
2. Shaft drivers.
3. Torque Limiters.
4. Gear Boxes.
5. Generators.

The criteria cited before, carries the next property after the element number 3;



As well we can see, on this last property, to figure out maximum values of Input Power, counteract less Efficiency. The reason, because the number of ratio and stages must to be higher, due to the power transmitted is considerable high as well the type of Gear box model must be the appropriate for this conditions. From Mechanical Engineering Design, we know that for high transmission of Power, implies high transmission of forces as well, as result greater concentration of stress.

The most suitable gear box, for these last establish requirements, are helical gear models, which ones has helicoidally angle between teethes, said angles reduce considerable the whole stress transmitted, as well improve the efficiency. The forces transmitted (Fn,Fa,Ft) tend to be constant between stages, only varies the moment between them and the diameters(D) from mayors until minors .

By other side, the Torque limiter is consider element key, due to reduce the N_{out}(rpm) and M_{out}(Nm) from each driver shaft to Gear Box.

Therefore, don't care how much highest the revolutions from and torque on driver wheels, due to be reduced by the Torque Limiter situated before the Gear box.

To carry out, the whole dimensioning process, I have calculated firstly the average of power transmitted from the two wire cables, which ones join the belt in each side. As well, we can observe on the next table below, I have increased the value at least 10%.

Table 4.2. Parameters of Lineal Movement by belt transmission.

From	T_{max}(N)=F_t	V_f(m/sec)	Pot(Kw)
Pulley on Land	44809,404	3,322	148,860
Tower	44809,404	12,528	561,388
		average→	355,124
		safety factor(↑10%)	**390,637**

After to know how much the power is transmitted along the belt in movement, I must show how will be the representation of forces transmitted outside the Machinery, see figure below;

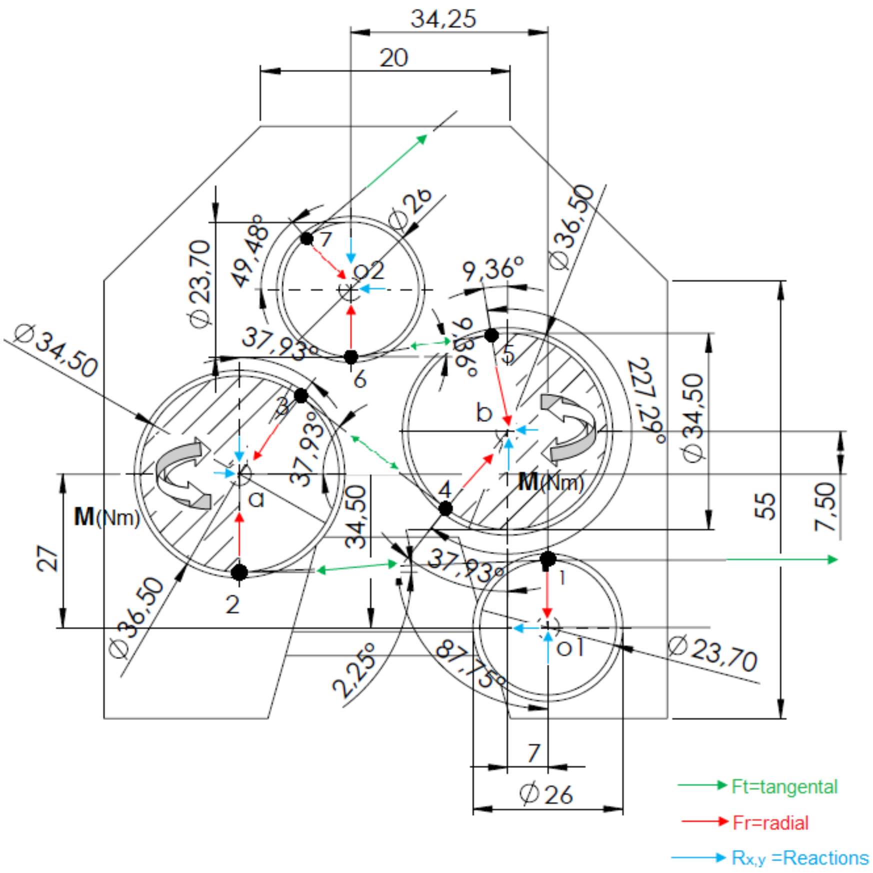


Figure 4.2. Representation of Forces Transmitted.

The corresponding analytical analyses of transmission development along each point show before, were carrying out point by point, as well was represented on last figure. The specific table results, with all values, which ones, I have calculated, are showed on **Appendix F**.

The most important values to extract from this **Appendix F** are the Moment transmitted on each shaft driver (**a&b**), which ones, are connect directly with the torque limiter, and then connect each Helicoidally Gear box. I will assume that from commercial industry, the Torque Limiter must fixed with a Moment of **19895,85 Nm(Mout)**, as well as the output speed revolutions will be a maximum of **120rpm**(**N**out).The 3D model are showed, below:

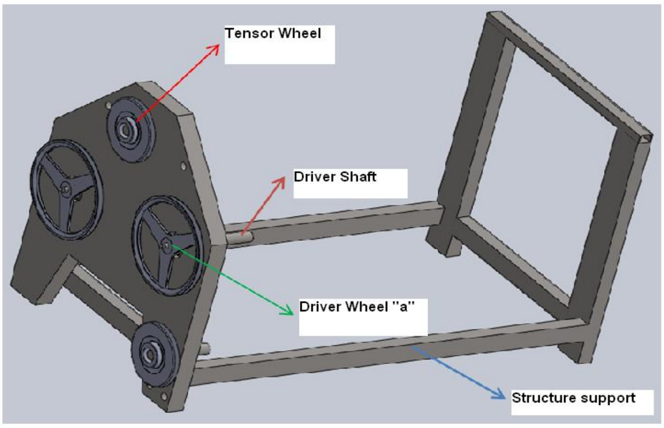


Figure 4.3. Parts of 3D-model Cape Verde design.

The commercial models, which ones I have determined, so such ideal for this device, have been;

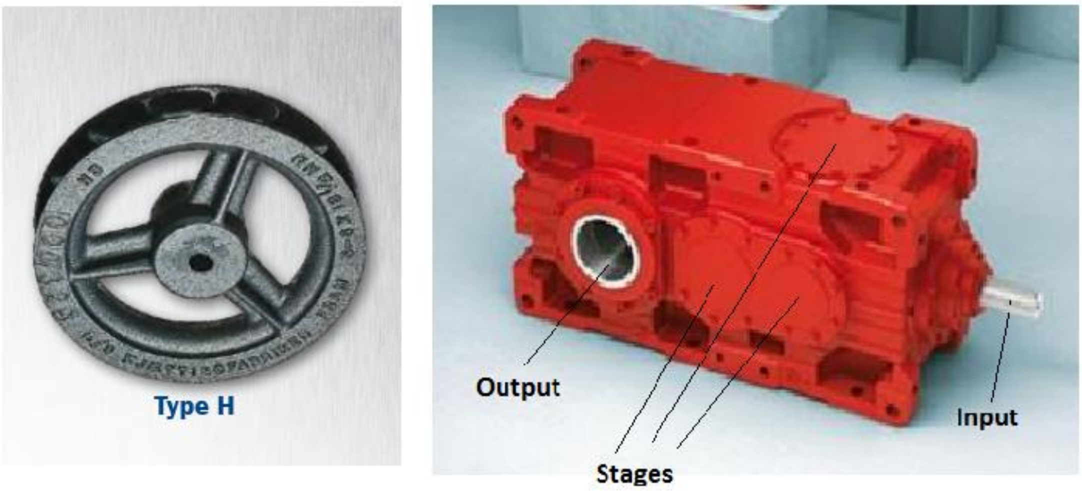


Figure 4.4. Driver Wheel and G.Helical X-series, adopted from [21] [4].

According to all cited before, the main results obtained, from the dimensioning process for this Machinery, are;

Table 4.3.Results of Dimensioning model "Cape Verde".

	Generator(Kw)	250
	n input(rpm)	1750
	n output(rpm)	120
	i(ratio)	14,58
	n^{o}stages	3
G.Model	**X-series**	
T1 input(Nm)	generator	1364,285
T2 output(Nm)	Driver "a"&"b"	**19895,833**
Effic.(%)	H	0,94
T1 input(Nm)		1282,428
Potinput (Kw)		**235**
losses↓	%	0,06

Hence, the maximum Power input take off from two generators, will be around **470Kw**.

4 [21] http://www.framlink.no *and* www.sew-eurodrive.com

4.2.2. Flexible Drive Line Device

As well I have explained along the Chapter 3, the second model of WEC requires others characteristic the design and dimensioning, due to the full linear movement described along 13m of stroke length established on the main body. The dimensioning and design adopted, was according to the requirements, established previously, such as WEC.

Mainly, respect to the mass moving up and down, which one originates high stress along the gear rack lines and well as it transmitted directly to helicoidally gear rack. The part which one, I will analysis, is the body2, which one is described a movement up-down around main body. The whole in movement up and down has mass of **14049,32Kg**, as well as the total Buoyancy determined from the last chapter was 2000Kg. Such as, the maximum stroke length described along the main body is 13m, from equation of lineal speed;

$$\boldsymbol{V\ \frac{m}{s} = Stroke\ length\ /Time}$$

I have determined, the corresponding Lineal Speed of Fall and Time, which one is 0.943m/sec and 14.846sec, and Up and corresponding time, 0.3767m/sec and 37.159sec.

By another side, if we share the total mass that this body 2 has, between the eight lines rack the corresponding force lineal force, on each one is **17227,980N**.

4.2.2.1 Dimensioning process of Machinery

As is evident, the power transmitted is going from outside to inside of the whole mechanism. Therefore, the entirely dimensioning must be carry out from outside to inside, as well. The order of elements which ones, are integrate;

1. Gear Rack.
2. Shaft Drivers.
3. Torque Limiters.
4. Generators.

According with this last order, I have calculated transmission system, from a commercial catalogue´s. I have decided to choose the gear rack from “Stober” [22], which ones are the most suitable gear racks for the model designed, according to previous estimations.

From catalogues date, the principal limitations, of this gear selected, are;

- Linear Force, maximum mass in movement supported (Fmax=25KN).
- Maximum Output Torque.
- Nominal diameter of drivers gear racks.
- Output revolutions (475rpm).

With this last limitations cited, I have ejected the theoretical estimations. See table below;

Table 4.4. Main parameters of movement fall and up.

Helicoidal_Stober g.rack ZTRS		Max.25KN		570		d =0,0541						
FALL → +	Nº lines rack	Force each rack (N)	Pot(Kw) linear mov.	M(Nm)	Pot(Kw) rot.	r (m)	V1fall(m/s)	V1fall (m/min)	ω (rad)	ω (rad/sec)	f (Hz)	n (rpm)
	8	17227,98093	**16,24610247**	466,0168842	**155,1274239**	0,02705	0,943006759	**56,58040556**	34,8616177	332,9039271	52,98331	**3178,99833**
		(safety = 31%)		safety(18,24%)								
UP → -	Nº lines rack	Force each rack (N)	Pot(Kw) linear mov.	M(Nm)	Pot(Kw) rot.	r (m)	V1up(m/s)	V1up (m/min)	ω (rad)	ω (rad/sec)	f (Hz)	n (rpm)
	8	17227,98093	**6,490801739**	466,0168842	**61,97802549**	0,02705	0,376759283	22,60555696	13,9282544	133,0050326	21,16841	**1270,1045**

As well, we can observe on last table, the limitation by the output revolutions (475rpm), is not complaint, due to the model will demand a torque limiter on each shaft driver, to keep a constant output to each gear box.

As well, I have determined range of output revolutions up and down along the gear rack was (3179rpm and 1210rpm). With these last dimensional inputs stabilize before, the whole output revolutions will be 475rpm, due to cited before, the driver shafts need to have torque limiter to reduce this output speed, as well to keep constant an output revolutions, along the movement described by the body2.

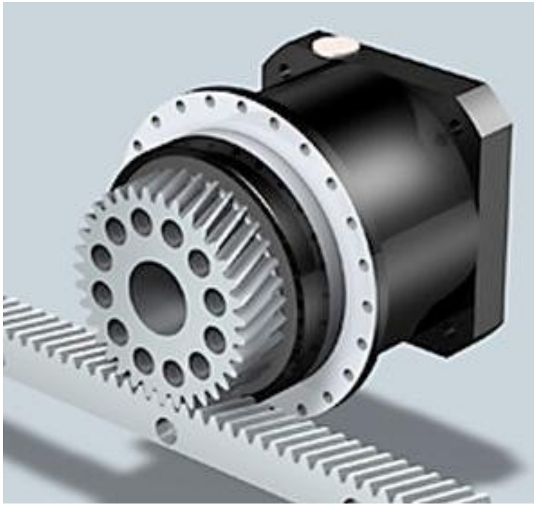


Figure 4.5. Gear Rack model, adopted from Stober[22] [5].

[5][22]http://www.directindustry.com/prod/stober/rack-and-pinion-drives-6037-796031.html

Therefore, with all described and cited before, the dimensioning process, as well the theoretical estimations are showed on the next table below;

Table 4.5. Results of Dimensioning model "Flexible Drive Line"

	Generator(Kw)	28,350
	n input(rpm)	1900
	n output(rpm)	475
	i(ratio)	4
	nºstages	2
G.Model	**Stober g.rack**	
T1 input(Nm)	Generator	142,5
T2 output(Nm)		570
Effic.(%)	η	0,93
T1 input(Nm)		132,525
Potinput (Kw)		**26,366**
losses↓	%	0,07

According with the theoretical estimation, with a total number of sixteen generators subsea from "Sicei" [6], the total amount of energy obtained offshore will be around **0,422MWatts.**

[6] [23] http:// www.sicei.com

4.3. Chapter Summary.

Along this chapter, the methodology has been addressed from theoretical calculations by a dimensioning process follow from a commercial catalogues, which ones have been the principal aim, to start the whole dimensioning process, such as to get better practical approximations. As well I have described, step by step in this Chapter, both devices demand different ways to dimensioning and assembly, according to the proposal of each device. The main goal of this chapter, from practical mechanical engineering point of view, was how each dimensioning process, was adapted previously from catalogues date, according with the demands of each device.

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23. http:// www.sicei.com

APPENDICES

APPENDIX A

Scatter diagram for the North Atlantic by DNV-RP-C205.

Hs (m) \ Tz (sec)	3,5	4,5	5,5	6,5	7,5	8,5	9,5	10,5	11,5	12,5	13,5	14,5	15,5	16,5	17,5	18,5
0,5	1,3	133,7	865,6	1186	634,2	186,3	36,9	5,6	0,7	0,1	0	0	0	0	0	0
1,5	0	29,3	986	4976	7738	5569,7	2375,7	703,5	160,7	30,5	5,1	0,8	0,1	0	0	0
2,5	0	2,2	197,5	2158,8	6230	7449,5	4860,4	2066	644,5	160,2	33,7	6,3	1,1	0,2	0	0
3,5	0	0	34,9	695,5	3226,5	5675	5099,1	2838	1114,1	337,7	84,3	18,2	3,5	0,6	0,1	0
4,5	0	0	6	196,1	1354,3	3288,5	3857,5	2685,5	1275,2	455,1	130,9	31,9	6,9	1,3	0,2	0
5,5	0	0	1	51	498,4	1602,9	2372,7	2008,3	1126	463,6	150,9	41	9,7	2,1	0,4	0,1
6,5	0	0	0,2	12,6	167	690,3	1257,9	1268,6	825,9	386,8	140,8	42,2	10,9	2,5	0,5	0,1
7,5	0	0	0	3	52,1	270,1	594,4	703,2	524,9	276,7	111,7	36,7	10,2	2,5	0,6	0,1
8,5	0	0	0	0,7	15,4	97,9	255,9	350,6	296,9	174,6	77,6	27,7	8,4	2,2	0,5	0,1
9,5	0	0	0	0,2	4,3	33,2	101,9	159,9	152,2	99,2	48,3	18,7	6,1	1,7	0,4	0,1
10,5	0	0	0	0	1,2	10,7	37,9	67,5	71,7	51,5	27,3	11,1	4	1,2	0,3	0,1
11,5	0	0	0	0	0,3	3,3	13,3	26,6	31,4	24,7	14,2	6,4	2,4	0,7	0,2	0,1
12,5	0	0	0	0	0,1	1	4,4	9,9	12,8	11	6,8	3,3	1,3	0,4	0,1	0
13,5	0	0	0	0	0	0,3	1,4	3,5	5	4,6	3,1	1,6	0,7	0,2	0,1	0
14,5	0	0	0	0	0	0,1	0,4	1,2	1,8	1,8	1,3	0,7	0,3	0,1	0	0
15,5	0	0	0	0	0	0	0,1	0,4	0,6	0,7	0,5	0,3	0,1	0,1	0	0
16,5	0	0	0	0	0	0	0	0,1	0,2	0,2	0,2	0,1	0,1	0	0	0

APPENDIX B

The table shows all parameters of wave theories, according to main inputs (H and T) of North Sea.

						Linear ther. Stokes' 3rd																			
Wave	**a**	**H**	**T**	**t**	**f**	**λ.**	**λ...**	**U_r**	**$H/(g*T^2)$**	**$d/(g*T^2)$**	**k**	**ω**	**θ**	**Cp.**	**C.**	**C...**	**Cg.**	**Cg.**	**H/λ**	**α**	**ω/k**	**H/d**	**λ.../d**	**d/λ...**	**$Pot_{sea\text{-}state}$**
1	0,35	0,7	3	2,25	0,333333	14,0517899	14,0517899	5,10361006	0,007928418	0,849473327	0,447145	2,094395	4,712389	1,420004	4,68393	4,68393	1,420004	0,5	0,049816	2,851873	4,68393	0,009333	0,187357	5,337398	0,7203
2	0,5	1	4	7	0,25	24,9809599	24,9809599	7,90366476	0,00637105	0,477828746	0,251519	1,570796	10,99557	1,53849	6,24524	6,24524	1,53849	0,5	0,04003	2,292354	6,24524	0,013333	0,333079	3,002287	1,96
3	0,65	1,3	5	13,75	0,2	39,0327498	39,0327498	11,4177849	0,005300714	0,305810398	0,160972	1,256637	17,27876	1,610192	7,80655	7,80655	1,610192	0,5	0,033305	1,907552	7,80655	0,017333	0,520437	1,921463	4,1405
4	0,8	1,6	6	22,5	0,166667	56,2071568	56,2071546	15,6298029	0,004530524	0,212368332	0,111786	1,047198	23,56194	1,657875	9,367859	9,367859	1,657875	0,5	0,028466	1,630549	9,367859	0,021333	0,749429	1,33435	7,5264
5	1	2	7	33,25	0,142857	76,503848	76,503561	18,5316281	0,004160686	0,156025713	0,082129	0,897598	29,84513	1,779237	10,92912	10,92908	1,779237	0,5	0,026143	1,497518	10,92908	0,026667	1,020047	0,980347	13,72
6	**2**	**4**	**8**	**46**	**0,125**	**99,9158337**	**99,9100591**	7,90148494	**0,00637105**	**0,119457187**	**0,062888**	**0,785398**	**36,12832**	**3,077387**	**12,48948**	**12,48876**	**3,077387**	**0,5**	**0,040036**	**2,29267**	**12,48876**	**0,053333**	**1,332134**	**0,750675**	**62,72**
7	3	6	9	60,75	0,111111	126,392775	126,342488	5,6157351	0,007550874	0,094385925	0,049731	0,698132	42,4115	4,072728	14,04364	14,03805	4,072728	0,5	0,04749	2,718932	14,03805	0,08	1,684567	0,593625	158,76
8	4	8	10	77,5	0,1	155,758288	155,505283	4,78542584	0,008154944	0,076452599	0,040405	0,628319	48,69469	4,878053	15,57583	15,55053	4,878053	0,5	0,051445	2,944996	15,55053	0,106667	2,073404	0,482299	313,6
9	5	10	11	96,25	0,090909	187,635613	186,78169	4,41854077	0,008424529	0,063183966	0,033639	0,571199	54,97787	5,568845	17,05778	16,98015	5,568845	0,5	0,053538	3,064601	16,98015	0,133333	2,490423	0,401538	539
10	6	12	12	117	0,083333	221,455369	219,323248	4,23074755	0,008494733	0,053092083	0,028648	0,523599	61,26106	6,198617	18,45461	18,27694	6,198617	0,5	0,054714	3,131745	18,27694	0,16	2,92431	0,341961	846,72
11	7	14	13	139,75	0,076923	256,547334	252,31837	4,113883	0,008444469	0,045238224	0,024902	0,483322	67,54424	6,80267	19,73441	19,40911	6,80267	0,5	0,055485	3,175826	19,40911	0,186667	3,364245	0,297244	1248,52
Average→														3,509463	12,39895										290,6715636

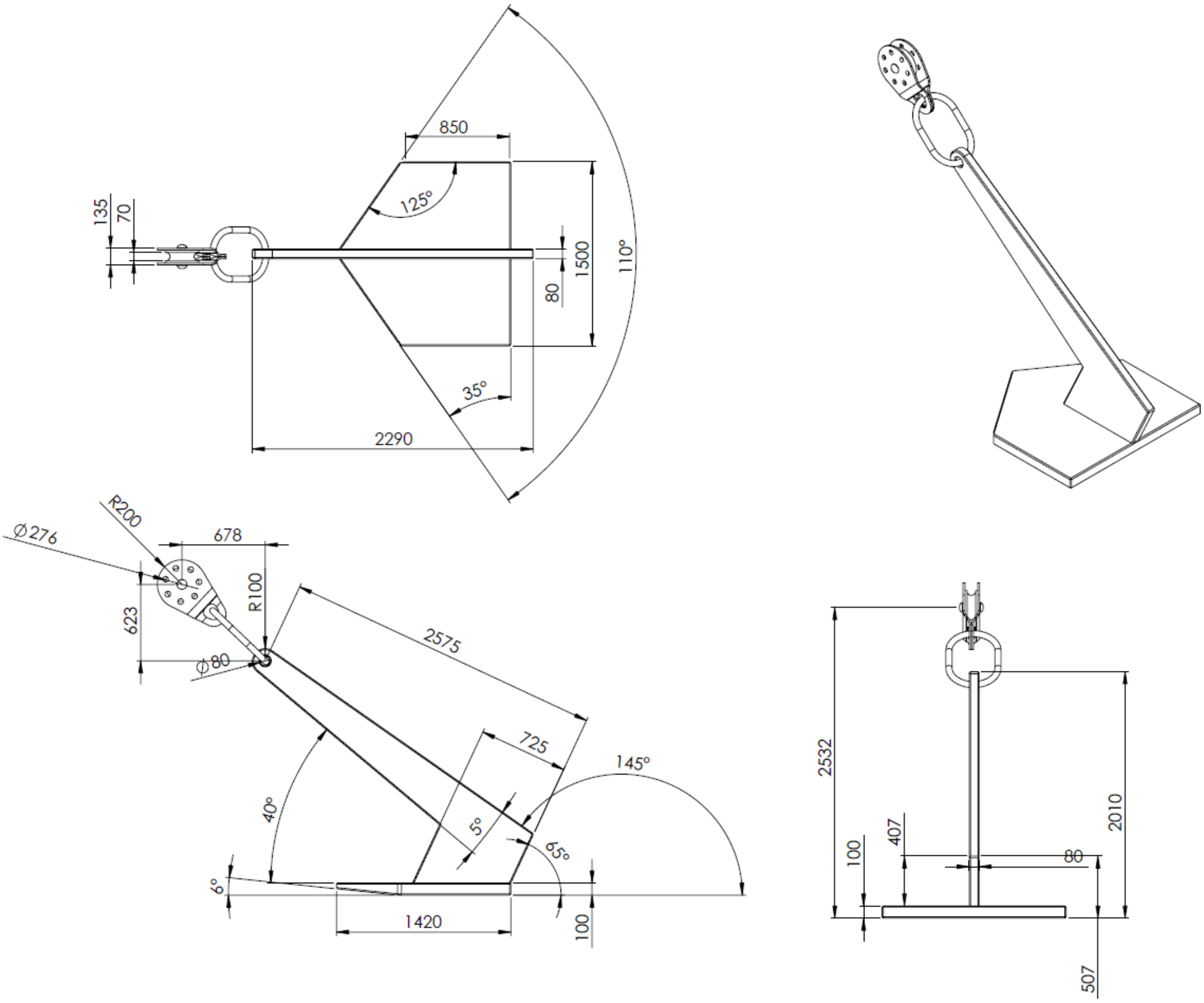
RELATIVE DEPTH	SHALLOW WATER $\frac{d}{L} < \frac{1}{25}$	TRANSITIONAL WATER $\frac{1}{25} < \frac{d}{L} < \frac{1}{2}$	DEEP WATER $\frac{d}{L} > \frac{1}{2}$
1. Wave profile	Same As →	$\eta = \frac{H}{2}\cos\left[\frac{2\pi x}{L} - \frac{2\pi t}{T}\right] = \frac{H}{2}\cos\theta$	← Same As
2. Wave celerity	$C = \frac{L}{T} = \sqrt{gd}$	$C = \frac{L}{T} = \frac{gT}{2\pi}\tanh\left(\frac{2\pi d}{L}\right)$	$C = C_0 = \frac{L}{T} = \frac{gT}{2\pi}$
3. Wavelength	$L = T\sqrt{gd} = CT$	$L = \frac{gT^2}{2\pi}\tanh\left(\frac{2\pi d}{L}\right)$	$L = L_0 = \frac{gT^2}{2\pi} = C_0 T$
4. Group velocity	$C_g = C = \sqrt{gd}$	$C_g = nC = \frac{1}{2}\left[1 + \frac{4\pi d/L}{\sinh(4\pi d/L)}\right]\cdot C$	$C_g = \frac{1}{2}C = \frac{gT}{4\pi}$
5. Water Particle Velocity (a) Horizontal	$u = \frac{H}{2}\sqrt{\frac{g}{d}}\cos\theta$	$u = \frac{H}{2}\frac{gT}{L}\frac{\cosh[2\pi(z+d)/L]}{\cosh(2\pi d/L)}\cos\theta$	$u = \frac{\pi H}{T}e^{\frac{2\pi z}{L}}\cos\theta$
(b) Vertical	$w = \frac{H\pi}{T}\left(1 + \frac{z}{d}\right)\sin\theta$	$w = \frac{H}{2}\frac{gT}{L}\frac{\sinh[2\pi(z+d)/L]}{\cosh(2\pi d/L)}\sin\theta$	$w = \frac{\pi H}{T}e^{\frac{2\pi z}{L}}\sin\theta$
6. Water Particle Accelerations (a) Horizontal	$a_x = \frac{H\pi}{T}\sqrt{\frac{g}{d}}\sin\theta$	$a_x = \frac{g\pi H}{L}\frac{\cosh[2\pi(z+d)/L]}{\cosh(2\pi d/L)}\sin\theta$	$a_x = 2H\left(\frac{\pi}{T}\right)^2 e^{\frac{2\pi z}{L}}\sin\theta$
(b) Vertical	$a_z = -2H\left(\frac{\pi}{T}\right)^2\left(1 + \frac{z}{d}\right)\cos\theta$	$a_z = -\frac{g\pi H}{L}\frac{\sinh[2\pi(z+d)/L]}{\cosh(2\pi d/L)}\cos\theta$	$a_z = -2H\left(\frac{\pi}{T}\right)^2 e^{\frac{2\pi z}{L}}\cos\theta$
7. Water Particle Displacements (a) Horizontal	$\xi = -\frac{HT}{4\pi}\sqrt{\frac{g}{d}}\sin\theta$	$\xi = -\frac{H}{2}\frac{\cosh[2\pi(z+d)/L]}{\sinh(2\pi d/L)}\sin\theta$	$\xi = -\frac{H}{2}e^{\frac{2\pi z}{L}}\sin\theta$
(b) Vertical	$\zeta = \frac{H}{2}\left(1 + \frac{z}{d}\right)\cos\theta$	$\zeta = \frac{H}{2}\frac{\sinh[2\pi(z+d)/L]}{\sinh(2\pi d/L)}\cos\theta$	$\zeta = \frac{H}{2}e^{\frac{2\pi z}{L}}\cos\theta$
8. Subsurface Pressure	$p = \rho g(\eta - z)$	$p = \rho g\eta\frac{\cosh[2\pi(z+d)/L]}{\cosh(2\pi d/L)} - \rho gz$	$p = \rho g\eta e^{\frac{2\pi z}{L}} - \rho gz$

Summary of linear (Airy) wave theory--wave characteristics.

(From the Shore Protection Manual Vol. 1 p. -2-17)

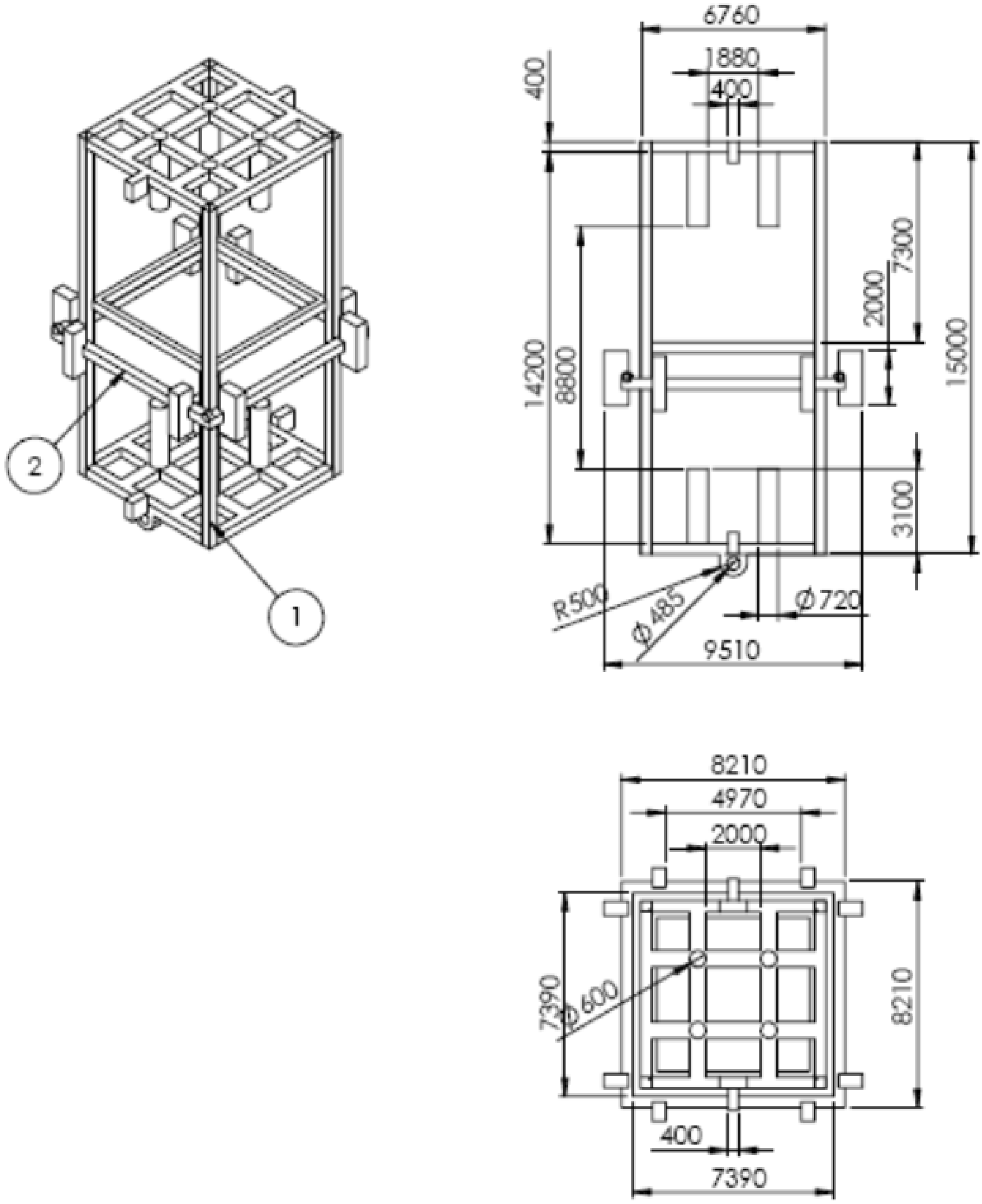
APPENDIX C

Draft Anchor Line and Snatch Block



APPENDIX D

Draft of Main Body and Body 2

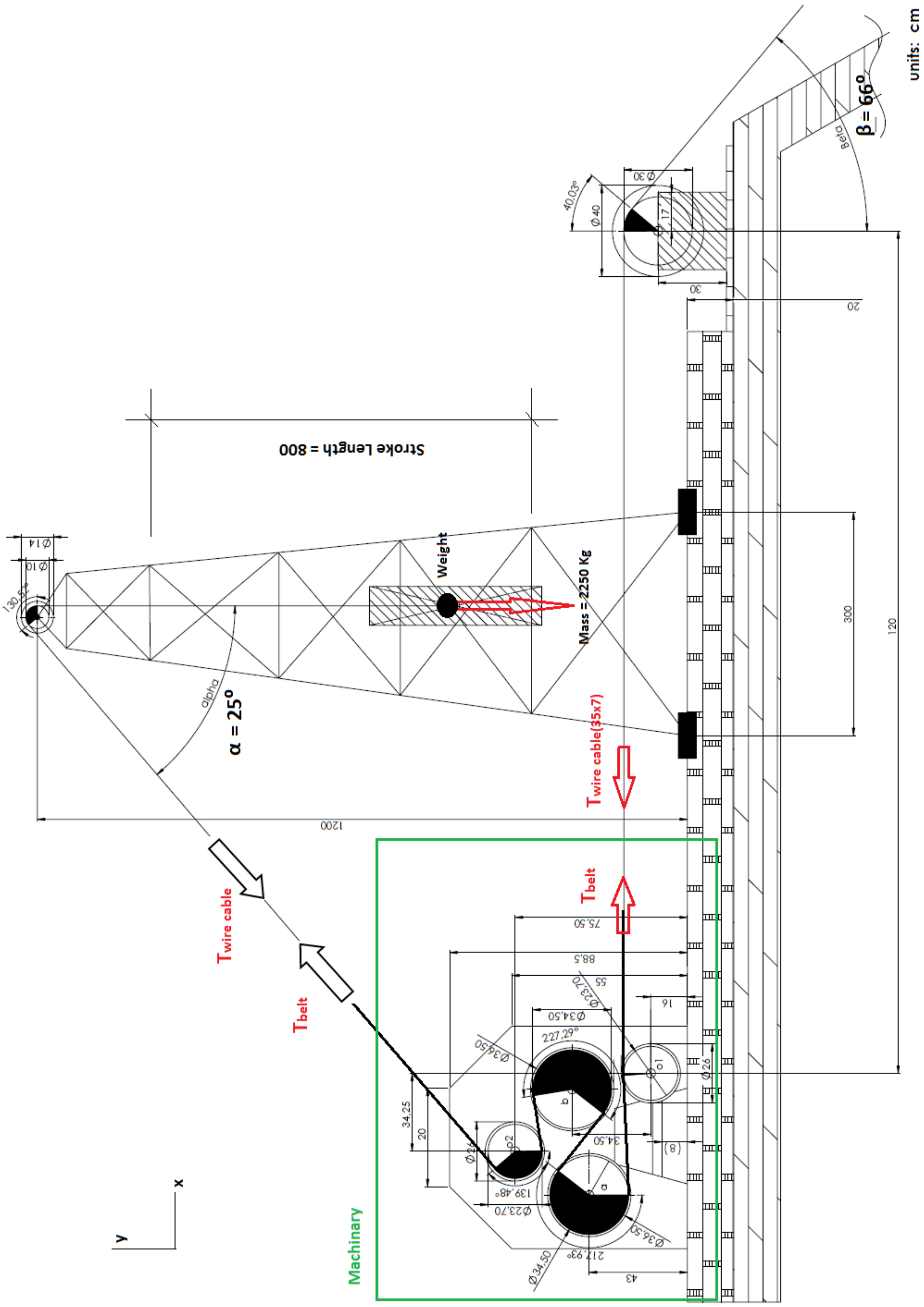


Units: mm

Main module(1) and Body(2)

APPENDIX E

Draft of Equipment Cape Verde on Land.



APPENDIX F

The table shows the values obtained from Cape Verde model.

				moment	Belts	radial	tangential	resultant
Point	**n_o(rpm)**	**Pot (Kw)**	**$D_{nominal}$ (mm)**	**$M_{(Nm)}$**	**$fk_{(fact.coeff.)}$**	**F_r(N)**	**F_t(N)**	**R (N)**
	Chain Tensor"Type H" =TENSOR "o1"							
1	120	**390,6371497**	237	31088,2065	1,75	**459108,5348**	**44809,4047**	461290,0708
	Chain Wheel"Type H"= DRIVER "a"							
2	120	390,32464	345	31063,33593	1,75	315135,2921	44774,85827	318300,236
3	120	177,0183631	345	14087,7114		172006,3	25905	173946,0729
	average→	**283,6715015**		**22575,52366**		**159030,3124**	**40524,23002**	164112,3197
	Chain Wheel"Type H"= DRIVER "b"							
4	120	177,0183631	345	14087,7114	1,75	142918,8113	25905	145247,5667
5	120	236,3765448	345	18811,63336		216086,43	32543,36	218523,2608
	average→	**206,697454**		**16449,67238**		**89849,46842**	**95329,69585**	130998,7706
	Chain Tensor"Type H" =TENSOR "o2"							
6	120	236,3765448	345	18811,63336	1,75	277808,9314	32354,84855	279686,6794
7	120	371,2172455	345	10731,07243		118800	17892	120139,7672
	average→	**303,7968952**		**14771,3529**		**199124,3131**	**122711,1576**	233898,5256

APPENDIX G

Representation of a Tower Model, Structural Profiles and Physical Properties.