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Exploration of Alternative Designs for the Pumping Structure of the Ocean Grazer

Master Thesis - Ocean Grazer

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Abstract

Renewable energy sources are becoming increasingly important to meet future energy demand. The Ocean Grazer is a concept developed by the Rijksuniversiteit Groningen that may contribute to achieving this future demand. The sealing system, which is keeping the seawater out of the base structure, is crucial for the functioning of the Ocean Grazer. The current sealing system is predicted to fail after just 193 days.

This study aims to explore alternative designs for the pumping structure which do not need the current sealing system. The problem has been tackled from a broad perspective at first where alternative pumping mechanisms have been evaluated. This analysis shows that only a diaphragm pumping mechanism might be a feasible alternative. Preliminary calculations, however, show that this is not feasible due to the large pressure differences that arise in the concept of the Ocean Grazer.

Next, the scope has been narrowed to the exploration of alternative sealing systems. A *local passive seal* alternative has been proposed where a flexible membrane separates the base structure from its external environment. A preliminary prototype has been created and all difficulties that arise from this concept, such as friction, elastic expansion, and fatigue have been explained. Also, calculations have been performed on the efficiency of the sealing system. From these calculations, it has been proven that the annulus area between the heaving rod and the walls can only be of a limited size to keep the efficiency loss reasonable. Next, simulations have been performed to analyze the elastic expansion of this design when EPDM is used. From these simulations, it has been proven that the material will fail to meet some of the requirements.

More research should be done on a hybrid material that is reinforced with a stronger material that can deal with the pressures applied to the material. This enhanced material might be similar to the structural membrane created by Trelleborg for the Symphony wave energy converter.

Acknowledgement

After 21 weeks of hard-working, I am proud to present my thesis "*Exploration of Alternative Designs for the Pumping Structure of the Ocean Grazer*". This thesis has been performed for the master study Industrial Engineering & Management at the Rijksuniversiteit Groningen. The past 21 weeks were challenging and hard, but also instructive and sometimes even fun. I have learned many new things during my thesis, not only theoretical but also organizational and practical. Lessons I will remember for the many years to come. The result lays in front of you on which I am extremely proud.

First of all, I would like to thank my first supervisor A. Vakis for the freedom he gave me during this project and all the feedback he provided me with during our meetings. Next, I would like to thank my second supervisor A.A. Geertsema for the time he spent on listening to the struggles I faced and for providing me fresh new directions I could pursue. I would also like to thank W. Prins for the many interesting, sometimes confronting, but always useful chats we had the last weeks, providing me new insights that always made sense. I would also like to thank K. Oenema for the many sparring sessions we had regarding both our projects, which were always helpful. Last but not least I would like to thank the coffee machine which was able to provide me with new energy, a fresh look, and the sometimes much-needed motivation to continue, every single day. Thanks.

I hope you enjoy reading my report and that my effort will be a contribution in helping the Ocean Grazer project forwards.

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Acronyms

EPDM Ethylene Propylene Diene Monomer.

MP^2*PTO* Multi-pump, multi-piston, power take-off.

OG Ocean Grazer.

RuG Rijksuniversiteit Groningen.

WEC Wave Energy Converter.

Part I
Thesis Introduction

'Doubt is the beginning of wisdom' - Aristotle

1 Introduction

1.1 A Brief History

Since 1880 electricity is becoming more and more important in society's day to day life. Nowadays, almost everyone is using electricity daily. According to the World Bank almost 90% of the world's population had access to electricity in 2016 (1), and in that same year more than 22.000tWh was consumed globally (2). There are various methods to generate electricity and traditionally this is done by using coal or gas. Such power plants are affordable, reliable and relatively safe, but are also producing a tremendous amount of greenhouse gas emissions. On top of that, these fossil fuels are exhaustible.

The energy transition is becoming more and more important, due to the growing awareness among the global society about global warming, especially after the Paris Agreement of 2015. In this agreement, 196 countries agreed that the increase in the global average temperature should remain well below 2 °C. The main force to realize this is to end the use of fossil fuels for energy production. In order to end the use of fossil fuels for the production of energy, several actions can be taken. The most predominant action is to switch from coal stations to renewable energy sources, such as wind and solar energy. Solar energy, for example, will be the main source of energy in the Middle East and North Africa in 2050 (3). Another consequence of the energy transition is that electric vehicles are becoming more and more popular. Governments are playing an important role in this increase in popularity by making it financially attractive for consumers to drive electric vehicles. In addition, governments can also take a more rigorous step. A good example is the government of Norway which has stated its ambition to only allow the sales of electric cars by 2025 (4).

So, on the one hand, the use of fossil fuels should be diminished for the generation of energy while, on the other hand, the demand for electricity is only increasing. On top of that, peak demands will only increase which puts pressure on the flexibility of the electricity grids. This creates a problem with how electricity should be generated and stored. Other sources of energy are possible to meet future energy demand. One could think of nuclear energy as a potential energy source. This energy source, however, raises much resistance among society as there is always a risk something goes wrong with catastrophic consequences. Other options are, as mentioned already, renewable energy sources. Although wind and solar energy are becoming more important, it might not be enough. When thinking of renewable energy, one potential source is not mentioned that often: the energy of the ocean.

Already at the end of the 19th century, people pursued the idea of generating power from the ocean in the United States of America (5). Although a few successes were achieved and many thousands of patents were filed, people lost interest due to many fatal failures. As the need for renewable energy is increasing, the idea of generating energy from the waves has made its way back into the interest of the people and is now widely studied among many companies and academic research groups. In North America, Japan and Europe alone over 1000 WECs have been patented (6). According to literature, using waves as a source of renewable energy offers significant advantages compared to other methods of energy generation. Some of these advantages are described below:

- WECs have a limited negative environmental impact in use. In general, offshore devices have the lowest potential impact (7)
- Waves can travel large distances without much energy loss (8)
- WECs can generate power up to 90 percent of the time, compared to 20-30 percent for wind and solar power devices (8)
- Sea waves offer the highest energy density among renewable energy sources (9)

Although there exist many different designs with a wide variation between them, WECs can be classified into three predominant types: *Attenuator*, *Point absorber*, and *Terminator*. Attenuators lie parallel to the predominant wave directions and are 'riding' the waves while a terminator device has the principal axis parallel to the wavefront and physically intercept waves (8). A point absorber is a WEC that possesses small dimensions relative to the incident wavelength. A floating structure moves up and down either on the water surface, or below it. Wave direction is less important for a point absorber device because of the relatively small size.

1.2 Ocean Grazer

The OG is one specific type of WEC currently developed and researched by the Rijksuniversiteit Groningen (RuG). The concept of the OG has been developed since 2014. Since then over 60 bachelor, master, and PhD students have graduated having worked on relevant topics regarding the OG so far (10). Although the first concept of the OG focused on a large scale WEC for the Atlantic Ocean, the current concept, the Ocean Grazer 3.0, is focusing to be implemented in the North Sea. Not only is the Ocean Grazer 3.0 able to generate energy from the waves, it is also able to store the energy so it can be used to generate electricity when needed. Figure 6 is showing this last concept of the Ocean Grazer.

Figure 1: *Ocean Grazer 3.0*

The floater blankets are moved upwards and downwards due to the heave of the waves. These floater blankets are attached to the pumping mechanisms through a rod. The rod is going into the internal underwater base structure of the OG through a seal to make sure that there is no ocean water leaking into the base structure. Due to the heave of the ocean waves, the internal working fluid is pumped from the base reservoir into a flexible bladder. Important to note is that the pressure inside the base structure is equal to the atmospheric pressure, while the internal fluid stored in the flexible bladder is exposed to the hydro-static pressure of the ocean, which depends on the depth. This creates an overpressure between the base reservoir and the flexible bladder which is used to store potential energy. In order to generate electricity, a valve is opened through which the internal fluid flows from the bladder reservoir back into the base reservoir. During the process, the internal fluid goes through a turbine which generates the electricity.

Figures 2 and 3 are showing a simplified picture of both the pumping subsystem and the flexible bladder reservoir subsystem. In figure 2, B_1, B_2, and B_3 are the floaters, while P_1, P_2, and P_3 are the pumps and T_1 is the turbine. A more detailed description of the pumps is given in section 2.3.1.

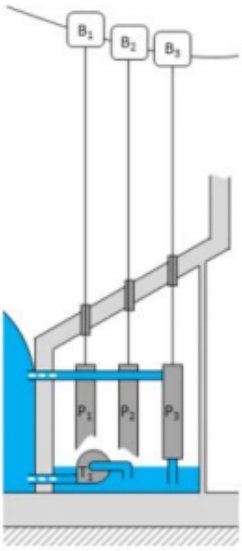


Figure 2: *Schematic drawing of the Ocean Grazer 3.0 including the floaters, pumping mechanism and turbine*

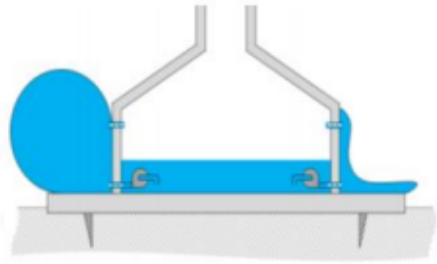


Figure 3: *Schematic drawing of the flexible bladder used to store the potential energy of the Ocean Grazer 3.0. The left part shows a completely filled flexible bladder while the right shows an empty bladder*

2 Problem Analysis

In this chapter the problem of this thesis is analyzed. First, the problem owners are explained. Second, the business problem that they have is mentioned and elaborated on. In the consecutive sections of the chapter the business problem will be analyzed in order to check whether it is a *reality problem*. This is done through empirical analysis. Based on literature three types of problems can be distinguished: a *perception*, *target*, and *reality* problem (11). When the problem owner has an incorrect perception of the system and its performance, it is called a *perception* problem. A problem that is based on unfeasible targets is classified as a *target* problem. Both these problems are not desired and therefore it is necessary to verify before starting a project whether the problem owner has is indeed a *reality* problem. This chapter ends with the problem statement, which is based on the business problem and the empirical analysis.

2.1 Problem Owners

For this project there are two problem owners, namely prof. dr. A. Vakis and MSc. M van Rooij. Both are closely linked to the OG project, but they are fulfilling different roles within the project. M. van Rooij is the chief technical officer (CTO) of the company created for the OG project. His responsibility is related to all technical aspects of the OG. As CTO, M. van Rooij is responsible that all technical components of the OG work accordingly and therefore he owns the problem when some technical aspects are not or partly not functioning. A. Vakis, on the other hand, is an associate professor at the Faculty of Science & Engineering at the RuG, and specialized in mechanical engineering, tribology lubrication, advanced manufacturing, renewable energy, and biomechanics. As an associate professor, A. Vakis is supervising most projects related to the OG, such as this project. He is responsible for the academic content related to the OG. Both persons are problem owners as the project is scheduled to have a working prototype in 2020. Therefore, much research is needed and many solutions to problems they face should be devised.

2.2 Business Problem

During the first meeting for this thesis, both M. van Rooij and A. Vakis explained the problem they face. Currently, a specially designed pumping system, which will be explained in section 2.3.1, is utilized to harvest energy from the ocean waves. Much effort has been put into the optimization of this system, while other completely different pumping systems never got much attention. For the current pumping system, seals are crucial to prevent any leakage of seawater into the structure of the OG. In a previous thesis this sealing is designed but similarly to the pumping system, other alternatives are not evaluated at this moment in time. A. Vakis, however, has the feeling that both the pumping and sealing system has certain downsides. Therefore he believes that it is worth investigating alternatives to eventually improve the performance of the OG. He proposes an alternative design where a flexible membrane can be used as a seal or pumping system.

2.3 Empirical Analysis

As already mentioned in the introduction of this chapter, the empirical analysis is needed to validate the business problem. In the next three sections, the business problem is analyzed on different aspects to validate that the problem is indeed a *reality problem* which is worth solving.

2.3.1 Current Pumping System

The pumping system which is used in the current concepts of the OG is the so-called $Multi-pump, multi-piston, powertake-off(MP^2PTO)$ WEC (12). The MP^2PTO WEC is the core of the OG and for a single OG device, multiple MP^2PTO WECs are employed. This system uses a modified point absorber design where the floating buoys actuate linear hydraulic pumps that are pumping the internal fluid such that hydraulic head is created, see figure 2. Much research is done on this pumping mechanism and based on literature it can be concluded that this system is able to extract energy from waves ranging with a height from 1 to 12 meter and periods of 4 to 20 s (13). On top of this, the efficiency is believed to be around 99% for a piston-cylinder separation of 100 μm (12). Three different pistons are included in the MP^2PTO WEC which can be activated depending on the energy level that a wave contains. The differences are based on the size, and therefore the energy required to lift the piston. With these three different pistons, in total 7 different combinations can be achieved. All different combinations are used for different sea states (with different amounts of energy) such that all energy from the waves is harvested. So for small waves, only the smallest piston is activated, as bigger pistons require more energy and therefore will not be lifted by such waves, while for the largest waves all three pistons are activated. The activation and de-activation of the pistons is regulated by a regulator which is located inside the base structure of the OG (14).

2.3.2 Current Sealing System

Crucial for the functioning of the MP^2PTO WEC are seals. Keeping the seawater out of the base structure of the OG is crucial as ocean water itself is highly corrosive and bio-fouling and therefore destructive for most components inside the reservoir structure. In a previous thesis, a proposal for this sealing system is given (15). This sealing system fulfills many requirements, such as having an efficiency of roughly 98%, and no leakage. The seal designed in this thesis is based on an already existing reciprocal sealing system where several types of seals are used to prevent any leakage of ocean water into the base structure of the ocean grazer. A final design of the seal is given based on a performance analysis where different seal setups are taken into consideration. Although the sealing system meets almost every requirement, early simulations show that the seals need to be replaced every 193 days. Although a tool is created which enables changing the different types of rings, there is no plan on how to reach the seals in order to change them.

The seals itself are not expensive, but hiring a dive crew that changes every seal, one by one, might be. As there is no other current solution available for changing the sealing systems, it is assumed that divers are necessary. According to research, a diver costs around $80 an hour (16). The working depth is the most important parameter when considering costs. Diving deeper than 40m can make a dive three to four times more expensive due to regulations (16). In addition, when

diving to a depth of 20-25m, a four-diver team can dive no more than 3h/day (16). A quick cost calculation can be made for the replacement of the seals. The following assumptions are made:

- An average windmill park in the North Sea consists out of 70 windmills
- One OG incorporates 70 seals
- It takes 15 minutes to replace one complete seal
- Costs for one diver is $80 per hour
- Due to depth, a four-diver team can dive no more than 3h/day
- Due to depth, total expenses increase threefold

A quick calculation shows that the operation costs per day are roughly $7,500 (assuming a working day of eight hours) and that in one day only 12 seals can be replaced. This entails that it takes almost 6 days to replace all the seals of one OG. Total costs for replacing all the seals of one OG is therefore estimated at roughly $45,000. This replacement operation needs to be done every 193 days and when the life expectancy of the OG is 20 years, the seals are replaced 38 times in total. Total costs, therefore, are $1,710,000 over the life expectancy of the OG. Also, two four-diver teams are needed full time to replace every seal on time for one average windmill park.

2.3.3 Design Process

Every product, whether small or big, has gone through different design steps before it is brought to market. Although literature suggests many different design steps, it suffices to say that a generic product development cycle includes phases such as the concept, conceptualization, preliminary design, and detailed design phases. It may come as no surprise that the costs of design changes in a later stadium are more expensive compared to earlier stages and therefore should be avoided as much as possible. Many graphs found in literature support this claim. One of those can be seen in figure 4 (17). From here it is concluded that when the product development reaches further phases, the costs of design changes increases exponentially.

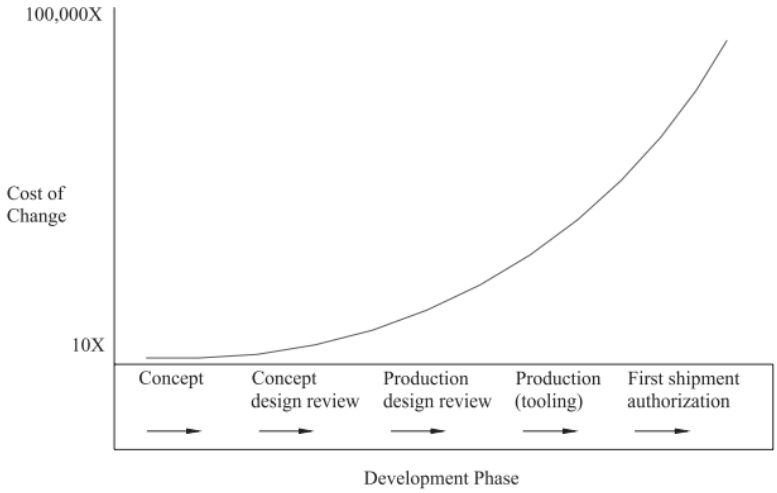


Figure 4: *Cost of changes made in the design related to the different product development phases (17)*

Not only the costs changes when a product development process reaches further stages, but the ability to impact a project changes also. In the first stages of product development, the ability to impact a project is at its highest, while this is decreasing over time. This is perfectly shown in the Mac Leamy graph. This graph clearly depicts the need to make changes as soon as possible in the development process (18).

2.4 Problem Definition

The current pumping system is showing great potential, but it cannot be concluded that it is a flawless concept, as it is now. Although early models and simulations are showing that the MP^2PTO WEC is working well, the sealing system can be seen as a weak spot. Based on the empirical analysis, it became clear that the seals need to be replaced every 193 days. As shown in the cost calculation in section 2.3.1, it can become extremely expensive to replace each seal that often. Another important aspect to note is that the OG is still in a relatively early stage of its development process. As can be seen in section 2.3.3, this is the best stage to make changes in the design as the costs will only increase when it is done later in time. Consequently, when considering alternative concepts for the OG this is the best time to analysis alternatives.

Based on the empirical analysis, we can thus conclude that the business problem is indeed a *reality* problem. It can not be seen as a *perception* problem as it is concluded that there are indeed some downsides to the current design. Therefore, based on the problems that are found above, the following problem statement is devised:

The current pumping structure of the OG has rods moving in and out of the base structure which requires a sealing system that needs expensive frequently reoccurring maintenance

3 System Description

Based on the problem devised above, taking the whole system of the OG as the focus system for this thesis would be too wide. Instead, only a specific part of the OG is taken as the focus system for this thesis. The system that is taken into consideration for this project is the *pumping structure* of the OG. The *pumping structure* is defined as a combination of the pumping system (which on itself exists out of a combination of pumping mechanisms, valves, and seals, etc.) and the sealing system that is needed to prevent leakage of ocean water into the base structure of the OG, see figure 5.

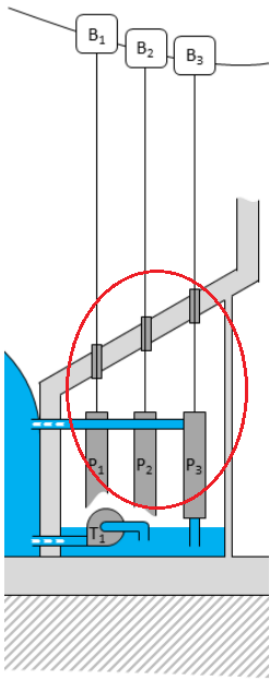


Figure 5: *Focus system for this thesis composed of the pumping subsystem and the sealing subsystem of the OG*

4 Research Goal

In the report so far the context is explained, the problem is elaborated on and analyzed, a problem statement is given and the focus system has been described. In this chapter the goal of this project is given first and in the second section the research questions are stated.

4.1 Goal Statement

Based on the first part of this report, the following goal is devised:

Find an alternative pumping structure that does not need the current sealing system such that the frequently reoccurring maintenance is prevented

4.2 Research Questions

Now that the goal of this thesis is clear, the following research questions are defined in order to reach that goal:

1. What are the requirements for the pumping structure?
2. Are there pumping mechanisms suitable for the OG that do not depend on the current sealing system?

   On which pumping criteria should the pumping mechanisms be evaluated on?
3. Are there alternative sealing systems applicable for the OG?

4.3 Scenarios

It is currently not yet known what the exact location of the OG will be. It is also imaginable that several OG's are located at different locations in the North Sea. Therefore, three different scenarios are evaluated in this thesis. All three scenarios represent a location somewhere in the North Sea, but differ in depth. Increasing depth means that the hydro-static pressure acting on the OG also increases. This is beneficial for the storage potential. After all, increasing the depth entails that the pressure difference between the flexible bladder and the inside structure also increases. As a result, more potential energy can be stored. On the other hand, increasing depth results in more severe conditions. More pressure is acting on all parts of the OG and therefore it should be designed accordingly. For this thesis depths ranging from 30m, 40m, and 50m are evaluated.

5 Stakeholder Analysis

It is important to define the stakeholders before starting with a project as it gives clarity on who to take into consideration and who not. Freeman defines a stakeholder as any group or individual who can affect or is affected by the achievements of the organization's objectives (19). Wieringa, on the other hand, has a stricter definition of a stakeholder and defines it as a person, group of persons, or institution affected by treating the problem (20). According to Wieringa, stakeholders are the source of goals and constraints of the project. For this thesis a stakeholder is defined as a person who is affecting, or affected by treating the problem.

Both M. van Rooij and A. Vakis are the most important stakeholders which are already explained in section 2.1, as they are the *problem owner*. Next to these two problem owners, no other stakeholders are worth mentioning. Although other students are working on projects that might be an influence on this project, their results will not be taken into consideration for this project.

Part II
Finding an Alternative Pumping Mechanism

6 Requirements

- *The pumping structure should pump internal working fluid from the base structure to the external flexible bladder.* By far the most obvious requirement, but also one of the most important requirements. Due to the large pressure differences, the pumping structure should be strong enough to pump the working fluid. Also, the pumping structure should be as efficient as possible. To set a quantitative requirement on this efficiency is a bit premature, as the aim of this project is to analyze potential alternatives. Due to time constraints it is most likely not possible to design an alternative in such depth that it is possible to make accurate statements regarding the efficiency.

- *The pumping structure should not have any leakage of ocean water into the internal base structure of the OG.* Saltwater is potentially catastrophic for mechanical components. Consequently, it is of utmost importance to keep the seawater out of the base structure in order to keep the maintenance low and life expectancy high.

- *The pumping structure should be adaptive and able to abstract all energy of varying wave types.* All waves differ not only in height but also in length. As a consequence, every wave has a different energetic potential. For the OG to work properly, all energy should be harvested from every wave and therefore the pumping structure should be adaptable.

- *The pumping structure should have a life expectancy of at least 20 years (maintenance is allowed).* Similar to the requirements stated for the older versions of the OG, the life expectancy should be 20 years. This does not mean that no maintenance at all is allowed. Nevertheless, this should be kept to a minimum.

- *No cavitation should occur within the pumping structure.* Cavitation is a phenomenon where small vapor-filled cavities occur in a liquid owing to a rapid change of pressure. These cavities, also called bubbles or voids, collapse when it is subjected to higher pressures and generate an intense shock wave (21). These shock waves can cause significant damage to components inside the pumping structure. Therefore, cavitation should be prevented at all costs.

- *The pumping structure should be able to handle waves varying from 1-6m.* Based on literature it is concluded that the maximum wave height in the North Sea is 8m on average (22). However, such wave heights are expected to occur only once every 10.000 years. Together with problem owner M. van Rooij it is decided that the OG should handle waves up to 6m in height. Waves smaller than 1m in height do not contain enough energy and are therefore insignificant. This minimum wave height is similar to the requirements of previous designs of the OG.

- *The pumping structure should go in safe mode when the waves are higher than 6m.* When waves are higher than the maximum wave height, the OG should stop working, to protect the pumping structure. Therefore a safety mode should be included in the design of the OG.

- *The pumping structure should be able to handle waves varying from 4-12s.* Based on literature it is proven that for a wave of 1m in height the wave period is 4s (23) while the wave period for the maximum wave height is 12s (22).

- *The pumping structure should be robust, i.e. withstand any external disturbances* The OG is located underwater and in the current designs it is not possible to lift the OG to the surface at will. Therefore the pumping structure should be robust in a high degree and should withstand all different environmental states (bio-fouling, currents, etc.)

7 Pump Evaluation Criteria

Before describing all available pumping mechanisms it is useful to first describe the criteria on which they are evaluated. To obtain these criteria, it is useful to first define which functions the pumping mechanism should fulfill. Note that the difference between a pumping *mechanism* and a pumping *system* is of great importance. The pumping mechanism is only the working principle of various pumps while the pumping system can incorporate multiple pumping mechanisms, sealing systems, etc. This chapter first describes the main functions that the pumping mechanism should fulfill. The second part of this chapter deals with describing all criteria that are devised based on the functional decomposition.

7.1 Functional Decomposition

A functional decomposition subdivides the main functional requirements into its respective sub and sub-subfunctions. Such kinds of diagrams contribute to understanding whether functions are connected and where interface connections might be (24). A functional block diagram will help to separate what the product needs to do versus how it gets done. Based on the functions it is possible to set certain criteria that are used to evaluate different pumping mechanisms.

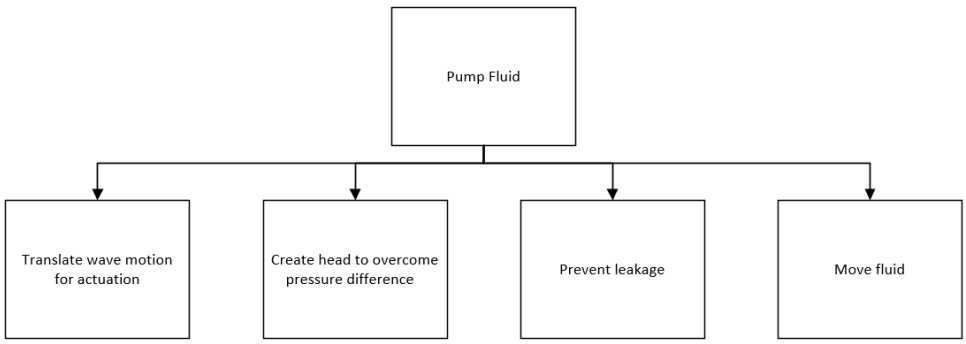


Figure 6: *Preliminary functional decomposition for a pumping mechanism*

The main function of a pump mechanism needs to fulfill is to pump fluid. To pump fluid, several sub-functions need to be performed. First, the reciprocating movement of the waves should be translated such that it can actuate the pump. Second, the pump mechanism should move the fluid from the base reservoir into the flexible bladder. As the pressure in the flexible bladder is higher compared to the pressure in the base reservoir, the pump mechanism should be able to overcome this pressure difference. Last, the pump mechanism should prevent leakage. Not only should the pump mechanism make sure that it is not leaking, the pump mechanism should also make sure that water is not flowing back into the reservoir when the pumps are not working. The four evaluation criteria used to evaluate all pumping mechanisms are described in detail in the next sections.

7.2 Pressure

The pressure is the first criteria on which each pumping mechanism is evaluated. As explained already, the pumping system needs to pump water from the reservoir, with atmospheric pressure, to a flexible bladder where the pressure is equal to the hydro-static pressure. It is assumed that the depth at which the OG will be located ranges from 30m to 50m. Therefore the total pressure difference between the reservoir and the flexible bladder is at least 3bar. Although current research focuses on how to increase the pressure in the flexible bladder, by adding air to the bladder, which might improve the efficiency, this is not incorporated into this thesis.

7.3 Sealing

As the current sealing system is one of the biggest downsides of the current pumping system, all alternative pumping techniques should be evaluated based on whether they would need a similar (or the same) sealing system. As the current pumping system is working well, it might not be useful to dive deeper in creating an alternative pumping system which is based on a pumping mechanism that will also need the current sealing system.

7.4 Driving Mechanism

Waves in itself have a reciprocating movement. As the buoys are moved because of the waves, the main movement is upwards and downwards. Although the movement of the buoys are not only vertical but also horizontal, meaning that in one cycle the path the buoy has traveled is elliptical, the OG only considers the vertical movement for the generation of energy.

There are many solutions for transforming a reciprocal movement into a rotary type of movement. This requires, however, extra moving components that are either placed outside or inside the base structure. Outside the structure is not desirable as it is then placed in seawater, where it is exposed to severe conditions. Translating the heaving motion into a rotating movement inside the base structure entails that the current sealing system should still be used, which does not solve the problem that we aim to solve.

7.5 Irregular Actuation

The condition at the North Sea changes over time. Sometimes the weather is calm, and the waves are not that high. Consequently, not much energy can be harvested from such waves. Sometimes, however, the weather is rough and the waves are much higher. In such a scenario waves contain much more energy that can be harvested. In both scenarios, the pumping mechanism should be able to work. The energy provided by the buoys should always result in a pumping movement, independent of the sea state.

8 Pumping Mechanisms

Pumps exist in many different types, shapes, and working principles. A literature study is performed to find all different pumping mechanisms. Based on this study it is concluded that all pumping mechanisms are divided into two main categories: *positive displacement pumps*, and *roto-dynamic pumps*. Both categories can further be divided into two subcategories. A clear overview of all pumping mechanisms is given in figure 7. In this chapter all different pumping mechanisms that are shown in figure 7 are described shortly.

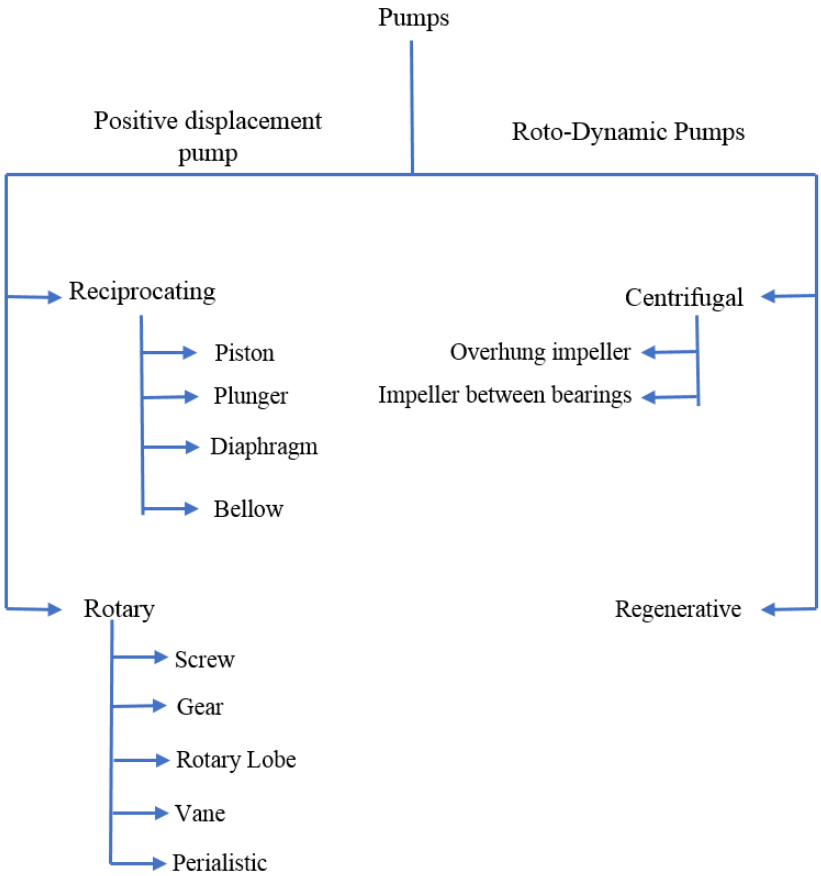


Figure 7: *Overview of all different pump mechanisms that are evaluated in this thesis*

8.1 Positive Displacement Pumps

Although centrifugal pumps are used for many applications, they are not always preferred. Sometimes a positive displacement pump should be used instead. According to literature there are some key application criteria that can lead to the preference of a positive displacement pump over a centrifugal pump. Some of these application criteria are: *high pressure*, *sealless pumping*, *low flow*, *high efficiency*, *self-priming*, and *low shear* (25). The working principle of different types of positive displacement pumping mechanisms is briefly explained in the next few sections.

8.1.1 Screw Pump

A screw pump is a positive displacement pump that uses one or multiple screws to move fluids. One of the best-known screw pumps is the Archimedes screw pump which is invented by the Greek mathematician and scientist Archimedes. It is probably the oldest pump to transport liquids (26).

A screw pump is a special type of a rotary positive displacement pump. In a screw pump the flow through the pumping elements is perfectly axial (27). As the liquid is forced to move circumferen-

tially in all other rotary pumps, the screw pump has several advantages. A screw pump is applied in a diverse range of markets such as the navy, marine, industrial oil burners, and crude oil industries. A screw pump can handle liquids width a range of viscosities, ranging from molasses to gasoline. Due to the relatively low inertia of the rotating parts in such a pump, screw pumps can operate at higher speeds than other similar pumps. According to literature, the maximum pressure of a single screw pump is 2000psi, which can increase until 4500psi when multiple screws are used (21).

Screw pumps are normally divided into single- or multiple-rotor types. A single-screw is also called a progressive cavity pump and has a rotor thread that is eccentric to the axis of rotation, see figure 8. Multiple-screw pumps, on the other hand, employ one driven rotor in a mesh with one or more sealing rotors, see figure 9.

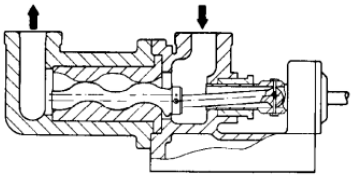


Figure 8: *The single-screw pump (27)*

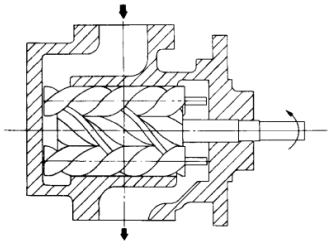


Figure 9: *Multiple-screw pump (27)*

8.1.2 Gear Pump

The gear pump is, next to the screw pump, one of the oldest pump of any type. Drawings for gear pumps dating back from the 16th century exist (27). Also, the gear pump is one of the most common type of a rotary pump as it can be used in a wide variety of applications. The fluid is moved by two gears that mesh to provide the pumping action, where one gear drives the other. As the gears make physical contact, it becomes a seal between the inlet and the outlet (27).

Two types of gear pumps exist and are classified as *internal* and *external* gear pumps. In an external gear pump, the gear teeth mesh about their outside diameter. Bearings on both sides of the gears are needed so that the gears never contact the outside wall. External gear pumps are capable of dealing with pressures up to 2500psi. These kinds of pumps are applied in marine and API-related industries (27).

Similarly to the external gear pump, internal gear pumps move liquid by the meshing and unmeshing of gear teeth. Internal gear pumps, however, have one larger gear where the gear teeth cut internally on the major outside diameter meshing with and driving a smaller externally cut gear. The advantages of an internal gear pump are that they have only a few moving parts, are relatively cheap and only utilize one seal (21). This type of pump generally will not work with abrasives inside the fluid that is being transferred. The internal gear pump has a maximum pressure capacity of roughly 17bar, which is lower in comparison with the external gear pump (27).

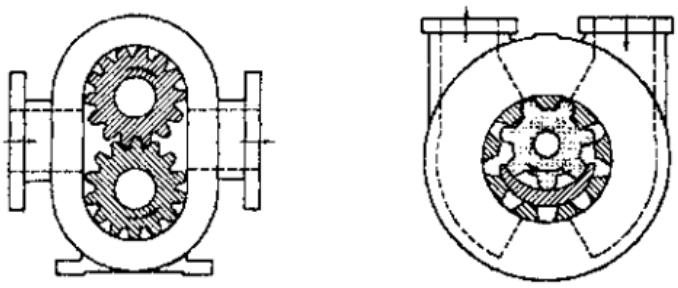


Figure 10: *Typical external (left) and internal (right) gear pump (27)*

8.1.3 Rotary Lobe Pump

There are many similarities between a gear pump and a lobe pump. In a lobe pump, the liquid is carried between the rotor lobe surfaces. As the rotor lobe surfaces are constantly in contact with each other, they provide a constant seal, as is done by the gears in the before mentioned pump. Unlike the gear pump, though, one lobe cannot drive the other. Therefore both lobes need to be driven such that they remain in sync with each other and need additional timing gears to do so (21). Similar to other mentioned pumps, lobe pumps can be either single- or multiple-lobe pumps. Lobe pumps are mainly used for pharmaceuticals and food applications. The maximum pressure where a rotary lobe pump can deal with is 450psi, according to literature (21).

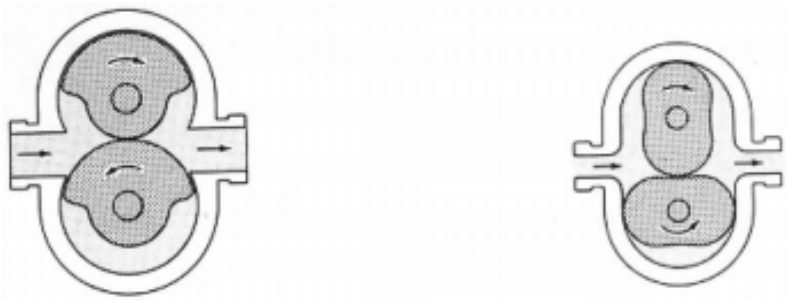


Figure 11: *Typical single-lobe (left) and multiple-lobe (right) pumps (27)*

8.1.4 Vane Pump

A vane pump is another rotary type positive displacement pump. In a vane pump, several vanes are connected to a rotor which is driven by a motor. There exist two basic types of vane pumps. The most common one is the rigid sliding metal vane pump. The other basic type consists of flexible or elastomeric vanes. In both types, the drive shaft is not exactly in the middle (21). All rigid vane pumps consist out of movable sealing elements, which move inwards and outwards such that they make constant contact with the outer surfaces, see figure 12. In this way, a constant seal is formed. For the flexible or elastomeric vane type, no movable sealing elements are needed as the flexible vanes deform such that a tight seal is formed between the vanes and the external housing (21). This can be seen in figure 13. The main advantage of vane pumps is that they have a rather simple structure, can deal with low viscosity liquids, can operate with mildly erosive liquids, and are self-priming (21). However, they cannot deal with highly viscous liquids and also are not able to handle fragile solids (21). Vane pumps are commonly used for low-pressure transfers of gasoline, kerosene and similar light hydrocarbons. (27). Sliding vane pumps can have a maximum pressure of 200psi which is much more compared to the flexible impeller. The latter type has a maximum pressure of only 60psi (21).

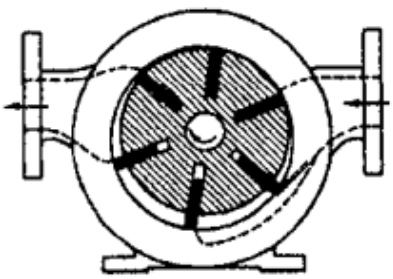


Figure 12: *Typical external vane pump (27)*

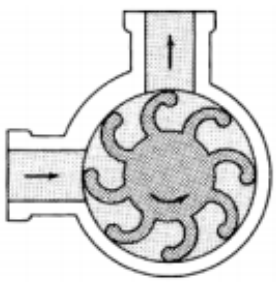


Figure 13: *Flexible vane pump (27)*

8.1.5 Perialistic Pump

A perialistic pump, also called a flexible tube pump or simply a hose pump, has a flexible tube located inside a circular housing. This tube is mostly made out of rubber, but can also be made out of other polymers (28). Rollers or cams, which are attached to the rotor, squeeze the tube during the movement. Due to this squeezing the liquid is drawn through the pump, see figure 14. This process is similar compared to when someone swallows, which is called peristalsis. This explains the name of this kind of pump (21). Again a seal has formed as the rotor contacts the outer surface which prevents liquid from leaking. In addition, before the roller reaches the outlet, the other roller closes the inlet of the tube. According to literature, a perialistic pump can have a maximum pressure of 220psi (21).

A perialistic pump has multiple advantages (28). The fluid that is being transported cannot be contaminated, and the pump mechanism does not need any cleaning. Also, as the liquid does not leave the tubing, these pumps are immune to abrasive media and chemicals. Another advantage is that these pumps are self-priming. There are, however, also several disadvantages. Due to wear the tubes need to be changed frequently and also calibration is needed to sustain accuracy. Also, the tubes are prone to leaking. Lastly, the flow rate is sensitive to varying differential pressure conditions. Perialistic pumps are used in many applications such as open-heart bypass pump machines, concrete pumps, pulp and paper plants, medical infusion machines, and pharmaceutical production (28).

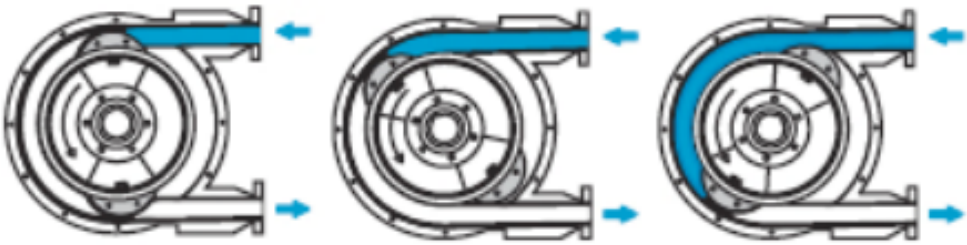


Figure 14: *Perialistic pump scheme flow (28)*

8.1.6 Piston Pump

In a piston pump one or more reciprocating pistons are used to move and pressurize the fluid. Although there exist many different types of pistons pumps, where the MP^2PTO WEC is one specific type, it suffices to say that fluid is pumped because of an increasing and decreasing pumping chamber volume. As the piston is operating within a cylinder, a sealing is needed on the outside of the piston such that there is no leakage within the pumping system (21). According to literature, a piston pump can work with a pressure of no more than 5000psi (21).

Piston pumps can be single or double-acting. In the case of a double-acting piston pump, the liquid is discharged during both the forward and return motion of the piston. In a single-acting piston pump, on the other hand, the liquid is only discharged during one stroke of the pump (27).

8.1.7 Plunger Pump

There are many similarities between a piston pump and a plunger pump. The main difference, however, is that a plunger is used instead of a piston. In general, plunger pumps are applied in higher pressure applications. Plunger pumps are capable of dealing with the highest pressures when considering all positive displacement pumps. In some very special applications, pressures up to 50,000psi can be achieved. Also, plunger pumps run at higher speeds than piston pumps.

8.1.8 Diaphragm Pump

Diaphragm pumps are a special type of reciprocating positive displacement pumps. A diaphragm pump features a flexible membrane and check valves that are used to move the liquid into and out of the pumping chamber (27). The reciprocating motion of the pump causes the flexible membrane to flex back and forth which results in a fluctuation of the volume of the pumping chamber. When the volume of the pumping chamber increases, the inlet valve opens and water is flowing into the chamber, see figure 15. When the flexible membrane moves back and thereby decreases the volume of the pumping chamber, the inlet valve closes while the outlet valve opens, pushing the water out of the pumping chamber, see figure 16. Such pumps can deal with a maximum pressure of 17,500psi (21).

Due to the flexible membrane, a diaphragm pump has several special properties (29). First, no seals are needed as the flexible membrane itself serves as a seal. Therefore, a diaphragm pump is well suited in applications where it is of utmost importance that there is no leakage. Second, a diaphragm pump does not need any lubrication as there are no mechanical components such as cylinders pistons, etc. Third, it is a simple and inexpensive construction meaning that no expensive and complicated maintenance is needed.

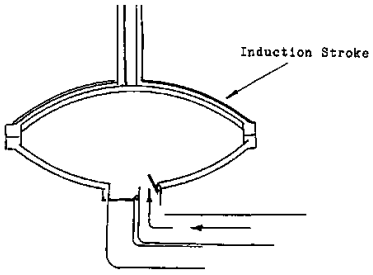


Figure 15: *Induction stroke of a diaphragm pump (30)*

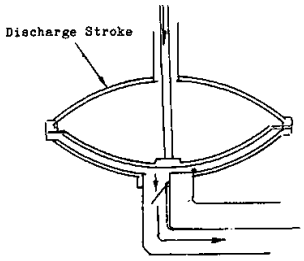


Figure 16: *Discharge stroke of a diaphragm pump (30)*

8.1.9 Bellow Pump

Similarly to the diaphragm pump, a bellow pump moves fluid by increasing and decreasing the pumping chamber volume. Different from the diaphragm pump, a bellow pump has a different shape. As there is no literature available on the performance and working principles of bellow pumps, it will be excluded from this research.

8.2 Roto-Dynamic Pumps

8.2.1 Centrifugal Pumps

A centrifugal pump is a device that is used to transfer liquid of various types. As the name already suggests, this type of pump relies on the centrifugal force to pump a liquid. Although there are many different types of centrifugal pumps, where the impeller can be either placed between bearings or at the end of the shaft, the working principle will remain the same. Therefore, no deeper explanation will be given on different sub-types of a centrifugal pump, as is done for the positive displacement pumps.

So, stripped of all the nonessential details, a centrifugal pump consists of an impeller, enclosed by a casing, which is attached to a shaft that rotates. All centrifugal pumps are driven by a motor. When the impeller rotates, the liquid is forced towards the discharge side of the pump. As the liquid leaves the centrifugal pump, a void or reduced pressure area is created at the inlet of the impeller. As this pressure is lower than the pressure at the pump casing inlet, the liquid is forced into the pump (21). When non-condensable gas, such as air, is located in the pipelines before the pump, the pressure reduction at the inlet of the impeller merely causes the gas to expand. As a result, no liquid is forced into the impeller inlet and no pumping action occurs. Only when the gas is eliminated, the pumping mechanism will work again. The process of removing the non-condensable gas is called *priming*.

Centrifugal pumps occur in many sizes and there exist pumps that can deal with several thousands psi of pressure. Therefore it is hard to say something on the maximum pressure. However, literature suggests that one should choose a positive displacement pump over a centrifugal pump when high pressure, high efficiency, and low velocities are required (21).

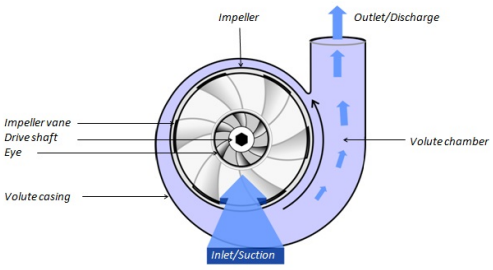


Figure 17: *Working principle of a centrifugal pump (31)*

8.2.2 Regenerative Pumps

A regenerative pump, which is also called a turbine or peripheral pump, is another type of roto-dynamic pump. In a regenerative pump, the impeller is the only moving part and has vanes on both sides of the rim that can rotate in a ringlike channel which is located in the casing of the pump. The fluid is not only able to discharge freely from the tip of the impeller, but is also able to recirculate back to a lower point of the impeller diameter (32).

This re-circulation, also named regeneration, is the main difference between the centrifugal pump and the regenerative pump. As the fluid can recirculate, an increased head is developed. However, as a regenerative pump relies on close clearances, it can only pump liquids that do not contain solid particles. Regenerative pumps are increasingly of interest as they are low in costs, compact and can deliver high heads (33). However, they are not efficient, which is usually lower than 50% (32).

9 Pump Mechanism Evaluation

In this chapter all before mentioned pumping mechanisms are evaluated against the criteria described in chapter 7. This is done based on the Pugh's method (24). In this method, one concept is selected as the reference concept on which all other concepts are compared with. For each criterion the concept is better (+), worse (-), or similar (R) to the reference concept. In this way, all concepts are rated on every criterion. After this rating is done, all +'s, -s and S's are summed and recorded at the bottom of the matrix. The matrix is given in figure 18. The evaluation of each criterion is explained in more detail in this chapter.

Alternative Pumping Mechanisms											
	Positive Displacement Pumps									Roto-Dynamic Pumps	
	Rotary					Reprocicating					
Criteria	Screw pump	Gear pump	Rotary lobe pump	Vane Pump	Perialistic Pump	Piston pump	Plunger Pump	Diaphragm pump	Bellow Pump	Centrifugal Pump	Regenerative Pump
Pressure	-	-	-	-	-	R	+	+	x	-	-
						E					
Sealing	S*	S*	S*	S*	S*	F	S	+	+	S*	S*
						E					
Driving Mechanism	-	-	-	-	-	R	S	S	S	-	-
						E					
Irregular Actuation	-	S	S	S	S	N	S	S	S	-	-
∑ +	0	0	0	0	0	C	1	2	1	0	0
∑ -	3	2	2	2	2	E	0	0	0	3	3
∑ S	1	2	2	2	2	NA	3	2	2	1	1
Total	-3	-2	-2	-2	-2	NA	1	2	1	-3	-3

+	better performance
-	worse performance
S	Similar performance

Figure 18: *Pugh's method where all pumping mechanisms are evaluated on all before mentioned criteria. The reference pumping mechanism is the piston pump as this is used in the current pumping system*

9.1 Pressure

As mentioned in chapter 7, the pumping mechanism should overcome the pressure as the fluid is moved from the base structure (with atmospheric pressure), to the flexible bladder (with hydrostatic pressure). The pressure depends on the location where the OG is placed and it is concluded that the pumping mechanism should overcome a relative pressure of 3bar, 4bar, and 5bar for each situation described earlier.

From the previous chapter, it can be concluded that almost all pumping mechanisms have a lower maximum pressure compared to the reference case. However, it is also concluded that almost all pumping mechanisms are able to overcome the required pressure differences regarding the three scenarios, except the flexible impeller vane pump. With a maximum absolute pressure of 4bar, it is not suited for a depth of 50m. In addition, literature suggests that it is a maximum pressure for which the flow rate will be minimal. So even with lower pressure it might not a viable alternative for the OG.

9.2 Driving Mechanism

The movement that is used to actuate the pumping system for the OG is purely vertical. Therefore the pumping mechanism should be able to translate this vertically reciprocating movement of the waves into a pumping movement. From the literature study it is concluded that both the roto-dynamic pumps and the rotary positive displacement pumps rely on a circular actuation. As the wave movement is mainly vertically reciprocating, an additional mechanism is needed to translate this reciprocating movement of the waves into a circular movement. As the aim is to have no mechanical components underwater outside the base structure of the OG, this additional mechanism should be placed inside the base structure. However, this entails that the same sealing system is still required. Only the reciprocating positive displacement pumps are actuated by a vertically reciprocating movement, and therefore more suited as an alternative for the current pumping system of the OG.

9.3 Sealing

This might be the most important criterion. An additional mechanism that translates the reciprocating movement of the waves into a circular movement to actuate the rotary positive displacement pumps and roto-dynamic pumps should be placed inside the base structure. This means that the movement of the waves should still be transferred from the environment into the base structure. Therefore, the same sealing system is still required. As the aim is to find an alternative pumping structure that does not require the sealing system, these alternatives are all not suited.

Regarding the reciprocating positive displacement pumps, the sealing system is still required for the plunger pump. For the diaphragm and bellow pumps, however, no sealing system is needed. One of the main features of such pumps is that the diaphragm and bellow functions as a seal. Therefore these two options are better alternatives for the OG when looking at the sealing criterion.

9.4 Irregular Actuation

Waves are highly irregular in terms of height, length, intensity, etc. Consequently, it is crucial that the alternative pumping mechanism can deal with this irregularity. For almost all evaluated alternatives it is concluded that they are as suited to deal with this irregularity as the reference mechanism, except for the screw pump and both roto-dynamic pumps. For the roto-dynamic pumps, the maximum pressure that the pump can deliver is dependent on the energy that is provided to the pump. When the waves are small and do not contain much energy, it might be the case that the pump is not able to overcome the pressure anymore, and no fluid will be moved. The same argument holds for the screw pump. When there are no waves or hardly any waves, the water will flow back along with the screws.

Although leaking might be prevented by adding additional valves, it makes the alternatives less desirable compared with the alternatives where no additional valves are needed.

9.5 Conclusion

From the analysis given in this chapter it can be concluded that only the diaphragm pump mechanism is a suitable alternative that is worth investigating further. As the diaphragm pump does not need the current sealing system, functions as a sealing itself, is able to overcome the required pressure differences, keeps working with irregular actuation, and directly transfers the reciprocating movement of the waves into a pumping movement, it might be a better pumping mechanism for the OG which can solve the problem stated earlier.

10 Feasibility of the Pumping Mechanism

In the previous chapters it is concluded that there is only one viable alternative pumping mechanism for the OG. This does not entail, however, that other pumping systems (composed out of pumping mechanisms described in chapter 8) cannot be used in the OG. As the aim of this research is to find a feasible alternative pumping structure where the current sealing system is not needed, we can only conclude that, except for the diaphragm pumping mechanism, there is no pumping mechanism that can contribute to this goal.

In this chapter the feasibility of a diaphragm based pumping system is evaluated. Preliminary calculations are given to find out whether such a pumping mechanism can be applied underwater. The main difficulty here is that the pumping system should overcome the pressure difference. This pressure is dependent on the depth at which the OG is located. Firstly, the mildest case is evaluated, assuming that the depth is 30m. In addition to the depth, the acceleration is also of great importance. To find the maximum acceleration of a wave, the wave is simulated with a sinusoidal function of the following form:

$$z_w(t) = \frac{H_w}{2} sin(\frac{2\pi}{T_w} - \frac{\pi}{2}) \tag{1}$$

where Z_w is the vertical displacement of the waves, H_w is the height of the waves, and T_w is the period of the wave. Taking two times the derivative of this function gives the acceleration of the wave. For a wave height and wave period of 6m and 12s respectively, the maximum acceleration is determined to be $0.6\frac{m}{s^2}$.

As the OG is situated underwater, a large mass of water is placed on top of the diaphragm. This mass should also be moved as the diaphragm is moving upwards. The maximum pumping force available in the North Sea is estimated at 0.1MN[1]. In order to calculate the maximum diameter that the diaphragm can have based on the maximum available force, newton's second law is used:

$$F = ma \tag{2}$$

By rewriting these equations, the maximum area of the diaphragm can be calculated as follows:

$$F_b = \rho_{sw} d A_d (g + a_w) \tag{3}$$

$$A_d = \frac{F_b}{\rho_{sw} d (g + a_w)} \tag{4}$$

$$d_d = \sqrt{\frac{4F_b}{\pi \rho_{sw} d (g + a_w)}} \tag{5}$$

where d_d, is the diameter of the diaphragm, F_b is the force of the buoy, ρ_{sw} is the density of seawater, A_d is the area of the diaphragm, d is the depth of the sea, g is the gravitational constant, and a_w is the maximum acceleration of the waves. Filling in the values assuming that the OG is located at a depth of 30m, the maximum diameter of the diaphragm is only 66cm.

Evaluating existing designs of diaphragm pumps, where the diaphragm itself is made out of specific types of rubbers, the ratio diaphragm length to stroke length is determined to be roughly 1 to 4 (34; 35). This implies that the maximum stroke length in the case of the OG is 16.5cm. Due to this geometrical constraint, a diameter of 66cm will never be enough to facilitate the vertical movement of 6m, which is one of the requirements given in chapter 6.

Note that this calculation is a rather extreme simplification of reality. In reality, the pumping force is not only needed to lift the fluid mass on top of the diaphragm. Instead, drag forces, viscosity, etc. should also be taken into consideration. However, this only makes the implementation of a diaphragm pump less feasible and is therefore left out. As a diaphragm cannot be used at a depth of 30m, it can also not be used at greater depths as the conditions are more severe.

[1] For the calculations of this maximum pumping force the reader is referred to chapter 14.3.1

In addition to the geometrical constraints, more constraints make a diaphragm mechanism not suitable for the OG. When the volume of the diaphragm expands, fluid is sucked into the pumping chamber. By decreasing the volume in the pumping chamber, the fluid is pressed out. For both the suction and discharging of fluid, energy is required. A wave, however, is only providing energy to the pumping structure in the upstroke, while at a downstroke no energy is provided. For the current pumping system, the pistons are moving down due to its own weight during the downstroke, while all the pumping (suction and discharging) is done during the upstroke. For a diaphragm, the energy of the wave can be either used to increase the volume of the pumping chamber (suction), or to decrease it (discharging). It is not possible to do both simultaneously. One can choose to use the energy of the wave to increase the pumping volume (suction), but then the fluid will not be discharged during the downstroke as the pressure of the flexible bladder is equal to the pressure in the pumping chamber. When using the energy of the wave for the discharge, there is no energy that can lift the flexible bladder.

Based on the calculations and explanations given above, it is concluded that a diaphragm pump is not suitable for the OG. The pressure difference between the inside of the base structure and the outside is too large for a diaphragm to work, given the maximum power that is provided by the buoys. Therefore it is concluded that there exists no alternative pumping mechanism (and therefore also no alternative pumping system) which does not need the current sealing system.

Part III
Alternative Sealing Systems

11 Sealing System Alternatives

Until now it is proven that there are no alternative pumping systems that can be used to solve the problem that is devised in chapter 2. Almost all evaluated mechanisms are not suited to prevent the usage of a reciprocal sealing system that is currently designed for the OG. However, one mechanism, the diaphragm mechanism, had some promising potential. Nevertheless, it is proven that this mechanism cannot be used in an underwater situation as too much of the available force from the buoy is needed to overcome the pressure difference.

There are, however, other solutions imaginable to prevent the usage of the current sealing system that allows the heaving motion of the rod moving in and out of the base structure of the OG without leakage. Instead of considering the pumping system, it is also possible to narrow the view and only look at the sealing system instead. Currently, a reciprocal seal is used, but previous research has shown that this design fails already after 193 days. A reciprocal seal is not the only seal imaginable. The diaphragm mechanism evaluated earlier incorporates one aspect that requires additional attention. In such a mechanism the flexible membrane is not only used to increase and decrease the pumping chamber, but it is also used to separate the pumping fluid from the external environment. Using such a flexible membrane for the OG might prevent the need for the current sealing system, as the flexible membrane is separating the external environment from the inside structure of the base structure.

In this chapter a few different passive sealing system alternatives are devised and explained. Based on an early evaluation the most promising alternative is chosen and designed further in the next chapters of this report.

11.1 Global Passive Seal

The first alternative incorporates the idea of sealing the entire top of the base structure with a flexible membrane. Note that the flexible membrane must be much larger than the total area of the top of the base structure, as the flexible must move up only locally. This idea is depicted in figure 19.

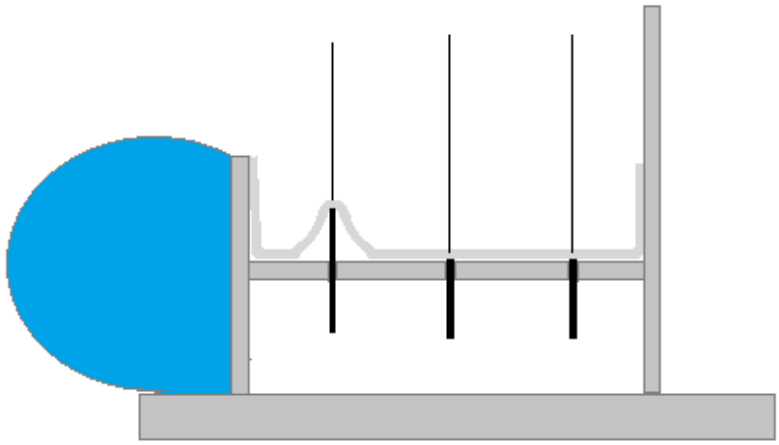


Figure 19: *Schematic drawing of the base structure of the OG where a global passive seal is incorporated. Due to the waves, the global passive seal will be lifted locally on the places where the buoy is connected to the seal. Underneath the seal, a bar is connected to the piston such that it is actuated by the waves. Note that the pumping system is not drawn in this figure*

In this design, the buoys are connected on top of the flexible membrane. Underneath this connection point, the pistons are connected to the flexible membrane. In this way, there is a constant separation between the inside structure of the OG and the external environment. The pressure working on top of the flexible membrane is equal to the hydro-static pressure while the inside pressure is equal to the atmospheric pressure. As a result, the flexible membrane will be pressed against the bar that connects the flexible membrane and the pistons so that a cone is formed. As the flexible membrane is large enough, multiple pistons can be moved up simultaneously while still separate cones are formed. By doing so the situation where a large area is lifted, as was the case in the evaluation of a diaphragm pump mechanism, is prevented.

11.2 Local Passive Seal

The local passive seal is an alternative to the above described *global passive seal*. In this alternative, a flexible bladder is only incorporated locally where the rod is heaving up and down. Similar to the first alternative, the flexible membrane is separating the inside of the base structure from the external environment. The flexible membrane is attached to both the base structure and the heaving rod. The length of the membrane allows the rod to move up and down freely for the required wave heights.

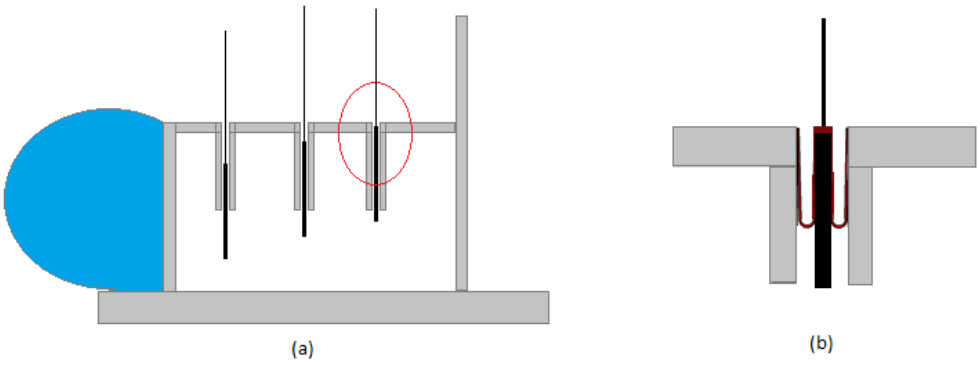


Figure 20: *(a) Schematic drawing of the local passive seal alternative. (b) Schematic drawing of one local passive seal. The seal is attached on top of the base structure and the rod is able to move freely in the vertical direction*

Due to the pressure difference between the external environment and the inside environment, the flexible membrane is pressed both to the sidewalls of the base structure and the heaving rod. Only the part of the membrane that lies horizontally between the base structure and the rod is pressed down due to the hydro-static pressure. When the rod is moving upwards, the flexible bladder will move along while the bend moves also upwards. As a result, there is a constant annulus area that is lifted upwards.

11.3 Harmonic Seal

The third alternative is based on the bellow pump described in section 8.1.9. Similarly to all other alternatives, the external environment is separated from the inside structure of the OG. Different from the other two alternatives is the shape. Instead of a flexible membrane, a structure is created that can shrink and expand vertically. It is comparable with an accordion. This alternatives also makes sure that the current sealing system is not needed anymore.

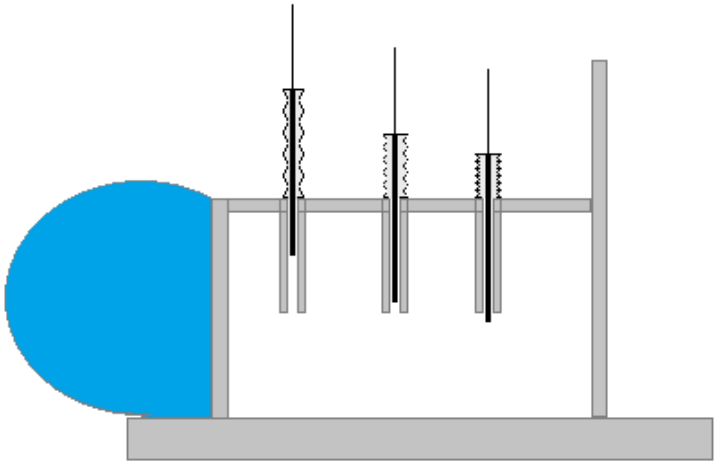


Figure 21: *Schematic drawing of a harmonic seal alternative. Also in this alternative, the buoys are connected on top of the harmonic seal. A rod under each harmonic seal is attached to the pistons. Note that the pumping system is not drawn in this figure*

11.4 Alternative Selection

Of the three alternatives described above, one should choose the best alternative for further analysis. When considering the three alternatives above, one can immediately conclude that a part of the force from the buoy is needed to lift the passive seal. As the hydro-static pressure is working on all flexible membranes, a force is needed to lift the membrane. As a result, the efficiency of the pumping structure will decrease. Therefore it is of utmost importance to keep the frontal area of the flexible membrane as low as possible.

Considering the three alternatives above, this frontal area that is lifted upwards is minimal for the second alternative. In the local seal alternative, the frontal area is constant during the heaving motion of the wave, as both the diameter of the rod and the diameter of the entrance in the base structure are fixed. This is not the case for the other two alternatives. Considering the global seal one can see that the diameter of the cone will increase during the upstroke. In addition, when two neighboring buoys are moving upwards, it might create a larger area that is lifted. For the third alternative, the frontal area will vary regarding the position in which it is in. On the top halves, the hydro-static pressure will press the seal downwards, while on the bottom halve it is the other way around. This design, however, is more complicated compared to the previous two and is therefore not preferred.

Crucial for the functioning of the seal is that the flexible membrane is not folded. Cracks in the material will occur more likely when it is folded completely. This might be problematic in the *global passive seal*. As mentioned above, it should be prevented that two simultaneously moving buoys also lift the area between their attachment points on the flexible membrane. To prevent this, more material is needed such that cones are formed locally only. In the downstroke, however, this material will then be pressed down due to the hydro-static pressure and will most likely fold completely. In the *local passive seal*, on the other hand, this complete folding will not occur. As the distance between the heaving rod and the side of the base structure is fixed, the material has a fixed minimum bending angle.

Based on the reasoning given above, the *local passive seal* is chosen for further investigation.

12 Local Passive Seal

12.1 Seal Requirements

- *The seal must work for temperatures between 6-16 degrees.* The temperature of the surrounding is of utmost importance for the functioning of the seal. The brittle-ductile transition temperature is only one example showing that temperature is crucial when designing a new artifact.
- *The efficiency loss cannot be greater than 2%.* The reciprocal sailing system already designed in a previous thesis had the requirement that the mechanical efficiency loss could not exceed 2%. Although it is proven that this seal fails after 193 days, this efficiency target was met. Therefore the aim is to also have a mechanical efficiency of 98% for the seal designed in this thesis. Note, however, that this is not a hard requirement. When it can be proven that the lifetime of the seal is significantly higher compared to the reciprocal seal, while the efficiency is lower, it can still be a viable alternative.
- *The seal must work for at least* 10^8 *cycles.* It is assumed that the average wave period for waves is 4-5s. This entails that in 10 years roughly 10^8 waves have passed an OG. As a result, the seal is heaving constantly.
- *The seal must be strong enough to withstand the hydro-static pressure for all three scenarios.* The material should not only not fail when subjected to the three different pressures depending on the depth, it should also be strong enough to withstand the many cycles that the material has to go through.
- *No complete folding must occur in the flexible membrane.* As a result of complete folding, stress concentrations will occur in the seal. As the seal should withstand many cycles, failure will occur more likely at the locations of the stress concentrations, and should, therefore, be prevented.
- *Seal must allow a vertical movement of 6m.* As already mentioned in the requirements for the pumping structure, the pumping structure should be able to harvest energy from waves with a height of up to 6m
- *The seal must be robust and withstand external disturbances.* The seal is constantly situated below the water surface. As no leakage can occur, the seal must withstand all external disturbances, such as the salinity, biofouling, etc.

13 Material Selection

Based on the requirements given in the previous chapter, it can already be concluded that there is only one class of material that can be used, a polymer. Both a ceramic or a metal are not suited due to the lack of flexibility and/or resistance to seawater.

There are three types of polymers, thermoplastics, thermosets and elastomers (36). The difference between the first two types is the response to changing temperatures. A thermoplastic softens when it is heated and becomes harder when it is cooled. This process is completely reversible and may be repeated. When the temperature is raised, secondary bonding forces are diminished which allows the relative movement of adjacent chains when a stress is applied. Also, thermoplastics are relatively soft. Examples of such polymers include polyethylene and polystyrene (36). Thermosets are permanently hard network polymers that do no soften upon heating. Such polymers have covalent cross-links between adjacent molecular chains. When the polymer is heated, these bonds anchor the chains together to resist the chain motions, and therefore the material does not soften when heated. Thermosets polymers are generally harder and stronger when compared to thermoplastics. Examples of thermosets include vulcanized rubbers and epoxies. Elastomers are polymers that allow considerable elastic behavior due to their molecular structure. Elastomers are lightly cross-linked polymers and the molecular chains between these links are free to move. Examples of elastomers are rubbers and silicones.

Considering the requirement of the *local passive seal* that no complete folding is allowed, two options for the material can be used:

- As the diameter of the rod is smaller than the diameter of the entrance in the base structure, the local passive seal can have the shape of a cone. The top diameter can be equal to the size of the entrance in the base structure, while the diameter of the bottom can be equal to the size of the heaving rod. This is only possible when the material is not too elastic. In operation, the seal should never touch the heaving rod as the diameter is increasing and therefore bigger compared to the diameter of the rod
- The seal can also have the diameter of the heaving rod. Due to the hydro-static pressure, the seal will then be pressed towards the walls of the base structure. In this way, no complete folding will occur. The material should be elastic to allow for this movement

The first option has the advantage that much stronger materials can be used. However, a simple preliminary experiment already showed that this option is not feasible. As the material bends, while it is also round, the diameter needs to increase. As a result, strain arises in the material that hinders the movement. In figure 22, the experiment is shown. A simple prototype of the seal is made and from this experiment it is concluded that it is not only difficult to bend a round material inwards. It is also extremely difficult to move the bend upwards or downwards.

Based on this early experiment it can already be concluded that the material used for the *local passive seal* should be a flexible elastic polymer, i.e, an elastomer.

Figure 22: *A simple experiment showing the feasibility of using a flexible non-elastic material for the local passive seal. It can be seen that the bend is problematic as the material is not able to keep the radius constant (a triangle-shaped figure is composed). In addition, it is not possible to move the bend upwards, or downwards*

Much research has already been done on which type of polymer is suited to be used in an underwater situation. Also within the OG research group, various polymers are evaluated to be implemented. From this evaluation there is one elastomer that is well suited for such applications, EPDM. Besides, EPDM is already used in similar applications. In Rampsol, a small city in the Netherlands, a huge inflatable rubber barrier is placed to protect Zwartsluis and Zwolle against raising water. For this huge project, EPDM is chosen as the best material for the barriers (37). However, in both situations EPDM is used where the pressure difference is not that significant as it is in this case. Therefore, research is needed to analyze whether EPDM is strong enough for such an application.

The aim is not only to prove whether specifically EPDM can be used for this application as there is a wide range of different sorts of EPDM, all having different properties. Instead, the aim is to analyze whether an elastomer can be used for the *local passive seal* in a more general sense. As many elastomers have material properties that are in the same order of magnitude, early simulations given in the next chapters are used to evaluate the difficulties and feasibility in a more general sense. The material properties that are used for EPDM are given in table 1. Due to convergence problems in the simulations, the maximum value for the Young's Modulus was chosen instead of the average value.

Table 1: *Material properties used in the COMSOL simulations which will be discussed in section 14.5.1 and 14.6.1*

Material Property	*Minimum Value*	*Maximum Value*	*Value used*
Density [kg/m^3]	800	1000	875
Young's Modulus [MPa]	2	10	10
Poisson's Ratio [-]	0.48	0.495	0.4875
Tensile Strength [MPa]	5.4	20.2	12.8

13.1 Maximum Allowable Stress

As stated in the requirements, the material should withstand 10^8 cycles. Due to this many cycles, fatigue may occur. Similar to metals, fatigue occurs at stress levels that are quite lower than the tensile strength. Fatigue behavior for a polymer is not as simple in comparison with metals. The behavior is much more sensitive to loading frequency. High frequencies and/or relatively large stresses can cause localized heating (36). As a result, failure may be due to a softening of the material rather than a result of typical fatigue processes. Research done to predict the fatigue properties of elastomeric materials and components are at a very early stage and currently partly of an empiric nature. Despite such differences and problems, fatigue data is plotted in the same manner for both polymers and metals and the shape is also similar. However, there are only a few Wöhler curves in literature due to the inordinate amount of time required to collect the data (38). As a result, a precise understanding of the durability of rubber does not yet exist.

For this thesis, however, it is of utmost importance to take this fatigue failure into account due to the many cycles that the material has to go through. Therefore an estimation is made on the maximum allowable stress that is tolerated for the material. This estimation is based on two aspects: a review of a typical Wöhler curve found in literature for a similar material, and the maximum allowable stress determined by Dunlop for their production of EPDM made conveyor belts.

As there is no literature available on a SN curve for EPDM, an SN curve is used of a similar material. Figure 23 is showing an SN-curve for a moderately filled synthetic polyisoprene, a material designed to be similar to natural rubber in structure and properties, which shows similar behavior compared to EPDM. Although the absolute values given in this graph are of no importance, it is interesting to see how much the maximum allowable stress can be after many cycles. CES EduPack 2014 gives a yield strength for synthetic polyisoprene of 25MPa. From this graph it can be seen that the allowable stress after 1E8 cycles (assuming that the line decays constantly) can approximately be 1MPa. So from this graph it can be concluded that the maximum allowable stress can only be 4% of the tensile strength.

Dunlop is a producer of conveyor belts. Many of their products are made out of elastomers, some of them are made of EPDM. Ing. R van Oijen, Manager Application Engineering at Dunlop stated that the maximum allowable stress in the conveyor belts made at Dunlop can maximally be 10% of the tensile strength of the material (39). Although a conveyor belt is a totally different scenario in comparison to the application of a local passive seal, it gives a good indication in the maximum allowable stress an elastomer can withstand when used for many cycles.

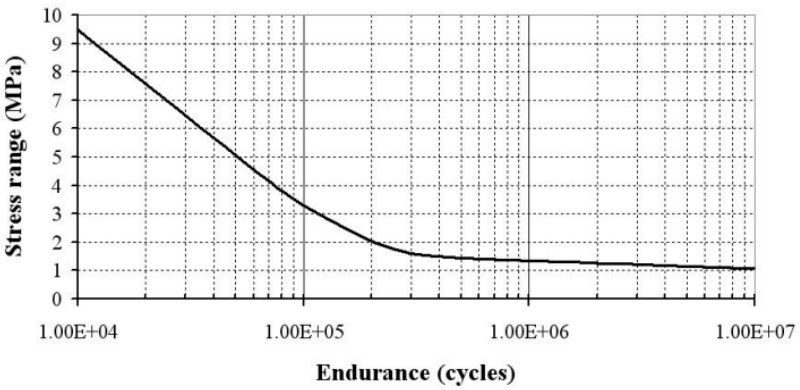


Figure 23: *SN curve for a moderately filled synthetic polyisoprene, a material designed to be similar to natural rubber (40)*

To conclude, the maximum allowable stress in the local passive seal is set to be maximally 10% of the yield strength, which is in this case maximum 1.3MPa.

14 Feasibility Local Passive Seal

In this chapter the feasibility of the l*ocal passive seal* is analyzed. This chapter starts with a detailed description of all the difficulties that arise when designing the *local passive seal* alternative. Next, a preliminary prototype is presented. In the consecutive sections, some of the difficulties mentioned in the first section are analyzed in greater depth. This chapter ends with a short conclusion on the feasibility of the *local passive seal* alternative when EPDM is used as the material.

14.1 Difficulties

Creating a flexible passive seal incorporates various difficulties. These difficulties are related to the efficiency of the sealing system, the elastic expansion of the material, the heaving bend in the material, and the viscous losses. This section will explain these difficulties in detail. Figure 24 shows a clear overview of where these difficulties occur. In reality many difficulties arise at the same moment in time. For example, the material expands radially and is strained vertically simultaneously while it is also pressed to the sides. On top of that, the seal is heaving constantly such that the forces acting on the seal are dynamic. It is, however, extremely difficult, if not impossible, to calculate the stress on the seal for all difficulties combined. Instead, every difficulty is calculated separately. In addition, due to time constraints, not all difficulties mentioned below are evaluated. Nevertheless, enough aspects are analyzed to give a conclusion on the feasibility of the local passive seal made of EPDM.

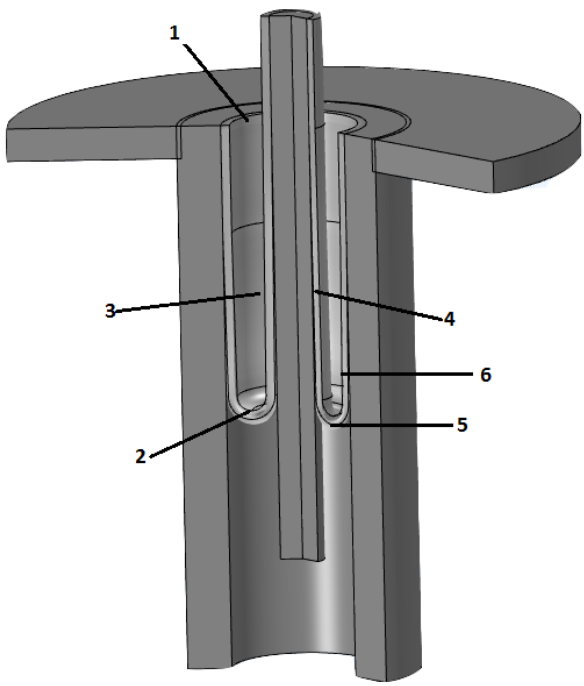


Figure 24: *A detailed figure of the local passive seal alternative. The dark gray bar in the middle represents the heaving rod. The dark gray outer structure is the entrance of the heaving rod in the base structure. The light grey domain between the rod and entrance represents the local passive seal. (1) The area mainly determining the efficiency loss of the seal. (2) The bending radius determining the maximum allowable thickness of the material. (3) The radial expansion resulting in stress in the material. (4) The friction between the walls and the material due to its elasticity. (5) The position of the bend that is not constrained to a wall, resulting in a free deformation. (6) Viscous losses due to moving walls and also due to the moving fluid volume*

14.1.1 Efficiency

The efficiency of the sealing system is of utmost importance for the functioning of the OG. The annulus area between the heaving rod and the walls is to a large extent determining the efficiency. The hydro-static pressure is applied to the entire surface area of the flexible membrane. Also, the hydro-static pressure works in all directions. Consequently, the size of this annulus area determines how much energy is needed to lift the water column positioned above. In addition, the fluid friction, drag forces and energy needed to elastically deform the material are also of influence on the efficiency. It is, however, believed, that such factors only play a minor role. The water column placed above the sealing system will determine the efficiency to the largest extent.

As mentioned already, the requirement is set that the efficiency loss may not be greater than 2%. However, this is not a hard requirement. When the life expectancy of the sealing system is improved significantly, one can allow for a higher efficiency loss. In addition, the weight of the pistons can be lowered when this sealing system is implemented. As it is now, the pistons move down through the water column due to its own weight. In the upstroke, this weight is lifted. When the local passive seal is implemented, the pressure difference between the two sides of the sealing system is pushing the piston down, such that the weight is not needed anymore. This might compensate for the efficiency loss.

14.1.2 Heaving Bend

One critical aspect of the *local seal alternative* is the bend that moves up and downwards along with the waves. The material should allow for this bending many cycles as mentioned in section 12.1. In addition, the amount of cycles that the material bends is not the same for every part of the local passive seal. One should know that the larger waves occur less frequently and as a result the material positioned further away from the equilibrium position are bent not as frequently compared to the material that is located around the equilibrium position.

Also, the bend determines the maximum allowable thickness. As mentioned already, the annulus area between the heaving rod and the walls determines the efficiency. From this area, it is possible to calculate the width between the two ends of the flexible membrane (which is explained in more detail in section 14.3.2). The radius of the bend is half of this width and from logic sense it can already be concluded that it is not possible to use an unlimited thickness.

14.1.3 Elastic expansion

The elastic expansion of the material is of utmost importance. The material does not only need to be able to bend on every vertical location. In addition, it should be able to expand in the radial direction. As mentioned already, the diameter of the heaving rod is smaller compared to the diameter of the entrance into the base structure while the material has the diameter of the heaving rod. Consequently, the material is strained in the radial direction towards the walls. Due to the strain, stress will arise inside the material that should not exceed the maximum allowable stress.

Also, when the material is elastic, the material will elastically expand in the vertical direction. As the material of the seal between the heaving rod and the wall is not constrained in any direction, it can elastically deform freely. As a result, large stress levels will occur in this part of the material. On top of that, significant wear is expected to happen. As the material expands vertically, it grinds against both the heaving rod and the outer walls. Combining this grinding with the fact that the hydro-static pressure is pressing on the material makes wear a significant problem.

14.1.4 Viscous Losses

Viscous losses are the last difficulty mentioned in this report regarding the *local passive seal* alternative. On the outside, the local passive seal is surrounded by seawater. The heaving rod moves the material up or downwards such that there is a speed difference between the material and the water. Consequently, viscous losses occur. These viscous losses, however, are complicated due to the movement of the entire water column. The position of the bend changes due to the heaving movement of the waves. Consequently, the water column between the two sides of the *local passive seal* moves also with a speed that is half of the velocity of the buoy.

14.2 Prototype

To get a first general impression of whether the *local passive seal* alternative might work, a preliminary prototype is created. The set-up of this prototype is given in figure 25. From this prototype, it can be concluded that the flexible membrane does indeed stretch radially until it touches the sidewalls. Next, it completely separates the external fluid (in the fluid tube) from the inside structure. Last, the elastic expansion in the vertical direction is significant. This preliminary prototype is only useful to get a general idea of the expansion of the material used for the seal. It is not usable for accurate measurements. The reader should realize that in the original concept, the flexible membrane is lifted by the buoys which are connected on the top part of the membrane. In this prototype, a tube is pushing the flexible membrane up. Although this is not comparable with the real situation, it still gives a good insight into the elastic behavior of the flexible membrane.

Figure 25: *A schematic figure of the prototype for the local passive seal alternative. Note that the prototype is placed vertically in reality, while in the figure it is shown horizontally. (1) The fluid tube to create enough pressure to expand the material. (2) Lid of the fluid tube that connects it with the tube that represents the sidewalls of the base structure. (3) Tube connecting the seal with the fluid tube. (4) Tube in which the seal is placed. (5) Location of the actual seal. (6) Tube representing the sidewalls of the base structure. (7) PVC sleeve connecting (6) with the guidance system of the internal heaving rod. (8) Guidance system. (9) Heaving rod*

Figure 26: *Set-up of the prototype in the OG laboratory*

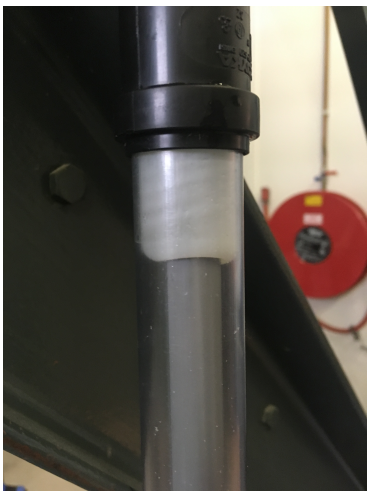


Figure 27: *Local passive seal prototype*

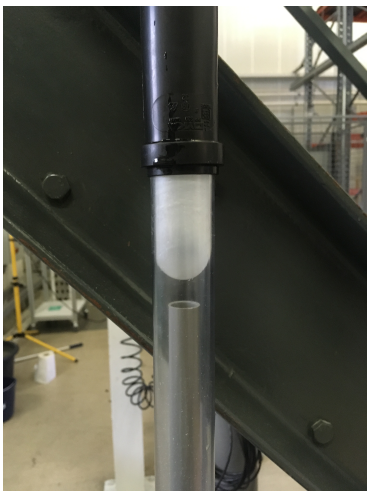


Figure 28: *Local passive seal prototype*

14.3 Efficiency

Efficiency regarding the flexible membrane used in the *passive seal* is defined as the force (or work) needed to move the flexible membrane. As can be seen in figure 24 there is a certain area that needs to be lifted due to the pressure difference between both sides of the membrane. To calculate the efficiency, the force delivered by the buoys should be estimated first.

14.3.1 Buoy Power

The following assumptions are made to estimate the force that a buoy can deliver:

- The buoy can have a maximum diameter of 20% of the wavelength for a wave of 1m in height, similar to the assumption made to the previous versions of the OG
- The maximum draft is 0.5m. The previous design of the OG could handle waves up to 12m. The maximum draft for this design was 1m. As the maximum wave height is halved for this design, the maximum draft is also halved
- A wave in the North Sea with a height of 1m has a wave period of 4s, based on data from an offshore wind farm near the Dutch city Egmond aan Zee (23)
- The wavelength for a wave of 1m is assumed to be 24.9m according to a mathematical model

Based on the assumptions given above it is possible to estimate the maximum pumping force that can be used for the pumping system:

$$d_{max} = 0.2 \cdot 24.9 = 4.98m \tag{6}$$

$$V_{max} = 0.25 \cdot \pi \cdot 4.94^2 \cdot 0.5 = 9.74m^3 \tag{7}$$

$$m_{waterdisplaced} = V\rho_{water} = 9.74 \cdot 1027 = 10,002kg \tag{8}$$

$$F_{buoy} = m_{waterdisplaced}g = 10,002 \cdot 9.81 = 98,120N = 0.1MN \tag{9}$$

14.3.2 Efficiency Loss Seal

Now that it is known what the maximum force delivered by buoy is, the efficiency of the seal can be determined. As mentioned already, this is determined by the annulus area between the heaving rod and the walls, and the pressure difference. The diameter of the heaving rod is 6cm, as determined in the thesis of O. van Hees in which the current seal is designed (15). This annulus area does not only changes when the width between the two ends of the material is increased, but it also increases with increasing thicknesses of the material (assumed that the width is equal). This can be seen in figure 29. Note that in this figure it is assumed that the material stretches completely to the sidewalls of the base structure.

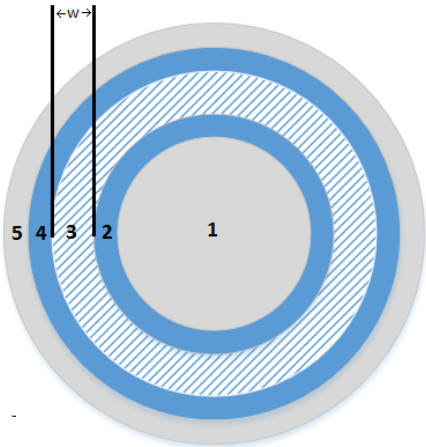


Figure 29: *Top view of the local passive seal. (1) heaving rod, (2) inside attachment of flexible membrane, (3) annulus area between the two sides of the material, (4) outside attachment of flexible membrane, (5) Base structure. The diameter of the base structure entrance depends on the diameter of the rod, the thickness of the material and the width between the two ends of the material*

Be aware that calculating the efficiency only based on the annulus area and pressure difference is only a rough estimation where many other aspects relevant for the efficiency are left out. As mentioned in section 14.1.1, the efficiency is also influenced by the friction induced by the water column, the energy needed for the elasticity, etc. However, it is assumed that the work needed to lift the seal because of the pressure difference is the major component determining the efficiency and is, therefore, a good indication to determine the dimensions of the local passive seal.

The following formula is applied to calculate the efficiency:

$$Effiency[\%] = \frac{W_{buoy} - W_{seal}}{W_{buoy}} \cdot 100 \tag{10}$$

where W_{bouy} is the work delivered by the buoy, which is simply the force determined above multiplied by the maximum wave displacement, and W_{seal} is the work needed to lift the seal. W_{seal} is determined as follows:

$$W_{seal} = \int_0^{h_{max}} A(d-x)dx \tag{11}$$

where h_{max} is the maximum wave height, A is the annulus area of the flexible membrane, ρ is the density of the seawater, d is the depth, and x is the vertical displacement of the flexible membrane.

Figure 30 is showing the graph where the efficiency is plotted against the annulus area. As given in the requirements for the *local passive seal*, the efficiency loss should not be greater than 2%. From this graph it can be concluded that the annulus area can therefore not be greater than $7000mm^2$, $5500mm^2$, and $4000mm^2$ for the varying depths of 30m, 40m, and 50m respectively. Decreasing the annulus area increases the efficiency and might, therefore, be beneficial. However, as will be explained in the next sections, there are some serious disadvantages when the annulus area is minimized.

Next to the efficiency, the width between the two ends of the material is also plotted in figure 30. As mentioned earlier, this width changes a bit for varying thicknesses when the annulus area is kept constant. For the widths in figure 30, the material thickness is left out and the width is determined by the following formula:

$$w = 1/2(\sqrt{4A/\pi + d_r^2} - d_r) \tag{12}$$

where w is the width between the two ends of the material, A is the annulus area, and d_r is the diameter of the rod (fixed at 6cm).

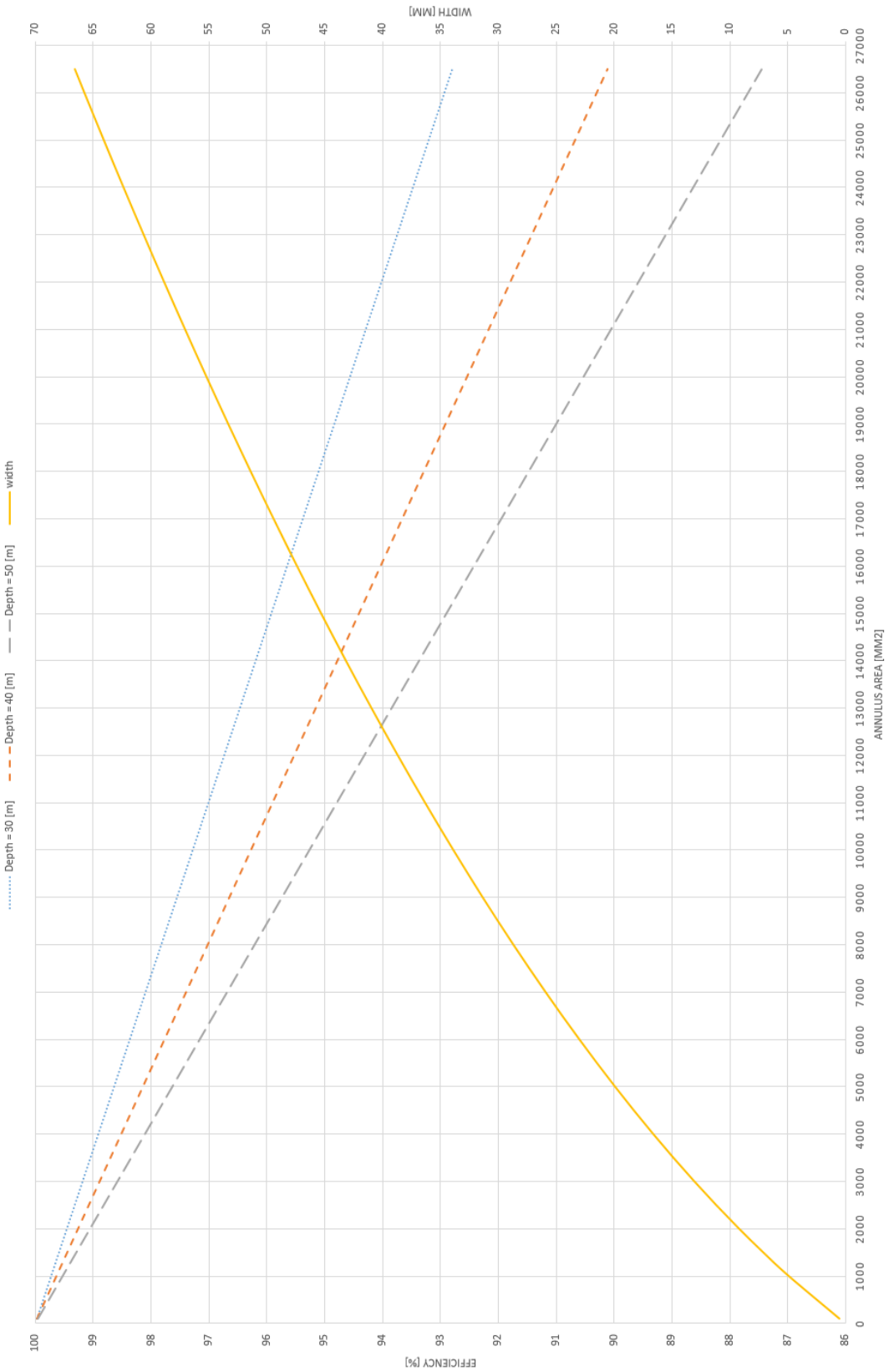


Figure 30: *Efficiency graph where the efficiency is plotted against the annulus area between the heaving rod and the walls. As the annulus area might be confusing, the width between the heaving rod and walls is also plotted against the annulus area. Note that this width changes a bit for varying thickness of the material*

14.4 Minimum Bending Radius

For the *local passive seal* to work, the material needs to bend constantly on every location depending on the position of the buoy on a wave. As already proven, the annulus area is limited to keep the efficiency loss reasonable. When a material bends, the neutral axis shifts towards the bend and as a result, more strain is located on the outside of the bend. As it is already determined in section 13.1, the maximum allowable stress in the material is set to be 1.3MPa.

To determine the maximum thickness of the material, the maximum allowable strain should be determined first. To do so, the stress-strain curve created by A. Claassen is analyzed (41). This graph is shown in figure 31 and depicts the stress-strain relation for EPDM test specimens. As mentioned earlier the maximum allowable stress is 1.3MPa and from this graph it can be concluded that the maximum strain can therefore not be larger than 18%.

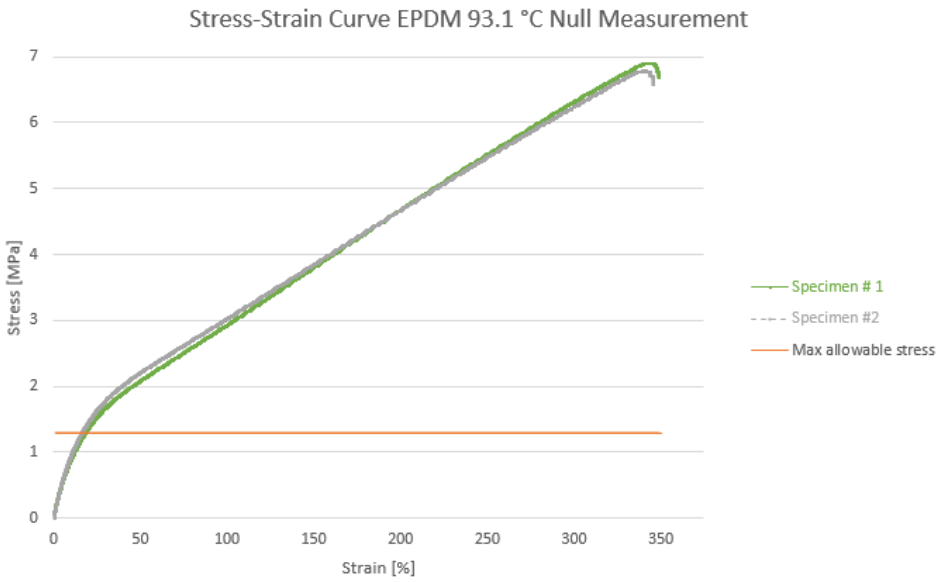


Figure 31: *Stress-strain curve for EPDM, adapted from (41). The orange horizontal line depicts the maximum allowable stress. From this graph, it is concluded that the maximum allowable strain is 18%*

The strain that occurs inside the material for the *local passive seal* is complicated. Not only does strain arise due to the bending. In addition, there is also a stress component in the radial direction. This radial strain, however, is neglected for the estimation of the maximum material thickness. Instead, it is assumed that the material is a sheet. From this assumption, the following formula is used to calculate the strain for varying thicknesses and annulus areas (42):

$$\epsilon_{max} = 1/2ln(1 + t/R_i) \tag{13}$$

where ϵ_{max} is the maximum strain, t is the thickness of the material, and R_i is the inner radius of the bend which is half of the width between the two ends of the membrane, see figure 29. As explained already, the width between the two ends of the material changes for varying thicknesses when the annulus area is kept constant. As a result, the width between the two ends of the membrane should be adjusted accordingly such that the efficiency loss is kept at 2%. The following formula is applied to calculate the width when the material thickness is included:

$$w = 1/2(\sqrt{4A/\pi + (d_r + 2t_b)^2} - d_r - 2t_b) \tag{14}$$

where A is the annulus area, d_r is the diameter of the rod, and t_b is the thickness of the bladder. In table 2 the maximum thickness of the material is given for varying annulus areas. Note that all the values of the thicknesses for areas until $1000mm^2$ are rounded to the nearest integer, while the values for the thicknesses for areas smaller than $1000mm^2$ are rounded to the first decimal.

Table 2: *Maximum material thicknesses for varying annulus areas. As the annulus area might be confusing, the width between the two ends of the material is also given. Note that the thickness for annulus areas larger than 1000mm^2 is rounded to the first integer. The thickness for annulus areas smaller than 1000mm^2 is rounded to the first decimal*

Annulus Area [mm^2]	*Width [mm]*	*Maximum Thickness [mm]*
7000	23.8	5
6000	21.4	4
5000	18.4	4
4000	15.6	3
3000	12.5	2
2000	8.8	2
1000	4.8	1
900	4.3	0.9
800	3.9	0.8
700	3.4	0.7
600	3.0	0.6
500	2.5	0.5
400	2.0	0.4
300	1.5	0.3
200	1.0	0.2
100	0.5	0.1

14.5 Radial Elastic Expansion

As mentioned earlier, the diameter of the whole *local passive seal* is equal to the diameter of the heaving rod. Due to the hydro-static pressure, however, the material above the bend on the outer side is pressed towards the sidewall of the base structure. As a consequence, the larger the annulus area between the heaving rod and entrance in the base structure, the more the material can strain radially.

Figure 32 is showing a typical stress-strain curve for a hyper-elastic material such as EPDM (43). From this graph, it can be concluded that the stress increases significantly for smaller strains and that it flattens out for larger strains. As the maximum allowable stress is only 10% of the tensile strength, it is most likely in the region where the stress is highly sensitive to the strain. Consequently, it is crucial to determine the stress inside the material when it is able to expand freely.

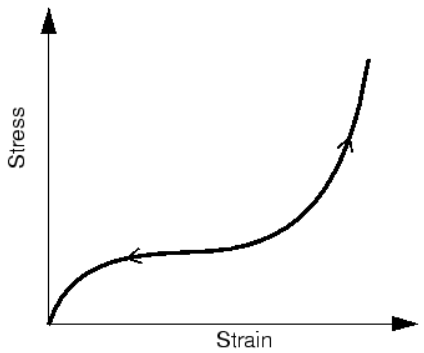


Figure 32: *Stress-strain curve for a typical hyper-elastic material, such as EPDM (43)*

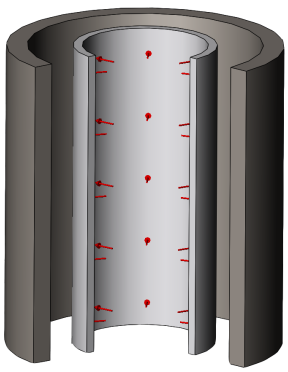


Figure 33: *Set-up of the COMSOL experiment performed to analyze the radial elastic expansion. Note that the dimensions are not representative of the simulations performed*

14.5.1 COMSOL Simulation

A COMSOL simulation is created to analyze the stress behavior of the material when it can expand from the inner diameter to the diameter of the entrance into the base structure. A stationary 2D-axis symmetric study is performed where a pressure of 3bar is applied to the material. A 3D study was too time consuming and has therefore not been performed. The set-up of this simulation is given in figure 33, while the details can be found in the appendix. The goal of this simulation is solely to analyze the stresses induced by the pressure when the material expands radially. Due to the volume conservation the material shrinks in length. In reality, the downward hydro-static pressure prevents this which will result in even higher stresses. This is, however, neglected in this simulation.

The simulation has been performed for all annulus areas mentioned in table 2 for varying thicknesses. From this simulation three conclusions can immediately be drawn:

- For a constant pressure and a varying thickness, the stress increases significantly for decreasing thicknesses when the material does not yet touch the sidewalls of the base structure
- For a constant pressure and a varying thickness, the stress is almost in varying when the material does touch the sidewalls of the base structure for decreasing thicknesses
- The thickest material possible always results in the lowest stress. As a result only the largest thicknesses are simulated [2].

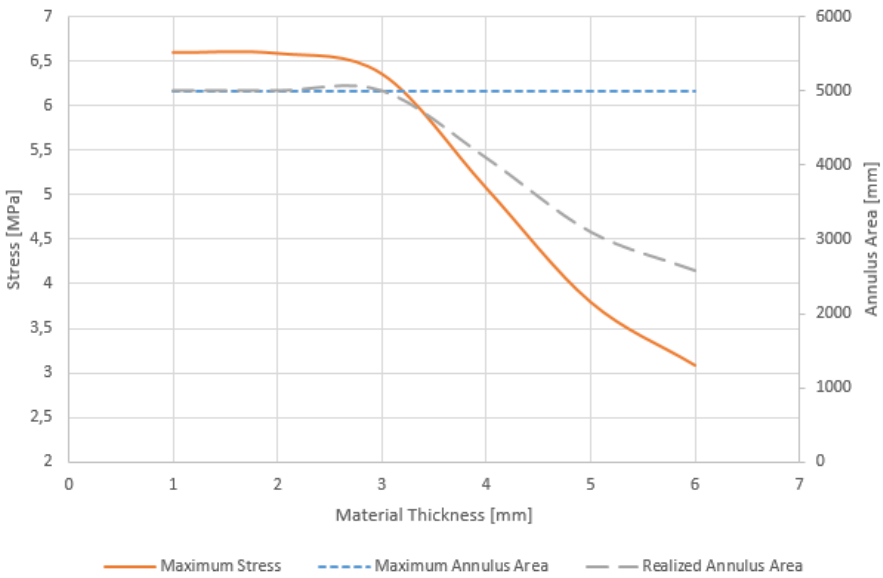


Figure 34: *Graph showing both the maximum stress and displacement in the material at a depth of 30m (relative pressure of 3bar) for an annulus area of 5000mm^2. The graph clearly shows the statements given above. The lowest stress occurs at the thickest material, and the stress remains almost constant for decreasing thicknesses when the material touches the walls of the base structure (realized annulus area = maximum annulus area). The realized annulus area is the area between the two ends of the material after the radial stretching. Note that the small peak where the realized annulus area is larger than the maximum annulus area is due to the inaccuracy of the excel graph. In reality the realized annulus area can never exceed the maximum annulus area. From table 2 it can be seen that for an annulus area of 5000mm^2 the maximum thickness can only be 4mm. The thicknesses of 5mm and 6mm are only included in this figure to support the claims stated above*

[2]Due to time constraints it was not possible to run the simulation for many combinations of thicknesses and annulus areas. As it was proven that the largest thickness resulted in the lowest stress, and when this stress is still larger than the maximum allowable stress, there is no point in simulating smaller thicknesses for the same annulus area

Table 3: *Results from the COMSOL simulation. The width indicates the radial displacement the material can deform before it touches the walls, while the displacement gives the values of the actual radial deformation for every simulation. Almost all stresses occurring in the material are too high considering the maximum allowable stress, except for annulus areas smaller than 700mm^2*

Annulus Area [mm^2]	*Material Thickness [mm]*	width [mm]	*Displacement [mm]*	*Maximum Stress [MPa]*	*Pass/ Fail*
7000	5	23.8	12.0	3.79	fail
6000	4	21.4	15.6	5.09	fail
5000	4	18.4	15.6	5.09	fail
4000	3	15.6	15.5	5.15	fail
3000	2	12.5	12.5	4.19	fail
2000	2	8.8	8.8	2.90	fail
1000	1	4.8	4.8	1.60	fail
900	0.9	4.3	4.3	1.47	fail
800	0.8	3.9	3.9	1.32	fail
700	0.7	3.4	3.4	1.21	pass
600	0.6	3.0	3.0	1.03	pass
500	0.5	2.5	2.5	0.87	pass
400	0.4	2.0	2.0	0.71	pass

Table 3 is showing the results of the performed simulations for all varying annulus areas where a pressure of 3bar is applied. As concluded earlier, the lowest stress is obtained for the thickest bladder thickness. For the largest annulus areas ($7000mm^2$, $6000mm^2$, $5000mm^2$) the material is not even stretched towards the walls of the entrance in the base structure (as can be seen in table 3, the displacement $<$ width). When the thickness is decreased, the material will eventually reach the walls of the base structure, but then the stress increases also. In table 3 it can be concluded that the stresses occurring in the material exceed the maximum allowable stress on almost every occasion. Only when the annulus area is $< 700mm^2$, the stresses due to this radial strain is lower than the maximum allowable stress. However, the material in these cases cannot be larger than 0.7mm.

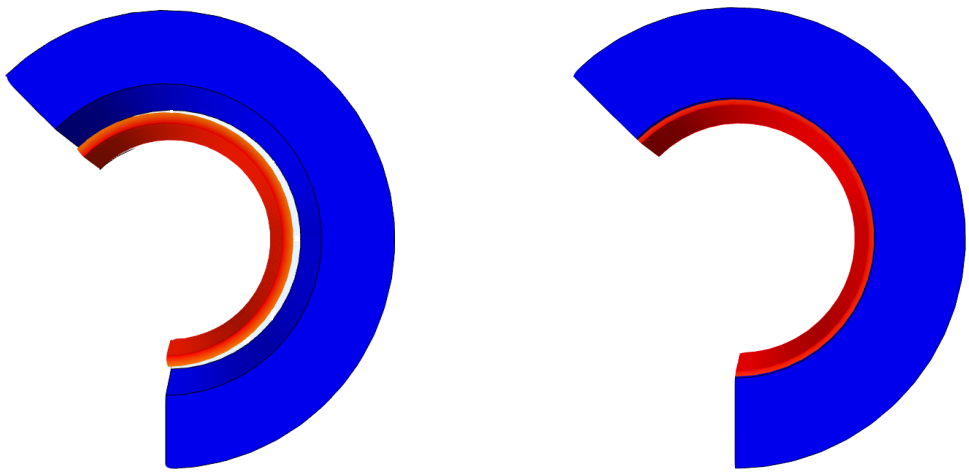


Figure 35: *Radial displacement for two different simulations, when a pressure of 3bar is applied. For an annulus area of 7000mm^2 and a thickness of 5mm (left), the material is not fully stretched towards the walls (displacement $<$ width). For an annulus area of 3000mm^2 and a thickness of 2mm (right), the material does touch the walls (displacement = width)*

14.6 Stress Concentration Bend

As concluded in the previous chapter, the allowable deformation of the material is crucial considering the maximum allowable stress. The bend in the material that is located between the heaving rod and the walls of the base structure, however, is not constrained by any wall. As a result, the material in this location can deform freely and therefore it is crucial to determine the stresses occurring in this bend.

14.6.1 COMSOL Simulation

Similar to the previous section, a COMSOL model is created to investigate the stresses acting in the bend of the *local passive seal*, to analyze the feasibility of this alternative. A stationary, 2D axis-symmetric, study is created where the pressure applied is constant. Also, to simplify the problem, the strain due to the radial expansion is neglected, i.e. it is assumed that the material touches both the heaving rod and the walls of the base structure without any strain inside the material. Similar to the other study a pressure of 3bar is applied to the material. The set-up of the experiment is shown in figure 36. For a detailed description of the simulation, the reader is referred to the appendix. Also, as the previous simulation has already proven that the annulus area cannot be larger than $700mm^2$, simulations are only performed on varying material thicknesses and annulus areas smaller than $700mm^2$.

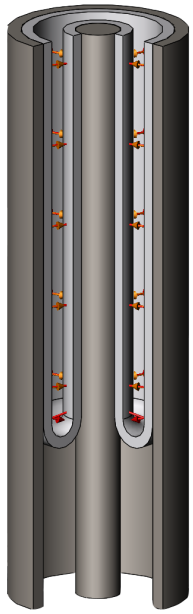


Figure 36: *Set-up of the COMSOL simulation used to analyze the stresses occurring in the bend of the local passive seal. Pressure is applied to the entire inside area of the flexible membrane while the material is constrained in movement by the heaving rod and the sidewalls of the base structure. Only the top part of the membrane is fixed, such that the material can deform in the vertical direction. Note that the dimensions used in this figure are not representative of the simulations, it is merely used to give a clear view on the set-up of the simulation*

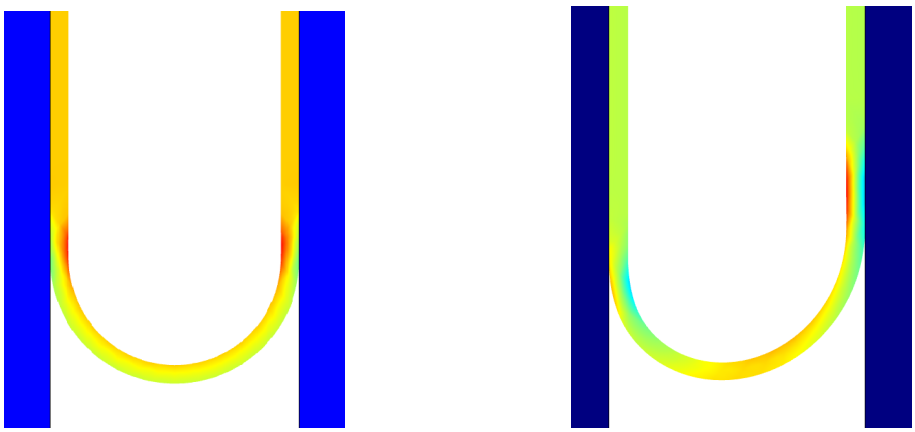
For this situation a 2D simulation is created first. The advantage of such a 2D simulation is that the computation times are much smaller compared to a 2D axis-symmetric study. The results, however, are deemed to be inaccurate. Table 4 is showing the results of a 2D and 2D axis-symmetric study for the same material thickness and annulus area between the heaving rod and the wall. One can see that the 2D simulation underestimates the stress levels in the material significantly. This makes complete sense as a 2D model assumes a width of 1m, neglecting the round shape of the material completely.

In addition, the location of the stress also differs. The 2D simulations show a symmetric stress distribution, while the 2D axis-symmetric study clearly shows that the stress levels are not symmetrically distributed, see figure 37. This is also because the model assumes that the 2D structure has a width of 1m, as if it is a plate. Consequently, the two sides of the bend are symmetric which results in a symmetric stress distribution. However, the simulation completely neglects the round shape of the actual structure, which explains why the actual stress distribution can never be symmetrical.

As a result, it is chosen to use the 2D axis-symmetric simulations to obtain more realistic results. A 3D study is not used as it is too time-consuming.

Table 4: *Difference between the maximum stresses occurring in the material for the 2D and 2D axis-symmetric studies for an annulus area of 5000mm^2*

Thickness Bladder [mm]	*Stress 2D [MPa]*	*Stress 2D axis-symmetric [MPa]*	*Difference [%]*
2	2.39	3.04	27
3	1.67	2.32	39
4	1.31	1.86	42
5	1.10	1.60	45
6	0.97	1.45	50



(a) *Result 2D simulation* (b) *Result 2D axis-symmetric simulation*

Figure 37: *Location of the maximum stress for the 2D (a) and 2D axis-symmetric (b) simulations for a thickness of 2mm and a width of 20mm (such that the annulus area is 5000mm^2). The maximum stresses are given in table 4*

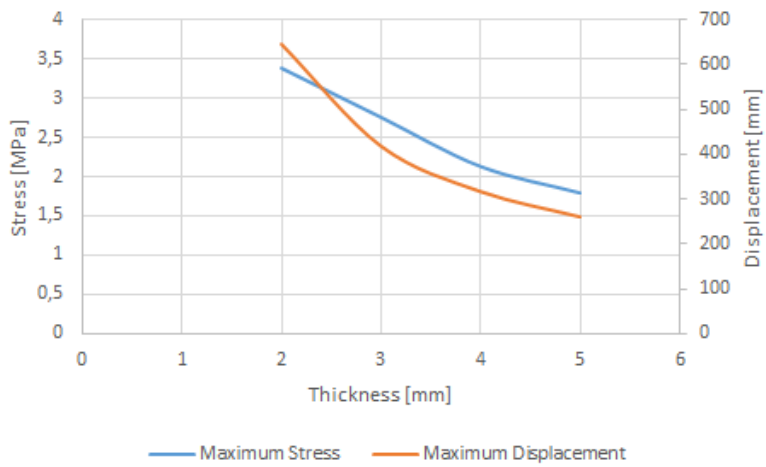


Figure 38: *Graph showing both the maximum stress and the vertical displacement regarding the bend of the local passive seal in the material at a depth of 30m (relative pressure of 3bar) for an annulus area of 7000mm^2 resulting from a 2D axis-symmetric study. The graph clearly shows that the stress is minimal for the largest thicknesses. Also, it concludes that the vertical displacement is significant*

Also, in this case, it is proven that the stress is minimal for the largest thickness, see figure 38. Therefore only the maximum thicknesses are simulated. The results are given in table 5. Also here it can be concluded that the stresses violate the requirement that the stresses cannot be larger than the maximum allowable stress. In addition, it shows that the material elastically deforms significantly in the vertical direction. As a result, friction between the sides and the material may become a significant problem. However, due to time constraints, this problem is not analyzed any further.

Table 5: *2D and 2D axis-symmetric results of the simulations analyzing the stress levels inside the bed of the local passive seal*

Annulus Area [mm^2]	*Material Thickness [mm]*	*Maximum Stress [MPa]*	*Vertical Displacement [mm]*
700	0.7	2.01	262
600	0.6	1.9	264
500	0.5	1.88	266
400	0.4	1.82	268
300	0.3	1.7	271

14.7 Conclusion

In the last four sections, several difficulties mentioned at the beginning of the chapter are analyzed. First, it is concluded that the annulus area between the heaving rod and the entrance into the base structure can only be of a limited size to keep the efficiency loss reasonable. Second, it is proven that the material has only a limited thickness for varying annulus areas. Based on two COMSOL simulations it is concluded that the stresses are simply too high and exceed the maximum allowable stress taking into account fatigue stresses. As a result, it can be concluded that the *local passive seal* alternative is not a viable alternative to be implemented in the OG when EPDM is used. Note that the simulations only calculated the stresses simulating a depth of 30m. As it is proven that the *local passive seal* will fail at this depth, it is also proven that it will also not work at depths of 40m or 50m as the conditions are more severe at these locations.

14.8 Improved Material - Symphony

The previous sections in this chapter analyzed various difficulties regarding the *local passive seal*. From this analysis, it is concluded that a pure elastomer, such as EPDM is most likely not able to withstand the hydro-static pressure for so many cycles. There are, however, methods imaginable to reinforce a material to make it stronger. In addition, the search for an elastic material that needs to expand radially is not completely new. Instead, another WEC, the Symphony, also needs a round elastic structural membrane that expands radially. Together with Trelleborg, a company specialized in the production of polymer products, a prototype for this material has been created and a report has been published (44).

The membrane designed by Trelleborg is a part of the prototype for the Symphony. To understand the function of the membrane, it is useful to first describe shortly how the Symphony WEC works. The symphony consists out of two parts which are separated by the membrane. Due to the hydro-static pressure of a passing wave, the outer cylinder is pushed down. As a result, the inner volume of the Symphony is decreased, which creates an internal water flow that goes through a turbine. Simultaneously, the inner pressure in the spring chamber is increased due to the decrease of the inner volume. As a result, when the inner pressure is larger than the hydro-static pressure, the outer cylinder moves up again. The internal flow is reversed and goes back through the turbine. The membrane acts as a seal to enclose the inner volume but also acts as a bearing for the cylinder. The inner volume changes due to the difference in width in which the upper and lower part of the membrane roll up and down. The prototype of the Symphony has a diameter of 1500mm. In this last section, the structural membrane created by Trelleborg is briefly analyzed.

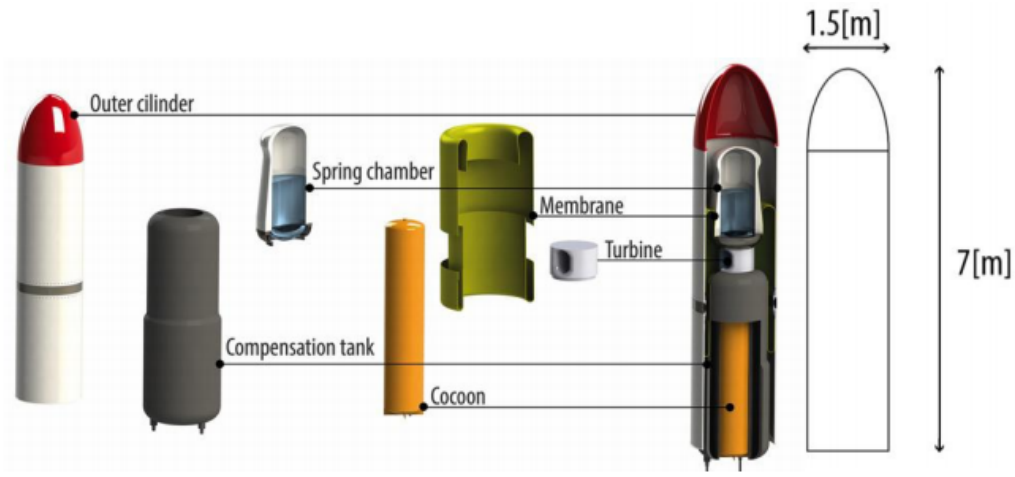


Figure 39: *Exploded view of the Symphony wave energy converter (44)*

First, they have not used EPDM as the material. Instead, they have chosen for natural rubber. Natural rubber is indeed a stronger material, as can be seen in table 6. Based on their report, natural rubber is chosen as it has the best properties regarding fatigue, hysteresis, durability and (sea)water resistance. More importantly, however, is that aramid is used to reinforce the structural membrane in a way that it is still able to expand radially.

The structural membrane is built from an inner and outer layer of natural rubber with two layers of aramid cord fabric in between under an angle of 10° with the axis of the membrane (44). The complete membrane has a thickness of roughly 10mm. The aramid fabric reinforces the membrane in the vertical direction. This is extremely helpful, as the simulation given in section 14.6.1 already has shown that the stresses in the bend are problematic. The free elastic expansion in the bend, as it is not constrained by any wall, result in these high stresses. When the bend is reinforced with aramid, the elastomer material will not deform that significantly anymore. As the structural membrane in the Symphony WEC should withstand pressure up to 20bar, it is safe to conclude that pressures ranging from 3-5bar are also no problem for the aramid fibers.

Table 6: *Material properties for EPDM, natural rubber and Kevlar 49, a specific type of aramid. All values are obtained from CesEduPack 2014 (45)*

Material Property	*Minimum Value*	*Maximum Value*	*Average Value*
EPDM			
Density $[kg/m^3]$	800	1000	900
Young's Modulus [MPa]	2	10	6
Poisson's Ratio [-]	0.48	0.495	0.4875
Tensile Strength [MPa]	5.4	20.2	12.8
Natural Rubber			
Density $[kg/m^3]$	1020	1200	1100
Young's Modulus [MPa]	2.1	8.4	5.3
Poisson's Ratio [-]	0.499	0.5	0.4995
Tensile Strength [MPa]	20	27	23.5
Aramid - Kevlar 49			
Density $[kg/m^3]$	1440	1450	1445
Young's Modulus [MPa]	117e3	130e3	123.5
Poisson's Ratio [-]	0.35	0.36	0.355
Tensile Strength [MPa]	2250	2750	2500

The dimensions of the structural membrane designed for the Symphony are completely different than the *local passive seal* analyzed in this chapter. Considering the largest annulus area for a pressure of 3bar, which is $7000mm^2$, the width cannot be larger than 25mm. Therefore, the diameter of the entrance into the base structure is 11cm, as the rod has a fixed diameter of 6cm. For the structural membrane of the Symphony, the outer diameter is 1500mm, while the maximum inside diameter is 1080mm, as can be seen in figure 40. As the inner diameter of the Symphony is much larger compared to the inner diameter of the *local passive seal*, the relative radial expansion is much higher for the *local passive seal* (38,8% & 83.3% increase in circumference for the Symphony and *local passive seal* respectively, given the values mentioned above). This is, however, not a problem as it is expected that natural rubber can deal with this radial expansion better. In addition, the width between the two sides of the material in the *local passive seal* can also be decreased. This is not only beneficial for the radial expansion, it is also favorable for the efficiency, see section 14.3.2.

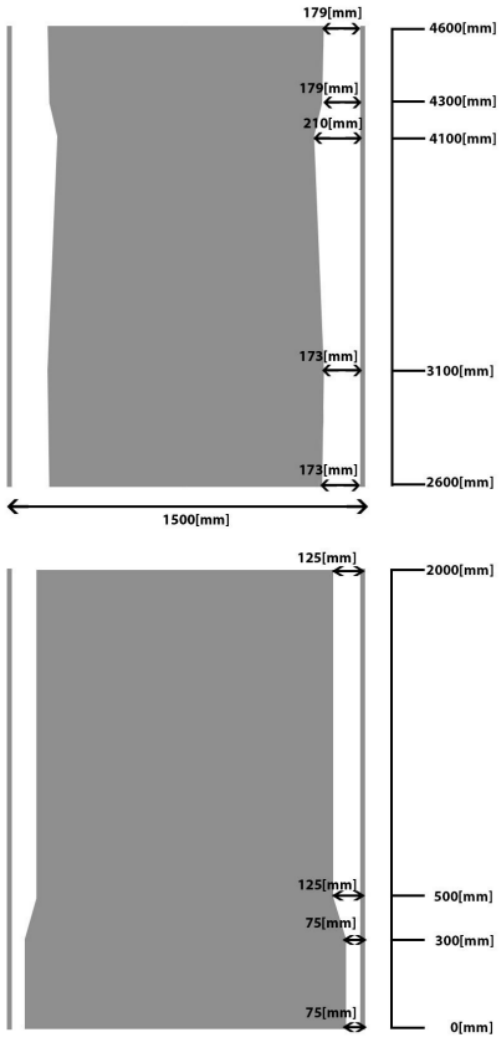


Figure 40: *Unfolded structural membrane designed for the Symphony WEC (44). The inner diameter of the material is 1080mm at its smallest while it needs to expand to an outer diameter of 1500mm*

The reader should be aware that this report was found only three weeks before the deadline for this thesis. This explains why the material provided in this section has not been used to analyze the feasibility of the *local passive seal* in the previous sections. The report dealing with the structural membrane created for the Symphony WEC, however, clearly shows that there are many future research directions. This might even result in the fact that the *local passive seal* is a feasible alternative after all. This is explained in more detail in section 15.2.

Part IV
Discussion & Conclusion

15 Discussion

This report deals with the search for an alternative pumping structure for the OG. The most important requirements related to the pumping structure have been stated first. Second, all pumping mechanisms, known in literature, have been analyzed. By making use of the Pugh's method all alternative pumping mechanisms have been evaluated based on specially defined functional requirements. By using this method only one viable alternative presented itself: the diaphragm pump. A diaphragm pumping mechanism has aspects that are in line with the problem owners' ideas. Indeed, the flexible membrane of a diaphragm pump mechanism both separates the inside structure from its outside environment, and also pumps the working fluid. Despite these promising potentials, it has been proven that a diaphragm pumping mechanism cannot work for the OG as the pressure differences are simply too large.

To continue, several alternative sealing systems have been devised, all incorporating a flexible membrane that is used to separate the inside of the base structure from the external environment. Based on a concise evaluation it has been chosen to investigate the local passive seal alternative in more detail. EPDM has been chosen as material due to its mechanical properties and because it already used in marine environments. Fatigue is a critical aspect to consider due to the many cycles the material has to withstand. The maximum allowable stress has been defined as 10% of the tensile strength of the material, for which average values from CES EduPack are used. First, a preliminary prototype has been presented. Next, all difficulties that arise when implementing the local passive seal alternative in the OG have been stated and four of these difficulties have been analyzed.

First, an estimation of the efficiency loss of the local passive seal has been given. The efficiency loss cannot be greater than 2%, similar to the requirement for the reciprocal seal. From calculations it has been concluded that the annulus area between the heaving rod and the sidewalls of the base structure cannot be larger than $7000mm^2$, $5500mm^2$, and $4000mm^2$ when the OG is located at a depth of 30m, 40m, and 50m respectively. Next, the maximum thickness of the material for the seal has been estimated. Two COMSOL simulations have been created to calculate the stresses inside the material when it elastically deforms due to the hydro-static pressure. Based on the simulations it has been proven that, although increasing the area between the rod and walls allows for increasing thicknesses of the material, the maximum stresses also increase. Making the material and the area smaller, however, does not result in stress levels that acceptable regarding the earlier determined maximum allowable stress. In addition, one might argue whether using a material with a thickness of only a few tenths of a millimeter is a wise decision regarding the robustness of the system. As the consequences of failure are catastrophic, one can make a strong argument that the material needs to be thicker instead. Therefore, it is proven that the local passive seal alternative cannot be used for the OG if it is made of EPDM. The fact that it is proven that EPDM cannot be used is not a total show stopper. As stated in section 14.8, there are still many options that can be researched to make this alternative a success after all.

15.1 Limitations

As for the limitations of this research, the main issue is related to working with polymers. Not only do there exist numerous different kinds of EPDM, but it is still not exactly clear how elastomers behave regarding fatigue. Consequently, some rough estimations are made. First, only estimates haven been used for the mechanical properties of EPDM, which are used in the COMSOL models. Second, the maximum allowable strain, that is used to calculate the maximum thickness of the material, is a rough estimation. This strain is based on one specific type of EPDM, which does not fully match the material simulated in COMSOL. In addition, the assumption that the material is a sheet that is bent instead of its actual shape is a rather extreme simplification of reality. Also, the stresses that occur inside the test specimens are only normal stresses. This is not comparable to reality.

It is important to note that even with the relaxing assumptions, the material fails to satisfy its requirements. Therefore, it can be concluded that such a sealing system would definitely fail under more realistic conditions. In addition, the severity of wear and friction are not even included in this report. Nevertheless, first thoughts suggest that the wear of the elastomer grinding against

the hard material of the rod and base structure, under a pressure of at least 3bar, is significant.

Regarding the COMSOL models, two stationary studies have been created. As a result, transient behavior is neglected. In a time dependent study, the stresses acting in the bend of the material, of which the position also changes over time, are included. In reality not only the local passive seal would be moving, but the water column inside would also move. As a result, the potentially interesting behavior of the fluid flows is neglected in both the models. In addition, the hydro-static pressure of the seawater is modeled as a constant pressure. Due to the movement of the water column inside the local passive seal, this assumption might not be realistic.

15.2 Further Research

This thesis focuses mainly on the implementation of a flexible membrane to solve the defined problem. This focus is in line with the desires of the problem owners who were interested in whether a flexible membrane might be useful for the OG. It is proven to a sufficient degree that implementing a flexible membrane to both pump the working fluid and separate the inside structure from its environment is not possible. Also, it is proven that the local passive seal alternative is not feasible when EPDM is used as material. There are, however, other alternatives imaginable where a flexible membrane is used as a seal. Such alternatives could be investigated further. Not only the two other alternatives mentioned in this report can be analyzed further, but there is also no doubt that one can think of other new alternatives that are worth investigating.

Regarding the simulations performed in COMSOL, there are also some future directions. The two performed simulations only analyzed two separate difficulties that arise: the stress due to the radial expansion and the stress inside the bend. A simulation can be created that combines these two elastic expansions to obtain more realistic results. Another significant difficulty is related to the stress that occurs in the material when the bend moves up and downwards. Currently, these stresses are approximated by estimating the bending stress, assuming as if it is a sheet. A next step could be to create a COMSOL model where the stresses that occur when the material bends are simulated. The set-up of this simulation is given in figure 41. This 2D-axis symmetric, time-dependent study can be performed to analyze what the stresses will be in the bend when an external pressure is applied to the material.

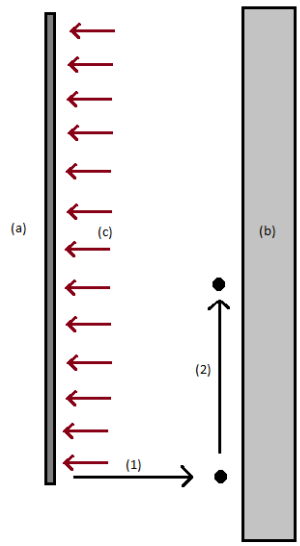


Figure 41: *Set-up of a COMSOL simulation to calculate the stresses in the heaving bend. (a) The flexible membrane, (b) a wall serving as a contact pair to restrict the movement, (c) the applied pressure on the membrane. (1) A prescribed displacement to move the material to the right. (2) A prescribed displacement to move the bend upwards, such that the bend is formed while the pressure is still applied*

As mentioned in section 14.1.1, the reduction in weight of the pistons can result in a better performance from an efficiency perspective. This is not taken into account in this project. More research should be done to which extent the weights of the pistons can be lowered. In addition, more research is needed to which extend this will be of influence on the efficiency. Also, more research should be done one allowing the efficiency loss to be greater than 2%. A larger annulus area between the rod and the wall allows for thicker material that can be used. Using a thicker material may reduce the stress levels inside the material.

Speaking of the material, the choice of material is only based on a preliminary evaluation where only average values are used. Taking a closer look at the material selection might result in a specific type of material that can be used better. If there exists a material that is flexible, elastic but also stronger and is better resistant to fatigue stresses, it might be useful for the *local passive seal*. As mentioned already in section 14.8, the reinforced material that has been used for the structural membrane of the Symphony WEC might be extremely useful. This material provides a clear solution to the stress concentration that occurs in the bend of the *local passive seal*, but it does not provide a clear solution to the stresses that occur in the radial direction. More research is needed on how natural rubber will deal with the stresses that occur due to this radial expansion. In addition, when the width between the two sides of the material is decreased, more research is needed on the behavior of the fluid film between these two sides. If possible, a collaboration with Trelleborg would be extremely useful to design a suitable material for the application of the *local passive seal*.

However, this is not the only solution imaginable to solve the problem devised at the beginning of the report. Other solutions are imaginable. Analyzing completely different sealing systems can be a future direction. One example can be the implementation of a rotational seal, similar to the seal used for the propellers of ships. The heaving movement of the buoy should be translated into a rotational movement to pass along the movement of the buoy to the pistons. As a consequence some mechanical components are required outside the base structure of the OG, surrounded by seawater that is potentially devastating for such mechanical components. Nevertheless, if such problems can be overcome, it gives you the advantage of implementing an already proven mechanism into the OG.

Another direction might be a more rigorous one. A future direction might be to analyze the possibilities to place all the mechanical components that translate the movement of the buoy into the movement of the piston inside the base structure of the windmill. By doing so, the complete sealing system at a large depth is prevented. There might still be some sealing needed, but when this is place nearer to the surface, the costs related to the maintenance of such sealing systems are expected to decrease significantly.

Last, it is worth investigating to find methods that make the replacement of the current seals cheaper. If there is a way where the seals can be replaced from the inside, no more divers are needed. As the divers are by far the most expensive part of the maintenance, this would reduce the costs significantly. Another solution might be to design the whole OG in such a way that it is possible to lift the entire base structure. The fact that the sealing system needs frequently reoccurring maintenance is not a problem on its own, the fact that it is so expensive makes it problematic. When the whole structure is lifted to the surface, the replacement of the sealing systems would be less costly.

16 Conclusion

In this research alternative designs for the OG have been analyzed. The requirements related to the pumping structure have been given and it is concluded that there is no alternative pumping system that can be implemented such that the need for a reciprocal sealing system is prevented. As the maintenance costs of the reciprocal seal are determined to be the weakest link of the design, it has been proven that there are no alternative pumping mechanisms that can reduce these costs because all require the same (or a similar) sealing system.

An interesting alternative for the sealing system has been presented. EPDM is chosen as the material to be used. The difficulties regarding this alternative have been explained in detail. From preliminary simulations, it has been proven that EPDM, or in a more general sense a pure elastomer, is not strong enough for this application. Nevertheless, the alternative has some promising potential and more research is needed to find a material, or a hybrid of multiple materials, that can be used for this proposed sealing system.

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Part V
Appendix

17 COMSOL model

Two COMSOL models have been created to analyze the stresses acting on the *local passive seal*. COMSOL Multiphysics is a general-purpose simulation software for modeling designs, devices, and processes in all fields of engineering. COMSOL allows its users to combine multiple physics to obtain accurate models that consider a wide range of possible operating conditions and physical effects.

The models created for this thesis uses solely the structural mechanics physics. As the OG is situated under water, one can argue that a combination of the structural mechanics and laminar flow physics would be appropriated. However, this complicates the model significantly. In addition, the function of the laminar flow physics would solely be to include the hydro-static pressure. This hydro-static pressure can also be included in the structural mechanics study by applying a pressure to the flexible membrane.

17.1 Model Bending Stress

17.1.1 Assumptions

The following assumptions are used:

- A constant pressure of 3bar is applied to the material
- A stationary study is performed, neglecting any dynamic behavior
- The strain that should be in the outer part of the membrane is left out. In addition, the strain that occurs due to the bending of the material is also neglecting. As a result, no strain is inside the material before the simulation is started

17.1.2 Geometry

For the 2D model of the local seal, the geometry depicted in figure 42(a) is created. The right bar represents the wall of the base structure, the left bar is the rod and the domain located between the two bars is the geometry of the flexible membrane. 42(b) shows the geometry of the 2D axis-symmetry simulation. The entire geometry is almost the same compared to the 2D case, except for the width of the rod. As the simulation is 2D axis-symmetric, the entire geometry is circled around the dashed red line. Therefore the width of the rod is halved for the 2D axis-symmetric case.

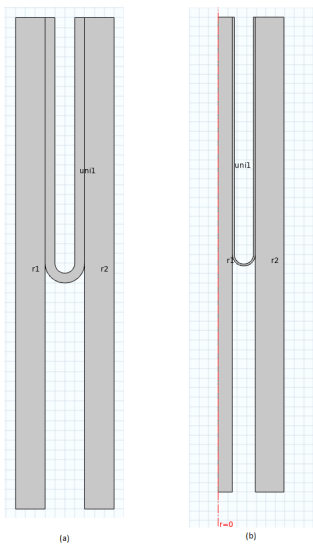

Figure 42: *(a) The geometry of the 2D study on the local passive seal. (b) The geometry of the 2D axis-symmetric study on the local passive seal. Note that the dimensions are changed to increase the visibility*

As mentioned in section 14.3.2, the thickness determines the exact width between the two ends of the material. Increasing the thickness results in a smaller width between the two ends, in order to keep the annulus area constant. This is adjusted for every simulation, where the width is calculated using the following equation:

$$w = 1/2\sqrt{4A/\pi + (d_r + 2t_b)^2} - d_r - 2t_b \tag{15}$$

where A is the annulus area, d_r is the diameter of the rod, and t_b is the thickness of the bladder.

17.1.3 Material

For the simulation, two different types of materials are used. For both the base structure and the rod concrete is used. For the flexible membrane, a blank material is created with the mechanical properties of EPDM that is defined in table 1. Although the rod will not be made out of concrete, it is not relevant for this simulation, as will be explained in the next section.

17.1.4 Physics

As mentioned already, the solid mechanics physics are used for this simulation. First, a fixed constraint is applied to both the base structure and the rod. As only the stresses within the flexible membrane are of relevance, the only function of both the wall and the rod is to constrain the movement of the flexible membrane. The top part of the flexible membrane domain is also fixed. Contact pairs are created such that the model recognizes the boundaries of all the separate domains. A boundary load applied on all inner edges of the flexible membrane domain is simulating the hydro-static pressure. Most importantly is the hyper-elastic material constraint that is applied to the flexible membrane domain. The first and second Lamé parameters are defined as follows:

$$\lambda = \frac{E\nu}{(1+\nu)(1-2\nu)} \tag{16}$$

$$\sigma = \frac{E}{2(a+\nu)} \tag{17}$$

where λ and σ are the first and second Lamé parameter respectively. In the context of elasticity σ is also called the shear modulus and it sometimes denoted by G. E, and ν is the Young's Modulus and the Poisson's Ratio of the EPDM respectively.

17.1.5 Mesh

A normal automatically generated mesh is applied on all domains, which, is in this case, a free tetrahedral mesh. As small deformations in the flexible membrane are expected, one could suggest implementing a moving mesh (ALE) in the simulation. However, COMSOL is able to automatically move the mesh in case of a deformation in the structural mechanics physics. The size of the mesh is dependent on the simulation and ranges from the setting *finer* to *extremely fine*.

17.1.6 Solver

For the 2D model, a fully coupled solver is implemented. Due to many convergence problems that arose when using the segregated solver it is chosen to use a fully coupled one instead. A fully coupled solver is more robust but also tends to use more capacity of the computer. As it is a relatively simple 2D simulation, however, this did not result in long computation times. For the 2D axis-symmetric simulation the segregated solver was used as the computation was already much longer compared to the 2D simulation. For both simulations are relative tolerance of maximally 0.01 was used to ensure accurate results. It is also tried to create a 3D model, but this took too much time (over 3 hours for a simplified model alone). An automatic highly nonlinear method is used in the solvers to control which damping factor is used in the damped Newton iterations. This method can be used for highly nonlinear problems and improved the convergence significantly

In order to improve the convergence rate of both simulations, an *auxiliary sweep* is implemented to ramp up the applied pressure. Comsol has difficulty in estimating the initial values before starting the simulation when a large pressure is applied at once. As a result, the simulation is unable to find realistic initial values and is therefore not able to converge. Because of the *auxiliary sweep*,

the simulations first run for a small pressure and then implements the solution of the previous step as initial values for the next step. Even when those steps are still too large, Comsol is able to automatically choose a smaller step instead. By doing this, the convergence of the simulations improved significantly.

17.2 Model Radial Expansion

17.2.1 Assumptions

The following assumptions are used:

- A constant pressure of 3bar is applied to the material
- A stationary study is performed, neglecting any dynamic behavior
- Only the radial expansion is taken into consideration, any vertical expansion is neglected.

17.2.2 Geometry

The geometry that is created for this simulation is shown in figure 43. The left domain shows the material, the right domain is the wall. The material is placed at a distance of half the width of the rod plus the thickness of the bladder from the axis of the revolution. The distance between the material and the walls is calculated with equations 15.

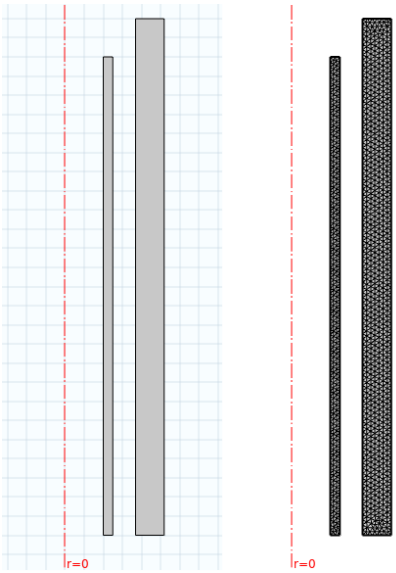

Figure 43: *The geometry of the COMSOL simulation to analyze the radial expansion (left). An extremely fine mesh is applied on the geometry (right). Note that the dimensions are changed to improve the visibility*

17.2.3 Material

The same material set-up is used compared to the previous simulation.

17.2.4 Physics

As mentioned already, the solid mechanics physics are used for this simulation. First, a fixed constraint is applied to the right domain, which is the wall. As only the stresses within the flexible membrane are of relevance, the only function of this wall is to constrain the movement of the flexible membrane. Contact pairs are created such that the model recognizes the boundaries of all the separate domains. A boundary load applied on the inner edges of the flexible membrane is simulating the hydro-static pressure. A prescribed displacement is applied on the top edge of the material domain which allows for a displacement that is equal to the width calculated with

equation 15. Most importantly is the hyper-elastic material constraint that is applied to the flexible membrane domain. Here the mechanical properties are used which are defined in table 1. The first and second Lamé parameters are defined as follows:

$$\lambda = \frac{E\nu}{(1+\nu)(1-2\nu)} \tag{18}$$

$$\sigma = \frac{E}{2(a+\nu)} \tag{19}$$

where λ and σ are the first and second Lamé parameter respectively. In the context of elasticity σ is also called the shear modulus and it sometimes denoted by G. E, and ν are the Young's Modulus and the Poisson's Ratio of the EPDM respectively.

17.2.5 Mesh

The same mesh is used compared to the previous simulations.

17.2.6 Solver

The same solver set-ups are used compared to the previous simulations