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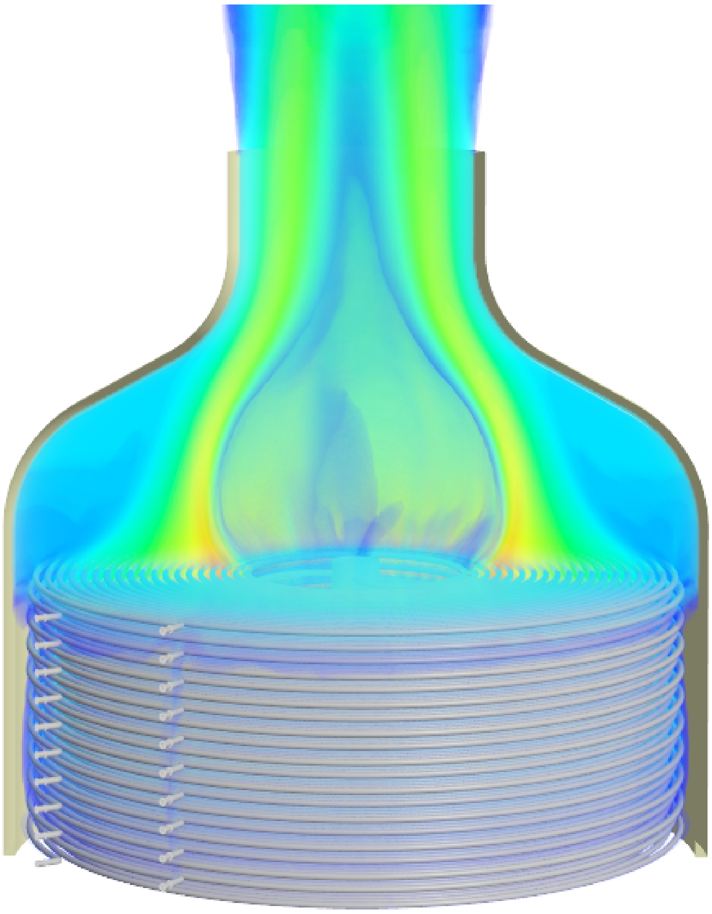

Optimization and Evaluation of Subsea Cooling for the Offshore Wind Industry

A Combined CFD and Market Evaluation for a Passive Subsea Cooler

Master's thesis in MPSES and MPQOM

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Cover: Rendering of Subsea Cooler from CFD Simulations.

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Abstract

With an expanding energy market and an increasing focus on developing green, reliable and sustainable energy, the demand has never been greater for new innovations meeting these criteria. Today, offshore wind stands as a front-runner amongst the renewable energy alternatives. For the technology to be truly competitive against fossil fuels, the method of transporting large amounts of generated power long distances must be improved. High Voltage Direct Current (HVDC) has proven to be a promising technology offering low transmission losses over greater distances. However, the conversion process from Alternating Current (AC) to Direct Current (DC) which takes place on offshore HVDC platforms is generating a significant amount of heat, resulting in a high cooling demand. Conventional cooling systems are energy consuming and utilize chemicals in the cooling process which are discharged to sea.

This thesis has investigated an alternative cooling solution, utilizing subsea cooling and a closed loop. The alternative cooling solutions is free of chemicals and was found to be far more energy-efficient in terms of operating cost compared to conventional cooling. The thesis also screened the future market for HVDC platforms in the North Sea and its cooling demand, by conducting a progressive market screening. It was concluded that the market is set to triple in terms of amount of HVDC platforms to be established and seven-fold in terms of installed converting capacity over the next seven years. A passive subsea cooler, operating in a closed loop utilizes natural convection and has a unique design to withstand shallow water conditions and lower coolant temperatures. Computational fluid dynamics (CFD) simulations have been performed to evaluate different cases of pipe arrangement and chimney designs to investigate the performance of the proposed subsea cooler. Manufacturing possibilities and their implications were also investigated together with a techno-economic discussion which presents various design alternatives.

The study has concluded that, compared to existing technologies in the market, the closed loop cooling system decreases overall operating energy consumption and removes chemical pollutants that previously have been the base case for cooling offshore HVDC platforms. CFD simulations were proven to be an important tool in testing the passive subsea cooler performance. It was identified that baffles and large temperature differences are crucial in enhancing a passive subsea cooler. The thesis also managed to reduce computational time for the full 3D simulation by 96%, to test the chimney design.

Keywords: offshore wind, passive cooler, high voltage direct current, offshore platform, computational fluid dynamics, market screening, subsea cooling

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Jonathan Blomberg, Gothenburg, May 2024
Elsa Täck, Gothenburg, May 2024

List of Acronyms

Below is the list of acronyms that have been used throughout this thesis listed in alphabetical order:

AC	Alternating Current
CAPEX	Capital Expenditure
CFD	Computational Fluid Dynamics
CuNi	Copper Nickel
DC	Direct Current
ESG	Environmental, Social and Governance
HVAC	High Voltage Alternating Current
HVDC	High Voltage Direct Current
ID	Inner Diameter
OD	Outer Diameter
OHTC	Overall Heat Transfer Coefficient
OPEX	Operating Expense
OWF	Offshore Wind Farm
SWLP	Sea Water Lift Pump

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1
Introduction

This chapter describes the background to the issue being investigated and how CFD can be applied. Additionally the problem is described, together with the purpose, aim, limitations and research questions the thesis aims to answer.

1.1 Background

In a time characterized of climate change and increasing green house gas emissions, the need for green, reliable and renewable energy has never been more significant. COP28 in Dubai concluded the first global assessment under the Paris Agreement which measured the progress towards the climate goals set by the agreement. Renewable energy and energy efficiency was highlighted as the most important aspects to mitigate climate change and it was agreed amongst the party members to triple global renewable energy capacity and double the rate of energy efficiency improvements by 2030 [4]. Among the renewable energy alternatives available today, offshore wind stands as a front-runner offering almost limitless opportunities. The International Energy Agency predicts that the global offshore wind industry is set to 15-fold its capacity over the next two decades turning it into a 1 trillion dollar industry, with potential for further growth as climate targets are becoming more ambitious, making the industry even more attractive for investors [5]. Offshore wind could be deployed in large scale without having an impact on land use, meaning that offshore wind turbines could be much larger, than onshore ones, allowing for an increase in energy output per wind turbine produced. The wind speeds offshore are generally higher and more reliable, offering a mitigated intermittence of the energy production, which currently is a challenge for renewable energy production.

Currently, offshore wind is expensive to build and operate, resulting in slimmer margins than for instance fossil fuels, such as oil and gas [6]. For offshore wind to be competitive against alternative energy sources, such as oil and gas it has to cope with transporting electricity long distance and avoiding severe energy losses in the process. Offshore wind farms (OWF) today are located more than 70 kilometres from shore and generate large quantities of alternating current (AC) power through its wind turbines. Transporting AC power such distances generates significant losses, resulting in a need to convert the electricity into direct current (DC) and ramp up the voltage before being transmitted to shore. The conversion process takes place on a high voltage direct current (HVDC) offshore platform located in close connection to the OWF and becomes significant as the positioning of OWF are becoming greater from shore [7]. When the electricity has reached shore, it is converted back to AC to be used by consumers. The conversion process on the platforms generates a byproduct of heat that has to be removed in order to not damage the conversion and transformation equipment. Since the platform is placed offshore in harsh environment the equipment has to be enclosed on the platform as to avoid saltwater damage which makes cooling by ambient air difficult. The function of the cooling system is therefore critical in order for the OWF to be operational. A failure in the cooling system would result in a shutdown of several wind turbines or the entire OWF, which is costly both in terms of lost revenue due to decreased production and excessive cost of sending service personnel to the platform. The only commercially available technology today to cool these platforms is by an open loop system which utilize seawater for cooling. This technology have a rather high operating cost, requiring electricity that has to be scavenged from the OWF. The system also uses chemicals that are discharge to sea which has proven to have an environmental impact. As an alternative to the open loop system a closed loop system utilising cooling through natural convection has been investigated. A new type of subsea cooler has therefore entered the preliminary design stages to serve as a passive cooler in a closed loop system, which is further investigated in this thesis.

1.1.1 Societal, Ethical, and Ecological Aspects

The development and implementation of a new subsea cooler used for offshore wind, will have an impact on the society as well as the environment. Offshore wind is a crucial energy source that will play a major role in the green transition. Innovative solutions that allow renewable energy production to stay competitive against fossil fuels is of great importance in order to achieve the green transition. Furthermore, the deployment of an innovative subsea cooler for offshore wind will not only represent a leap forward in the green transition but also offer the benefit of new jobs being created in local communities and economic development.

1.1.2 CFD Background

Computational Fluid Dynamics (CFD) is a field of fluid mechanics that uses numerical analysis to solve and analyze problems involving fluid flows. By employing powerful computational techniques, CFD allows engineers to simulate the behavior of fluids under various conditions, providing detailed insights into the interactions between fluids and surfaces [8].

CFD is particularly valuable tool for evaluating the performance of a subsea cooler and has been used by company in previous subsea cooler projects. It enables visualization and analysis of fluid flow patterns around and through the cooler, helping to identify areas of high resistance that can impact efficiency. Additionally, CFD simulations model the thermal performance of coolers by analyzing heat transfer processes, allowing for predicted temperature distributions and ensure optimal operating temperatures. Moreover, CFD helps in optimizing the cooler's design by simulating different configurations and operating conditions. Different geometries and materials could be tested virtually, leading to improved designs that maximize cooling efficiency in harsh offshore environments without extensive physical prototyping. Traditional testing and prototyping methods are often expensive and time-consuming, especially for subsea equipment. CFD reduces the need for physical prototypes by providing accurate virtual simulations, significantly cutting down on development time and costs. By predicting potential performance issues and failure points, CFD also helps in identifying and mitigating risks early in the design process, enhancing the reliability and safety of subsea coolers [8].

1.2 Problem Description

With major energy companies rapidly shifting their focus from fossil fuels, which have been the main energy source for decades, to green and renewable energy, there is a growing demand for new innovations to drive the energy market forward. Subsea coolers has previously been developed for oil and gas applications which are placed in much deeper depths than offshore wind which requires much stricter standards, adding additional cost to the cooler. Furthermore, the cost of offshore wind is comparatively high and therefore all of the components of an OWF has to be evaluated in order for it to be a competitive energy alternative. Since the subsea cooler and its associated system is in a concept phase, the actual performance of the cooler and the system is still not determined. This also applies to the manufacturing aspect of the cooler where there also is a need for further investigating. In order to make the technology commercially viable, the concept has to be proven and evidence of why this technology is a better alternative have to provided.

1.3 Purpose

The overall purpose of the thesis is to have a dual perspective. Meaning that both commercial aspects such as market analysis and manufacturing possibilities and its cost implications is investigated together with the technical aspect that CFD is offering by investigating the cooler performance. The final aim is then to merge the findings to provide a conclusion whether development of such a subsea cooler utilising a closed loop cooling system is worth it or not. The aim is that the conclusion is based on both perspective, meaning that it rests on both the market outlook, the associated cost savings with a closed loop system and how the design of the cooler performs. To arrive at this conclusion the thesis aim to use CFD to try different designs and evaluate how that effects parameters such as, outlet temperature, heat transfer etc. Additionally the thesis aims to conduct a technical market analysis to arrive at an estimated market outlook for HVDC platforms and its cooling demand and the cost implications of a subsea cooling system compared to a conventional cooling system. Finally, the aim is to investigate the manufacturing possibilities from a cost perspective.

1.3.1 Limitations

The thesis will focus on the cooler being used for offshore wind applications requiring offshore HVDC platforms, meaning that the inlet and outlet temperatures temperature are determined with this application in mind. Regarding the market analysis it is limited to only focus on the North Sea. Regarding the cooling system, it is currently in a concept stage meaning that data are used for similar application and not specifically for HVDC platforms. Additionally, a limitation is the intermittency of wind power, since the production is not continuous, electricity prices can vary significantly between full load hours based on demand from consumers, which is not considered in this thesis, were average electricty prices are used.

For the CFD analysis any structural components will not be included in the simulations in order to reduce computational time. Another limitation of the thesis the OHTC is based on the mesh area, which depends on the surface refinement and amounts of points on the circle, which means the area is smaller than in reality. Articles found and investigated mostly discuss heat exchangers with air on the outside of the tubes, which deviates from the thesis cooler.

1.3.2 Research Questions

How can CFD facilitate the optimization of a new design of subsea cooler, and how can different design aspects affect the performance?

- To what degree can the testing and evaluation of a subsea cooler model be improved by simplifying its features to reduce computational time in CFD simulations?
- How can a subsea cooler be optimized when investigating the relationship between design and performance?

What are the key market dynamics and factors influencing the successful entry of the new subsea cooler in the wind power market, and how can strategic marketing and positioning be leveraged to ensure a sustainable competitive advantage in the evolving market landscape?

- To what extent can existing technologies and knowledge be innovatively applied to meet the demands of the new changing market?

1.4 Thesis Outline

The thesis contains chapters and sections based on the content required for the reader to have an understanding for the structure of the thesis. Each chapter introduces a new stage in the thesis, and the sections are designed to divide the information into logical order, beginning with the larger market, and successively working towards the technical aspects and the product specifications.

The thesis is initiated by a **review of current HVDC platforms and its associated cooling system** located in the North Sea and a deeper investigation about alternative cooling systems to and the subsea cooling system in detail, which serves as a foundation for the cost comparison between the systems.

The **Theory** chapter describes the underlying theory for heat transfer and heat exchangers necessary for the CFD analysis and the computational models and equations that are important considerations such as solvers and mesh theory.

The **Methods** chapter is initiated by general considerations for writing a thesis such as a literature review and data collection. This is then followed by the method of the market and cooling system analysis. The design basis of the thesis is described followed by the construction, meshing, and simulating set-up for the CFD analysis, which is concluded by the model design parameters. Finally the costing estimation is described.

Results and Discussion chapter similar to the previous chapter is initiated by the market analysis and mapping results, followed by the cooling system analysis results. For the CFD analysis the mesh refinement study is presented, followed by some preliminary CFD results and the model validation. The final cooler design is then presented and discussed together with the product costing and weight estimation results.

In the **Conclusion** chapter, the thesis summarize its findings in regards to the market, CFD and manufacturing possibilities.

2

A Review of Current HVDC Platforms & Its Cooling System

As an initial step of the thesis a review over current offshore HVDC platforms and its associated cooling systems was conducted. The current market is used for comparison between the market screening at a later stage in the thesis.

2.1 Offshore HVDC Platforms

The need for HVDC offshore platforms has emerged over the last couple of years, as more and more OWF have been established further from shore where there is an increasing need for transmitting generated electricity longer distances [9]. The development in technology and increased demand of green energy has made OWFs grow and significantly increase its generated electricity. The biggest advantage presented by HVDC technology lies in its significantly reduced transmission losses compared to HVAC systems. This feature is of great importance, particularly in the transportation of substantial quantities of electricity [10]. The wind turbines that makes up the OWF are generating AC, which creates a need to convert the generated AC electricity into HVDC which is more suitable for long transportation of electricity to shore [7]. Hence, offshore HVDC platforms has to be installed in close proximity to the OWF and collect the electricity from several wind turbines were it is converted into DC and the voltage is increased. The platforms are connected to large HVDC interconnected subsea export cables that are transmitting the electricity to an Onshore Converter Station (OCS), where it is being converted back into AC to be used by consumers. The platforms are placed in harsh conditions and have to cope with all sorts of weather, which makes it crucial to ensure high standards for the equipment to guarantee reliability.

2.1.1 Operational HVDC Platforms

The majority of OWF established today are using HVAC to transmit electricity to onshore consumers. The reason being that wind turbines generate AC and the electricity does not have to converted and could be transmitted directly to shore. The advantage is that the current does not have to be converted to DC, hence there is no need to establish offshore converting infrastructure such as HVDC platforms.

The rather high investment cost of building the HVDC infrastructure required is

Figure 2.1: Reference image of HVDC platform [1]

avoided and many cases that is the cheaper alternative [7]. In recent years however, more and more OWF operators have chosen to opt for DC rather than the AC, which is mainly due to the fact that OWFs are being established further from shore, which means that the electricity has to be transmitted further distances. Today there are nine operational HVDC platforms in the North Sea that are all are owned by the Dutch energy company TenneT. The platforms are serving multiple wind farms in the North Sea outside Denmark's, The Netherlands's and Germany's coast and transmitting HVDC to OCS where the electricity is being converted back to AC and fed into the grid. The OWFs providing power to the HVDC converter platforms are of comparatively smaller scale, typically ranging between 200 to 450 megawatts (MW), in contrast to the larger-scale wind farms which are planned and developed. The distance to OCS are ranging from 130 - 200 km, and the wind farm capacity is for each associated platform are ranging from 400 - 900 MW [3]. The platforms are positioned in the DolWin-, SylWin-, HelWin-, and BorWin Cluster, which is outside Germany's northwest coast in the North Sea. The HVDC platforms are listed in Table 2.1, where the distance to onshore converter station (DTOCS), capacity and commissioning year is listed. The analysis over current HVDC platforms shows that the the average DTOCS is 165 km and installed conversion capacity is 6,845 GW, which yields an average of approximately 760 MW. The average depth of where these platforms are positioned was also investigated and where found to be approximately between, 40-70 meters. This is however limited by the foundations for anchored wind turbines and in line with the design depth of the cooler.

Table 2.1: Operational offshore HVDC platforms [3]

HVDC Platform	DTOCS (km)	Capacity (MW)	Commissioned
DolWin Alpha	165	800	2015
DolWin Beta	135	916	2016
DolWin Gamma	160	900	2017
SylWin Alpha	205	864	2015
HelWin Alpha	130	575	2015
HelWin Beta	130	690	2015
BorWin Alpha	200	400	2015
BorWin Gamma	160	900	2019
BorWin Beta	200	800	2015

2.2 Cooling Systems for Offshore HVDC Platforms

The conversion of AC to DC for offshore wind farms is necessary to mitigate energy losses in the transportation process and enables for wind energy to become a cost competitive alternative. In the conversion process a significant amount of heat is generated which has to be drawn away in order to not damage the equipment, which creates a cooling demand [7].

2.2.1 Current Cooling Systems

Current cooling systems on HVDC platforms for offshore wind utilize an open loop cooling solution, where cold water is pumped up to the platform where the cooling process takes place and then discharged back to sea. The cool sea water is passing through a topside heat exchanger positioned topside on the platform where the hot cooling medium is cooled and pumped to the cooling consumers, i.e. converters and transformers on the platform. To maintain a cold stream of sea water to the platform a Sea Water Lift Pump (SWLP) is used, which continuously pumps sea water up to the heat exchanger on the the platform. The cooling system is therefore consisting of two separated loops connected by the heat exchanger, one open sea water loop and one closed cooling medium loop, located on the platform. The open-loop system is the most effective way of cooling HVDC platforms and the only system in place for operational platforms today. Allowing for the most reliable, effective and cost-efficient solution for AC to DC conversion offshore [7]. In order for the sea water loop to function properly and mitigate required maintenance, the sea water is filtered before it being pumped into the heat exchanger. Coarse filters is placed after the inlet, where sand grains and other marine species, such as plankton or fish is filtered out before reaching the heat exchanger. These filtration systems are often backflushed to decrease maintenance and allow for continuous operation to ensure no unwanted objects are entering the heat exchanger, which may result in a blockage of the sea water loop. When the filtration system is backflushed, some of the marine species caught in the filters are returned back to the ecosystem. However, many

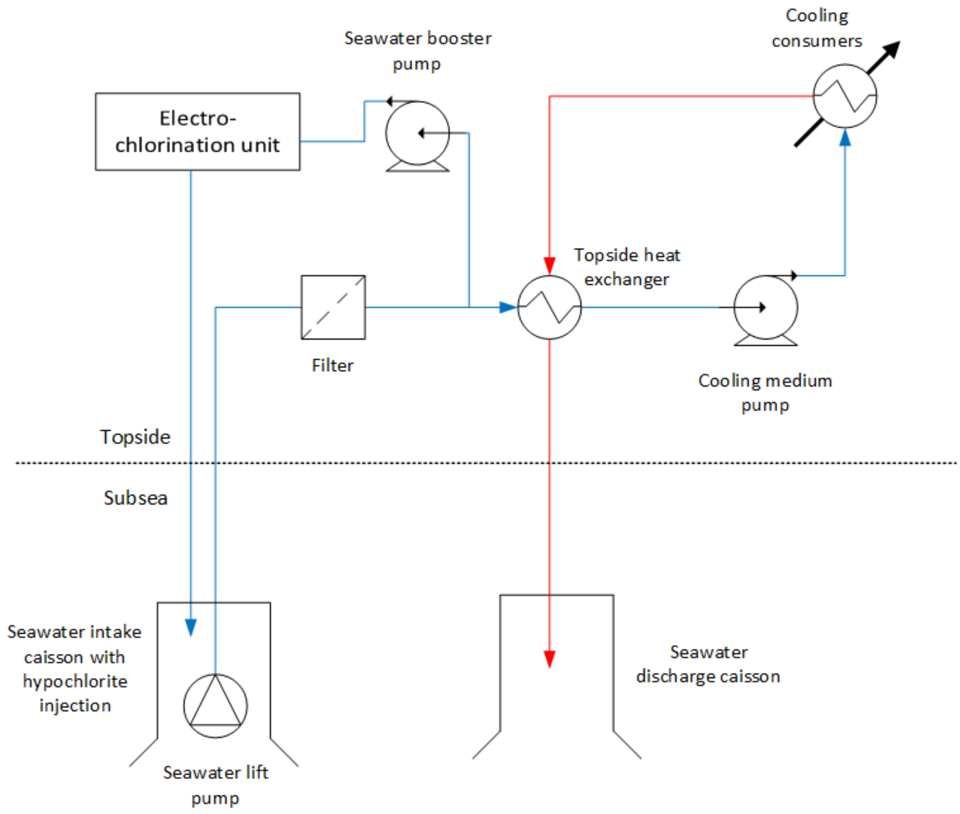

Figure 2.2: Conventional cooling loop

marine species are not able to mature and reproduce once it has been flushed out of the filtration system [7]. Figure 2.2 is a schematic over the cooling loop and its components.

To mitigate the risk of biofouling in the open cooling loop, a dose of sodium hypochlorite is injected into all areas where marine biofouling is at risk of occurring and preventing marine growth in exposed areas from starting. The sodium hypochlorite The hypochlorite is produced in the electrochlorination units, by passing electric current through the sea water. To perform the electrolysis of sea water to produce a chlorinated solution, the sea water has to be filtered, which is already done by the filters earlier in the loop. Then the sea water is passing through an electrolyzer cell's channel of decreasing thickness, where one side is a cathode and the other an anode. The electrolysis takes place when a low voltage DC is applied producing sodium hypochlorite and hydrogen gas [11]. After the sodium hypochlorite has been used in the cooling loop it is discharged to sea. Equation 2.1 is showing the chemical process of producing sodium hypochlorite through electrochlorination.

$$NaCl + H_2O + ENERGY \Rightarrow H_2 + NaOCl \tag{2.1}$$

To control the flow of cold sea water entering the heat exchanger sea water booster pumps are used. In many platforms today three sea water booster pumps are used, where one is a spare for redundancy. The booster pumps are variable speed pumps, meaning that the out put rate could be controlled in order to be aligned with the cooling need on the platform. Since the sea water is variable throughout the year, being warmer in the summer and colder during the winter, the pump rate of supplying sea water to heat exchanger has to be adjusted accordingly. During most of the time, only one sea water booster pump is active and the second one is activated only in cases when the sea water is exceptionally warm. The increased flow rate of cold sea water will increase the cooling capacity of the system to cope with warmer sea water temperatures.

2.2.2 Subsea Cooling System

A subsea cooling system is a new concepts when it comes to cooling HVDC platforms and is utilizing a closed loop rather than an open seawater loop as in the conventional cooling loop. The overall philosophy of the system is to move the entire cooling process subsea, by using one subsea cooler. Instead of lifting seawater to the platform the entire cooling process is taking place subsea by utilising the natural convection and the cooler seawater to draw away heat from the cooling medium. Subsea cooling would mean that, several components could be eliminated, which includes, pumps, the electrochlorination unit and the filter which is illustrated in 2.2. The electrochlorination unit is eliminated since there is no risk for biofouling when using a closed loop, since the system is not continuously supplied with fresh seawater. This would mean that no sodium hypochlorite is discharge to sea and a mitigated environmental impact. A subsea cooling system would also free up space on the actual platform, otherwise occupied by the pumps and the electrochlorination unit. The only pump present in the subsea cooling system is the cooling medium circulation pump that would be placed topside on the actual platform. There also no need for an intake caisson nor a discharge caisson that is needed in the conventional cooling system. The passive subsea cooler illustrated in Figure 2.3 is the same cooler as described in Figure 2.3 that is placed on the seafloor. The hot cooling medium coming from the cooling consumers would enter through the inlet header, pass through the piping bundle were it is cooled and then be pumped up to the cooling consumers again and closing the loop.

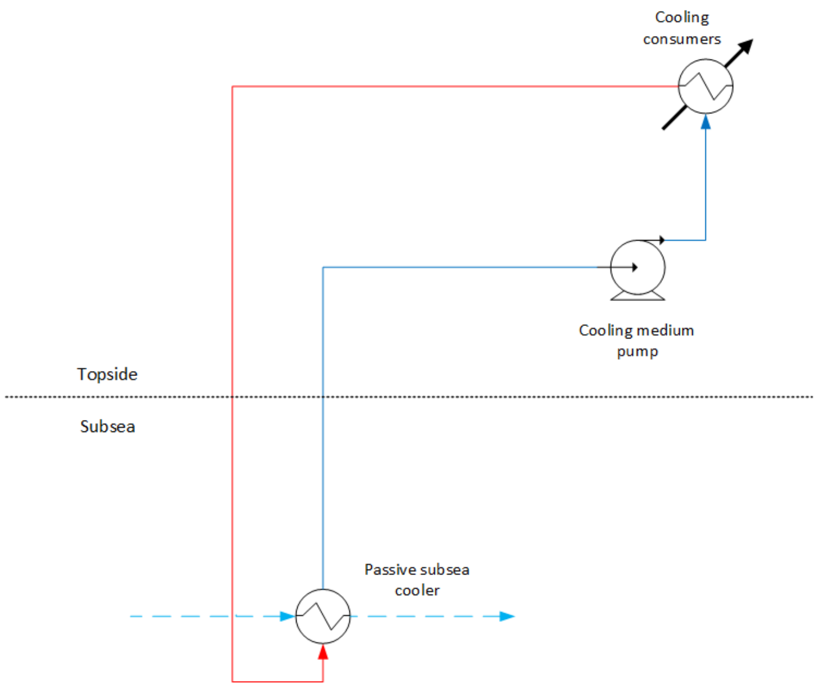

Figure 2.3: Subsea cooling loop

2.2.3 Alternative Cooling Systems

The corrosive nature offshore limits the alternatives to other cooling solutions than the one described under 2.2.1. The electrical equipment has to be enclosed to stay protected, from salt water and lightning, which limits air cooling capabilities. Commonly air is used to cool the water in a closed-loop system, by having a large bank of electric powered fans cooling the water. When taking the cooling requirements and limited space on the platform into account for such a solution, it would be hard to motivate an air cooled solution over an open looped water cooled. Having several fans is also resulting in an excessive amount of movable parts, which requires continuous maintenance and electric consumption that would have to be scavenged from the wind farm, resulting in decreased output from the wind farm to shore. Other cooling systems uses refrigerant gases to draw heat from the water and dissipates it to ambient air. The refrigerant gases is often chlorofluorocarbons and hydrofluorocarbons that has to replenished over time and the dissipated heat from the water has to be removed by using fans which is power consuming and not suitable to highly corrosive offshore conditions. This solution would also most probably result in a higher overall cost of operating the cooling system [7]. By investigating several different alternatives to cool the conversion process on the HVDC platforms, it was concluded that the current open loop cooling system and the subsea cooling system are the most suitable.

2.3 The Subsea Cooler

The passive subsea cooler illustrated in Figure 2.3, is illustrated in more detail by a 3D model provided by company, see Figure 2.4. This initial 3D model is used throughout the thesis and the figure displays the names of each part of the cooler to have consistent terminology usage throughout the thesis.

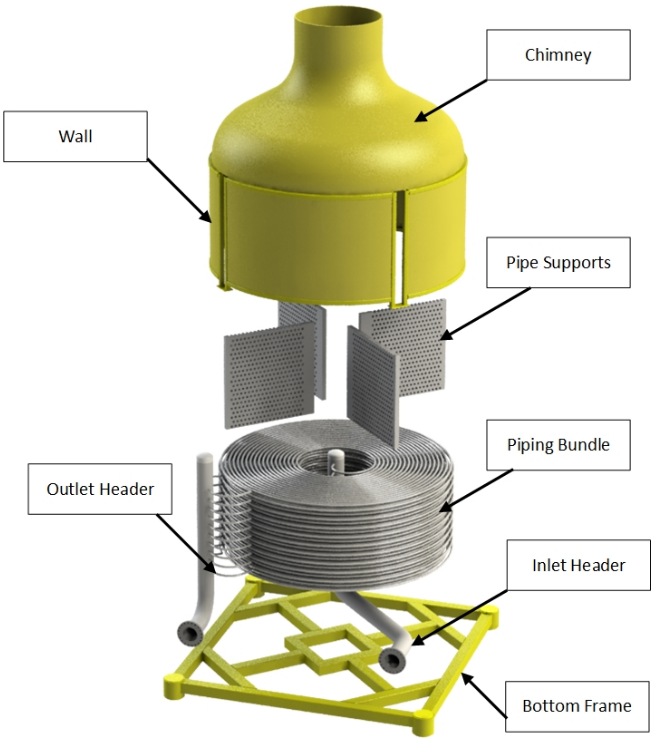

Figure 2.4: Exploded view of subsea cooler

Each part of the cooler has been specified below:

- **Chimney** refers to the final part sitting on top of the wall support. This part will be tested in the technical design and manufacturing possibilities will be investigated.
- **Wall** refers to 4 round sheets attached through the wall support. This part will be simulated as one sheet.
- **Pipe Support** will not be investigated in this thesis.
- **Piping Bundle** refers to the 20 **pipe stacks** that together forms the part transporting the cooling medium through the heat transfer process. This part will be tested in the technical design and manufacturing possibilities will be investigated.
- **Inlet Header** refers to the pipe transporting the cooling medium into each pipe stack. Pumped from the HVDC platform.
- **Outlet Header** refers to the pipe transporting the cooling medium out of each pipe stack. Pumped to the HVDC platform

- **Bottom Frame** will not be investigated in this thesis.

An important consideration for the thesis is that the seawater being the cooling medium for this cooler is referred to as seawater. While the fluid inside the pipes being the cooled medium becomes the cooling medium when looking at the application which is why henceforth the fluid inside the pipes are referred to as the cooling medium or fluid.

3
Theory

This thesis investigated a subsea cooler with CFD, which requires a lot of background understanding of transport phenomena. This chapter is meant to act as a guide for the thermal, and fluid dynamic problems that is solved, as well as suggesting conventional design requirements for a thermodynamic CFD problem.

3.1 Heat Transfer

The two modes of heat transfer relevant for the thesis was conduction and convection. Thermal conductivity is molecular interaction where with higher energy (temperatures) the greater motion of a molecule affects the energy of adjacent molecules with lower energy levels. Which will be applied to the pipes and specified as a solid physics model. Convective heat transfer occurs with energy exchange between a surface and an adjacent fluid [12]. There are two types of convection free and forced, forced convection will be used in the cooling fluid domain, and can be defined as the fluid (at a temperature) flowing in a tube (with a different temperature) at an average velocity imposed by an external agency such as a fan or pump. The relation between conduction and convection can be described in terms of boundary layers in which the closest boundary to a stationary surface the fluid will be laminar, while turbulence will cause bulk mixing of the fluid between regions at different temperatures. There are a few significant parameters in convective heat transfer, the Prandtl number being the ratio of the molecular diffusivity of momentum to the molecular diffusivity of heat, and the Nusselt number being the ratio of conductive thermal resistance to the convective thermal resistance of the fluid [12].

In order to evaluate the convective heat transfer coefficient there are four methods; dimensional analysis which require experimental results; exact analysis of the boundary layer; approximate integral analysis of the boundary layer; and analogy between energy and momentum transfer. Generally overall heat transfer is described with temperature difference and through a series of thermal resistances due to the following:

- the thermal resistance due to heat transfer from the process fluid to cooler tube
- the thermal resistance due to conductive heat transfer through the pipe walls
- and the thermal resistance due to heat transfer from the cooler tubes to the seawater

Creating heat transfer coefficients which for convection is the limiting factor compared to the conduction. The inner heat transfer coefficient varies with fluid composition and velocity. For dense process fluid, the outer heat transfer coefficient is generally significantly smaller, which means it would govern the overall heat transfer [13]. While the inner and outer thermal resistances may be comparable for less dense fluids. The sum of the total thermal resistances between two fluids determines the value U or the overall heat transfer coefficient (OHTC). In order to relate this coefficient to heat transfer of an heat exchanger, the temperature difference must account for both the inner and outer heat transfer mediums, which is why the Log Mean Temperature Difference (LMTD) is used as defined below:

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} \tag{3.1}$$

Where ΔT_1 is the difference between the hot fluid inlet and the cold fluid outlet temperature, and ΔT_2 is the difference between hot fluid outlet and cold fluid inlet. With this information the heat transfer can be calculated with the following equation:

$$Q = UA\Delta T_{LM} \tag{3.2}$$

Where Q is the heat transfer, A is the heat transfer area, and U is the OHTC.

Natural convection or free convection as Incropera [14] refers to it as, occurs due to buoyancy forces within the fluid inducing natural currents to move through the fluid, compared to forced convection which is externally imposed. The buoyancy occurs with a fluid density gradient and a body force that is proportional to density, usually gravity. Since density of both liquids and gases are heavily dependent on temperature. This will be used for the seawater domain. Generally the Reynolds number is used on a fluid element to measure the ratio of the inertial to viscous forces, in a forced convection problem. However for free convection the Grashof number is used on a fluid to measure the ratio of the buoyancy forces to the viscous forces, which when no forced convection occurs will have the only effect (together with Prandtl) on the Nusselt number to characterise the convective heat transfer.

3.2 Heat Exchanger Design

The simplest type of heat exchangers are a concentric tube with hot and cold fluid moving in the same or opposite direction. There are also heat exchangers where the fluid moves in cross flow over finned and unfinned tube banks. The most common industrial heat exchanger is the shell and tube where baffles are of importance to increase the convection coefficient of the shell-side fluid. Baffles induces turbulence and a cross flow velocity component relative to the tubes, additionally the tubes are supported by the baffles reducing flow-induced vibrations. [14]

The shell and tube heat exchangers are active heat exchangers where a fluid is pushed through both the tubes and the shell from an external source. There are however passive heat exchangers where natural convection is utilised based on the temperature difference of the pipe fluid to the ambient surrounding fluid. For the subsea industry this is a useful tool as seawater has an immense cooling power. The active heat exchangers are currently the base case for offshore HVDC platforms as discussed in 2.2.1, where the seawater is pumped through the heat exchanger. This thesis however aims to introduce passive heat exchanger as a replacement for these as to reduce pump operations.

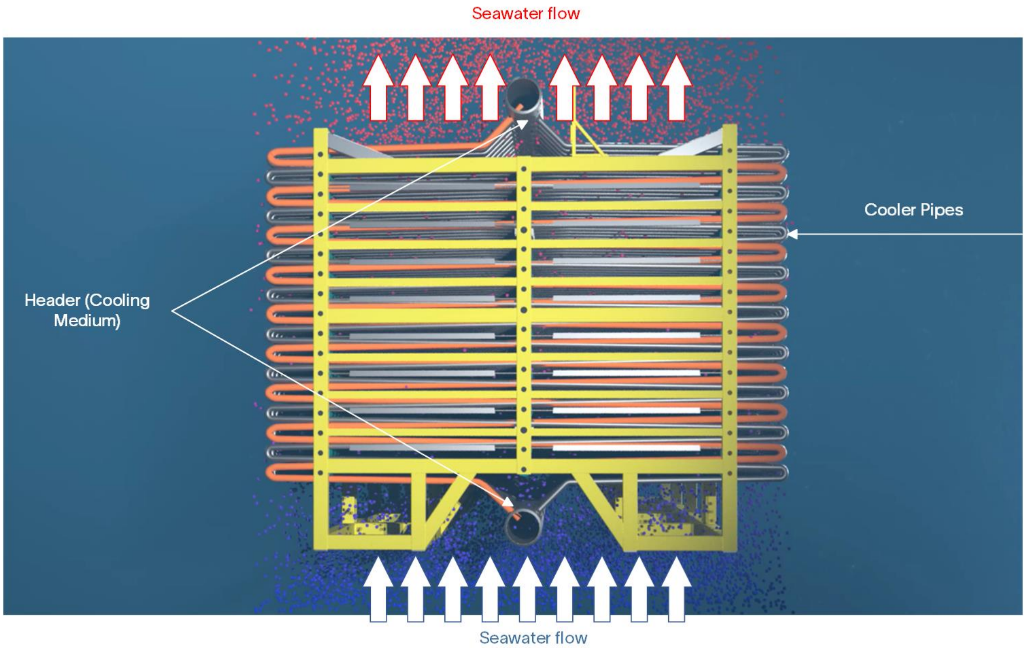

Figure 3.1: Reference passive cooler

Passive heat exchangers are investigated by Liu and Sakr [15] in order to find the parameters that greatly enhance the overall thermal performance. Generally for convection enhanced heat transfer is achieved with increasing effective surface area and residence time of the heat transfer fluid. There are several techniques to achieve this, passive heat exchangers are based on these principles. Techniques to increase heat transfer of passive coolers range from surface treatments and extensions, swirl flow devices and addatives to the fluids. However the conclusion of this investigation was that for turbulent flow passive techniques such as ribs, conical nozzle and ring are generally more efficient.

3.2.1 Pipe Arrangements

The article by Khan, Zou and Yu [16] explored the characteristics of air-side heat transfer and pressure drop of staggered twisted tube bundles in cross-flow. Where the adjusted parameters can be referred to as transverse tube pitch S_T, longitudinal tube pitch S_L, diagonal tube pitch S_D and longitudinal tube rows A_1, which are clarified in figure 3.2.

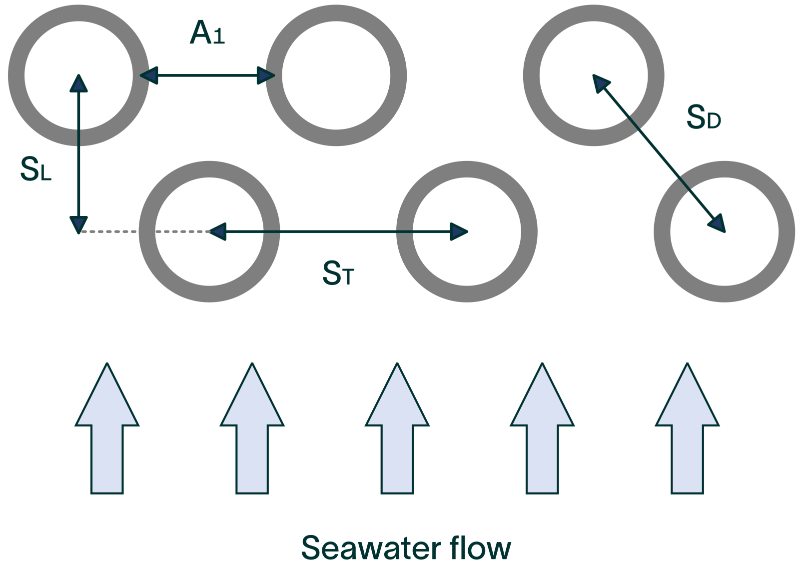

Figure 3.2: Example of tube arrangement parameters

It was found when increasing the transverse tube pitch (from the base case being 2*Diameter) the Nusselt number and pressure drop decreases. The authors verified this finding with three separate studies which suggested that the turbulence flow increased with the increased pathway between the tubes, which gave lower maximum velocity, thus smaller transverse tube pitch is more appropriate with consideration of higher flow resistance. The effect of increase in longitudinal tube pitch increases the Nusselt number and pressure drop which coincided with another study found. As the longitudinal and diagonal tube pitches increases, the air distribution flows perpendicularly to the twisted oval tubes, thus more proportional movement is allowed near the wall and mixing close to the tubes are enhanced.

With more tube rows a notable increase was observed in both Nusselt number and pressure drop. Due to the increased surface area the performance of heat exchangers are dependent on the number of tube rows, which has been validated by two other studies [16].

3.2.2 Chimney

The work conducted by Kumar [2] investigated the physics of natural convection of air over a finned tube kept in a small chimney utilizing transient 3D numerical simulations. The simulation model assumed the Boussinesq approximation was valid, which generally is the case for lower temperature cases. The geometry of the model can be seen in figure 3.3.

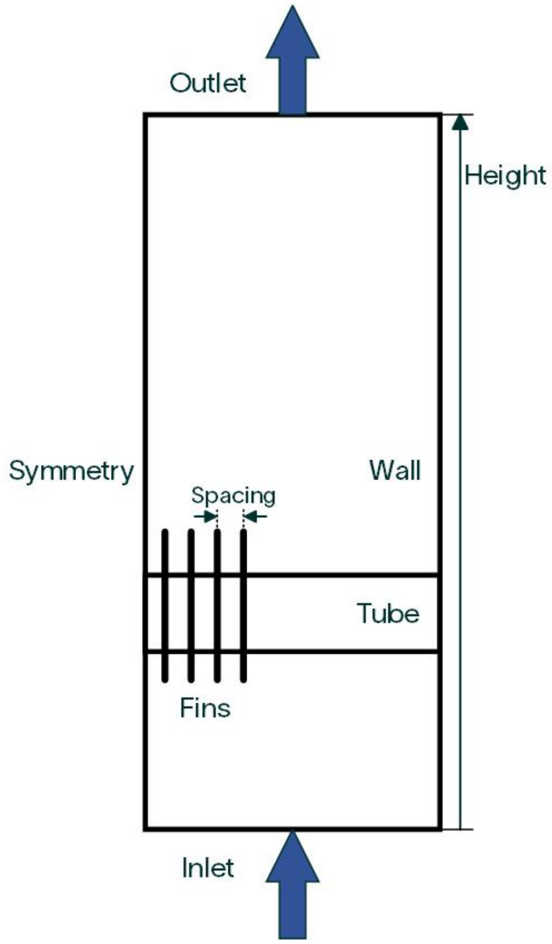

Figure 3.3: Example of the chimney set-up by Kumar [2]

The investigation of the chimney was in two parts, the first showed the effects of the chimney height which as the height increased resulted in greater heat transfer coefficient. The second part was how fin spacing affected the chimney effect and showed the smaller spacing between the tubes the chimney effect decreased. The chimney creates two major effects the first increases the driving force, and the second is a decrease in the air outlet temperature, with an increase in chimney height. The paper also investigated the transient behaviour of natural convection in the chimney, where the heat transfer is high initially, and then drops before reaching steady state. Which indicates that the simulations for natural convection should be more conservative.

3.3 Physics Models

Computational fluid dynamics (CFD) is used to describe continuity, momentum, energy and species transfer between fluids. Traditional modelling base heavily on empericial and semi-empirical models, which often are not reliable for new process conditions. Therefore new design equations and parameters in existing models must be determined in order to accurately describe the changing equipment or process conditions outside the validated experimental database. Therefore CFD is a useful tool in describing continuity, momentum and energy transfer in a steady state simulation. However in order to solve turbulence in steady state, transport equations are required.

Single-phase laminar flow can be simulated accurately as well as for turbulent flows the simulations can be reliable. However for most cases and as is the case for this thesis complex turbulent flow are currently difficult to predict. Due to the properties of turbulence the Navier-Stokes equations are used to describe the flow however even with super computers it is difficult to solve these complex engineering problems. Especially with turbulence fluctuations where DNS and LES modelling would not handle the large amount of data, therefore most simulations use the Reynolds averaged Navier-Stokes (RANS) methods [8].

The paper by Paul [17] deep dives into different CFD models to accurately simulate tube bundles. It was observed that the transverse turbulent intensity was significantly higher than the stream wise turbulent intensity. The overall performance of the models observed the k-based two equation models appeared closer to the experimental data than the particular second moment closure model (LRR-IP), especially the k-ε model were in agreement with the measured values. For the mesh generation k-ε and LRR-IP models showed less grid sensitivity than k-ω and SST models. Based on the discussions presented by the article and a previous course [8] it is known that for inside pipes where the walls are of interest the k-ω turbulence model is preferred, while surrounding the pipes the k-ε models are used. Deciding which k-based transport equations to use for each region was mainly based on internal knowledge.

3.3.1 EB k-ε

The elliptic blending turbulence model solves the turbulent kinetic energy, k, and dissipation rate, ε, transport equations, the normalised (reduced) wall-normal stress component φ, and the elliptic blending factor α in order to determine the turbulent eddy viscosity. The elliptic relaxation was proposed by Durbin for Reynolds stress models [18]. The model was simplified for industry applications by Manceau and Hanjalic [19].
The transport equations:

$$\frac{\partial}{\partial t}(\rho k) + \nabla(\rho k \bar{v}) = \nabla \cdot [(\frac{\mu}{2} + \frac{\mu_t}{\sigma_k})\nabla k] + P_k - \rho(\varepsilon - \varepsilon_0) + S_k \tag{3.3}$$

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \nabla(\rho \varepsilon \bar{v}) = \nabla \cdot [(\frac{\mu}{2} + \frac{\mu_t}{\sigma_\varepsilon})\nabla \varepsilon] + \frac{1}{T_e} C_{\varepsilon 1} P_\varepsilon - C^*_{\varepsilon 2}\rho(\frac{\varepsilon}{T_e} - \frac{\varepsilon_0}{T_0}) + S_\varepsilon \tag{3.4}$$

$$\frac{\partial}{\partial t}(\rho \varphi) + \nabla(\rho \varphi \bar{v}) = \nabla \cdot [(\frac{\mu}{2} + \frac{\mu_t}{\sigma_\varphi})\nabla \varphi] + \rho\frac{\varepsilon_0 \varphi_t}{k_0} + P_\varphi + S_\varphi \tag{3.5}$$

$$\nabla \cdot (L^2 \nabla \alpha) = \alpha - 1 \tag{3.6}$$

3.3.2 SST k-ω

The k-ω turbulence model solves the transport equations for the turbulent kinetic energy, k, and the specific dissipation rate, ω which is the dissipation rate per unit turbulent kinetic energy $\omega \propto \frac{\varepsilon}{k}$ to determine the turbulent eddy viscosity.

$$\frac{\partial}{\partial t}(\rho k) + \nabla \cdot (\rho k \bar{v}) = \nabla \cdot [(\mu + \sigma_k \mu_t)\nabla \omega] + P_k - \rho \beta^* f_{\beta^*}(\omega k - \omega_0 k_0) + S_k \tag{3.7}$$

$$\frac{\partial}{\partial t}(\rho \omega) + \nabla \cdot (\rho \omega \bar{v}) = \nabla \cdot [(\mu + \sigma_\omega \mu_t)\nabla k] + P_\omega - \rho \beta^* f_{\beta^*}(\omega^2 - \omega_0^2) + S_\omega \tag{3.8}$$

The SST variant model addresses a problem of sensitivity to free-stream/inlet conditions in the standard model remedied by Menter [20]. Where the following transport equations have an additional non-conservative cross-diffusion term containing the dot product $\nabla k \cdot \nabla \omega$ in the ω production term.

$$P_\omega = G_\omega + 2\rho(1 - F_1)\sigma_{\omega 2}\frac{1}{\omega}\nabla k \cdot \nabla \omega \tag{3.9}$$

3.4 Boundary Conditions

Boundary conditions are used to define the fluid domains, and the inlet, outlet and wall conditions are equally important for the simulation results as the differential equations. The usual condition for non-inlet/outlet boundaries is the wall boundary, which uses the 'no-slip condition' stating the relative velocity between the wall and fluid is zero[8]. For heat transfer applications the wall can be defined according to heat transfer type, temperatures or heat flux, in this thesis the conjugate heat transfer condition is used as well as fixed temperature.

Another important consideration regarding the Navier-Stokes equations is the initial guess. Since the equations are non-linear, the better initial conditions the better the solution will converge, demanding less computational time. If the start guess is out of order and the problem has several solutions the solution will diverge, which usually requires transient simulations to be solved.

Stagnation inlet is used as an inflow condition used for both incompressible and compressible flows. It refers to the condition in an upstream imaginary plenum where the flow is completely at rest. For incompressible flows Bernoullis equation is used to relate total pressure, static pressure and velocity magnitude. Such as the boundary pressure given by:

$$P_s = P_{t,spec} - \frac{\rho}{2}|v|^2 \tag{3.10}$$

The velocity magnitude is extrapolated from the interior of the domain:

$$|v| = |v|^{ext} \tag{3.11}$$

The velocity vector is given by:

$$v = |v| \cdot \theta_{spec} \tag{3.12}$$

For a non-isothermal simulation the static temperature is calculated based on:

$$T_s = T_{t,spec} - \frac{|v|^2}{2C_p} \tag{3.13}$$

Where the density at the boundary face is then updated according to the static temperature:

$$\rho = \rho(T_s) \tag{3.14}$$

The total and static enthalpies are given by:

$$H_s = H_s(P_s, T_s) \tag{3.15}$$

$$H_t = H_s + \frac{|v|^2}{2} \tag{3.16}$$

The **mass flow inlet**, is for applying a specified mass flow at a inflow or outflow. The specified total mass flow rate is distributed over all faces of the boundary and calculates a uniform mass flow rate on each face.

The **pressure outlet** imposes the working pressure, the boundary pressure can be considered the static pressure of the environment which the fluid enters.

3.5 Important Considerations

When using Star CCM+ to simulate CFD problems many important considerations have to be taken into account. Which includes that the turbulence models use the All y+ wall treatment physics to describe the turbulent behaviour at walls. Other considerations are listed in the chapters below.

3.5.1 Solvers

In order for the program to solve the transport equations, it must be selected if the flow equations should be solved for each component in a segregated, or coupled manner. Segregated meaning the flow equations are solved for each component pressure then each component velocity and are linked together the momentum and continuity equation with a predictor correction factor. While the coupled solver solves the conservation equations for mass and momentum together using a pseudo-time marching approach. The coupled solver require more memory however is more applicable for compressible flows and high Rayleigh number for natural convection. The segregated solver is actually designed for constant density flows, it can still be applied to compressible or low Rayleigh number natural convection flows. [21]

Since the thesis will model a large 3D problem with a high computationally heavy load, the segregated solver is more applicable. Even though the coupled solver might converge in less iterations it requires a lot more fine tuning to achieve stability. The industry (internal) experience for this type of product (and temperature differences we have) have shown that the segregated solver converges faster but tend to under predict the thermal performance of the cooler. This means that the results will be on the conservative side and an increased performance is expected in reality.

3.5.2 Natural Convection

Natural convection is the driving gravitational forces on a fluid due to density differences in the domain. The density of the fluid is most often affected by temperature changes due to heat transfer. For this thesis this will be the driving force for moving seawater through the cooler. In order to accurately describe this phenomenon stagnation inlet is applied, as well as some common practices highlighted by Siemens Star CCM+ team, which also has been verified in industry applications [22]

- Ensure y+=1 at the heat transfer regions of the seawater domain.
- For liquid flow a temperature dependent density should be used where polynomial density in function of temperature is the most convenient way, thus polynomial in T is utilised.
- Selecting the solver is important, as coupled solver would be more robust for natural convection even if segregated solver is more conservative. The coupled solver is more difficult to use however recommended for high Rayleigh numbers.
- More common practices are mentioned however based on this thesis these three are the most applicable for our model

3.5.3 Mesh Theory

Creating a mesh in CFD is a complex procedure, but usually included in a commercial software packets for mesh generation. For 3D unstructured generation builds the mesh from different elements (tetrahedra, hexahedra, pyramids, prisms and docecahedra). An important consideration for the mesh generation is the walls which is why most applications require a surface remesher to be established before the volume mesh can start. In order to make sure the mesh is accurately representing the domain Andersson [8] suggests that using different grid spacing in the critical regions of the grid is a good way to avoid divergence of the results. Especially for flows with boundary layers which requires a dense mesh close to the wall, while regions far away from the mesh can be coarser. Andersson suggests these decisions will be based on intuition and previous knowledge, which for this thesis has been a crucial part in the decision making of the mesh generation. As the company has many years of experience with testing different settings and discovering what is reliable and not.

4
Methodology

This chapter presents the general methods used for the thesis and the methodology regarding the market screening, cooling system analysis and how CFD was used. Design basis values for the CFD is also presented in this chapter.

4.1 Thesis Structure

Since one of the initial aims of the thesis was to have dual perspective, combining both CFD and commercial aspects for the subsea cooler that where to be developed, the philosophy behind the methodology was to initially have a broad focus that would narrow down to focus on solely the cooler as the thesis was progressing. The broader focus of the thesis consisted of screening competitive technologies that would pose a threat against the evaluated subsea cooling technology. The result of the screening resulted in a cost comparison between HVDC and HVAC that was used to screen the market for suitable HVDC platforms and establish the demand and cooling capacity.

Following the market screening, a CFD analysis was conducted, investigating the cooler performance of the subsea cooler, by looking at the effect of adding baffles and how well the chimney performed. The conventional cooling system was also compared to the closed loop cooling system, utilising subsea cooling. Where the focus was on overall power consumption of the two systems and environmental impact. The remaining focus of thesis was aimed towards the subsea cooler based on results provided by CFD, where the original design was evaluated in terms of manufacturing possibilities. The entire cooler was split up into its sub-assemblies where every sub-assembly was treated independently in order for the analysis to be as comprehensive and structured as possible. In other words, the piping bundle was treated as one independent part as well as the shell and chimney encompassing the cooler bundle.

4.2 Literature Review

To develop a deeper understanding of the topic of HVDC platforms, CFD analysis and subsea coolers a literature review was conducted. The literature review served as a basis for the thesis by integrating and comparing already established information around the topic with the results provided by the thesis. For a literature review to become trustworthy and possess a high level of credibility different approaches can be taken [23]. For the thesis a semi-systematic literature review was selected as the most suitable approach, due to the somewhat lack of literature covering especially subsea cooling for HVDC platforms. Subsea cooling for HVDC platforms is a new concept, hence the amount of published articles being limited. The semi-systematic literature is more suitable when the research questions are narrower and there is not enough literature available to conduct a systematic literature review [23].

The semi-systematic literature review was initiated by keyword searches as "Offshore HVDC platforms", "Natural convection", "subsea cooling", etc., across Google Scholar and Chalmers Library as the main database. The first step was to screen titles and abstracts of selected articles to determine if they were relevant for the research. Secondly, if the articles were found relevant, they were further investigated and used in the thesis. In many cases the references in the selected articles were also used to dive deeper into the topic and gain even better accuracy for the research.

4.3 Data Collection and Data Processing

The data used throughout the thesis consisted of both quantitative and qualitative data. The data typically origin from either published articles that were found relevant or from industry experts within the collaboration company through interviews. To gain information and data from within the company was crucial to provide reliable results, since the topic in many aspects were in an early stage and therefore more difficult to find in published articles. This methodology strategy of using both qualitative and quantitative data was used to leverage the strengths of both types of data to address the research objectives to the largest extent. Since the aim of the thesis was to provide hands on numerical results with the use of CFD and costing estimates the mix of the two data types was required and is also a methodology that is widely used among several researchers [24]. Regarding data collection, the vast majority of the thesis relied on secondary data gathered from published sources, such as academic literature, industry reports from relevant companies within the offshore industry and government publications. Data used for the cost comparison between HVAC and HVDC was gathered from recent published bidding contracts. The reason secondary data was used as the main source of data was mostly due to the complexity and time consumption of gathering primary data. Since the evaluated cooler is intended for offshore subsea conditions it would require traveling to these locations to gather primary data for the design basis, which was outside of the limitations of the thesis. The same reasoning is applied to the market analysis and cost evaluation where the only data attainable is provided from other companies. However, the data was cross-checked between several sources and in many cases the average of numerical data was used to maintain credibility of the results, which is in line with what academic papers treating research methodology is suggesting [25]. The collected data was processed and capture in Microsoft Excel throughout the thesis. Meaning that all of the calculations for the market screening and cooling system analysis was done by this software. The same software was used to store the results of all the simulations, to use for comparison at a later stage in the thesis.

4.4 Market Screening

The research objective of the market analysis was to establish an understanding of the predicted market of size for HVDC platforms. The market screening was conducted as a progressive market screening, meaning that the scope of the market screening was continuously narrowed down to finally arrive at the HVDC platforms that were of interest for the thesis [26]. Since the HVDC market is correlated with the expansion of OWFs the initial data collection was aimed at gathering data over planned OWF. The research area was later narrowed down to focus solely on OWFs that planned to use HVDC instead HVAC. The market analysis had a prospective perspective meaning that it emphasized a focus on the future, anticipating the potential outcomes and trends forming an outlook on the market for HVDC platforms. The data collection regarding information of planned OWF wind farms was accessible mainly through a database named "TGS 4C Offshore" that compiled data over OWFs that are operational, planned and under construction. TGS 4C Offshore worked like an interactive map highlighting areas where different OWFs were positioned or planned to be positioned. The status of the OWF was indicated by different colours. Where green meant fully operational, orange under construction and light blue and purple that the consent for building an OWF had been authorised. The areas marked by a darker blue colour was specified as development, which meant that development was planned for post 2030. See Figure 4.1 for a snapshot of the map. When using the database, information about the various OWF would show up once an area was zoomed in and selected.

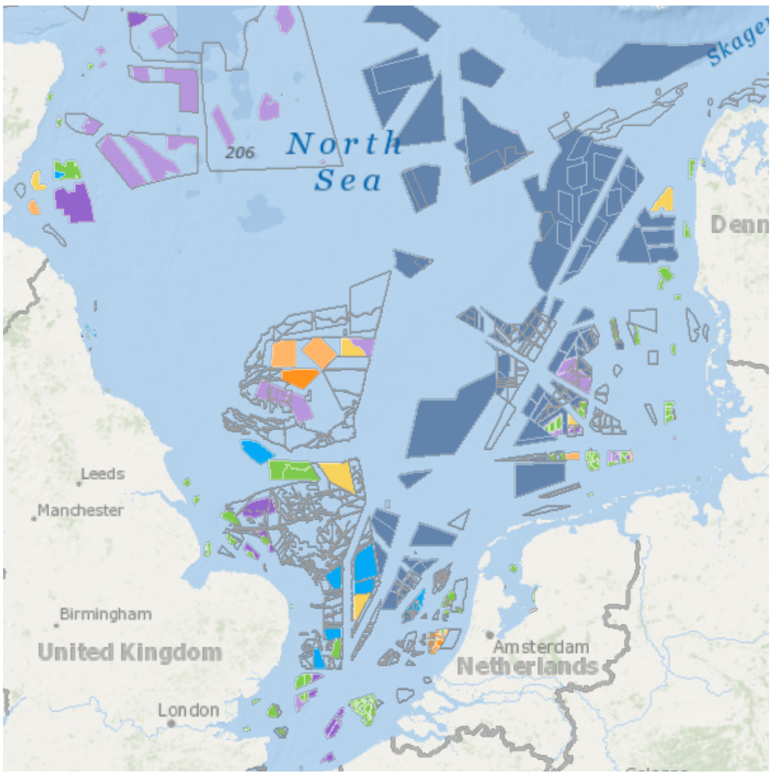

Figure 4.1: Snapshot of TGS 4C offshore interactive map

Regarding the planned OWFs that were of most interest for the screening it was not specified which transmitting technology the wind farm was planning to use. To provide an estimate of which OWF that may be suitable for HVDC, research between HVAC and HVDC was conducted, with the aim of finding a break-even distance for when HVDC was more cost competitive than HVAC. The break-even distance established under Section 5.1.4 was then used a filter to screen out the OWFs that was closer to shore than the break-even distance and therefore assumed to be opting for HVAC rather than HVDC.

4.4.1 Cost Analysis Comparing HVAC and HVDC

In this section of the methodology chapter, a comprehensive market screening was undertaken to examine current HVDC offshore platforms and forecast future projects, to establish a market outlook for the technology. Specifically, the investigation focused on the North Sea region. The aim was to gather insights into existing HVDC platforms in the North Sea and understand why HVDC was preferred over HVAC. This information served as a basis for estimating the future demand for HVDC platforms.

During the literature review, it was observed that HVDC's primary advantage over HVAC is its ability to maintain transmitting voltage over very long distances, which is especially important for offshore wind. To determine which OWFs would benefit economically from HVDC rather than HVAC, a break-even distance was established. Previous studies suggested various distances at which HVDC was more cost-effective, but they focused on different applications and made assumptions that was not updated according to today's climate. For instance, the electricity prices and discount rate used was no longer relevant. Therefore, a new analysis specifically targeting HVDC platforms for OWFs and comparing them with two types of HVAC transmitting technologies from a cost perspective, was conducted. The break-even distance was the result of collecting cost data and technical data from previously built platforms in the North Sea. The cost data for subsea cabling was collected through various published bidding contracts. Data on the costs of other relevant infrastructure necessary for the analysis of both technologies was gathered from academic papers that had conducted similar studies in the same research area. Regarding electricity prices, the average electricity price between 2021 and 2023 for countries with coast to the north sea, excluding Norway was used. Norway was excluded since the majority of the energy mix consisted of hydro power and the price of electricity generated by wind power not being as relevant as other countries bordering to the North Sea.

Initially the costs was split up between investment costs (CAPEX) and operating costs (OPEX). Where the investment costs covered cost associated with building the infrastructure required for both of transmitting technologies evaluated. The operating cost consisted of transmitting losses, discounted over the life time of the OWF. Equation 4.1 was used to establish the present value of the future yearly OPEX to get a justified cost, considering the fact that these cost will occur in the future and has to be put into the perspective of alternative investments.

$$PV = \sum_{n=1}^{30} CF_n \cdot \frac{1-(1+r)^{-n}}{r} \tag{4.1}$$

The sum of the two cost components made up the total cost and the final break-even distance. Microsoft Excel was utilized for data processing, calculations and visualization. The break-even distance served as a criterion to filter OWFs in the TGS 4C Offshore database, identifying those most likely to adopt HVDC technology. A list of HVDC platforms from its associated OWF meeting the distance criteria was compiled as a bar chart as the outcome of the market analysis. The outcome was further analysed by computing the average converting capacity for the identified projects and converting it into cooling demand.

4.5 Cooling System Analysis

The cooling system analysis was focusing on both the environmental impact of the system and the actual power consumption between the systems. To determine the power and cost saving of choosing a closed loop rather than an open loop that pumps sea water to the platform the different eliminated components for the closed loop was further investigated. Firstly, the eliminated sea water lift pumps was analyzed where calculations was made in order to find out how much energy that was required to pump the sea water up to the platform where the heat exchanger was positioned. The circulation pumps was also investigated, together with the electrochlorination unit to determine the total energy consumption of the cooling system which resulted in how much the electricity that would have to be scavenged from the wind farm in order to keep the cooling system functional. Equation 4.2 was used to determine the work required to lift the sea water or cooling medium in both of the cases.

$$P = \dot{m} \cdot g \cdot h \cdot \eta \tag{4.2}$$

To calculate the energy consumption regarding the electrochlorination unit, equation 4.3 was used. T_E represent the total energy to produce sodium hypochlorite at a specific mass flow, expressed in watts. E_{prod} is the energy required to produce one kilogram of sodium hypochlorite through electrochlorination for using seawater and is expressed in kWh/kg. The energy consumption was based industrial equipment's commonly used in cooling systems utilizing seawater as cooling medium, were biofouling mainly occurs.

$$T_E = \dot{m} \cdot E_{prod} \tag{4.3}$$

To represent the cost associated with the required work, the same price of electricity established under the market analysis was used, see Table 5.2. The same values were used to maintain consistency with the previous results provided by the thesis, thereby ensuring uniformity and comparability throughout the analysis. The yearly energy consumption was based on the assumption that the system was constantly operational, 24 hours a day and 365 days a year. In reality the system would probably have some kind of downtime, due to maintenance or failure. But since the aim of the thesis was to compare the two systems and an added scope of investigating the up-time of the two systems was considered to be outside of the limitations.

4.6 Design Basis

The design basis for this project was based on research and discussions with experts. It was divided into material, cooling water loop, and seawater. Since the aim of the thesis was to design a cooler applicable for as many places as possible, the design and testing simulations were done in worst-case conditions. Even though anchored wind farms are usually around a maximum depth of 100m, the cooler should not be excluded to these designs as floating wind farms should also be considered, which have a maximum depth of 300m. Another constraint regarding the design of the cooler was a height restriction, which was considered for the chimney design.

4.6.1 Material

Since the material of the piping tubes are requested to move away from Cu-Ni, stainless steels are appropriate replacements, even though their heat transfer properties are known to be worse, but better corrosion resistance. Stainless Steel has been used as base case for material and was used throughout the whole thesis, due to its high corrosion protection.

Table 4.1: Pipe material specification

Parameter	Value	Unit
Density	8000	kg/m^3
Specific heat capacity	502	J/kg-K
Thermal Conductivity	15.9	W/m-K

The pipe dimensions were designed after a 300m ocean depth, to handle all types of wind farm applications.

4.6.2 Cooling water Loop

The cooling water loop consisted of water with 10% MEG. The properties of the medium was provided by internal company knowledge, and the cooling medium was not further evaluated in this thesis. The mass flow of the medium was assumed based on internal knowledge, since the only constraint was the ability to fill each pipe stack without back pressure. The inlet and outlet constraints for the converter platforms were discussed with experts and found to differ between products and companies, however a range was provided and used as guideline. A temperature difference of 10-20K should be assumed between inlet and outlet. For the outlet (to the cooler) the range was 25-30°C, while for the inlet the range was 35-45°C. With these parameters the specifications were established as in table 4.2, with the aim to create a worst-case scenario.

Table 4.2: Cooling fluid specification

Parameter	Value	Unit
Inlet Temperature	35	°C
Desired Outlet Temperature	25	°C
Mass flow	2.2	kg/s-pipe

4.6.3 Seawater properties

Due to the subsea cooler being designed for worst case scenario all parameters were the max or min of possible outcomes, which means no seawater velocity, a high ambient temperature and salinity. The properties of seawater was provided by internal knowledge, and the specification can be seen in table 4.3.

Table 4.3: Seawater specification

Parameter	Value	Unit
Ambient Seawater Temperature	20	°C
Current velocity	0	m/s
Salinity	35	%

4.7 Building the Model

The dimensions of the coolers were provided by company in a Solidworks assembly, with multiple solidwork parts, some were considered unnecessary for the CFD analysis (see chapter 2.3) and were removed or simplified as to reduce computational time. The simplified parts included the large bend to the outlet header and the outlet header, while the walls and inlet header were removed in the seawater domain.

In order to build the model, trial and error was utilised, where the thesis attempted to build the model from scratch at first, however due to the amount of parameters affecting the inlet and outlets of the pipe geometry, it was decided to use the provided pipe parts. Since the pipes are staggered and the inlet and outlet of the pipe are placed differently. The two different pipe stacks were extracted as parts in Solidworks, and assembled without any other part, to then be patterned to the amount of pipe stacks required for the pipe bundle for each simulation.

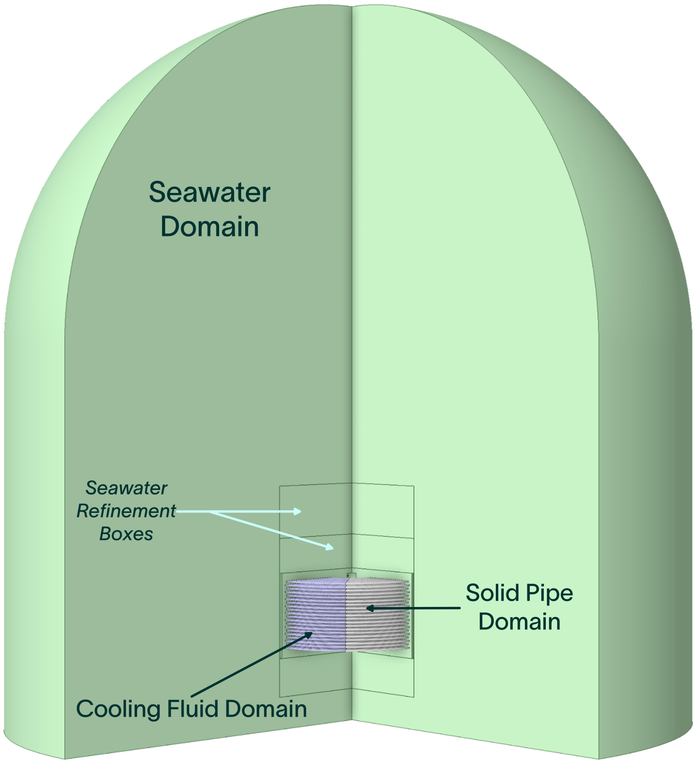

Figure 4.2: Subsea cooler domain setup

The Solidworks assembly with the pipe bundle were then imported into SpaceClaim 2023 R2, which was company standard to import to 3D CAD in Star CCM+. A scene specifying each domain can be seen in figure 4.2. The volume inside the pipes were added through volume extract, and the seawater domain was sketched out and using the 'pull' tool created around the pipe bundle.

4.8 Meshing the model

The mesh controls were based on a previous study on the cooler performed by company. The automated mesh utilised surface remesher, polyhedral mesher and prism layer mesher. To specify the multitude of domains with specific requirements the default controls were lowered slightly however all surfaces were specified to a custom control listed in table 4.4.
These settings were used for the model, however some parts were more critical than others which was why refinement boxes were used as well. The placements of these can be seen in figure 4.2 where the custom size was applied slightly smaller than the seawater. Another refinement box was created directly in Star CCM+ to simulate the plume from the outlet of the cooler.

There were some issues with the mesh from the previous study due to the precarious design of the pipes, the mesh inside and surrounding the pipes were piercing each other therefore the surface size had to be reduced. The mesh settings specified above was finer than previous study, however to check the sensitivity to grid

Table 4.4: Automated mesh custom controls

Surface	Custom Control	Value
1. Cooling Fluid 2. Seawater Pipes Surface	Target Surface Size	Base Size
	Number of Prism Layers	4
	Prism Layer Total Thickness	2.8416mm
Solid Pipes	Target Surface Size	Base Size
	Prism Layer	Disabled
Seawater: Cooler Walls Inlet Header	Target Surface Size	+50%
	Number of Prism Layers	2
	Prism Layer Total Thickness	10mm
Seawater	Target Surface Size	0.4m
	Number of Prism Layers	2
	Prism Layer Total Thickness	0.2m

refinement a mesh study was conducted. Additional prism layers were applied and compared to the original mesh, to see the effects of smaller y+ value inside the pipes would improve or hinder the heat transfer to the seawater. A further refinement of the target surface size was also performed to the effects of refinement outside the pipes. Since the surface size governs how well the circular pipe is built with tetrahedral shapes, if the tetrahedral are smaller more area would be added to create the mesh more accurately describe the pipe surface. Therefore the heat transfer area was expected to increase with smaller surface size, which as introduced in chapter 3.5.3, would benefit the heat transfer to the seawater.

4.9 Simulation Setup

The simulation setup varies depending on what was simulated and was specified in the sections below, however the physics of each medium remains the same. Unless specified no parameters were changed from the default settings of the physics models applied. All physics are three dimensional, steady state and using a segregated solver.

Cooling water medium was specified as a Fluid continuum and the SST K-ω model was set as turbulence model based on recommendations from company. The SST k-omega works well with adverse pressure gradients and was unsuited for the seawater region as it tends to over predict turbulence in regions with large normal strain. The discretization scheme used was specified by the turbulence models default settings. The liquid properties of the medium were modelled using polynomial functions dependent on temperature according to the design basis. The region for the cooling water has mass flow inlet and pressure outlet as boundary conditions, with the no-slip wall condition however contact interfaced with the solid pipe material with a conjugate heat transfer condition.

The pipe material was Stainless Steel for which a solid physics continuum was used, with constant material properties specified in the design basis. The solid pipe region was connected towards the cooling fluid and seawater regions using conformal interfaces that model the conjugate heat transfer across the boundaries.

For the seawater surrounding the pipes it was decided to use EB k-ε turbulence model based on company experience when modeling natural convection for this type of application. As the flow around the pipes were the important areas as well as the turbulence modelling being more robust. As the cooling fluid material properties, polynomial temperature based functions were used to simulate the natural convection, as well as the all y+ wall treatment being of importance. The regions for the seawater utilises stagnation inlet and no slip wall for the seabed. It also had an interface towards the solid pipes.

4.9.1 Pipe arrangements

In order to get started with simulations it was decided to test a simplified model with only 4 pipes with varying parameters. Due to the computational demand of the full size cooler it was decided to test these configurations on a reduced model.

The SolidWorks 3D-model that formed the basis of the design did not contain any fluid domains. As such, the external seawater domain as well as internal cooling water domain was designed in Spaceclaim. This model was then imported into Star ccm+ where the computational model was built. This model included 4 pipe stacks and a cylindrical seawater domain with the same radius as the cooler walls. The volume extruder mesh function was used to create larger domains above and below the imported pipes. The domain was changed two times as issues with convergence occurred, instead of having a cylindrical seawater domain, a cone was placed above the pipes as to create a smaller outlet surface. This issue was due to the pressure outlet boundary as the outlet surface was to large to find a steady state solution and large areas of back flow occurred which was solved by reducing the outlet size. The two domains can be seen in figure 4.3.

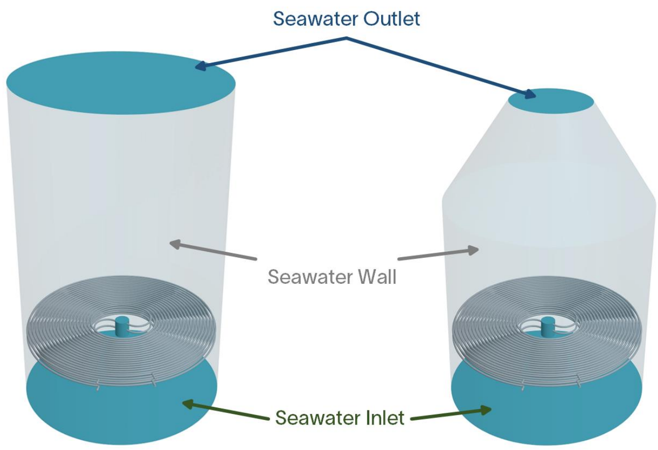

Figure 4.3: Pipe arrangement seawater set-up

Two designs were thought of to test the first simulations. The first adding more spirals towards the inlet header to increase heat transfer area and prevent channeling. The second increasing and decreasing the longitudinal pitch, as suggested in the literature study. The design tests can be seen in figure 4.4.

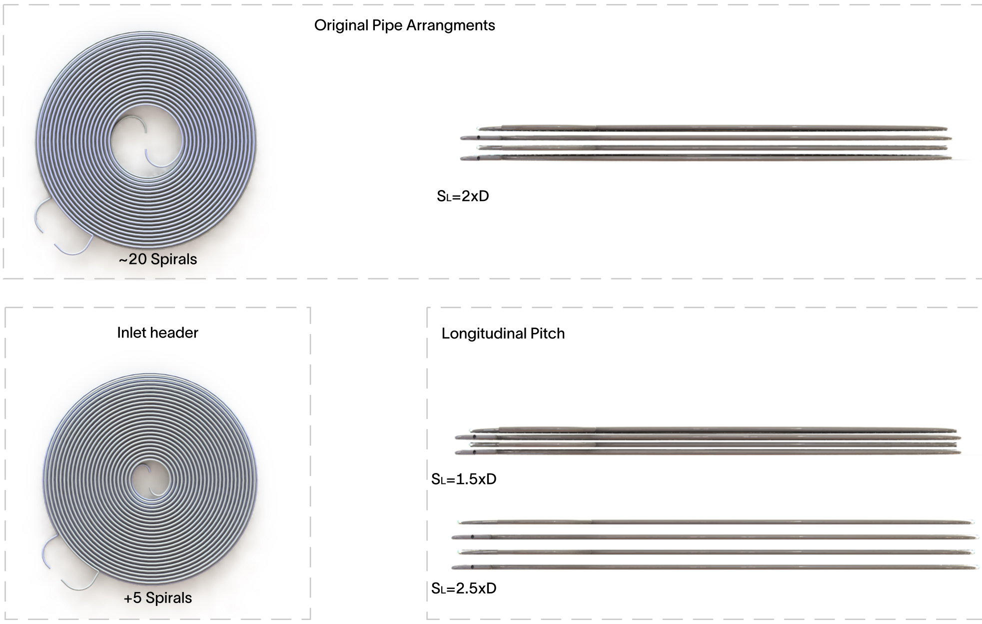

Figure 4.4: Pipe arrangement designs

4.9.2 Full 3D model

In order to verify results of the simplified models the thesis conducted three large scale simulations, one with only walls, one with the chimney, and one with chimney and baffles.

The large simulation model utilised the pipes built in Solidworks and created an assembly with 20 staggered pipes. The assembly was then imported to SpaceClaim where walls, inlet header, seawater domain, and cooling water in pipes were added. The seawater domain was created as a cylinder with a diameter of 4 times the cooler's, and the same height with half a sphere above with a radius of 2 times the cooler.

The boundary conditions in Star CCM+ was the same as described in chapter 4.9, the cooling water had mass flow inlet and pressure outlet. Figure 4.5 displays all boundary conditions applied to the large domain, where cooling fluid inlet and outlet refers to all pipes in the bundle, as the boundary condition was the same for each inlet and outlet. The seawater has stagnation inlet on the sphere and cylinder walls, the bottom circle was considered the seabed which has the wall no-slip condition. The solid pipe has only the wall condition with conjugate heat transfer to both fluid sides.

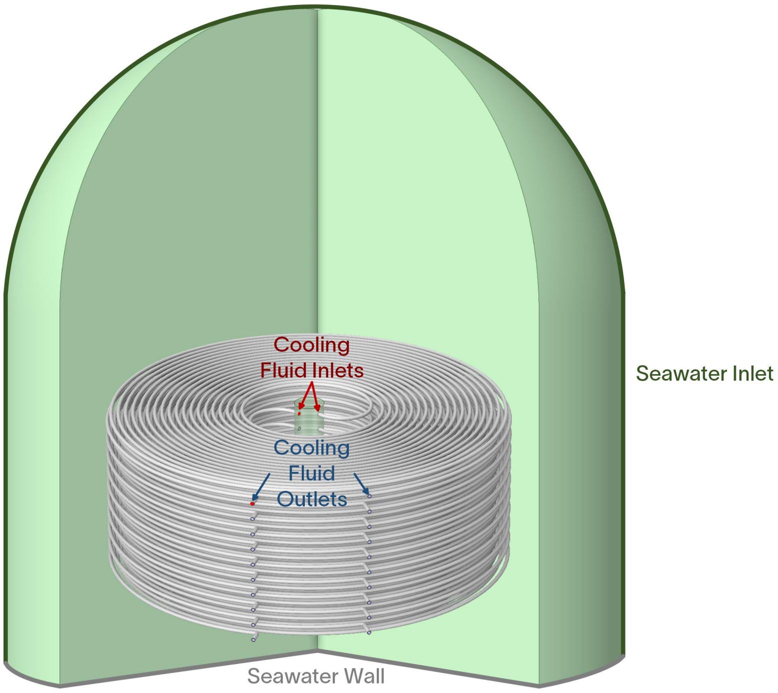

Figure 4.5: Boundary conditions on the cooler

The subsea cooler 3D simulations evaluated the following three cases to benchmark trends:

- Wall: Only containing the walls
- Chimney: Containing the walls and chimney
- Baffles: Containing the walls, chimney and baffles around the inlet header

As stated in chapter 4.7 when the walls and chimney were included they were subtracted from the seawater domain as no information except the seawater flow around the geometry was necessary for the cooler performance.

4.9.3 Model Simplifications

The full 3D model represents a significant computational demand due to the size of the cooler, the computational mesh included above 120 million cells. In order to resolve the physics of the cooler and provide results with a sufficient level of accuracy it was necessary to do model simplifications. As one of the research questions stated, the thesis attempted to investigate simplification methods in order to design the chimney, as this would only require the seawater domain.

At first the chimney was attempted to be designed based on a 2D model of the domain, however after verification the design was invalidated, and another approach of taking a wedge of the 3D model was utilised. In order to do this the thesis only simulated the seawater domain, and assigned temperatures to the outer heat transfer surfaces. The temperatures were extracted from simulation "wall" according to figure 4.6, where the top two and bottom two pipe stack surface average temperatures were reported from the seawater surface around the pipes. These were assumed to represent the average temperature of each half spiral surface, as there were not much difference in between pipe stacks (y-direction).

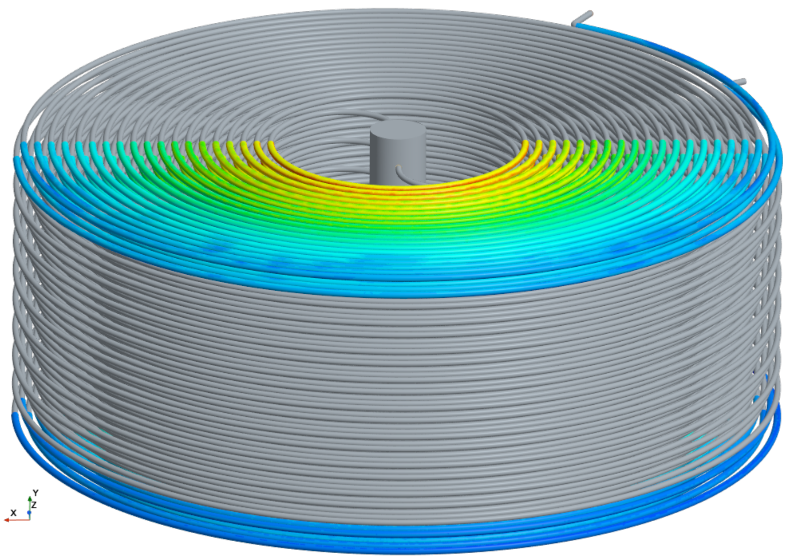

Figure 4.6: Temperature profiles from simulation wall

These temperatures weren't meant to be accurate they were meant to provide a fixed temperature value, when designing the chimney, in order to see trends to assess the heat transfer to the rest of the seawater domain. To design the best chimney.

4.9.3.1 2D model

The chimney design, was decided to be designed based on a 2D model for only the seawater domain. Since the cooler pipes are spirals the tubes and cooling medium cannot accurately be represented, the temperature profiles were applied to the seawater surfaces. The challenge arose when applying the temperature data on each seawater hole surface, where firstly all surfaces were named from top 1-19 and from the inlet header 0-20 as a matrix, see figure 4.7 creating 390 surfaces to be assigned a temperature. This was done in the seawater region through 4 boundaries for each pipe data set, with surface sub-groupings filtering for each pipe surface into Pipe.x.y, which was created with a java script to avoid having to filter and name each subgroup 100 times. The thermal specification was set to fixed temperature and a Java script was utilised which looped the temperatures and assigned them to the correct surface based on their names.

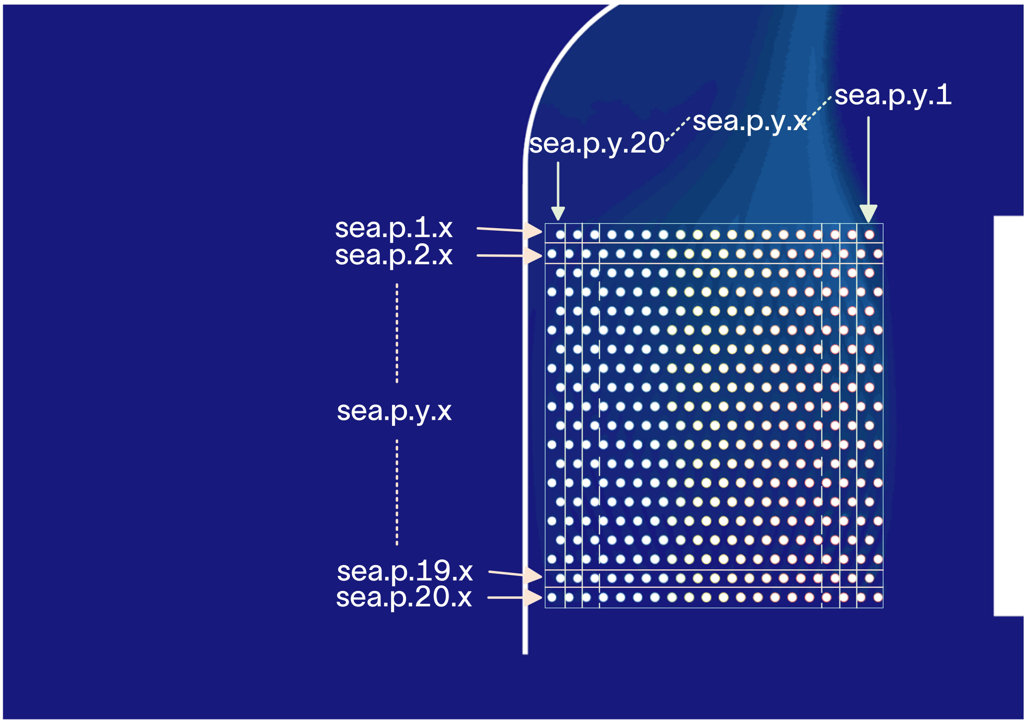

Figure 4.7: 2D model surface naming matrix

It is important to note that the 2D model in star ccm, is not purely 2D it accounts for one cell depth. To model 2D in star ccm+ a two dimensional surface representation was imported which is then extruded 1 mesh cell in the z direction, in order to create a 2.5D. To adjust and design the chimney with ease it was decided for it to be designed in 3D CAD in Star CCM+ with a sketch and cut extrude. In order to verify this simplification a copy of the simulation "Wall", simulation "Chimney" and simulation "Baffles", was created and the trends were compared.

4.9.3.2 Wedge model

The wedge model was the second attempt of simplifying the designing of a large domain simulation. For this case the original geometry for the full simulation was cut at first a 11.25 degree angle, and then a 22.5 degree angle. A similar simulation approach as was used for 2D was used for the wedge model, meaning only the seawater domain was considered and a temperature profile was specified on the pipe surfaces.

To simulate the heat transfer from the pipes, the thesis decided against renaming all 390 surfaces as was done in the 2D model, instead the top ten pipe rows were renamed pipes1 and the bottom ten pipe rows were named pipes2. Two temperature profiles were imported to Star CCM+ as tables, table T1 and table T2, which can be seen in figure 4.8. These temperatures were assigned to each pipe surface based on its radius according to a field function. Even though this function was not exact since it applied different temperatures around the pipe the differences were considered negligible since the alternative to do as the 2D model would take too much time.

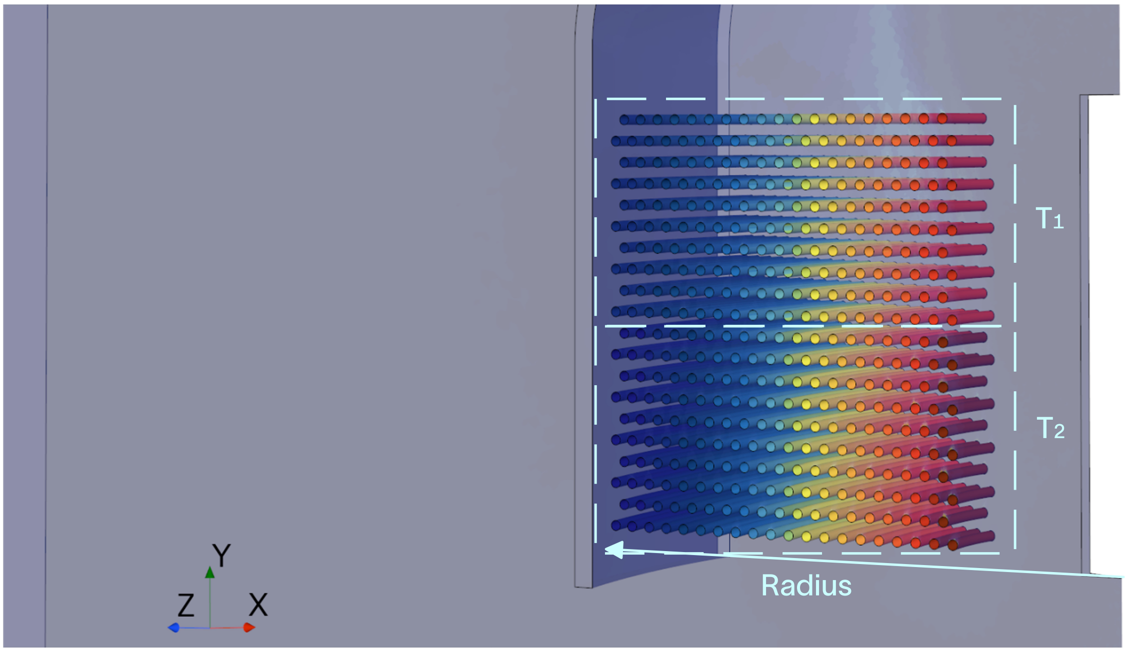

Figure 4.8: Temperature distribution on pipe surfaces

The wedge model were verified in the same matter as the 2D model with three simulations according to simulation "Wall", simulation "Chimney" and simulation "Baffles".

4.10 Final Model Design

In order to test cooler the following parameters were changed and evaluated.

4.10.1 Chimney Design

When designing the chimney the following parameters were tested as can be seen in figure 4.9. Where the chimney visible on the left hand side was the original design and Sim refers to the simulation number that was run, which changes a certain constraint.

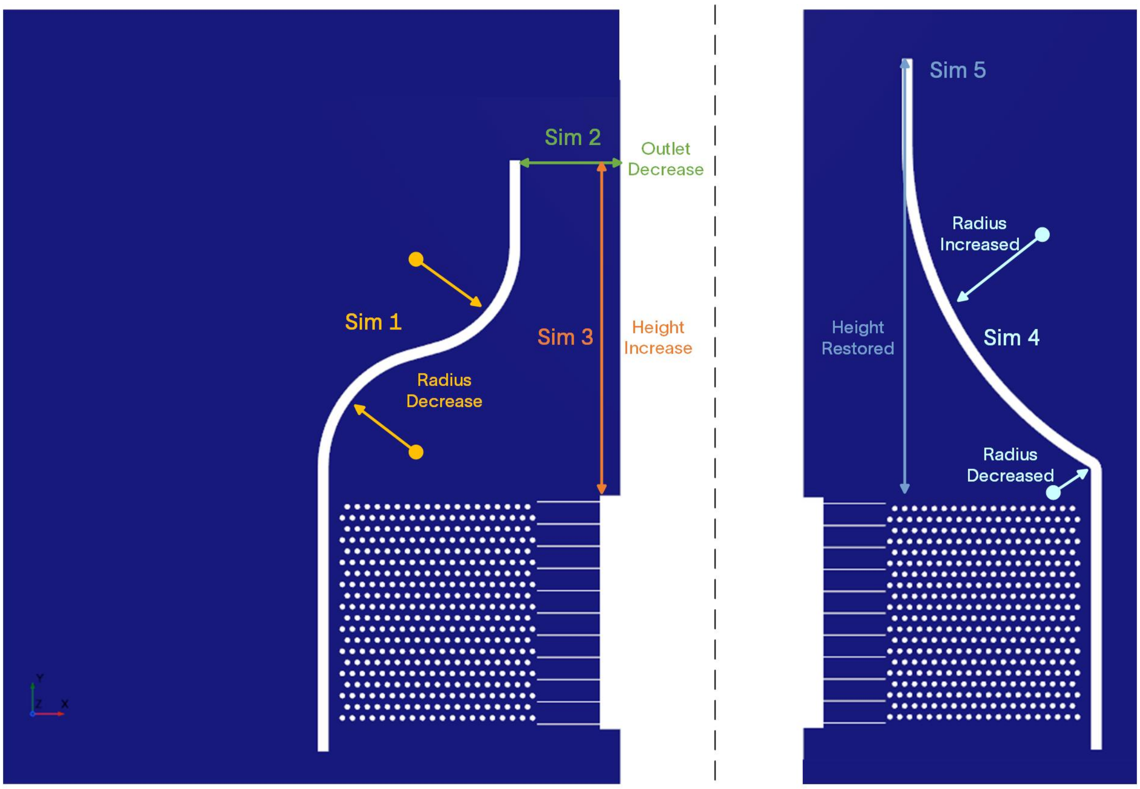

Figure 4.9: Wedge tests designs

These tests were designed based on the streamlines scalar presented in chapter 5.5.2, where in the first tangent arc of the chimney flow gathered and swirling regions occurred, thus these tests are designed to attempt to prevent these.

- Sim 1: Decreasing the radius of both tangent arcs
- Sim 2: Increasing the chimney height and decreasing the chimney outlet
- Sim 3: Increasing the chimney height (with smaller radius of the bottom arc)
- Sim 4: Increasing the chimney height, decreasing the bottom arc, and increasing the top arc
- Sim 5: Decreasing the bottom arc, and increasing the top arc

4.10.2 Design Improvements

Based on the outcome for the 2D and wedge simulations full 3D model simulations were performed to test the robustness of the cooler. Based on those full 3D simulation results it was decided to attempt to remove some pipe spirals.

Based on the result presented in chapter 5.5, concerns regarding no change in effect with chimney or with only walls were raised. Also the outlet temperature in the pipes did not differ much from each other from the last three spirals, the thesis wanted to test if removing five outer spirals would increase the chimney effect. Another parameter to change was the inlet temperature of the cooling fluid as this could increase the chimney effect but increase the outlet temperature out of the desired range. Therefore the following models were tested and compared using the same simulation set up as chapter 4.9.2:

- **"No Walls"** Simulation with only tube bundle
- **"Baffles"** Simulation with walls, chimney and baffles
- **"BafflesT45"** Simulation with walls, chimney and baffles with inlet temperature of cooling fluid increased to max of 45 °C
- **"HTA70"**Simulation with walls, chimney and baffles with 5 spirals removed from the outside
- **"HTA70T45"** Simulation with walls, chimney and baffles with 5 spirals removed from the outside inlet temperature of cooling fluid increased to max of 45 °C

4.10.3 Post-processing

The post processing of the models was based on reports for multiple scalar or vector functions pre-defined in the Star CCM+ database.

The Overall heat transfer coefficient was calculated using LMTD however by convention, for this type of heat exchanger, the inlet and outlet temperature was assumed to be the same. Giving the following definition of LMTD,

$$\Delta T_{LM} = \frac{T_{cw,in} - T_{cw,out}}{\ln \frac{T_{cw,in} - T_{ambient}}{T_{cw,out} - T_{ambient}}} \tag{4.4}$$

Where $T_{cw,in}$ is the cooling water inlet temperature, $T_{ambient}$ is the seawater ambient temperature, and $T_{cw,out}$ is the surface average outlet temperature of all the pipe outlets.
Which was then used for the following equation

$$OHTC = \frac{Q}{A\Delta T_{LM}} \tag{4.5}$$

Where the report for heat transfer between the sea and pipe is Q, the mesh area report of the OD pipe is the heat transfer area A. The result is U or OHTC, which in this report is presented as a normalised value.

Another parameter that had a special set-up in the simulations were the measure of mass flow of seawater through the cooler. This was computed through a mass flow report, based on a plane section created from a threshold derived part with the cooler walls diameter.

All the scenes extracted from the simulations to be compared were normalised. The simulation with the highest and lowest scalar function value was applied to all scenes to be compared. This was true for all scenes except those presented in Chapter 5.7.1, since the timing of the simulations caused all the scenes to be extracted before sim 5, the scenes are normalised but based on sim 4, even though sim 5 had higher/lower scalar values.

4.11 Manufacturing Possibilities

When assessing the product cost from a manufacturing perspective of the subsea cooler the input from the CFD analysis was used. To determined which sub-assemblies which was of most interest to investigate. Since the CFD analysis investigated the piping bundle and different configurations of the pipe stacks and what effect the chimney had on the heat transfer these two sub-assemblies was further investigated from a manufacturing perspective. It was evident that the cooler design would allow for a lower total amount of welds and therefore literature researching welding and cold bending was used as empirical material to be assessed on the cooler of what cost implications could be expected when a large amount of welds was replaced with a continuous bend. The 3D model was examined in SolidWorks together with other subsea coolers with a rectangular design, similar to the one illustrated in 3.1.

A similar methodology was followed for the manufacturing investigation regarding the chimney design. The chimney was also examined in SolidWorks and literature was reviewed in order to provide empirical evidence behind how the chimney could be manufactured. Results from the CFD was also used as a basis for the discussion whether implementation of a chimney was worth it or not. Input in form of internal knowledge regarding the manufacturing of both the chimney and the pipe bundle was also used.

5

Results & Discussion

In this section the result from all parts of the thesis is presented and discussed.

5.1 Market Screening

The following section provides the results of the technical market analysis for offshore HVDC platforms. The analysis was initiated by a break-even cost analysis between the two most common transmission technologies, namely, HVDC and HVAC. The outcome of the analysis consisted of a break-even distance when HVDC became the cheaper option followed by the identified projects that are most likely to opt for HVDC technology in the North Sea.

5.1.1 Cost Analysis Comparing HVDC and HVAC

The review of current transmission technologies suggested that two alternative methods available for transportation of electricity from OWFs to shore, is either by HVDC or HVAC [27]. The HVDC is a point-to-point connection where the equipment consists of one offshore converter in close proximity to the OWF, were electricity generated by the wind turbines is directed to the to the platform and one onshore converter, converting the electricity back into AC where it is being feed into the grid. The HVDC subsea export cables could stretch hundreds of kilometers and provide and maintain its transmitting voltage over very long distances. HVAC, on the other hand, requires intermediate reactive compensation stations to mitigate voltage instability and transmission inefficiencies which contributes to a higher reliability of the HVAC system [28]. These intermediate reactive compensation stations have to be placed with an equal spacing between each other, when transmitting over a certain distance. The disadvantage of having compensation stations becomes more evident when offshore conditions are considered, since new platforms and associated infrastructure have to be established for every station, which quickly contributes to an higher overall cost of the system [10].

By evaluating the total cost associated with both HVDC and HVAC transmission system, both the operating expenses as well as the capital expenses have to be into account. The operating expenses in this case is assumed to be the cumulative cost of electricity losses, discounted over the lifetime of an OWF. The capital expenses consists of the investment cost required for infrastructure for both technologies. Specifically, infrastructure costs include the cost associated with establishing the

HVDC platforms, the cost of subsea export cabling and the cost of OCS. The assessment framework is exemplified using an OFW boasting a capacity of 1 GW, providing a comprehensive and illustrative model for understanding the financial implications of HVDC and HVAC transmission systems within the offshore renewable energy sector. The framework is working as a supporting analysis to provide a better understanding on the market outlook for HVDC platforms.

5.1.2 CAPEX

The cost assessment for subsea HVDC cables are based on recent bidding contracts from four operational HVDC platforms and their associated HVDC export cable (Sylwin Alpha, Dolwin Gamma, Helwin Beta, Borwin Gamma). The average cost of subsea HVDC cables accounts to 1.93 M€/GW·km considering these four already operational platforms, see Table 5.1. The transmission voltage is set to be 320 kv and the cables are XLPE (cross-linked polyethylene) insulated. The offshore converters station cost are assumed to be 220 M€/GW and the onshore one 92 M€/GW [10].

Table 5.1: Cost data from bidding contracts for established HVDC platforms

HVDC platform	**DTOCS**	**Cost (M€)**	**Cost (M€/GW·km)**
SylWin Alpha [29]	205	250	1.41
DolWin Gamma [30]	160	350	2.34
HelWin Beta [31]	130	200	2.23
BorWin Gamma [32]	160	250	1.74

The CAPEX for HVAC also consist of two parts, namely the HVAC subsea cables and the intermediate compensation stations, which contributes to reducing power losses when transported long distances [33]. Two type of AC scenarios are investigated, namely AC1 and AC2. AC1 is based on a cable made of three-core 1400 mm2 aluminium which requires intermediate compensation stations for every 90 km of cable length. In other words, a cable length of 180 km can be achieved with one intermediate compensation station. The AC2 scenario is based on a single circuit 2000 mm2 copper cable, requiring intermediate compensation stations for every 100 km of cable length [10]. The cost of cable for AC1 used in this examples is rounded to 2.0 M€/km and for AC2 3.0 M€/km. The cost of building an offshore structure for an intermediate compensation station is more complicated, since it depends on a number of factors such as, water depths, current and building method. Therefore, an average cost of about 45-50 M€ for platforms built in relatively shallow waters (20-60 meters) presented by, is assumed for the platforms [34]. These estimates will be used in below calculations as well, hence the curve is discontinues in Figure 5.1, since intermediate compensation stations have to be added for each distance.

The AC cable cost are 3.0 M€/km for AC2 which was based on a 2000 mm^2 copper cable and 2.0 M€/km for AC1 which was based on a slightly thinner 1400 mm^2 aluminium cable [10].

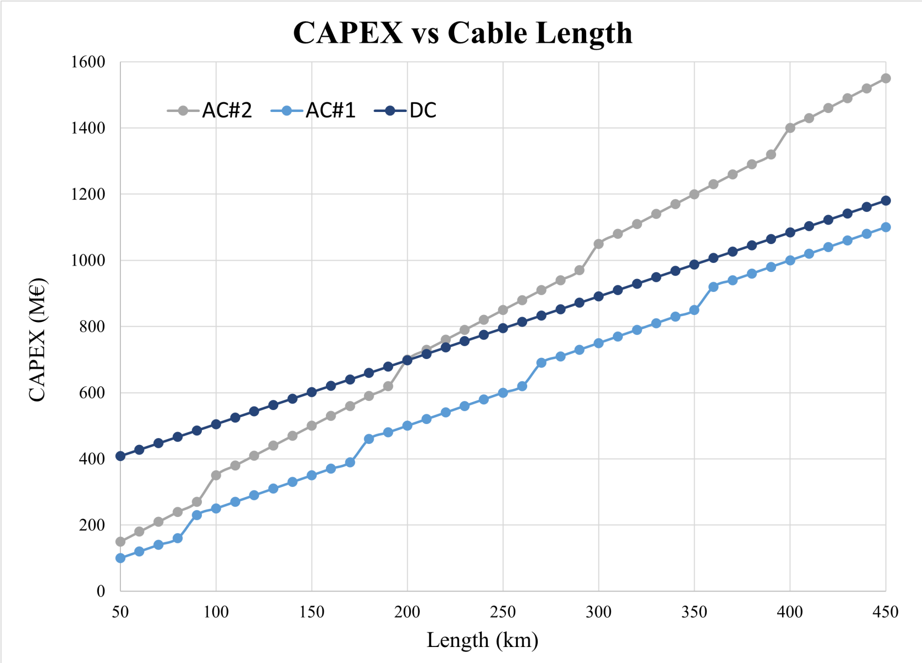

Figure 5.1: Investment cost compared to total cable length

Regarding solely CAPEX you would have to place an HVDC platfrom about 360 km from shore in order for it to be economically beneficial. The low cost of AC2 cable combined with relatively inexpensive offshore intermediate compensation stations results in a combined lower total CAPEX than the HVDC scenario.

5.1.3 OPEX

The operating cost analysis between HVAC and HVDC is based on the power flow calculations as the difference between input and output of the entire system, to determine the percentage losses over longer distances [10]. The HVDC losses is linear since there is no need for intermediate compensation stations and the losses increase as the distance increase. For both of the AC cases the losses are slightly decreased as the electricity is passing through the compensation station, hence the sharp decrease in cost in Figure 5.2 at the distances were compensation stations are positioned. The percentage power losses are then multiplied to represent the actual transmission losses for an OFW with a capacity of 1 GW. To provide representative values that could be put into context of what the actual losses mean in terms of loss of revenue for a potential wind farm operator, the losses are summed over the entire lifetime of the OWF. To represent the losses in terms of actual cost the average electricity prices for relevant countries with direct access to wind power generated in the North Sea and transmitted to onshore consumers. The electricity price used in the analysis is the average Price of Electricity (PoE) from 2021 - 2023 expressed in €/kWh and are presented in Table 5.2. The PoE for the selected countries is the final value used for the OPEX analysis. The average lifetime of a wind turbine was found to be around 30 years, therefore this value is also used in the calculations when investigating the total losses over the lifetime of the OWF [37, 38]. The discount

Table 5.2: Average PoE North Sea countries

Country	Average PoE (€/kWh)
Netherlands [35]	0,475
Belgium [35]	0,435
Germany [35]	0,413
Denmark [35]	0,381
UK [36]	0,13
Average	0,3667

rate for large scale renewable energy projects is dependent on several factors, such as risk associated with the investment, time horizon and opportunity cost. For the analysis a discount rate of 5% was selected which were in line with similar studies conducting investment decision models for OWFs in Europe [39]. Equation 4.1 was established to compute the present value of the losses discounted yearly over 30 years and then summed. As mentioned in 5.1.1, the OPEX in this analysis is solely made

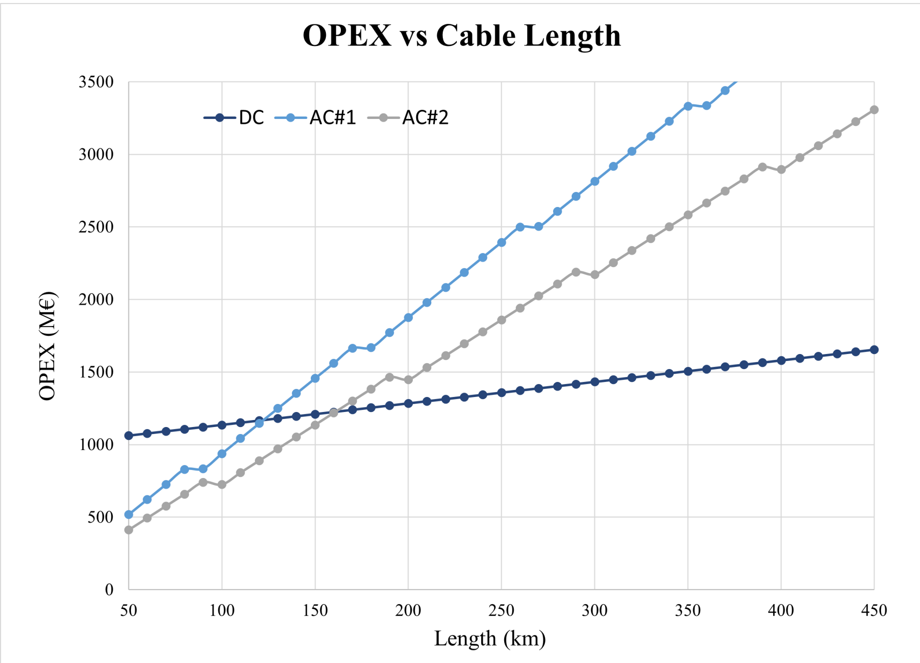

Figure 5.2: Operating expenses compared to cable length

up from the transmission losses. This is mainly due to the fact that the losses has the biggest impact from a cost perspective, which is clearly indicated when looking at the absolute cost represented on the y-axis. Where the cost of transmission losses is quickly becoming the dominant factor when comparing to investment cost as the cable length increases. The slope of the graph in Figure 5.2 for the range of 100 to 200 km is summarized in Table 5.3, which represents the losses per km in M€. From the table it could be concluded that DC is about five and a half times more cost-efficient when transmitting over this range of distance.

Table 5.3: OPEX per km in range 100 to 200 km

Transmission technology	OPEX/km
AC1	7,24
AC2	9,38
DC	1,48

5.1.4 Total Cost

With the two cost graphs illustrating the relationship between how CAPEX and OPEX increase as the cable length increase established, the total cost for the OWF could be determined. The two costs are added together and the result is shown in Figure 5.3. The total cost graph is suggesting that a distance of about 175 km is required in order for HVDC to be beneficial from a cost perspective. When looking at the cost in absolute figures represented by the y-axis, it is worth noting that the OPEX is dominating factor when comparing the two cost components and the investment cost quite quickly becomes less relevant as the cable lenght increases. Furthermore, the total cost is increasing substantially as the distance exceed the break-even point of approximately 175 km, which would indicate that a potential wind farm operator would loss out on a significant amount of revenue if opting for HVAC for further distances from shore in the long run, even though the investment cost is lower. The AC1 alternative is however becoming the cheaper option even though it is having a higher investment cost than AC2, which is due to its lower transmission losses. Since the losses is based on the PoE calculated in Table 5.2,

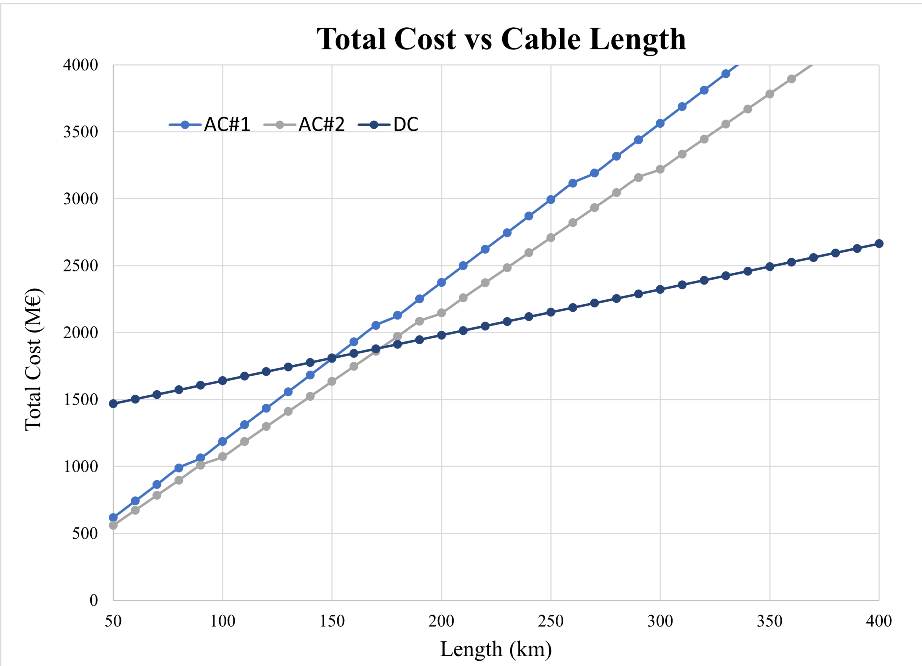

Figure 5.3: Total cost over lifetime compared to cable length

this parameters was found to have a significant impact on the break-even distance. The PoE is therefore, determining the OPEX factor and the break-even distance in the end to a large extent. If for instance the PoE electricity in the future would be much lower that would mean that it is less expensive suffer from transmission losses and the higher investment cost of established HVDC infrastructure would be more significant. Estimating PoE as far into the future as seven years is associated with significant uncertainty. Such an estimation would need to consider several factors, including geopolitical and macroeconomic aspects, which were considered to be outside the scope of this thesis.

5.2 Identified HVDC Platforms

The break-even distance of 175 km established under section 5.1.4, was used as the decision parameter when determining which OWF projects that were relevant when estimating the demand for the subsea cooler. The prerequisite was that the cooler was that the OWF most likely would opt for HVDC technology which would create the cooling demand that the subsea cooler would be designed for. In the process of selecting HVDC platform in regards to the search criteria it was found that the calculated break-even distance was in line with which technology OWF had opted for earlier. For instance it only differed about 10-20 km for established platforms placed 160-190 km from shore between those who opted for HVDC rather than HVAC. The identified HVDC platforms that are planned for completion between 2025-2031 are summarized in Figure 5.4, where the converting capacity is shown on the y-axis and the average converting capacity is highlighted by the light grey line. It was found that several major OFW projects were planned for completion in mid 2025 to 2031. In total 35 individual platforms where identified positioned in various locations of the North Sea. By comparing the amount of platforms to what was found under Section 2.1.1 it could be concluded that the market is set to more than triple in size over the next seven years, in terms of number of platforms planned for construction. Total installed conversion capacity is set to seven-fold, going from 6 845 MW to 47 960 MW over the same time period based on the review over the current state and the progressive market screening. The limitation of only looking at projects that had a completion date up until 2031, was due to the fact that later projects had not been authorised yet and was referenced to as "development zones" in the TGS 4C Offshore database, which meant that the development in these zones was not confirmed yet and therefore not included in the thesis.

Another interesting aspect shown by the market screening was that, even though the capacity of the OWFs are planned to increase over the couple of years, the converting capacity for the HVDC platforms are not following this trend. To cope with the rapid increase in OWF capacity several HVDC platforms are installed instead of having one HVDC platform with much higher converting capacity. This in turn means that the average converting capacity of the identified platforms are similar to the once that are already installed. The cooling demand on the platforms could in general be assumed to be equivalent to 1% of the converting capacity. This assumption results in an average cooling demand for the investigated platforms shown in Figure 5.4, to be around 13,7 MW. Furthermore, the depths of where the identified HVDC platforms were planned was also investigated and found to in the range 15-60 meters, which is also very much in line with what as assumed under Section 4.6. To conclude on the results from the market screening it is evident that the market for HVDC platforms are quickly growing and a crucial technology in order for offshore wind to be competitive. Since the market screening only looked at relevant projects from 2025 to 2031, which is quite a short time horizon considering the construction time of establishing a fully operation OWF, there is potentially even more HVDC platforms that are planned on being constructed after 2031. Meaning that the size of the market is potentially a lot larger with even more HVDC platforms suitable for subsea cooling than shown by the analysis conducted by the thesis. This is resting on the fact that offshore wind is set to increase rapidly over the coming years and the only available space of placing the wind turbines would be further from shore which would require DC transmission technology and HVDC platforms [5]. It could be concluded that the driving force behind establishment of more HVDC platforms is the expansion of OWF requiring an export cable lengths that exceeds 175 km. For the North Sea it was found that that is the case for several OWF that are planned for construction, hence the increased demand for HVDC platforms.

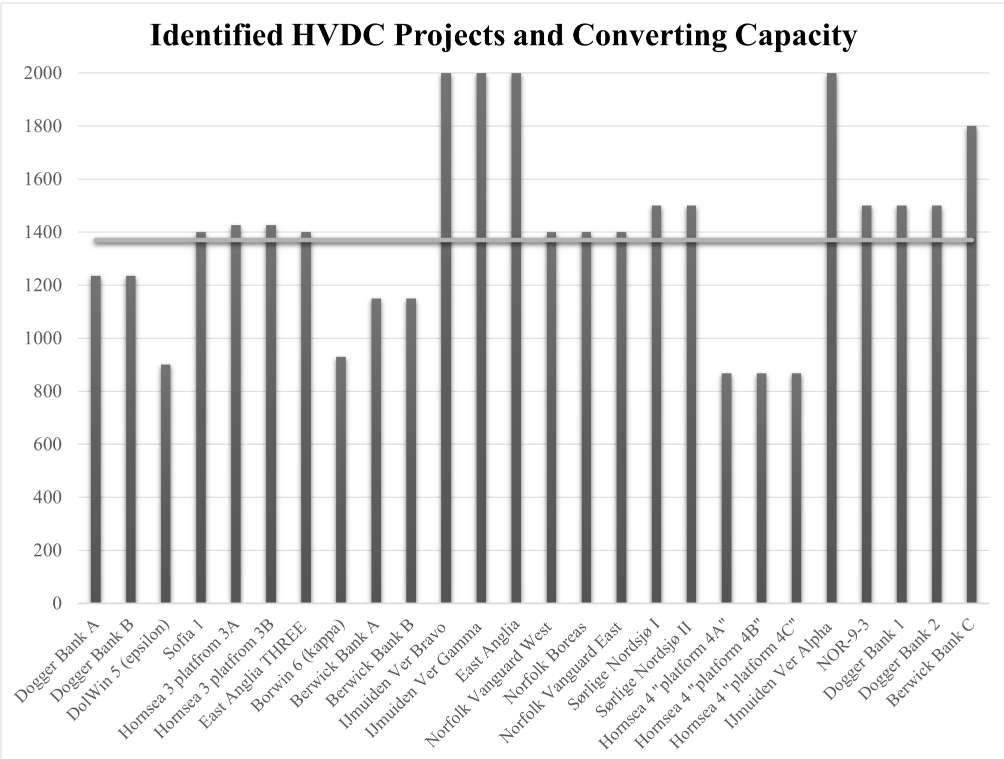

Figure 5.4: Identified HVDC projects and average converting capacity

5.3 Cooling System

The results from the cooling system analysis is broken up into the power savings for each eliminated component in the system, namely the sea water lift pump and electrochlorination unit and compared to the closed loop solution utilizing subsea cooling rather than lifting sea water to the topside heat exchanger. The result is at first presented individually for both the sea water lift cooling system and then for the subsea cooling system. Followed by a comparison of the two systems, where both energy savings and environmental impact is discussed.

5.3.1 Sea Water Lift Cooling

For the sea water lift cooling system the two main energy consuming components, namely the electrochlorination unit and the sea water lift pump was further investigated in terms of energy consumption. The results are presented below.

5.3.1.1 Sea Water Lift Pump

The required pump work for lifting sea water to the topside heat exchanger in the conventional open cooling loop was first evaluated. The required pump work for was calculated using Equation 4.2. The value for the mass flow was taken from reference project Beacon Wind where it was stated that a mass flow of approximately 1500 m^3/h was used. The lift height was approximated using dimension data from already established HVDC platforms and assuming that the sea water is lifted from 15 meters above the seafloor [7]. The efficiency of the pump was assumed to be the standard value for variable speed [40]. The input data is presented under Table 5.4.

Table 5.4: Input data for sea water lift pump calculations

Mass flow	416,67 kg/s
g	9,82
Height	45 m
η	0,8

The work required to lift sea water was calculated to be approximately 230 kW, which yields a yearly energy consumption of 2 014 GWh of renewable energy that is scavenged from the OWF. This result rests on the assumption that the topside heat exchanger necessitates a consistent mass flow rate equivalent to the one utilized in the calculations. This assumption was considered valid, given that the initial mass flow value represented the yearly average. The yearly cost of supplying the sea water lift pump with electricity using the price of electricity established in Table 5.2 equates to approximately 703 000 €.

To investigate the power consumption of the electrochlorination unit it was found that a concentration of 1 ppm of sodium hypochlorite would drastically prevent biofouling and therefore this value was selected as the base for the power consumption calculations. To achieve a concentration of 1 ppm of sodium hypochlorite with a seawater mass flow of 416 kg/s a continuous production rate of 0,416 g/s is required. A typical electrochlorination unit package usually consumes around 4,5 kWh of DC per kg of sodium hypochlorite produced. This yields a total energy of consumption of 1,872 W, which gives a yearly cost of approximately 5 700 €.

5.3.1.2 Electrochlorination Unit

The literature review concluded that the process of producing sodium hypochlorite from seawater requires additional energy in the shape of DC that has to be passed through an elecrolyzer's cell, which is done by a electrochlorination unit. To prevent biofouling a concentration of 1 ppm of sodium hypochlorite is generally required, which is also the value used in the equations to determine the power consumption of the process [41, 42]. The general power consumption of producing sodium hypochlorite from seawater was found to be 4,5 kWh/kgNaOCl [43]. This value is based on other industries that also uses seawater cooling in an open-loop were a electrochlorination is required to prevent biofouling. For instance, nuclear power is a great example of this. Since the It is worth noting that this is the average power consumption. Since the system is dynamic and continuously adjusted to match the cooling demand on the platform, the flow of cold seawater through the heat exchanger may be different depending on the ambient temperature which in turn affects the amount of sodium hypochlorite that needs to be produced.

To achieve a concentration of 1 ppm the electrochlorination units needs to produce a continuous flow of 0,416 g/s, which represent $\dot{m}$ in Equation 4.3. E_{prod} is set at 4,5 kWh/kgNaOCl, which results in a total energy output of 1,872 W and a yearly energy consumption of 16,5 MWh. Finally, the yearly cost of operating the electrochlorination unit could then be calculated and was found to be approximately 5 725 €. The yearly total quantity of disposed sodium hypochlorite to sea is 13 tons, which could have an impact to marine species around the platform which is mentioned under Section 2.2.1 and is further discussed under the comparison of the two systems, see Section 5.3.3.

5.3.2 Subsea Cooling

The same equation used to determine the power consumption for pumping sea water to the platform is used for the circulation pump in the subsea cooling system to calculate the pump work required. Since the cooling process is taking place subsea, the cooling medium is the only fluid that needs to be pumped from the cooler placed on the seabed to the platform. This results in a greater height that the cooling medium has to be lifted compared to the sea water cooling loop. However, since it does not have to lift water from the seabed. The cooling medium and mass flow is defined under section 4.6.2 and since MEG and water have similar densities the cooling medium's density was assumed to be equal to 1000 kg/m^3. The combined mass flow through the cooler was assumed to be equal to the sum of the mass flow through each pipe, which is also defined under section 4.6.2. The height parameter is defined for a platform with a height of 25 meters over the surface and a depth of 50 meters which gives a total height that needs to be covered of 75 meter. The selected height is somewhat conservatively chooses to be as applicable as possible, meaning that the power calculations is applicable to the tallest platforms and the deepest depths were these platforms are placed today. Input data for power calculations regarding the cooling medium pump in the closed loop system is presented in Table 5.5.

The result of the calculations showed that the required pump power was 40,5

Table 5.5: Input data for cooling medium pump

Mass flow	44 kg/s
g	9,82
Height	75 m
η	0,8

kW which yields a yearly energy consumption of 354 MWh and a yearly cost of 123 750 €, assuming the same PoE as determined under 5.2. Since the case study is considering the circulation pump to be the only pump present in the subsea cooling system, the power consumption of this pump is also the power consumption of the entire system.

5.3.3 Comparing the Cooling Systems

The results from calculating the energy consumption of the two alternative solutions, shown that the sea water lift cooling system was by far the most energy consuming. The total yearly energy consumption including both the seawater lift pump and the electrochlorination unit was 2030 MWh and the total cost 708 850 €. Meaning that, selecting a susbea cooling system utilizing a closed cooling loop and natural convection through seawater rather than an open loop system would result in a cost reduction of approximately 83%. As shown by Figure 2.3 & 2.2, there are even more components that are eliminated in the sea water lift cooling pump, that also could be eliminated, that could be further evaluated which would provide an even more accurate result. However, since the aim of the thesis is to build a business case around the cooling system and its associated subsea cooler and that the calculations from solely looking at the electrochlorination unit and sea water lift pump showed that you could achieve drastic cost reduction. It was decided that these preliminary steps points towards the conclusion that it is a far more efficient system from an operational cost stand point. The additional height that you would have to lift cooling medium in the subsea cooling system did not affect the total energy consumption as in comparison to the sea water lift pump since the mass flow is much lower. However, the seawater lift system does not require any subsea cooler at all, which means that the cost associated with the actual cooler would is not existing. You could therefore argue that the CAPEX of installing the subsea cooling system would be higher.

Another advantage of the subsea cooling system is the eliminated need of disposing sodium hypochlorite to sea. The European Union states in their risk assessment report regarding disposal of sodium hypochlorite that a predicted no effect concentration is 0,04 $\mu g/L$ (0,00004 ppm) for saltwater species [44]. This concentration is substantially lower than the one used by the seawater lift system and it could therefore be concluded that the operation of the system is posing a threat to the marine environment around the platform. Due to the health and environmental concerns several alternatives have been considered and many companies are trying to move away from using sodium hypochlorite. Since subsea cooling is not using any sodium hypochlorite, this could be used as a marketing aspect in addition to the decreased operating cost in order to promote the system.

5.4 Mesh Refinement Study

In order to ensure the models are independent of the mesh settings, a mesh refinement study was conducted. The mesh settings presented in table 5.6, were designed to test the inside and outside refinements. Based on the y+ value the inside of the pipes did need some refinement, however due to the large size of these simulations the thesis time limit did not allow for a deeper investigation.

Table 5.6: Mesh settings for refinement study

Settings		Original Mesh	Prism Layers	Surface Size
ID	# layers	4	10	4
	Surface Size	Base Size	Base Size	-25%
	Cell Count	39764404	83860131	68459843
	y+	14.66	1.41	14.84
OD	# layers	4	10	4
	Surface Size	8	8	6
	Cell count	71469000	124265050	100671362
	y+	0.82	0.09	0.83
Total Cell Count		1.27E+08	2.24E+08	1.95E+08

The mesh of the whole cooler can be seen in figure 5.5, where quite large cells were used in the outer parts of the seawater domain with smaller sizes where the expected plume would occur, and the smallest cells are around the pipes. This figure shows clearly, the importance of refinement boxes, and magnified views of the mesh around and inside the pipes.

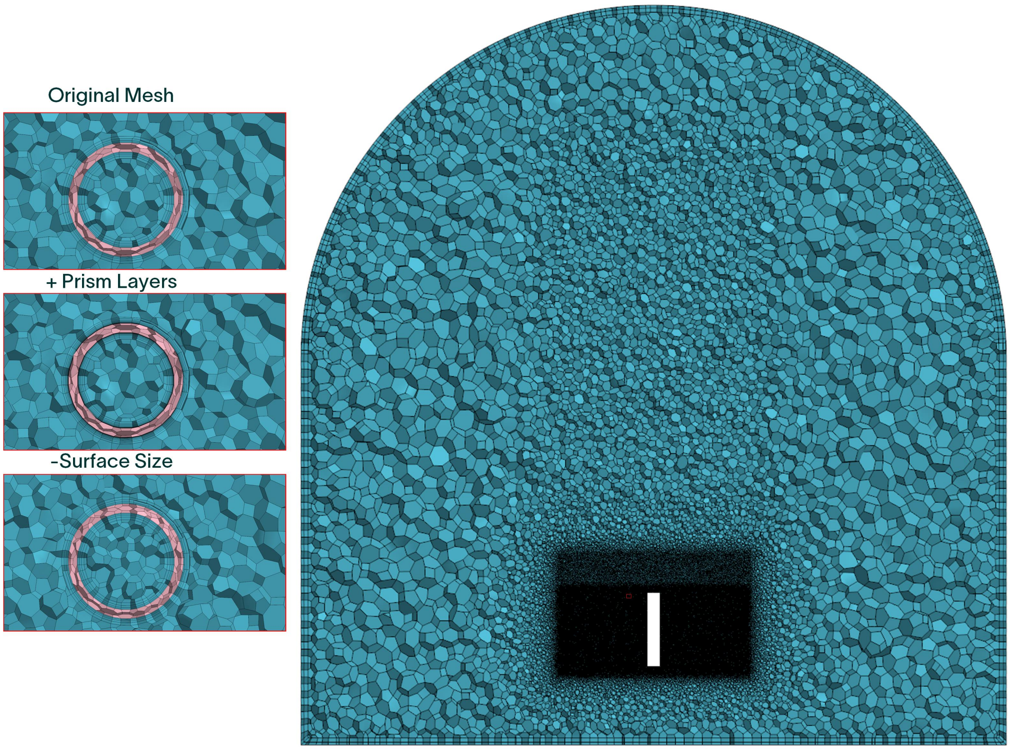

Figure 5.5: Mesh scenes of the seawater domain and of the refinements of the pipes

Figure 5.6 shows the difference in surface size of the pipes, to highlight the different estimations in mesh area, where a decreased surface size provides an increased resolution of the actual pipe geometry. In order to ensure the mesh study to be a fair comparison, the same amount of iterations and processes were used when running the simulations.

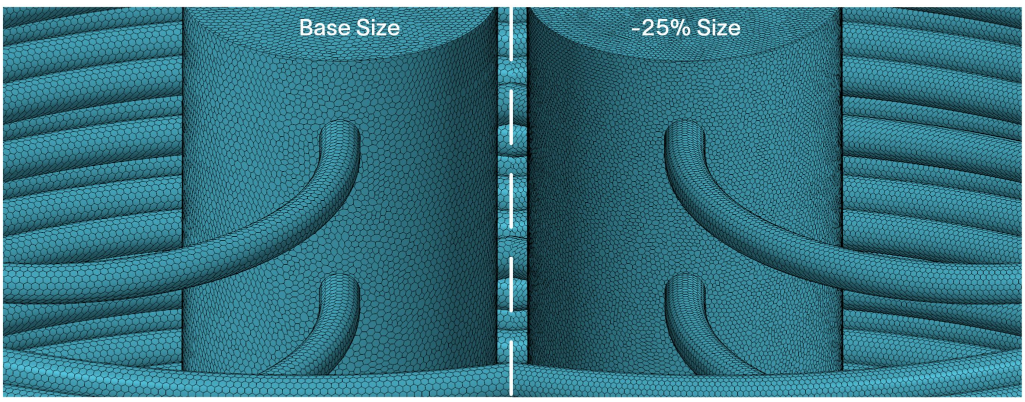

Figure 5.6: Mesh refinement scene of the surface mesh for 8mm and 6mm Surface size

5.4.1 Mesh Results

The result from the mesh study can be seen in table 5.7, there were differences between adjusting the two settings. For prism layers the OHTC decreased which was attributed to the ID y+ value decreasing, the difference compared to the original mesh was around 2.5%. The refinement of the surface size had the opposite effect the OHTC increased by 1.2%, which is due to the round pipes having better resolution with more points per circle. Which gives a better estimation of the heat transfer area of the mesh.

Table 5.7: Mesh refinement result

Parameters	Original Mesh	Prism Layers	Surface Size
Outlet Temperature [°C]	23.82	23.92	23.72
Heat Transfer [MW]	2.02	1.99	2.04
OHTC [W/m^2-K]	REF	-2.50%	1.19%

The computational load of the simulations can be compared from table 5.8. Where the prisms layers had the heaviest computational load with around 40% increase in computational demand compared to the original mesh. The mesh refining the target surface size had around 35% increase in computational demand. Both these values were a great indicator that the mesh couldn't have been refined more due to the time limitation.

Table 5.8: Mesh study computational time results

Parameters	Original Mesh	Prism Layers	Target Surface Size
Total Solver CPU Time (s)	1.19E+07	1.95E+07	1.76E+07
Total Solver Elapsed Time (s)	9.64E+04	1.58E+05	1.43E+05
Solver CPU Time (s)	2.74E+03	4.44E+03	4.25E+03
Solver Elapsed Time (s)	22.23	35.97	34.47
Iterations	4000	4000	4000

To conclude the mesh study, the thesis assumes the results proving the original mesh to be acceptable with a percentage difference of below 3%. Especially considering the reduction of 40% of the computational time, using the original mesh setting was preferable. In addition the simulation set up was designed conservative with the segregated solver, which based on internal knowledge tends to underestimate passive coolers performance. As well as the design being based on worst case scenarios, the cooler performance can be expected to increase enough to accept the original mesh.

5.5 Initial CFD Results

In this section the results from the first simulations completed by the thesis are presented. Due to the large cooler size, the thesis had to create an understanding of what would be required in the simulation set-up and mesh creation in order to ensure less show stoppers. Thus pipe arrangements were tested with a simplified model before creating a full scale model.

5.5.1 Pipe Arrangements

The pipe arrangements are described in chapter 4.9.1, these were designed to test the set-up and some standard parameters known to increase heat exchanger performance. The results from the first simulation set-ups are presented below in table 5.9, the solution with more spirals towards the inlet header had the largest overall heat transfer coefficient and lowest outlet temperature due to an increase in heat transfer area. There was virtually no difference between original and change in longitudinal pitch, suggesting the literature based on air was not translatable to seawater.

Table 5.9: Pipe arrangement results

Parameter	Original	Add Spirals	Smaller S_L	Larger S_L
OHTC [W/m^{2}K]	REF	1.47%	-0.74%	-0.71%
Heat Transfer Sea [MW]	0.409	0.426	0.408	0.408
Outlet Temperature [°C]	23.68	23.23	23.72	23.71
Sea Mass Flow [kg/s]	290.01	292.37	289.14	291.24
Sea mass flow Center [kg/s]	39.12	16.42	38.19	37.09
Mass flow Channeled[%]	13.49%	5.62%	13.21%	12.73%
Pressure Drop [bar]	1.44	1.55	1.42	1.42

Scalar fields were used to visualise the velocity streamlines through the pipes for each case see figure 5.7 and 5.8. The streamlines for the original pipe arrangement as well as change in longitudinal tube pitch had large turbulent eddies created in the center of the domain behind the inlet header. While the one with additional spirals had more uniform flow through the pipes.

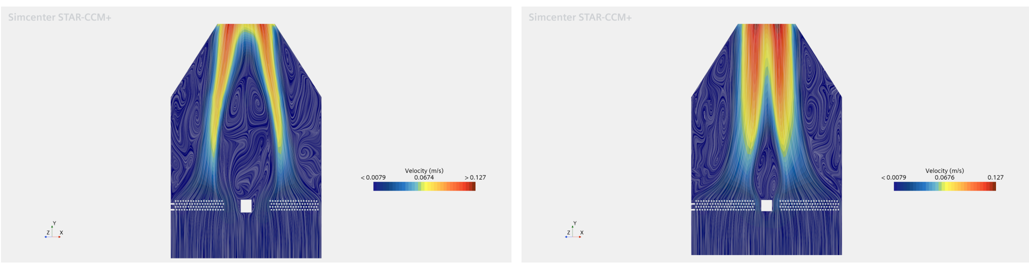

Figure 5.7: Scenes of velocity streamlines for a) original pipe arrangement b) additional spirals

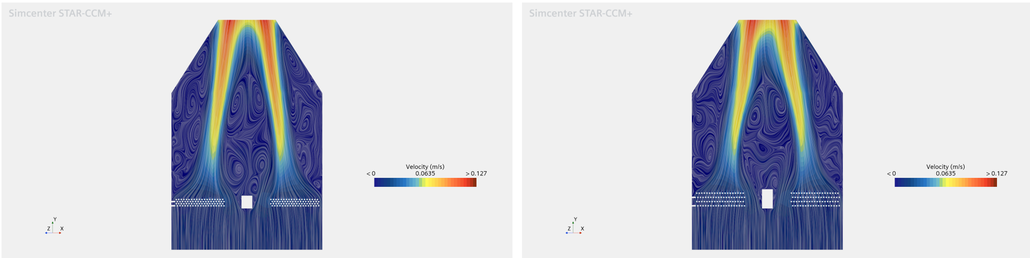

Figure 5.8: Scenes of velocity streamlines for a) smaller S_L b) larger S_L

Since the results indicates that a change in the longitudinal pitch did not change the cooler performance, it was not investigated further. This result deviates from articles presented in chapter 3.2.1 where air was used as the cooling medium outside the pipes. Which can be explained by the density difference of water and air, as explained in the limitations. The additional spirals towards the inlet header indicated that some form of baffles should be added to the cooler since channeling occurs around the inlet header. However based on some internal discussions the additional spirals would pose manufacturing concerns and were not investigated further.

5.5.2 3D model

In order to test the full 3D model, according to chapter 4.9.2, three versions of the cooler was chosen. The preliminary results from these three simulations are presented in table 5.10. All simulations indicates the cooler can manage the cooling fluid outlet temperature requirement, all being well below 25°C. Simulation "Baffles" had the best performance with higher OHTC and heat transfer, while the seawater mass flow through the cooler was lower. This was due to the removed channel between the inlet header and inner pipe spirals. As both simulation "Wall" and "Chimney" has 20% of the mass flow channeled through the middle of the cooler, and perhaps even more since this report was based on the cooler inlet. Therefore the baffles creates an obstacle which reduced the amount of seawater able to get through the pipes.

Table 5.10: 3D model results

Parameter	Walls	Chimney	Baffles
OHTC [W/m^{2}K]	REF	0.13%	4.85%
Heat Transfer Sea [MW]	2.08	2.08	2.12
Outlet Temperature [°C]	23.48	23.48	23.25
Sea Mass Flow [kg/s]	373.52	365.40	123.08
Sea mass flow Center [kg/s]	75.95	70.46	
Mass flow Channeled[%]	20%	19%	
Pressure Drop [bar]	1.42	1.42	1.43

The scalar scenes can be seen below in figure 5.9. The results indicated that there was a negligible difference between having only walls and adding the chimney on top. Which suggests that the chimney effect was not achieved, and as discussed with pipe arrangements baffles were necessary to ensure as much seawater as possible went through the pipe stacks. Another interesting parameter was the low pressure drop based on the cooling water inlet and outlet, this would be expected to increase once the inlet and outlet headers are simulated, however this suggests the pipe dimensions would be more than enough to handle the internal pressures.

5.6 Model Validation

The thesis wanted to investigate what could be done to reduce computational time, since it was estimated before hand that the full 3D model of the cooler would include a large domain. The thesis considered many options to simplify the cooler size, such as periodic models, and a 1D code to simulate the pipe fluid coupled with the 3D model in Star CCM+. However due to the cooler pipes being spirals and not constant in size around, the solution would not be able to estimate the geometry of the cooling fluid pipes.

5.6.1 3D model

In order to have reference points for feasible results the thesis conducted three large 3D simulations to verify the simplification models. The results that were able to be compared to the simplified models are presented in table 5.11.

Table 5.11: 3D model validation results

Parameter	Wall	Chimney	Baffles
Sea Mass Flow [kg/s]	373.52	365.40	123.08
Sea mass flow Center [kg/s]	75.95	70.46	
Mass flow Channeled [%]	20.332%	19.284%	
Heat Transfer Sea [MW]	2.078	2.081	2.12

The velocity scalars of the 3D model are presented in figure 5.9 the section plane selected for these scalars are the same as for the simplified models. It was visible that both for Wall and Chimney a higher velocity was experienced between the tubes and inlet header, thus being channeled, rather than going through the pipe stacks. The flow swirls after the inlet header before creating the plume causing a flow separation, this was removed by adding baffles, where much smaller swirls are visible. The thesis did not see much channeling between the outer wall and pipe bundle which otherwise could be a concern.

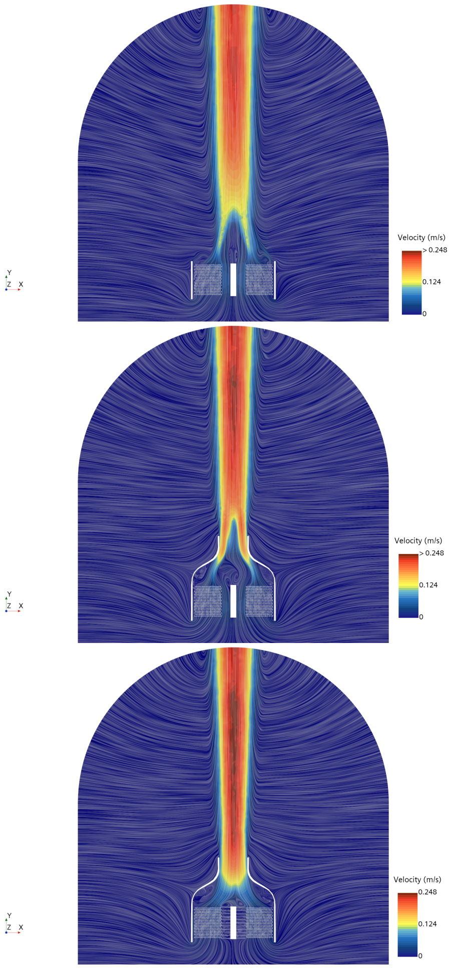

Figure 5.9: 3D Scenes of velocity streamlines for a) Wall b) Chimney c) Baffles

5.6.2 2D model

Table 5.12: 2D model comparative results

Parameter	Wall	Chimney	Baffles
Sea Mass Flow [kg/s]	53.66	69.59	44.41
Sea Mass Flow Center [kg/s]	25.85	33.53	-
Sea Mass Flow Channeled [%]	48.2%	48.2%	-
Heat Transfer [MW]	0.238	0.281	0.320

Since only the seawater domain was simulated as explained in chapter 4.9.3.1, the comparative parameters were the heat transfer from the applied temperature as well as the mass flow flowing through the cooler, where the relationship can be compared to that of the full 3D model. The results presented in table 5.12, shows the chimney and baffles to increase heat transfer, which deviated from the trends presented in the

3D simulation. Another concern was the amount of mass flow channeled representing almost 50% of the domain, which was not accurate.

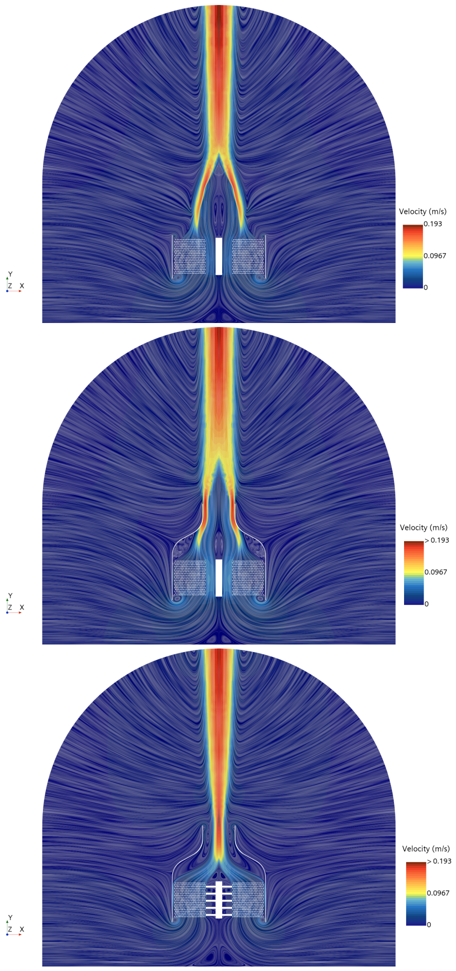

Figure 5.10: 2D simulations scenes of velocity streamlines for a) Wall b) Chimney c) Baffles

The velocity scalars can be seen in figure 5.10, where it became apparent that the model was not accurately representing the 3D model. Simulation wall had a much larger flow separation and two swirling regions were visible after the inlet header. There were also larger swirls at the inlet of the cooler around the walls, which were present in the 3D model, however not at this scale. The same trend can be seen for simulation chimney, where the flow had a larger separation, in addition with the chimney the flow created large re-circulation zones. However the largest difference occurred when applying the baffles, where the flow separation shows similar to the 3D model, however flow was entering at the chimney outlet which was not an accurate representation of the trends. This was the basis for invalidating the 2D model. Since the 2D model was invalidated when compared to the 3D models trends, the computational results was not presented, however the 2D model was a lot quicker than both the 3D and Wedge model.

5.6.3 Wedge model

The wedge model presents more reasonable results when compared to the 3D simulations, where there were negligible difference between simulation wall and simulation chimney, while when adding baffles there's improvements in the heat transfer. The mass flow through the cooler was also more accurate since around 20% was channeled through the centre similar to the 3D results.

Table 5.13: Wedge model comparative results

Parameter	Wall	Chimney	Baffles
Sea Mass Flow [kg/s]	22.76	21.96	21.38
Sea Mass Flow Center [kg/s]	4.39	4.05	-
Sea Mass Flow Channeled [%]	19%	18%	-
Heat Transfer [MW]	0.123	0.122	0.131

The velocity scalar for the wedge model can be seen in figure 5.11. There are still flow separations in the streamlines but not of the magnitude as experienced in the 2D model.

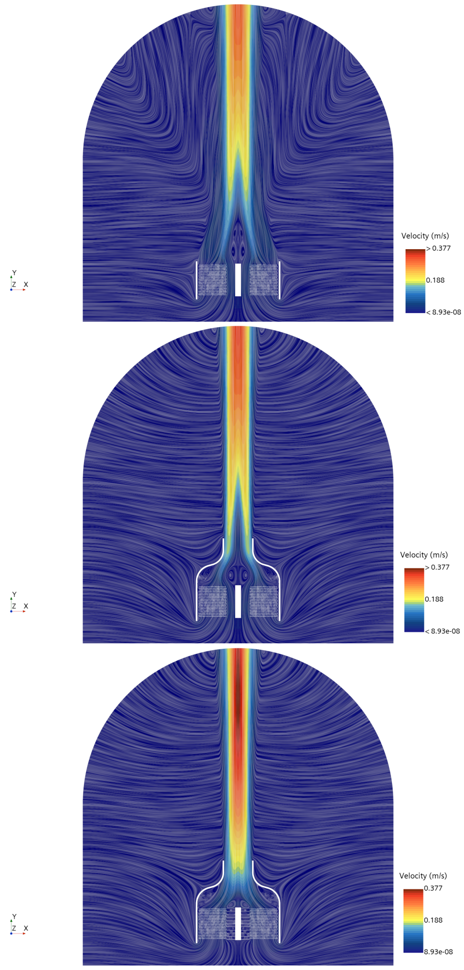

Figure 5.11: Wedge simulations scenes of velocity streamlines for a) Wall b) Chimney c) Baffles

5.6.4 Computational models discussion

Based on the trends displayed in the 3D model it was concluded that having only walls or with the chimney made no difference in the performance of the cooler. This trend was crucial to verify the results of the simplified models.

For the 2D model this trend was not experienced, and invalidated the data collected from these models, even though computational time was reduced, the difference between walls and chimney was too large to find a trend to normalise the values. The reason for the 2D model not working was due to the circular effects not being captured in and around the cooler. When modelling in planar 2D the simulation assumes the plane continues infinitely thus does not capture the round geometry of the model.

The wedge model was more accurate due to its capturing of the circular flows. However the centre of the model might not have been the most accurate since the cells were quite small. This was evident since the thesis begun by creating a wedge of 11.25 degrees which gave approximately the same trends as the 2D model showed. Therefore a 22.5 degree model was tested, which showed the trend of small difference between wall and chimney, while an increase in heat transfer were experienced with baffles. The velocity scalar also showed longer (in y-direction) flow separation, however both models had the highest seawater velocity in the baffles simulations.

Table 5.14: Computational time for 3D and wedge model of simulation "Baffles"

Parameters	3D	Wedge
Total Solver CPU Time (s)	1.45E+07	5.88E+05
Total Solver Elapsed Time (s)	1.72E+05	3.69E+04
Solver CPU Time (s)	2.05E+03	62.11
Solver Elapsed Time (s)	21.394	3.888
Iterations	7000	10000
Cell Count	1.30E+08	4.61E+06

In order to verify that the computational time was reduced, system reports were run on simulation "baffles" for the verified wedge model and the 3D model, presented in table 5.14. It was important to note that the wedge model was always run on less processes with more iterations than the 3D model, making the comparison unfair. Nevertheless the results proved the CPU time was reduced by around 96% and the elapsed time by around 80%.

5.7 Final design

The results presented in this section is based on the tests described in chapter 4.10, which includes the chimney design based on the wedge models validated previously, and the Design improvements made to test on the full 3D model. Since an aim of the thesis was to investigate parameters that could improve the cooler performance.

5.7.1 Chimney design

Based on the wedge model verification the chimney parameters were tested and gave the results presented in table 5.15. The results show no change in small scale parameters affected the chimney, therefore an increase in the chimney height were tested, however this did not improve the heat transfer or the sea mass flow significantly. Which can be verified against the model validation showing at a minimum an increase by approx, 10 kW would have a significant impact on the 3D model. The significant change occurred when the outlet size were reduced, where the heat transfer and mass flow was half of the original design.

Table 5.15: Chimney design results

Parameters	Baffles	Sim 1	Sim 2	Sim 3	Sim 4	Sim 5
Sea Mass Flow [kg/s]	21.38	21.39	11.17	23.09	23.06	21.31
Heat Transfer [MW]	0.131	0.131	0.094	0.135	0.135	0.131

The velocity scenes of all designs are presented in figure 5.12. Where the original and Sim1 shows negligible difference in the flow behaviour, while Sim2 proves that choking the chimney outlet caused larger residence time and re-circulation zones in the chimney which decreased the performance. When increasing the chimney height a small increase in performance occurred as Sim3 and Sim4 shows, and the streamlines scenes shows fewer re-circulation zones in the chimney, which also was the case for Sim5, even though the performance did not improve from the original.

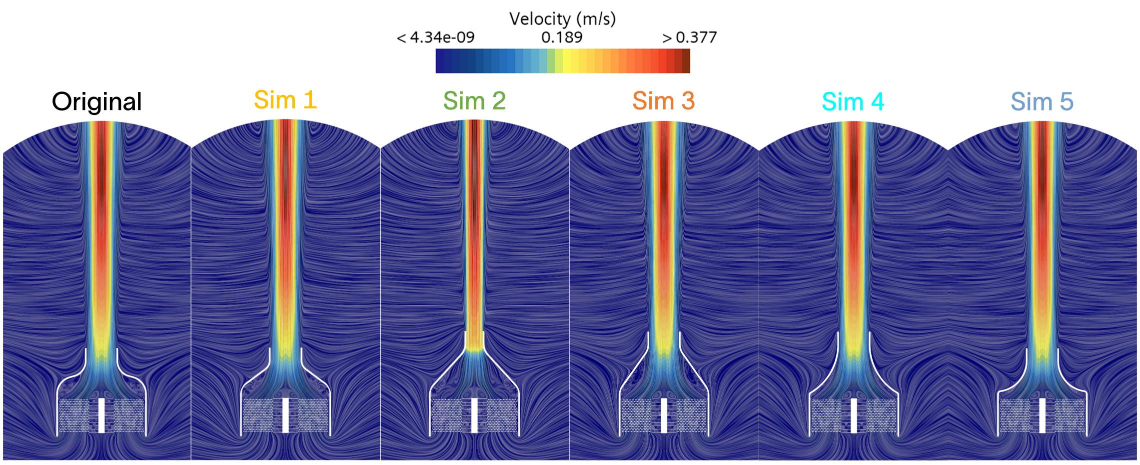

Figure 5.12: Wedge simulations scenes of velocity streamlines for all Chimney Sims

5.7.1.1 Chimney Design Discussion

Since the cooler has a height constraint Sim 1 and Sim 5 was designed to test what parameters could improve the chimney within that constraint. However since no parameter had any effect on the heat transfer the thesis tested increasing the chimney height which improved the sea mass flow and heat transfer slightly, however not enough to claim the chimney effect was experienced. Therefore the thesis concludes the coolers temperature differences between seawater and cooling medium caused a too small density difference for the buoyancy force to be driven through the cooler.

5.7.2 Design Improvements

Given the relatively small impact on thermal performance by including a chimney on the cooler, a number of alternate design changes were evaluated, as described in the Method chapter 4.10.2.

These alternative design results are presented in table 5.16, where the case "No Wall", meaning removing the chimney, walls and baffles, resulted in a reduced heat transfer and consequently a higher outlet temperature. The mass flow however was not able to be estimated as there was no longer a logical "inlet" to the cooler.

Increasing the inlet temperature of the cooling fluid however caused a drastic increase in OHTC and heat transfer as the seawater mass flow increased, as there was a larger ΔT to remove. However this also increased the outlet temperature which was expected even though it was still below the requirements discussed in the design basis.

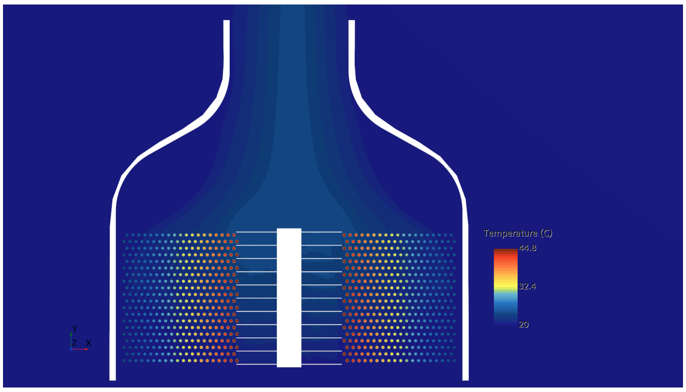

Figure 5.13: Simulations scenes of temperature profile for BafflesT45

Figure 5.13 suggested perhaps not as many pipe spirals were required, and identified the temperature difference between the last 3 spirals were negligible. Simulation HTA70 was created by removing 5 pipe spirals, which decreased the heat transfer as heat transfer area were reduced which increased the OHTC. However as expected this also increased the outlet temperature of the cooling fluid even though it still was below the requirement. In order to test the robustness of this design, Simulation

HTA70T45 was run where increasing the cooling medium inlet temperature resulted in the outlet temperature increased even though it was still below 30 degrees.

Table 5.16: Design improvement results

Parameter	No Wall	Baffles	BafflesT45	HTA70	HTA70T45
OHTC [W/m^{2}K]	-6%	5%	17%	10%	23%
Heat Transfer Sea [MW]	2.02	2.12	3.70	1.84	3.24
Outlet Temperature [°C]	23.82	23.25	24.53	24.84	27.07
Sea Mass Flow [kg/s]	NA	123.08	147.97	105.21	127.28
Pressure Drop [bar]	1.42	1.43	1.40	1.01	0.98

When considering the velocity streamlines scenes, in figure 5.14, it was interesting to see that without walls the flow goes directly to the cooler center and contributes to a type of channeling. However compared to figure 5.9 simulation "Walls" the same type of vertices seem to be formed behind the inlet header.

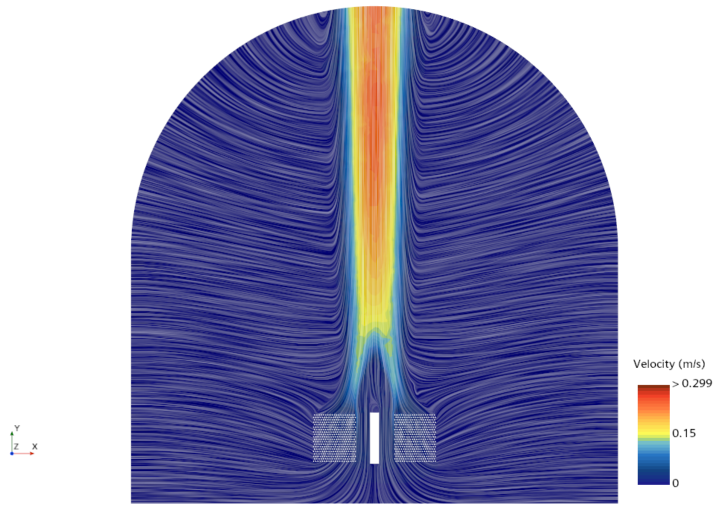

Figure 5.14: Simulations scenes of velocity streamlines for No Wall

Figure 5.15 shows simulation "Baffles" and "BafflesT45", which shows a larger velocity was achieved with increased temperature differences, however the re-circulation zones were still present.

Figure 5.16, re-iterates the trend of higher velocities with larger temperature differences. However these designs had much fewer re-circulation zones in the chimney as compared to the standard cooler size.

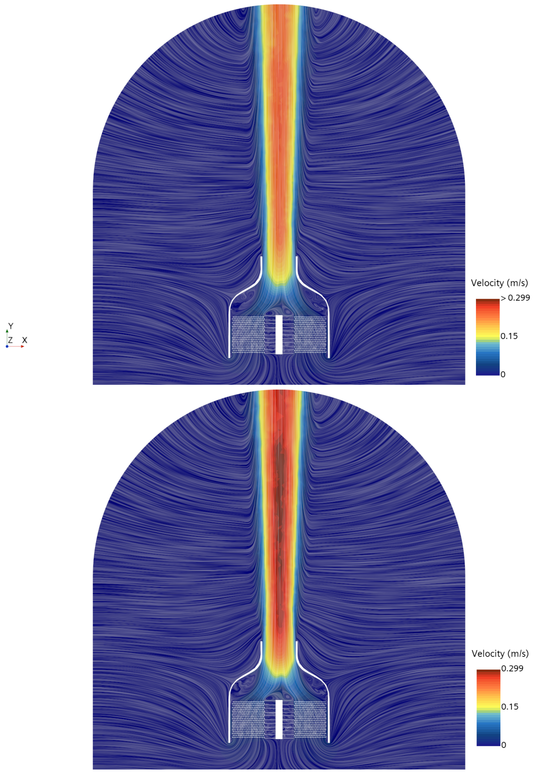

Figure 5.15: Simulations scenes of velocity streamlines for a) Baffles b) BafflesT45

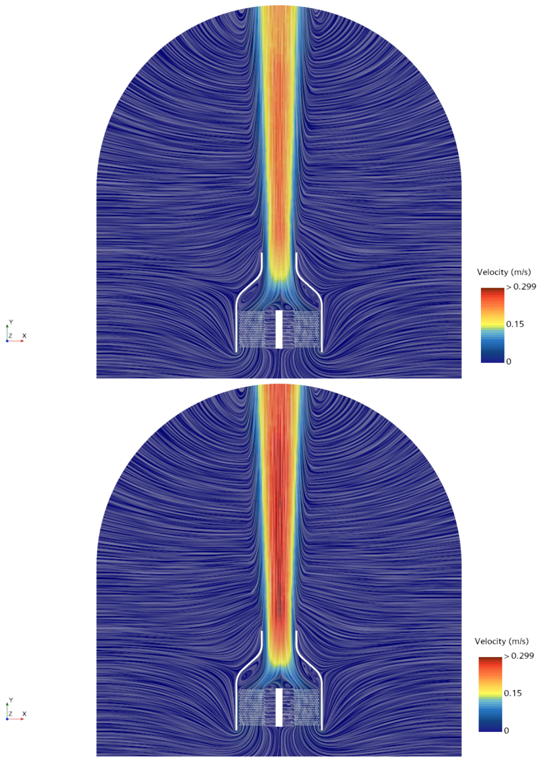

Figure 5.16: Simulations scenes of velocity streamlines for a) HTA70 b) HTA70T45

5.7.2.1 Design Improvements Discussion

The results from the design improvement process provide indications of trends that could improve the cooler performance. It also gives grounds for discussions as of which design was of large significance for the product.
The results show that there were several options, the first was including chimney and baffles or removing both. As discussed in chapter 5.5.2 there were no use including the chimney or walls without baffles as a large part of the seawater was by-passed or channeled through the centre. However when removing the walls, chimney and baffle components the cooler performance decreased. As the simulations were designed based on worst case scenarios for a general passive cooler, there might be different worst case scenarios for "No Walls" and "Baffles". Arguably applying high currents in the seawater could improve the "No Walls" performance since more mass flow would be driven through the cooler, while for simulation "Baffles" higher currents could cause no seawater being pushed out of the chimney. Another design could be considered by removing the chimney there would be more space for pipe stacks, the cooler could double in size, increasing the heat transfer area. This introduces other challenges such as the pressure drop and mass flow of cooling fluid, as a larger pump would be required, even though this was not evaluated in this study it was grounds for future work.

The other options to remove some pipe spirals and decrease the diameter of the cooler to attempt to achieve the chimney effect proved possible shown by simulation "HTA70". Although the amount of spirals removed were over estimated the seawater mass flow did not decrease too much however the thesis cannot conclude with only these results that the chimney effect were experienced. Nevertheless the results showed an outlet temperature within the specified range which seems promising since even though the outlet temperature is tied to the mass flow of the cooling fluid, during the set conditions this design was upheld.

It is also worth discussing that the thesis attempted to test the cooler with seawater currents, however this result was not presented as the model could not converge. This was due to the circular geometry of the cooler and seawater domain. Where vortex shedding occurred around the walls and chimney which is a transient dependent phenomenon, which cannot be solved with a steady state solver. Which suggests an issue for this passive cooler, however as almost all structural components surrounding the cooler were removed for simplification purposes this might not be an issue for the as-built cooler. In order to accurately simulate the current effects the thesis believes a transient simulation would be necessary which within the time limitation of writing the thesis would not have been possible.

5.7.3 Final design Discussion

The CFD work performed for this subsea cooler assumed a worst case scenario regarding ocean conditions. No ocean currents and high ambient temperatures are not usual conditions in the North Sea, however the aim was to prove that the cooler could perform under these conditions and design the cooler accordingly. To provide a more comprehensive understanding of the cooler an OHTC map would have to be created, meaning mapping out the thermal performance versus different cooling fluid flow rates and temperatures. As was visible when changing the cooling fluid temperature, the cooler's heat transfer increased drastically. It is possible that the benefit of having the chimney will be more prominent at high temperature inlet conditions and low seawater currents due to the passive nature of the cooler. However this would have to be tested against simulations of the "No Wall" cooler with the same conditions. Nevertheless, seawater conditions would also be the case for seawater currents as without walls currents would drive more seawater mass flow through the cooler improving its performance, however with the chimney or walls less seawater would flow through the cooler depending on temperatures, as the velocities imposed by the buoyancy forces might not be comparable to the velocity magnitudes from the ocean currents. As mentioned previously the currents effect on the cooler could not be evaluated in this study, and would require a transient simulation as a steady state could not be found.

Another important parameter was the mass flow of the cooling water through the pipes, this was not evaluated in this thesis. However during the preliminary simulations of the cooler a too low mass flow was accidentally set which resulted in a much lower outlet temperature and decreased heat transfer performance. Which was due to the fluid residence time in the pipes being much longer. Showing the cooler sensitivity to the mass flow being a governing parameter for the cooler. The mass flow is also greatly tied to the cooling output, if 5 MW cooling is required for the HVDC platform compared to an usual 3 MW this can be achieved through increasing the mass flow which is a variable parameter used in the industry.

The CFD simulation presented in the thesis were performed using a number of conservative assumptions, such as using a segregated solver approach. This in combination with the design basis consisting of a quite extreme operating scenario means an increased cooler performance is expected for an as-built cooler operating at normal conditions. An alternative CFD solver approach would be to use a coupled solver. This could have the potential to provide faster convergence with increased accuracy however the solver requires substantially more tuning, using an iterative process, meaning it was outside the scope of the thesis.

5.7.3.1 Further CFD Work Discussion

Based on these discussions the thesis has many suggestions of more work that needs to be done in order to select a design:

- Simulation "Wall" should be simulated with baffles to act as a comparative point for the chimney design.
- Design of the chimney should be tested with lower seawater temperatures as to pin-point at which conditions the chimney effect occurs.
- The design with reducing the diameter of the cooler (reducing the total) should also be considered and compared with lower seawater temperatures.
- Simulation "No wall" should also be simulated with the same conditions as the chimney.

- Simulation "Wall" (with baffles) and "No Wall" should be simulated with additional pipe stacks based on the height constraint, as to compare performance to the chimney simulations.
- All these should be simulated with worst case scenarios of seawater conditions with currents and temperatures.
- The simulations should be tested with a coupled solver, as to determine the level of conservatism and attempt to reduce computational time of the 3D models.

Based on those results a design could be chosen together with cost estimations. With this chosen design an OHTC map could be created with varying cooling medium mass flow and seawater temperatures, to attain an overview of the cooler performance.

5.8 Manufacturing Possibilities

The results from investigating the manufacturing possibilities and its implications of the cooler design is split up under different sections treating the sub-assemblies that makes up the cooler as independent parts. The cooler is divided into piping bundle, which contains both the small bore piping constituting the pipe stacks and the headers, which are connected to the centre and the outside of the spirals. Both the inlet and outlet header is connected to each of the small bore pipes. The walls and chimney is encompassing the piping bundle and the pipe supports. The subsea cooler and its sub-assemblies are presented as an exploded view in Figure 2.4. The sub-assemblies such as bottom frame and pipe supports are not investigated further from a manufacturing perspective, this is due to them being excluded during the CFD analysis and was assumed to not have any effect on the cooler performance.

5.8.1 Piping Bundle

The geometry of the piping bundle is somewhat unique when compared to conventional subsea coolers and heat exchangers. The most common design to build a cooler is in a rectangular shape with straight pipes that are connected with an 180 degree bend, see reference cooler illustrated in Figure 3.1. The bend is often refereed to as an elbow and is welded in to place forming a rectangular shape consisting of straight pipes and elbows. This way of manufacturing is generally quicker since the assembly time is low, when pipe segments quickly could be welded together. However, the cost of welds is quickly becoming significant when a large amount of elbows needs to be welded on. When talking to industry experts with several years experience in the subsea industry it was shown that the welding was often considered to be the cost driver when manufacturing subsea coolers. This is mainly due to the fact that the heat transfer area of a cooler or heat exchanger is directly correlated to the performance of the cooler, which makes the material cost parameter more difficult to adjust. An alternative is to select a cheaper pipe material with similar or better heat transfer properties, which decrease the overall cost of the cooler. A cheaper material is often associated with lower quality, which would indicate the risk of compromising on the quality. Since the cooler is placed subsea this would result in expensive maintenance and a potential failure in the cooler would mean that the entire OWF would have to be stopped, since it would pose a severe risk of damaging the components on the platform if operated without a cooling system. Therefore, a compromise in material selection to decrease cost is really not an option in order to be compliant with subsea standards. To reduce the overall cost of the cooler, the focus has been aimed towards decreasing the amounts of welds rather through a unique spiral design, rather than focusing on material selection. A spiral shape of the piping bundle would indicate a lower amount of welds since a continuous bend of the pipes are utilized rather than a sharp bend in the shape of elbows. The cost of bending is generally much lower than the cost of welding, offering a shorter manufacturing time per unit of pipe [45]. The CFD analysis also found that the pipes on the outside of the piping bundle had a lower effect on the heat transfer and therefore it was tested to removes these pipes to investigative any chimney effect would be present with a decreased diameter. The results could not conclude on the chimney effect but would have a positive effect on the cost implications of the cooler. This in turn could be connected to the challenge of offshore wind today being costly and innovative technologies is needed in order for it to become cost-competitive against fossil fuels. A reduced manufacturing cost of the cooler would suggest that it being suitable for this industry and would contribute in how offshore wind could decrease its costs.

5.8.2 Chimney

The purpose of adding walls and chimney to the cooler was that the heat transfer would be more predictive. In other words, the cooler would be less prone to differences in heat transfer due to changing seawater currents. However, as discussed under Section 5.7.1 the effects of the chimney in terms of heat transfer was very low. The thesis investigated the manufacturing possibilities and cost implications of adding a chimney to the cooler. Throughout the process it was found that some kind of glass reinforced plastic could be used as material for manufacturing the walls and chimney. This would allow for the entire walls and chimney to be manufactured by one-piece moulding to achieve its desired shape [46]. In a scenario were multiple coolers would be manufactured with the same type of chimney, the mould could be used several times and the cost of making the mould could therefore be split between all coolers. This would imply a low manufacturing cost, that decreases over time as more and more chimneys would be produced. When comparing this with the potential manufacturing cost of the pipe bundle which is planned to be built in stainless steel the overall cost of the chimney and walls was found to be comparatively low. This makes the argument of implementing a chimney stronger, even though it did not had a significant effect on the cooling performance. However, this is further discussed under Section 6.3 since the thesis did not provide enough evidence to make a unanimous conclusion on whether it is worth implementing the chimney or not.

5.8.3 Techno-Economic Discussion

Based on the CFD discussion, the chimney did not provide significant performance improvements for the worst-case scenario. However if the chimney is found in further work to be improved with lower seawater temperatures, it could be argued that the chimney should be implemented. This argument is resting on the finding that the chimney design would be relatively cheap in comparison to the cost of the piping bundle. However for seawater conditions that have operations within the tested temperature range, the chimney would be unnecessary when looking at the height constraint, instead adding a tube bundle would perhaps be a better solution. Even though the pipe bundle is driving the cost more pipe stacks would improve the heat transfer more warmer seawater temperatures. In warmer seawater temperatures, an alternative could therefore be to add as many pipe stacks equal to the cost of the chimney and removing the chimney. This would offer a solution that probably enables a higher heat transfer and at the same time maintains the same cost for the cooler as if it would have a chimney.

6

Conclusion

This chapter of the thesis presents the concluding findings of the thesis and how they contribute to answering the research question. Finally, further work outside of the thesis is discussed.

6.1 Concluding Remarks

Based on the market screening conducted for the North Sea, the thesis can conclude on the need for HVDC platforms for the coming seven years, in the growing offshore wind market. The key findings were that HVDC is becoming cost competitive for operators once they are requiring an export cable lengths that exceeds 175 km. The number of platforms that are planned to be placed outside of this distance are set to more than triple and that the installed capacity is set to seven-fold. The cooling demand is determined converting capacity and was found to be an average of 13,7 MW, which is a slight increase from today's platform but not in line with the increase in OWF capacity. The findings from the comparison in operating cost between subsea cooling and conventional sea water lift cooling found that cost savings of up to 83 % could be achieved by looking at the most energy consuming components. It is important to note that this figure is an estimation investigating solely pump work and power consumption of producing sodium hypochlorite. However, it could still be seen as an indication that you could achieve significant cost savings by finding alternatives to conventional sea water lift cooling.

The CFD analysis proved that verifying longitudinal pitch of pipe rows (stacks) did not enhance performance for seawater applications, neither did the chimney design. However the chimney still contributed to a larger heat transfer when including baffles, as compared to without chimney or walls. The thesis could then conclude that 3 design options are available for future testing:

- Cooler with only tube bundle
- Cooler with walls, chimney and baffles
- Cooler with a reduced number of pipe bundles

The thesis also found that the chimney and walls design was comparatively cheap in relation to the piping bundle. This would indicate that an added chimney would not affect the overall cost of the cooler significantly which makes the implementation of a chimney easier to motivate.

6.2 Research Questions

For the conclusion the thesis aimed to answer the research questions and can present the following:

How can CFD facilitate the optimization of a new design of subsea cooler, and how can different design aspects affect the performance?

The thesis was able to create, mesh and simulate the subsea cooler, based on the previous experience gathered within the company. Due to the nature of the cooler the attempts to optimise the design proved more difficult than anticipated. Both the assumptions researched previously regarding chimney design and tube arrangements proved futile for the passive cooler.

To what degree can the testing and evaluation of a subsea cooler model be improved by simplifying its features to reduce computational time in CFD simulations?

The computational time for the CFD simulations were investigated in two ways, the first assuming a 2D model could capture the trends of the cooler, which was invalidated due to the two dimensional assumption of the plane continuing infinitely failed to capture the circular nature of the cooler. The second attempt proved successful since the wedge model captured the circular relationships, it was able to capture the trends shown in the 3D model while reducing the computational time by 96 percent.

How can a subsea cooler be optimized when investigating the relationship between design and performance?

The thesis identified that baffles were an important design aspect for passive coolers, since the flow preferred to bypass the cooler tubes as to take the easy way out, the performance was significantly enhanced by the addition. Another important parameter discovered was the temperature differences between the cooling fluid and seawater as the natural convection approach cannot support the chimney design.

What are the key market dynamics and factors influencing the successful entry of the new subsea cooler in the wind power market, and how can strategic marketing and positioning be leveraged to ensure a sustainable competitive advantage in the evolving market landscape?

The thesis has investigated the market dynamics by conducting a market screening with the basis of an established break-even point of when HVDC is becoming cost competitive. These HVDC platforms has been further evaluated in terms of its cooling demand and in order to achieve a successful entry into this market the subsea cooling system needs to meet this demand. To ensure a sustainable competitive advantage the subsea cooling system was further investigated which resulted in findings suggesting a significant reduction in operating cost compared to the conventional cooling system. This in turn, could be connected to the slimmer margins faced for offshore wind were cost savings are of great importance for it to be competitive against, for instance, fossil fuels. Additionally, the environmental effect of the sodium hypochlorite concentration was concluded, which could be used as marketing point and highlight the decrease in environmental impact when choosing subsea cooling. The thesis investigated several areas that could contribute to a sustainable competitive advantage, such as the demand for subsea cooling and its alignment with the needs of offshore wind. The findings from these aspects together suggest that subsea cooling may offer a sustainable competitive advantage in the industry.

To what extent can existing technologies and knowledge be innovatively applied to meet the demands of the new changing market?

Subsea cooling has been around for several years, but its applications has been more aimed towards the oil and gas industry. Since the thesis provided evidence that the technology also could be used for renewable energy such as offshore wind it could be argued that existing technology with a focus on cost-reduction could be used to meet the demands for a market changing towards greater demand in the renewables sector.

6.3 Further Work

In order for the cooler to compete with the existing solutions on the market a metaocean data analysis has to be conducted for each OWF prospect region. As the assumption for the seawater temperature was based on the ability to utilise the cooler in as many places as possible. Thus in order to translate the heat transfer ability accurately to the regions. The CFD analysis concludes that for this assumption the chimney does not matter as the buoyancy force was too small to cause the chimney effect. However when changing the temperature difference there was an almost 45 percent increase in performance. Which sparks the discussion of how can the cooler be designed according to prospect regions of the market. Perhaps the chimney would be necessary for the north sea, while adding tube bundles are a better solution for warmer climates such as South-East Asia. However, to conclude on these statements a market screening would have to performed in other regions of the world as well, such as South-East Asia to establish the demand.

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