FACULTY OF SCIENCE AND TECHNOLOGY

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ABSTRACT

Pre-commissioning in subsea oil and gas fields ensures the readiness for operation and integrity of the valves, pipelines, and various components installed on the seabed. A critical task in this process is the pressure testing of pipelines, which involves flooding them with fluids such as freshwater, seawater, or water-glycol mixtures. The testing fluid, initially stored in vessel tanks, is typically at ambient temperature. The high insulation in the subsea pipelines creates a significant temperature difference between the internal fluid used for pressure testing and the pipeline surroundings. This condition creates a false impression of pressure decrease due to temperature stabilization. Consequently, it becomes challenging to accurately identify leakages during pressure testing, resulting in prolonged offshore campaigns.

This study aims to investigate different solutions for efficiently cooling large amounts of water within a reasonable time frame under different spread configuration cases. Five different alternatives were studied: 1) Installation of a subsea spool with low insulation to allow for natural and forced convection cooling from the sea currents. 2) Single heat exchanger on the vessel deck. 3) Water chiller on the vessel deck. 4) Producing seawater using a desalination system and pumping the water from the seabed. 5) Theoretical reviews of developed patents for subsea heat exchangers.

Heat transfer calculations indicate the first two alternatives won't cover the objectives. First, The required length of the spool piece renders this alternative unfeasible. As for the heat exchanger, using seawater as a cold fluid limits the process final temperature. Two technologies are found to be more suitable for this application due to their extensive utilization in marine applications. First, the installation of a system similar to an RSW chiller, commonly used in fishing vessels to cool seawater and preserve the catch. Capacity calculations for this project result in average ranges for this type of equipment. Alternatively, the installation of a desalination system provides a different approach to the problem, however, the outlet temperature of the water is limited to the minimum achievable seawater temperature, which, in turn, is restricted by the capacity and installed depth of a subsea pump.

Considering the advantages of using a chiller system on a vessel, an economic evaluation was performed using quotations for this system. The interviewed providers offered the option to purchase the system instead of renting when compared to the assumed daily rate of a diving vessel, acquiring a unit like this becomes profitable. The initial investment can even be recouped in a single project, and operational time in offshore campaigns will be significantly reduced while decreasing at the same time the company's CO2 emissions.

ACKNOWLEDGEMENTS

First, I would like to express my love and gratitude to my parents and my brother for their pure love and their unconditional support in this journey. Thanks for encouraging me to be a good person and for being always an inspiration, you taught me to fly high and that no dream is ever too big. To my friends in Colombia, thanks for always being my safe place and for being just a call of distance. To my new Colombian family and new friends in Stavanger, thanks for crossing my path and allowing me to share it with you.

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To Norway and its nature, for giving me such profound enlightenment moments, for teaching me so many lessons, and for showing me other ways of living and enjoying life.

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ABBREVIATIONS

- **EHTF** Electrically Heat Traced Flowline
- **FCG** Flooding, Cleaning and Gauging
- **HP** High Pressure
- **HT** High Temperature
- **ID** Inner Diameter
- **ILT** In-line Tee
- **IMR** Inspection, Maintenance and Repair
- **LMTD** Log Mean Temperature Difference
- **MSDS** Material Safety Data Sheet
- **MEG** Mono Ethylene Glycol
- **NTU** Number of Transfer Units
- **NORSOK** Norsk Sokkels Konkurranseposisjon
- **OD** Outer Diameter
- **OPEX** Operational Expenses
- **PLET** Pipeline end Termination
- **PLR** Pig Launcher and Receiver
- **PIG** Pipeline Integrity Launcher
- **RSW** Refrigerated Sea water
- **PiP** Pipe in Pipe
- **TIM** Tie in Module
- **XMT** Christmas Tree

NOMENCLATURE

ΔT Difference of Temperature between the two Fluids

Δx Length of the pipeline

$\dot{W}$ Ratio of power input per unit mass of refrigerant flowing

μ Dynamic viscosity of the Fluid

A Heat Transfer Area

Cp Specific Heat Capacity

D_i Internal Diameter of the Pipeline

D_o Outside Diameter of the Pipeline

f Friction factor of the pipeline

h_i Internal Convection Coefficient

h_o External Convection Coefficient

k Thermal Conductivity of the Material

L_{o} Distance from the pipe inlet to the point of interest

$LMTD$ Log Mean Temperature Difference

M_{h} Mass of the hot fluid in heat exchanger

NTU Number of Transfer Units

Nu_{laminar} Nusselt number in laminar flow conditions

Pr Prandtl Number

q Heat Transfer Rate

q_{conv} Heat Transfer Rate Due to Convection

q_{max} Maximun Possible Heat Exchange Rate

r_{i} Inner Radius of the pipeline

Re Reynolds Number

T_{i} Temperature of the Internal Fluid

U_{o} Overall Heat Transfer Coefficient

V Velocity of the Fluid

v_{pig} Pig Velocity

CHAPTER

ONE

INTRODUCTION

Due to the increasing global demand for oil and gas resources, oil companies have been compelled to explore and develop reservoirs situated in more challenging environments and extreme conditions (Moreno Trejo, 2012). In such cases, when conventional production methods prove inadequate, implementing subsea field developments is imminent. These subsea systems are positioned underwater, beneath the sea surface

A subsea field comprises various vital components, including wellheads, Christmas trees, manifolds, pipelines, and risers. Wellheads serve as the primary interface between the subsea reservoir and the production system, providing control and access to the well. Christmas trees, commonly mounted on top of the wellheads, regulate the flow of oil and gas during production operations. Manifolds act as distribution hubs, facilitating the efficient flow of hydrocarbons from multiple wells to processing facilities.

Pipelines, a critical element of subsea systems, enable the transportation of oil and gas over significant distances, connecting the subsea fields to onshore or offshore processing facilities. Risers, on the other hand, provide a conduit for the upward movement of hydrocarbons from the subsea wells to the production platforms or floating vessels on the surface.

Some of the components above mentioned are shown in Figure 1.0.1, which illustrates the interplay of these components, highlighting the complexity involved in the design and operation of such systems. This configuration is carefully engineered to ensure the safe and efficient extraction of oil and gas resources from challenging underwater environments.

Pre-commissioning is an activity performed after the installation of these components, prior to the handover of the system to the client, the main purpose is to ensure the subsea system, facility or equipment is fit for purpose, tested, and in a state where it is ready for commissioning-startup. Typically operations performed during pre-commissioning include flooding, cleaning and gauging of pipelines pressure test, leak test of subsea mechanical connections and dewatering of the system.

When extremely long tie back projects takes place specially in NCS due to the low temperatures in the seabed, thermal challenges become even bigger and the need of active heating solutions takes place to maintain the temperature in the line above the

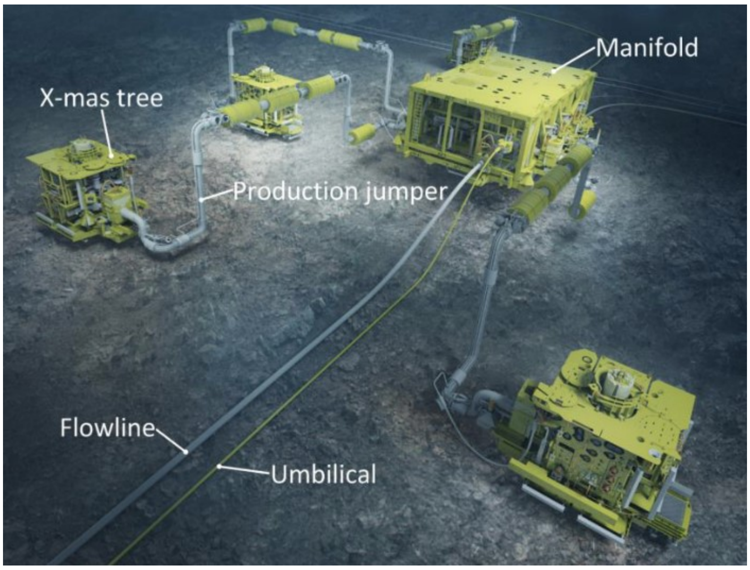


Figure 1.0.1: Subsea Production System Overview (AkerSolutions, 2023)

wax appearance. Pipe in Pipe technology has become an essential part in the subsea fields. This technology minimizes heat losses due to the inclusion of insulation with very low thermal conductivity between the inner pipe and the flow line. An illustration of a typical configuration of a Pipe in Pipe flowline is shown in Figure 1.0.2

The condition in the pipelines, combined with the temperature difference between the water used for flooding and the seabed temperature, leads to a prolonged stabilization period for the temperature. This, in turn, can have an impact on the pressure stabilization during the pressure test campaign. Consequently, offshore operations may experience significant waiting times as the temperature needs to stabilize before the pressure test results can be valid. As Figure 1.0.3 illustrates, depending on the volumes and the length of the pipeline, this process can take even up to 10 days.

1.1 Objectives

The main objective of this study is to investigate solutions to cool water during flooding, cleaning, and gauging operations of subsea assets, especially pipelines. The different methods analyzed must be evaluated according to heat transfer rates, flow rates, and operational parameters such as deck space, time, and overall cost of the cooling spread. Solutions that depend on the natural convection from the surrounding seawater are interesting.

The arrangement of the spread or placement of the equipment in the process will have an effect on the conditions used to perform heat transfer calculations. Also, the temperature profile chosen is based on NCS conditions and sea states from previous projects.

1.2 Structure of the report

A brief outline of each chapter is presented below:

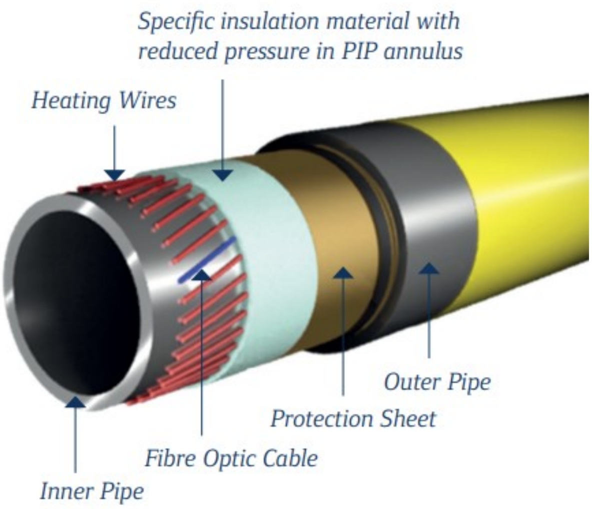


Figure 1.0.2: Pipe in Pipe Flowline (Subsea7, 2016)

Chapter 2. Theory review of thermodynamics, heat exchangers, and refrigeration systems, available and well-known concepts and configurations. Review and description of methods for calculating heat transfer rate and heat transfer coefficients in flow inside tubes. Theoretical review of methods to calculate heat transfer coefficients in external flow around cylinders. General calculations on overall heat transfer implementation of both conduction and convection.

Chapter 3. System overview of the general arrangements in subsea field development, possibly installed equipment and pre-commissioning operations. Definition of study cases for heat transfer calculations and revision of available equipment and typical vessel configuration.

Chapter 4. Solutions Assessment. Each of the proposed solutions in this chapter will have a Description of possible concepts for cooling of process streams, calculations of the water cooling with different mediums (MEG or water), different flow rates and different volumes, and finally, an evaluation of operational and financial constraints, advantages and limitations of the system.

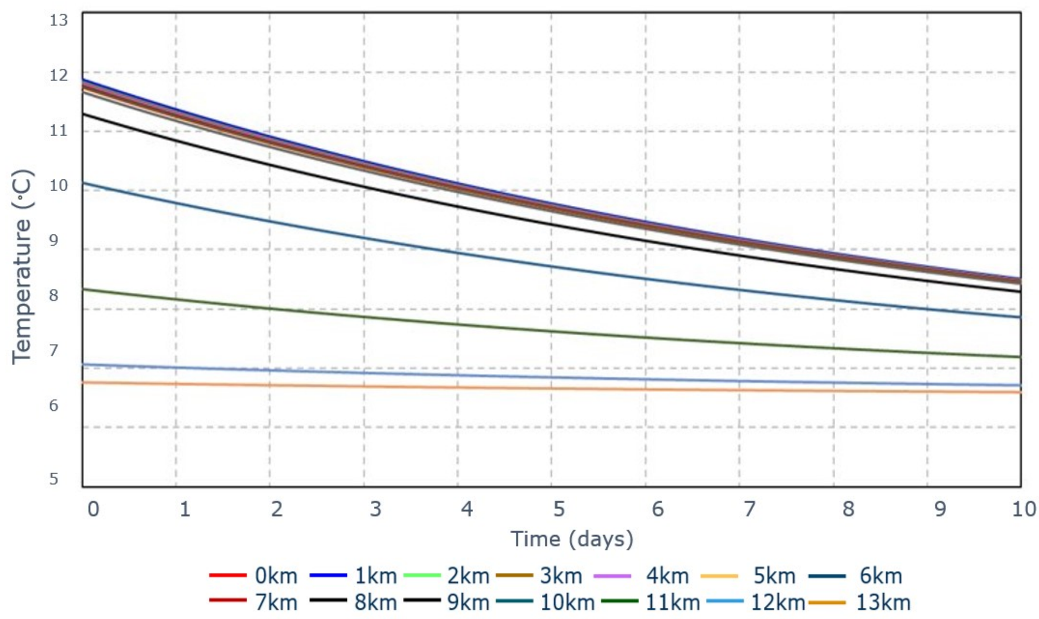
Chapter 5. Economical evaluation of the chosen equipment for cooling using cost savings metric and cost-effectiveness ratio.

Chapter 6. Conclusion and outlining of design of the cooling system.

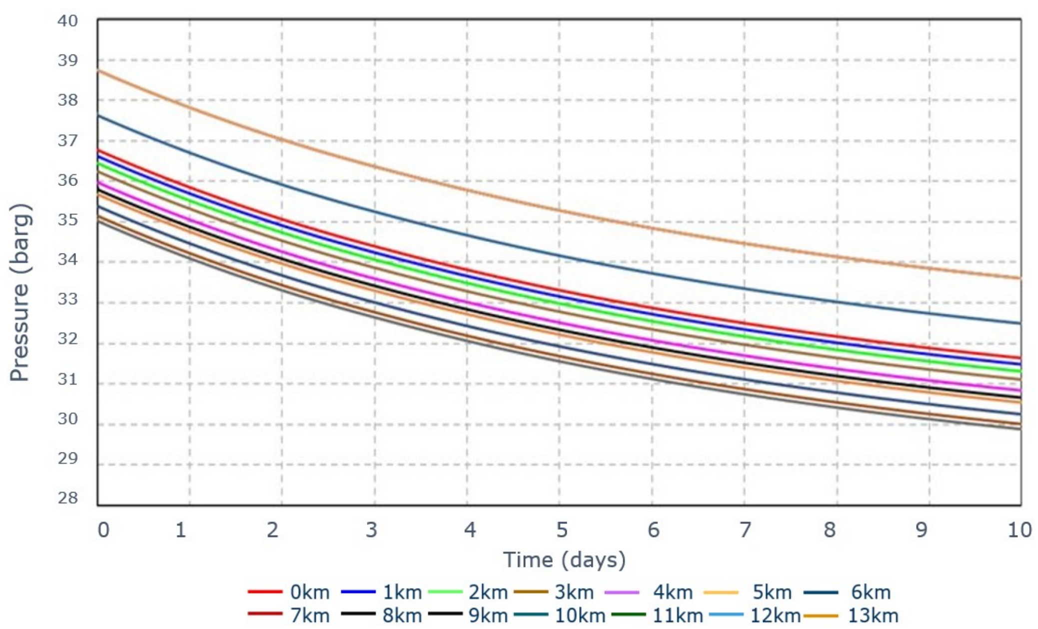
1.2.1 Stakeholders

This master thesis will be developed in collaboration with Subsea7, a global engineering, construction, and services company that delivers offshore projects for the oil and gas industry. The company provides a range of engineering services, including support in concept definition, design, procurement and fabrication, installation and commissioning, inspection, repair, and maintenance services (IMR) related to subsea fields.

A review of several technical documents from the company on existing pre-commissioning projects is used to establish study cases. Contact with the providers will be done on behalf of Subsea7 to investigate available technology and equipment in the market.



(a) Temperature trends during cool down 100% freshwater.



(b) Pressure Trends during cool down 100% freshwater.

Figure 1.0.3: Figure (a) shows the temperature decreasing tendency once the line is flooded. Figure (b) shows the pressure trends once the line is flooded

CHAPTER

TWO

THEORY

The comprehension of heat transfer mechanisms is essential for understanding the intricate processes involved in effective cooling techniques. By studying the transfer of heat around cylinders and plates, we aim to gain insights into the underlying principles governing the exchange of thermal energy in these configurations. This knowledge will provide a solid foundation for developing optimized cooling solutions.

Furthermore, an exploration of the refrigeration cycle will be conducted in this chapter. The refrigeration cycle serves as the core principle behind various cooling systems, enabling the extraction of heat from a specific space or object. Within this section the essential considerations and present relevant formulas pertaining to each stage of the refrigeration cycle will be presented. Comprehending the intricacies of this cycle, will enhance the understanding of the overall cooling process and pave the way for improved cooling system designs.

2.1 Heat Transfer Modes

Heat transfer is the study of the movement of thermal energy from one system to another as a result of a temperature difference. This process is an essential part of many physical phenomena and has a variety of applications in the offshore industry, especially in subsea fields where production fluids usually at high temperatures are flowing through pipelines installed at the bottom of the sea. Heat transfer in pipelines with the surroundings can occur by one or a combination of the following three known heat transfer modes, convection, conduction and radiation.

2.1.1 Conduction

Heat transfer is mainly governed by the Fourier law of heat conduction. Conduction is a mode of heat transfer due to temperature difference and it occurs across a stationary medium that can be solid or fluid. It does not involve the mass flow or mixing of fluids. The rate of heat transfer is given by:

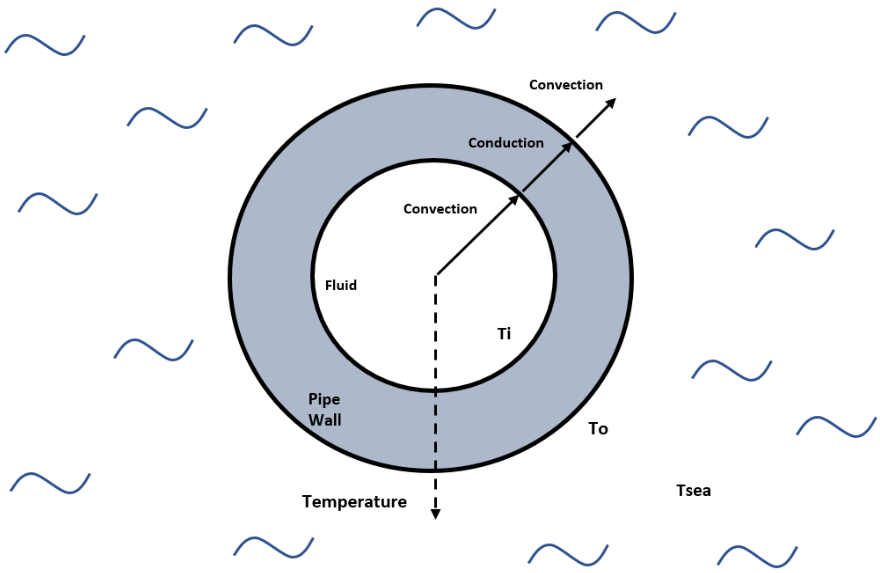


Figure 2.1.1: Heat exchange mechanisms in subsea pipelines

$$q = -kA\frac{\Delta T}{\Delta x} \tag{2.1}$$

This equation states that the rate of heat transfer Q through a material is proportional to the temperature difference across the material ΔT, the surface area A through which the heat is being transferred, and the thermal conductivity k of the material. Δx represents the distance over which the temperature difference occurs. The negative sign in the Equation indicates that heat flows from higher-temperature regions to lower-temperature regions.

Since heat transfer in conduction occurs through a material when a temperature gradient is established within it, the nature of the material is one of the most important drivers in this method since it determines the thermal conductivity k expressed in W/(mK). The highest values correspond to electrical conductors, followed by alloys of metals and nonmetals. The conductivity of liquids falls below these materials. Some reference values for this property are resumed in Table 2.1.1 and can be used in Equation 2.1 to perform heat flux calculations across the material

Material selection is therefore an important part of the design of heat transfer solutions and subsea pipelines. Whether in some applications, it is necessary to have a high heat transfer rate. In others like the operation of subsea pipelines for the transportation of fluids, the low temperature in the seabed makes it necessary to have good insulation materials and materials with low conductivity. Metallic materials are usually used, including carbon structural material, and varying between carbon steel, low alloy steel, and stainless steel.

Table 2.1.1: Examples of thermal conductivity k at 279 K

Material	**Thermal Conductivity** W/(mK)
Aluminium	204.2
Carbon Steel 1% C	43.3
Chrome Steel 20% Cr	22.5
Chrome Nickel Steel	12.8
12% Cr Stainless Steel	16
Cooper	378
90/10 CuNi	45
Titanium	16
Water	0.60
Air	0.026

2.1.2 Convection

Heat convection occurs between a fluid in motion and a bounding surface when they are at different temperatures. Even though the material of the surface won't affect the heat transfer, the shape and the material of the surface will affect the flow and this will have an impact on the heat transfer.

When considering a fluid flowing on a surface, a region called a hydrodynamic boundary layer is developed in the fluid where the velocity varies from zero in the surface to a finite value U_∞ associated with the flow velocity. If convection due to temperature difference is happening, the formation of a thermal boundary layer is expected, where the temperature will vary from Ts in the surface at T∞ in the outer flow.

This heat transfer mode is classified according to the nature of the flow. The convection will be forced when it depends entirely upon the results of an external influence like a pump and it will be called free or natural convection when the flow is caused by buoyant forces generated by the heating or cooling of the fluid.

The heat flux by convection y is defined in Equation 2.2:

$$q = h(T_s - T_\infty) \tag{2.2}$$

Where the convective heat flux (q) expressed in W/(m^2) is proportional to the difference between the temperatures of the surface and the fluid. h represents the convection heat transfer coefficient in W/(m^2K) which depends on the conditions of the boundary layer, influenced by the geometry of the surface. The heat flux in W will be obtained by integrating the surface area of the object being heated or cooled (in m^2)

$$q_{\text{conv}} = hA(T_{\text{surface}} - T_{\text{fluid}}) \tag{2.3}$$

The values of the convection heat transfer coefficient h is essential for heat flux calculation in model boundary conditions, and it is obtained experimentally by conducting tests in which the rate of heat transfer between a solid surface and a fluid is measured under controlled conditions. There are also analytical and numerical methods that can be used to estimate the value of the heat transfer coefficient based on theoretical models and simulations. Typical values are shown in 2.1.2

Table 2.1.2: Values of the Convective Heat Transfer Coefficient

Process	**h** W/(m^{2}K)
Free convection	
Gases	2 - 25
Liquid	50 - 1000
Forced convection	
Gases	25 - 250
Liquid	100 - 20.000
Convection with phase change	
Boiling or condensation	2500 - 100.000

Convection calculations will vary depending if the flow is internal and external.

Internal Convection

This occurs between the fluid flowing inside the pipeline and the internal surface and depends on the fluid properties, velocity of the flow, and pipe diameter(Bai & Bai, 2011). It is important to determine the type of flow inside the pipe using the Reynolds number. Depending on this, the adequate correlation can be selected for determining the Nusselt Number

Reynolds number can be determined with Equation 2.4.

$$Re = \frac{\rho V D}{\mu} \tag{2.4}$$

A laminar flow is considered with Re $<$ 2100 and a fully developed turbulent flow with Re $>$ 10,000 and between this can be considered as the transition region.

The Prandtl number is a dimensionless number that characterizes the relative magnitudes of momentum and thermal diffusivities in a fluid, and it is defined in Equation 2.5 as the ratio of the kinematic viscosity to thermal diffusivity.

$$Pr = \frac{Cp_f \mu_f}{k_f} \tag{2.5}$$

It is an indicator of how quickly momentum and heat are transferred within a fluid. In refrigeration applications, A low Prandtl number means that the refrigerant is a better conductor of heat than a fluid with a high Prandtl number. Or that a low-Prandtl fluid requires a smaller heat transfer surface area to achieve the same heat transfer rate as a high-Prandtl fluid. (ASHRAE, 2006)

Equation 2.6 was proposed by Dittus and Boelter as a correlation to determine the Nusselt number in turbulent flow of single-phase fluids with smooth surfaces relates the Nusselt Number to the Reynolds number and the Prandtl number (F.W. Dittus, 1930)

$$Nu_{turbulent} = 0.023 Re^{4/5} Pr^{n} \tag{2.6}$$

This Equation is suitable for Pr between 0.7 and 160, Re>10000 and L/D>10. It is accurate when difference in temperature are moderate. Is worth that several modern correlations have been developed that could reduce the error obtained from the correlation in their simpler form. The coefficient n=0.3 for cooling applications

The principal difference between these two flows regarding heat transfer takes place in the turbulent flow, where an additional mechanism of heat transfer in the azimuthal and radial directions must be considered. Commonly named Eddy transport. This condition provides a better transfer of energy across the flow at a given axial position than in laminar flow. (Subramanian, 2014)

If the flow is laminar, hi can be calculated using Hausens Equation (Hausen, 1943):

$$Nu_{laminar} = 3.66 + \frac{0.668 \frac{D_i}{L_o} RePr}{1 + 0.4 \frac{D_i}{L_o} RePr^{2/3}} \tag{2.7}$$

Where Lo is the distance from the pipe inlet to the point of interest. When this relation approaches 0, the Nusselt number can be estimated as 3.66.

The heat transfer in the transition region tends to have more uncertainty. One of the most used correlations was proposed by Gnielinski (Gnielinski, 1976), it incorporates the friction factor f of the pipeline, which can be calculated using the Moody diagram.

$$Nu_{transition} = \frac{\frac{f}{8}(Re - 1000)Pr}{1 + 12.7 \frac{f}{8}^{1/2} (Pr^{2/3} - 1)} \tag{2.8}$$

After determining the Nusselt number with one of the appropriate correlations above, it is possible to calculate the heat transfer convection coefficient using Equation 2.9 with k expressed in W/(mK)

$$Nu_D = \frac{h_i D}{k} \tag{2.9}$$

Table 2.1.3 contains a range of reference values for internal convection coefficient in different mediums, for turbulent flows. These values can be used to compare with the results on simulations.

External Convection

For external convection coefficient calculations, the Hilpert correlation is widely used in subsea applications:

$$Nu_o = C Re_o^m Pr_o^{1/3} \tag{2.10}$$

The non-dimensional numbers must be calculated for external conditions, where C and m are correlation constants dependent on the Re number range and listed in Table 2.1.4

Table 2.1.3: Typical internal Convection Coefficients for Turbulent Flows (Gregory, 1991)

Internal Convection Coefficient, hi	
Fluid	W/(m^{2}K)
Water	1700-11350
Gases	17-285
Oils	55-680

Table 2.1.4: Hilper correlation constant for different Re numbers ranges. (Hilpert, 1933)

Reo	**C**	**m**
$4 \times 10^{-1} - 4 \times 10^{0}$	0.989	0.330
$4 \times 10^{0} - 4 \times 10^{1}$	0.911	0.385
$4 \times 10^{1} - 4 \times 10^{3}$	0.683	0.466
$4 \times 10^{3} - 4 \times 10^{4}$	0.193	0.618
$4 \times 10^{4} - 4 \times 10^{5}$	0.027	0.805

For liquid systems, when the velocity of the surrounding fluid is less than 0.05 m/s natural convection will dominate the system and a reference value of 200 W/(m^{2}K) (Bai & Bai, 2011)

Water flowing at high velocities can experience flow instabilities such as turbulence, eddies, and vortices. These instabilities can result in non-uniform flow and temperature distribution, reducing the overall efficiency of the heat transfer process.

The flow of water can cause temperature variations within the fluid itself due to mixing and turbulence. This can result in non-uniform temperature distribution across the heat transfer surface.

2.1.3 Radiation

This term comprises the transfer of heat through electromagnetic waves (or photons) from a hot surface to a colder surface. It occurs without a physical medium or any direct contact between surfaces. The radiation originates from a special source such as the sun or from other surfaces to which the surface of interest is exposed.

Radiation is not a significant mode of heat transfer in water, as water is a poor absorber and emitter of radiation. Instead, it is an important aspect of heat transfer analysis of onshore oil and gas transmission lines. and can be ignored inside the pipelines due to the low temperatures in the subsea systems

2.1.4 Overall Heat Transfer Coefficient (U Value)

This parameter, determines the heat transfer in all the system (pipelines), it incorporates the thermal conductivity of the material, but also the geometry of the system. The U value considers the thermal resistance of the pipeline, the convective heat transfer of the fluid inside the pipeline and the heat transfer coefficient in the surroundings of the pipeline. This can be calculated either using theoretical models or experimental data.

For doing thermal design of pipelines, it is needed to consider both steady state and transient heat transfer analysis. Considering that the temperature of the fresh water flowing inside the pipeline will decrease as it flows along the pipeline due to the heat transfer to the pipe-wall to the surroundings environment. Conduction

Altogether, convection is the dominant mode of heat transfer in water, especially in large bodies of water such as oceans and lakes.

The following process regarding heat transfer is applicable for subsea systems:

- Convection between the internal fluid of the pipeline to the pipeline surface
- Conduction across the material of the pipeline, buried or insulated pipelines, convection also takes place in the insulation layers or soil where the pipeline is buried.
- Convection from the external surface of the pipeline to the surrounding fluid.

The Overall heat transfer coefficient is a parameter that determines the heat transfer in a system of pipelines. It takes into consideration the thermal conductivity of the materials, (conduction) and the geometry of the system (convection)

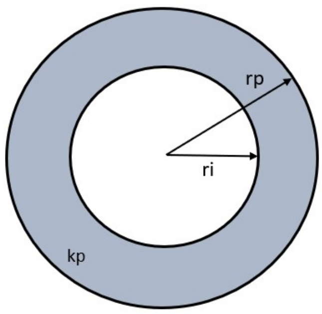


Figure 2.1.2: Parameters to use in the Overall Heat Transfer Coefficient Equation

$$\frac{1}{U} = \frac{r_p}{h_i r_i} + \frac{r_p ln(\frac{r_p}{r_i})}{k_p} + \frac{r_p}{r_o h_o} \tag{2.11}$$

2.2 Heat Transfer of MEG and water-MEG Mixtures

Subsea applications involve complex systems and components that operate under challenging conditions, such as high pressures and low temperatures. One of the main concerns in these applications is the formation of hydrates or the accumulation of lattice-like structures, which can cause severe flow assurance problems in gas production lines with water content

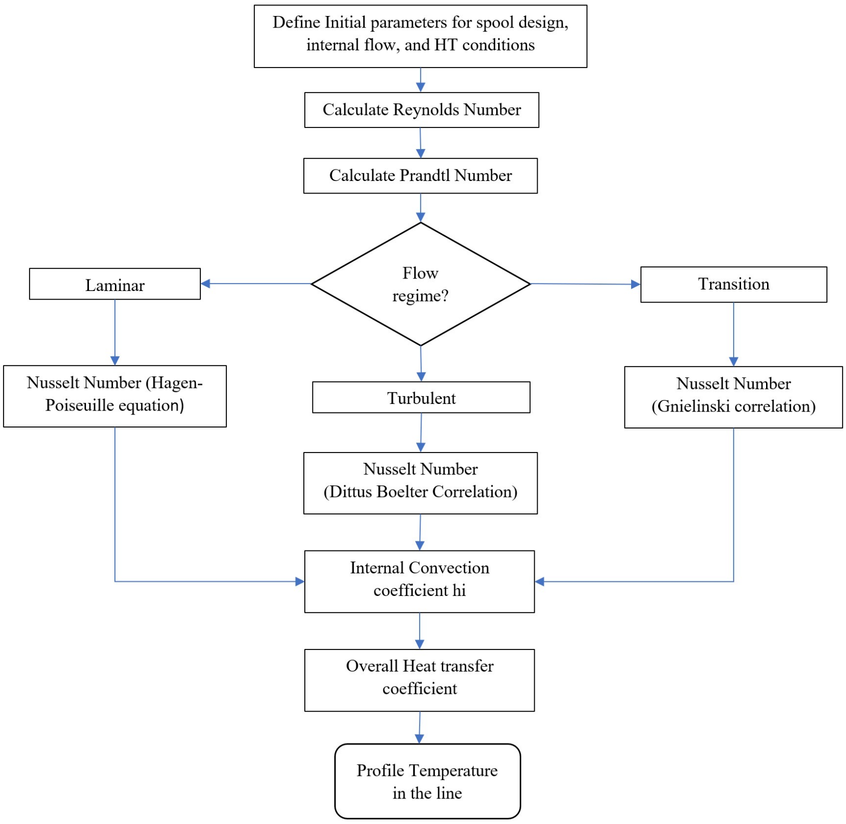


Figure 2.1.3: Heat Transfer Analysis Process Flow Diagram

MEG is a highly effective anti-freeze agent that can be used to lower the freezing point of water. By injecting MEG into the gas production lines, the formation of hydrates can be prevented, ensuring a smooth and uninterrupted flow of gas and facilitating the starting operation of the pipelines. I pre-commissioning, testing the lines with MEG or water-MEG mixtures allows for the evaluation of the injection system's effectiveness and can identify any potential issues that need to be addressed prior to delivery of the system to the client.

As it is a common fluid used in these activities, it is important to describe the heat transfer properties, which vary according to pressure and temperature, compared to water, MEG has a lower thermal conductivity and specific heat capacity, but it also has a lower viscosity and higher boiling point, which make it suitable for use in systems that operate at high temperatures or require low pumping power.

The thermal conductivity of MEG is about 0.22 W/mK at 20°C, which is lower than that of water (around 0.6 W/mK at 20°C). Meaning that MEG is less effective at conducting heat compared to water, and as a result, it may require a larger heat transfer surface area or longer residence time in a heat exchanger to achieve the same heat transfer rate as water.

The specific heat capacity of MEG is about 2.25 kJ/(kgK) at 20 °C, which is lower than that of water (around 4.2 kJ/kgK at 20°C). This means that MEG requires less heat energy to increase its temperature by a certain amount compared to water, and as a result, it may have a faster heating or cooling response time in some applications. Figure 2.2.1 shows the specific heat capacity for different Water-MEG mixtures, decreasing proportionately to the MEG amount in the mixture.

Thermodynamic properties of water-MEG mixtures can be estimated using linear interpolation as shown in Figure 2.2.1

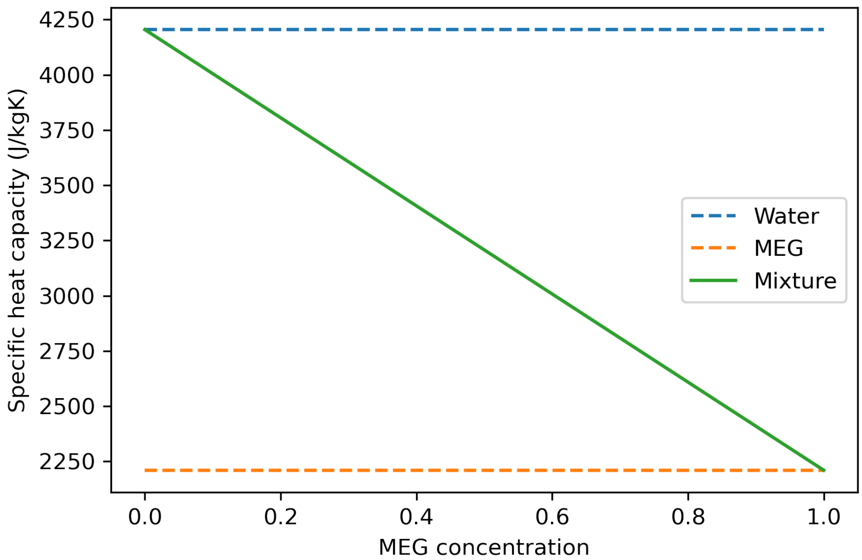


Figure 2.2.1: Specific Heat Capacity for Water-MEG mixtures.

Table 2.2.1: Heat transfer coefficients of water-MEG mixtures (ToolBox, 2005)

Specific Heat W/m^{2}K							
Ethylene Glycol Solution (%)	**Temperature (°C)** -20	-10	0	10	20	30	40
0			4.203	4.195	4.189	4.185	4.183
10			4.071	4.079	4.087	4.096	4.105
20			3.918	3.935	3.951	3.968	3.985
30		3.742	3.764	3.785	3.807	3.828	3.85
40	3.542	3.589	3.595	3.621	3.647	3.674	3.7
50	3.35	3.381	3.412	3.442	3.473	3.504	3.534
60	3.145	3.179	3.214	3.249	3.284	3.319	3.354
70	2.925	2.953	3.002	3.041	3.08	3.119	3.158
80	2.691	2.733	2.776	2.819	2.862	2.904	2.947
90		2.489	2.538	2.582	2.828	2.674	2.721
100		2.231	2.281	2.33	2.38	2.429	2.479

Regarding viscosity, it is lower compared to water, which means that it flows more easily through pipes and channels, and requires less pumping power. MEG also has a much higher boiling point (about 198°C) compared to water (100°C), which makes it suitable for use in high-temperature applications without evaporating or boiling off.

Table 2.2.2: Density of water-MEG mixtures (ToolBox, 2005)

Density - $kg\,m^{-3}$					
Mass Fraction of MEG in solution	**Temperature** °C				
	-8	**-4**	**0**	**20**	**40**
0			1000	998	992
0.1		1019	1018	1014	1008
0.2	1038	1037	1036	1030	1022
0.3	1056	1055	1054	1046	1037
0.4	1075	1073	1072	1063	1052
0.5	1093	1092	1090	1079	1067
0.6	1112	1110	1107	1095	1082

Overall, the choice between MEG and water as a heat transfer fluid depends on the specific requirements of the application, its important to determine the working temperatures in the flow line or system to determine the amount of Water and MEG in the mix.

Table 2.2.1 and 2.2.2 (ToolBox, 2003) have different values on meg - water mixture properties at different temperatures that can be used for studies involving MEG mixtures.

2.3 Heat Exchanger Theory

A heat exchanger is a device that transfers thermal energy from one medium to another, without mixing of the fluids occurring. They are used in many industrial applications and the basic working principle is the difference of temperatures between the two fluids. Heat transfer occurs through a conductive surface that separates them. Types and materials have several impacts on the performance, the selection of an appropriate heat exchanger will depend on the type, material, operating conditions, and working parameters.

2.3.1 Selection of a Working Fluid

The selection of the working fluid is one of the most important parts of cycle design in heat exchangers. It will especially affect their efficiency and financial constraints.(Tchanche et al., 2009) The economics on the heat exchanger will be linked mostly on the thermodynamic properties of the cooling fluid.

The principal properties to asses a cooling fluid are: Low specific volumes, high efficiency, amount of pressure in the heat exchanger, cost, toxicity, ODP and GWP (Bahrami et al., 2022)

For the application of this project, it is desirable to rely on natural and forced convection from the seawater, therefore the solutions presented in the following chapters will aim to evaluate the efficiency of heat transfer using this as a working fluid. (Lee et al., 2021) Also, it is desirable to explore the properties of different cooling fluids such as refrigerants and even cryogenic fluids.

To mitigate these impacts and constraints, it is important to carefully design the heat exchanger, taking into consideration the flow rate, water velocity, and other operational parameters. Additionally, proper maintenance, cleaning, and monitoring of the heat transfer surface can help reduce fouling and improve the efficiency of the heat exchange process. (Kakaç & Çengel, 2002)

2.3.2 Heat Exchanger Design and Performance

For the design of heat exchangers, the system is divided in nodes that will analyze the individual change during cooling. Many heat exchangers use the increase of the contact area as the main working principle, therefore a cautious selection of geometry must be carried out. To predict the performance of a heat exchanger, a relation between the total heat transfer rate to quantities like fluid temperature in the inlet and outlet, overall heat transfer coefficient and total surface must be carried out. For beginning this analysis, two procedures for performing heat exchange calculations are explained. Both methods can be used and should deliver similar results, but the NTU method can be easier to implement for some applications.

When water is flowing, it can carry impurities and contaminants that can deposit on the heat transfer surface and reduce its effectiveness. This effect, called fouling, can increase the thermal resistance and reduce the rate of heat transfer. During the normal functioning of equipment, fouling layers can appear due to impurities, rust, and other reactions between the surface and the fluid. This deposition can increase the resistance to heat transfer. Some correlations account for this introducing the fouling factor. (Incropera & DeWitt, 2002)

Log Mean Temperature Difference LMTD Method

The LMTD method is a commmon approach for calculating the heat transfer rate in a heat exchanger, when the fluid inlet temperatures are known and outlet temperatures are specified. It uses two relations obtained by applying overall energy balances to the hot and cold fluids. (Incropera & DeWitt, 2002)

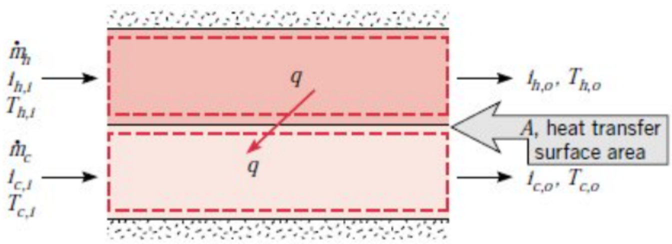


Figure 2.3.1: Straight tube and shell Heat Exchanger (Incropera & DeWitt, 2002)

The application of the Steady Flow Energy Theory for this case is shown in Equation 2.12 and represents the total heat transfer rate between the hot and cold fluid. There must be assumed negligible heat transfer between the heat exchanger and the surroundings, as well as any potential and kinetic energy changes.

$$q = m_h c_p, h(T_h, i - T_h, o) \tag{2.12}$$

$$q = m_h c_p, c(T_h, o - T_h, i) \tag{2.13}$$

Equation 2.13 can be used to relate the total heat transfer rate q to the difference of temperature between hot and cold fluid. i represent the fluid enthalpy. Also, there is the assumption that the fluids are not undergoing a phase change. But since the delta temperature varies with the position, it is desired to use an appropriate mean temperature difference.

$$q = UALMTD \tag{2.14}$$

Where U is the overall heat transfer coefficient in W/m^2/K, A is the heat transfer surface area in m^2, and LMTD is the logarithmic mean temperature difference.

Two important assumptions are to be considered:

- Constant overall heat transfer coefficient over the surface.
- Fluids well mixed and with constant properties.

The Log Mean Temperature Difference (LMTD) formula is then given by:

$$\text{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} \tag{2.15}$$

Where ΔT_1 is the temperature difference between the hot fluid inlet and the cold fluid outlet, and ΔT_2 is the temperature difference between the hot fluid outlet and the cold fluid inlet.(Incropera & DeWitt, 2002)

NTU Method

This is a method used preferably when only inlet temperatures are known. It is used to analyze heat transfer processes in heat exchangers using a dimensionless parameter, relating the fluid transfer to three main parameters that are: fluid flow rate, temperature difference, and property of the fluids in the heat exchanger.

For the application of this method, it is important to start with determining the maximum possible heat exchange rate, using the following equation:

$$q_{max} = C_{min}(T_{h,i} - T_{c,i}) \tag{2.16}$$

With this, it is possible to define effectiveness as the ratio of the actual heat transfer to the maximum possible heat rate in the heat exchanger

$$\epsilon = \frac{q}{q_{max}} \tag{2.17}$$

The number of transfer units is a dimensionless parameter defined as

$$NTU = \frac{UA}{C_{min}} \tag{2.18}$$

The effectiveness-NTU (Number of Transfer Units) relations are mathematical expressions used to calculate the heat transfer effectiveness of a heat exchanger. Several effectiveness-NTU relations are commonly used, depending on the configuration and geometry of the heat exchanger and the specific operating conditions.

For a shell and tube heat exchanger the following formulas can be applied to calculate the effectiveness and the NTU (Incropera & DeWitt, 2007)

$$\epsilon = \frac{1 - e^{-NTU}}{1 + R - e^{-NTU}} \tag{2.19}$$

2.3.3 Counter Flow and Parallel Flow

Firure 2.3.2 illustrates the main difference in heat transfer for this two types of configurations. In parallel flow, the fluids both flow in the same direction so the temperature difference is large at the beginning but diminishes along the length until both fluids are in equilibrium at the same temperature, then no more heat can be transferred

However, in counter flow design, the two fluids flow in opposite directions resulting in a lower inlet temperature difference, but, as the two fluids travel through the heat exchanger in opposite directions, the fluid constantly es newer fluids which are at higher or lower temperatures, causing the nearly constant temperature difference along the entire length of the heat exchanger making much harder for the two fluids to reach the same temperature. The thermal energy is being refreshed constantly, allowing more time for more heat to transfer over and for smaller and more efficient heat exchangers to be designed

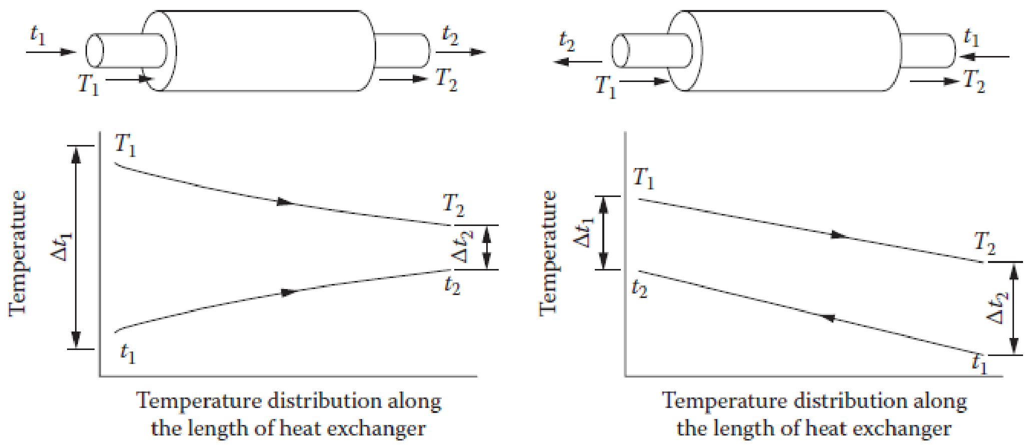


Figure 2.3.2: Temperature distributions in a Parallel flow (left) and counter flow (right) heat exchanger (Thulukkanam, 2013)

2.3.4 Shell and Tube Heat Exchanger

This is the most widely heat exchanger used in industrial processes. It consists of a bundle of tubes located inside a shell. In cooling applications, the hot fluid flows through the shell, and the cold fluid flows through the tubes. The number of tubes is determined by the mass flow rate and velocity of the working fluid. The shell and tube design provides a large heat transfer area typically of 50 - 500 m^2/m^3, therefore efficient heat transfer between the two fluids, it also allows for easy cleaning and maintenance. They are offered for a wide variety of temperatures and pressures.

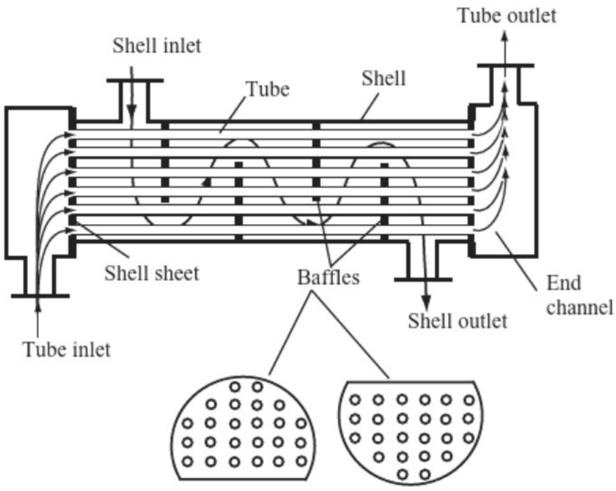


Figure 2.3.3: Typical configuration of straight tube and shell Heat Exchanger (Kaviany, 2014)

The heat exchanger in 2.3.3 is the simplest form of this type which involves single tube and shell es. the coolant is circulated on the shell side through a pump, extensive research has been done with this type of heat exchanger, which represents an advantage as multiple models and software exist to modulate the process in this equipment. Studies regarding materials should be done depending on the type of coolant used, and the addition of a pump must be considered in the economic evaluation.

Shell and tube type heat exchanger is economically better than the plate and fin-tube heat exchangers. Materials in the heat exchanger for this use case must consider corrosion due to seawater, Cu-Ni90-10 and Cu-Ni 70-30 are good materials

The calculation of the LMTD in a shell-and-tube heat exchanger is more complicated than in a simple counter flow heat exchanger. One approach is to use a correction factor, F, to account for the non-uniform temperature distribution along the length of the shell-and-tube exchanger. The corrected LMTD, LMTDc, is given by:

$$LMTDc = FLMTD \tag{2.20}$$

2.3.5 Plate Heat Exchangers

In this type of heat exchanger, the fluids are completely separated by the plates and they don't mix. Gaskets and plates are used to separate flow medium and prevent them from mixing. The gaskets are adhered to one side of the plate only, and two covers one on each side of the plate stack. The inlets and outlets are mounted to the fixed plates. The fluids typically flow in opposite directions

The plates are usually made of stainless steel but the brazing which joins each of the plates together is usually made from copper they can vary depending on the fluids that are being circulated through. The heat capacity on a plate heat exchanger can be modified by adding or removing plates to the design

They are classified in three groups: gasket, spiral and panel type.

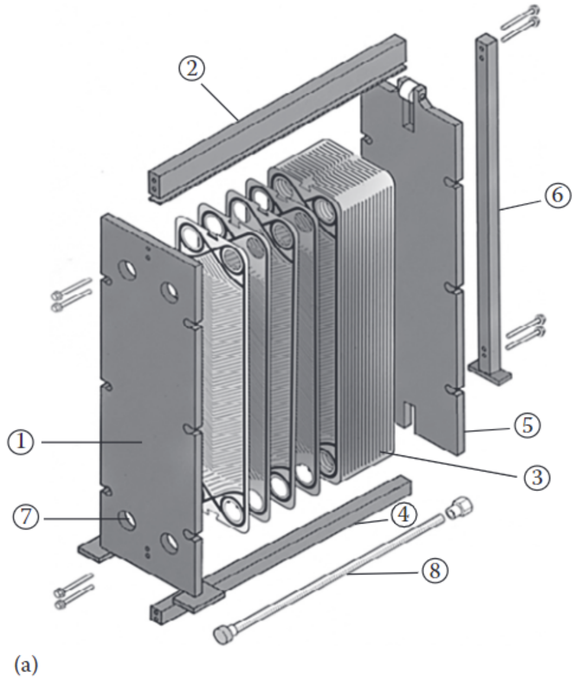


Figure 2.3.4: Plate Heat Exchanger Configuration (Thulukkanam, 2013)

Brazed plates heat exchangers are used in small applications, the main difference relays in the configuration, where the plates are brazed together, making one unit without the use of gaskets. The braze and the alignments of the plate dictates which of the channels each fluid can flow into. It is better against leaks and can have higher efficiencies than the gasket type. Cleaning is more difficult for cleaning and maintenance and the whole unit must be replaced if damaged. The plate has a fish-bone pattern

Microplates heat exchanger are the lattest technology on heat exchangers, providing great efficiency to date. They can be either gasket or brazed plate type, and the biggest change comes in the plate design, which uses small dimples patterns which allows the fluid to spread across the plates much more evenly whcich maximizes the heat exchange area and increases the turbulency, which also increases the heat transfer. This improvement has allowed to design lighter and smaller heat exchangers. They reduce the refrigerant charge. but it is still a compact unit with hard maintenace and full replacement if damaged.

2.4 Refrigeration Systems

This type of arrangement is composed of a compressor, a condenser, an evaporator, and a refrigerator. It makes use of a working substance refrigerant, typically a gas that circulates through the system and is exposed to several phase changes through the process of compression and expansion. In the cycle, It goes from a low-pressure gas to a high-pressure liquid, the final stage consists of an evaporator, where the cooling of the system takes place.

Figure 2.4.1 shows a typical cycle in a refrigeration system, the main four stages are labeled with numbers.

The unit of the refrigeration effect is Watt or Kilowatt, but the standard unit of refriger-

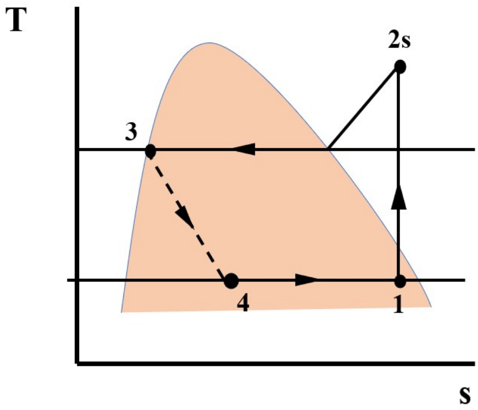


Figure 2.4.1: Temperature-Entropy diagram for vapor-compression refrigeration cycle.(Moran et al., 2014)

ation is a ton of refrigeration, meaning the amount of heat to be removed from a mass of fluid to go from one temperature to another in a fraction of time. and is usually denoted in Btu/h, Btu/min, Kj/min or kW

2.4.1 Compressor

In stage 1, The compressor compresses the refrigerant gas and increases the pressure and temperature. The mass and energy balance for a control volume enclosing the compressor, where Wc corresponds to the rate of power input per unit mass of refrigerant flowing (Moran et al., 2014):

$$\frac{\dot{W}_c}{\dot{m}} = h_2 - h_1 \tag{2.21}$$

The compressor is one of the main components in a refrigeration system, and it is also one of the most energy-consuming equipment. Therefore many studies on developing new and more energy-efficient technologies. The compressor should resist dynamic load changes and high pressures. Types of compressors can include:

- Piston
- Rotary Rolling Piston
- Rotary Van
- Rotary scroll
- Semi-hermetinc

The type of compressor will affect the choice of the refrigerant, depending of the range of pressure that this equipment is handeling.

2.4.2 Condenser

At stage 2 the condenser releases heat to the surrounding air or water and condenses in a liquid. The heat transfer occurs from the refrigerant to the cooling surroundings, This rate of heat transfer can be obtained by:

$$\frac{\dot{Q}_{out}}{\dot{m}} = h_2 - h_3 \tag{2.22}$$

2.4.3 Expansion Valve

In the next stage, The liquid flows through an expansion valve and reduces pressure and temperature Selecting the appropriate expansion valve for a refrigeration system depends on several factors, including the type of refrigerant, the cooling load, and the specific requirements of the system.

The refrigerants enter the expansion valve, it expands until it reaches the pressure of the evaporator. It exits the valve as a two-phase liquid vapor mixture (Moran et al., 2014). This process is commonly denominated as throttling.

2.4.4 Evaporator

This is also a key part of refrigeration systems, this is the system responsible of removing the heat of the space, it is a heat exchanger technology.

The flow goes through an evaporator, absorbing the heat from the interior of the refrigerator and turning it into a low-pressure gas. This evaporator is made of a series of tubes and coils that provide sufficient area for heat transfer to occur.

When the refrigerant passes the evaporator, It is possible to calculate the rate of heat transfer per unit mass of refrigerant whit equation 2.23

$$\frac{\dot{Q}_{in}}{\dot{m}} = h_1 - h_4 \tag{2.23}$$

This rate of heat transfer is commonly denoted as refrigeration capacity (Moran et al., 2014) usually expressed in kW or a ton of refrigeration

2.4.5 Refrigerants

Refrigerants are one of the most important features of a cooling system. This fluid absorbs the heat and cools the desired medium. Today, refrigerants are classified into three groups: saturated, unsaturated, and natural working fluids. Many refrigerants in the first group are in decreasing use due to their high environmental impact, toxic products, and flammable properties, which make them unwanted and banned in regulations. Therefore, natural working fluids are preferred as refrigerants. (Cengel & Boles, 2018)

Refrigerant 134a is considered to be environmentally better to replace other type of CFC refrigerants like R-22. Other refrigerants as Some unsaturated refrigerants such as hydrofluorolefins (HFOs) R-1234, R1234e, and R-513A have been used for similar applications. Ammonia (NH3) has been used in many vapor compression refrigeration systems it is receiving interest again, to replace CFC refrigerants. Hydrocarbons such as propane and methane and Co2 are also being investigated to be used as refrigerants.

An important property used for heat transfer calculations in refrigeration systems is the pressure - enthalpy. The selection of a refrigerant can be based on the suitability of its pressure and temperature relationship in the range of the particular application. (Moran et al., 2014) The type of material is also influencing the selection of the refrigerant. For example, materials like copper, zinc, and their alloys ‘cannot be used with R-717 due to corrosion.

CHAPTER

THREE

SYSTEM DESCRIPTION

To conduct accurate feasibility calculations for the proposed technologies in this study, it is crucial to establish relevant study cases. Pre-commissioning is a critical activity for subsea projects, each with unique requirements tailored to the field. In this section, an overview of a typical system based on a previous project from the company is presented and used as a baseline to establish the basic parameters and requirements for the heat transfer calculations made for each of the studied technologies.

3.1 Pre-commissioning Operations

Pre-commissioning operations ensure the preparedness of newly installed or modified subsea facilities for commissioning. It takes place in every project related to subsea assets after their installation and involves a series of tests and procedures which include flooding, cleaning, gauging, leak testing, pressuring, and dewatering of the system.

Subsea field developments in the Norwegian Sea are complex systems containing a number of pipelines, valves, and equipment installed in the seabed. When development plans for the field take place, it is usual that they require tie-ins into the existing subsea facilities, along whit additional configurations such as chemical injections, hydrate inhibitors, and electrical power distribution systems. The pipelines and spool pieces installed in this project can be manufactured in different diameters and sizes. This design will be provided by the client company, based on the expected production of the subsea wells and the development plan of the field, following API 17B standards.

The different activities in precommissioning scope include:

Pig loading and recovery: A Pipeline Inspection Gauge (Pig) is a device used in pipelines to perform different functions, such as calibrating, cleaning, flooding the lines or dewatering. They are handled with a pig launcher and a pig receiver. The pig loading is performed at the mobilization site prior to vessel mobilization.

Flooding, cleaning, and gauging: After installation of the pipeline, it shall be cleaned and gauged. The purpose of the operation is to flood the line in preparation for a tie-in and hydro testing as well as to sweep any dirt ferrous debris left in the pipeline product of fabrication. The gauge plate will confirm that no major obstructions or ovalities exist in the line after installation. These operations must be performed with chemically treated fresh water, making sure that no air is introduced to the system. Velocities of the pig during the operation must be maintained between 0.5 m/s to 1 m/s

Pressure / Leak test: It is done to different equipment, usually to production/injection lines and jumpers, immediately after tie-in operations. The system is pressurized through a high-pressure cap. Pressure test as a base case, is performed in combination with DNVGL-ST-F101 and BS ISO 13628-11:2007, usually with a hold period of 24 hours. Figure 3.1.1 illustrates a typical vessel arrangement to perform a pressure test in the flow line and jumper system. In this example, a downline hose is connected to the PLR at the PLET at well 6 and the system then is pressurized. It is important to note that when the pipelines are installed, they are air filed, so when fluids are pumped from the vessel, there are high pressures involved in the pumping systems that can go from 30 bar to 90 bar.

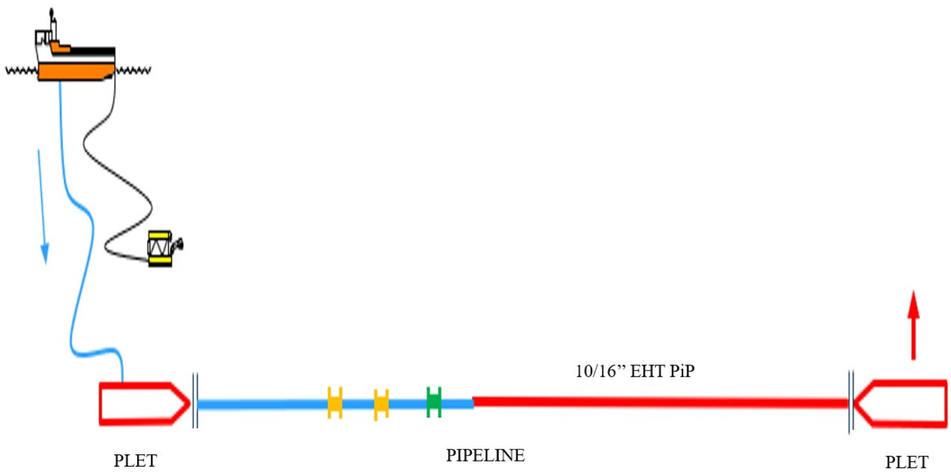


Figure 3.1.1: Typical operation in a Pressure Test of a production System

During the hold period, the pressure test will be monitored from the vessel as a base case. However, if there is a temperature difference between the flooded line and the surrounding environment, the process of temperature stabilization can have an impact on the pressure test. For pipelines with a low U-value, this stabilization process can take up to several days. This means that in worst-case scenarios, the vessel may have to return to shore and then come back again, resulting in significant costs for the operation.

Dewatering and N2 Packing: After the pressure testing of the flow lines and jumpers, the flow line must be dewatered. It is usually done from the PLET by the displacement of dewatering pigs with a MEG batch between them (concentration of MEG to be determined by the client). If it is required in the project, nitrogen will be injected into the lines. A valve arrange solution must be in place to prevent accidental flooding of the pipeline during dewatering operations.

Flushing during tie-in: If a tie-in is part of the scope of the project, the lines or jumpers must be flushed with MEG through the open end at the PLET, to remove any seawater that has come into the system.

3.2 System Components Specifications

DNV-OF-F101 Contains the required parameters for designing of production flow lines that include Nominal size, Pipeline length, design pressure, material, Inner diameter, Outer Diameter

The project used as a reference case for this study includes flexible jumpers of 6" and 10" diameter and a 10" to 16" EHTF Pipe in pipe production flow line. The flow line jumpers are connecting the TIM and PLET to the manifold valves. An in Line Tee is in place to facilitate the tie-in of the new system.

Table 3.2.1: Design Specifications of 6" flexible Jumpers (Subsea7)

Flexible jumpers 6”			
	Well A	**Well B**	**Well C**
Nominal size [inch]	6	6	6
Jumper length, total [m]	122	122	99
Design pressure @30m	330	330	330
Content Condensate	Condensate	Condensate	Condensate
Carcass material	Duplex 2205	Duplex 2205	Duplex 2205
Water depth. Min-Max [m]	409 - 410	406 - 407	419 - 420
Outer Diameter [mm]	426.3	426.3	426.3
Inner Diameter [mm]	152.4	152.4	152.4

The design specifications of the flow line and the jumpers are noted in Table 3.2.1, for the 6 inch flexible jumpers that connect the wells to the PLET and the ILT,

Table 3.2.2 for the specifications of the 10 inch jumpers used to connect the TIM to the facilities and Figure 3.2.3 for the design specifications of the principal flow line. The material used for the fabrication of the jumpers is Duplex 2205, this is a type of stainless steel with desirable properties for subsea environments due to its high stress and corrosion resistance. As for the thermal properties, this material has a thermal conductivity on 13-20 W/m-K, lower than many other materials in the market.

Table 3.2.3 contains the specification of the 10 in electrically heat traced flowline (EHTF). This type of flolines are used to mitigate the formation of hydrates and wax when producing hydrocarbons. Which minimizes the production downtime. This lines tend to have very low U values, making difficult the heat loss of the fuid contained in the pipeline.

Table 3.2.4 resumes the dimensions of the components in the subsea system that will be pressure tested. The total volume of water for all the components rounds the 900m^3.

Table 3.2.2: Design Specifications of 10" flexible Jumpers (Subsea7)

Flexible jumpers 10”			
	PLET @ TIM	**Facility 1**	**Facility 2**
Nominal size [inch]	10	10	10
Jumper length, total [m]	137	155	174
Design pressure @30m	330	330	330
Content Condensate	Condensate	Condensate	Condensate
Carcass material	Duplex 2205	Duplex 2205	Duplex 2205
Water depth. Min-Max [m]	326 - 335	323 - 335	324 - 335
Outer Diameter [mm]	499	499	499
Inner Diameter [mm]	254	254	254

Table 3.2.3: Specifications of pipe in pipe production flowline

10” Production Flowline	
Nominal size [inch]	10
Pipeline length, total [m]	20 245
Design pressure @30m above MSL [bara]	330
Content	Condensate
Material	DNV MWP 450 C UNS31603
Water depth. Min-Max [m]	320 - 430
Outer Diameter [mm]	273.1
Inner Diameter [mm]	235.31

Table 3.2.4: System details of the production System

Item	**Size (in)**	**ID (mm)**	**Lenght (m)**	**Volume (m3)**
EHTF Flowline	10 to 16	235.6	20245	880.3
TIM	12	254	20	1
Jumper PLET – Well C	6	152.4	99	1.8
Jumper ILT –Well B	6	152.4	122	2.2
Jumper PLET - Well A	6	152.2	122	2.2
Jumper PLET - TIM	10	254	137	6.9
Jumper TIM – Facility 1	10	254	155	7.9
Jumper TIM – Facility 2	10	254	174	8.8

This volume will be used as a reference to construct study cases in this section, as the total volume and the time available will determine the flow rate in each of them.

3.3 Vessel Arrangements and Available Equipment

The vessel used for the reference project is a light construction subsea vessel, reaching depths of 3000m. The main features are resumed in 3.3.1. The solutions presented in this study will mainly be limited due to the vessel dimensions and capacity to host the equipment on board.

Table 3.3.1: Vessel characteristics

Vessel Characteristics		
Base Facts	Length overall	98.1 m
	Breadth	21.5 m
	Deck Space	875 m2
	Deadweight	4200 ton
	Accommodation	85 persons
Capacities	Fresh water	1000 m3
	MEG tanks	800 m3
	Ballast Water	2900 m3
Cranage	Main lifting facility	150 Te SWL @ 15m
	Additional lifting facility	3 Te SWL @ 15m

The rented equipment can vary from project to project, but it is usual to have already one or two spread solutions in the vessel for pumping the necessary fluids in the pipelines. The spread solutions or pumping arrangements usually include:

Pressure test Pump DHDA33: This pump has a maximum flow rate of 31 L/min and a maximum pressure of 1800 bar, with an integrated stroke counter. It can be used with water, hydraulic, and some type of chemicals. The vessel has 2 pumps installed on the deck.

PP-60: Pump used for pigging has 3 preset flow rates: 47 L/min @ 665 bar (60Hz), 35 L/min @ 880 bar (60Hz) and 25 L/min @ 1180 bar (60Hz)

PP-104: Pump used for pigging and chemical cleaning, with a maximum pressure of 9 bars and 3 different flow rates of 130 L/min @ 8,6 bar (60Hz), 600 L/min @ 8 bar (60Hz) and 1050 L/min @ 6,5 bar (60Hz) and the inlet connection of 4" and outlet connection of 2"

3.4 Met-ocean Conditions

The met ocean conditions of this project must be considered as they will affect the heat transfer calculations. As this development was lacking of a direct archive of measured met-ocean parameters, the close location to other developments allowed to use sources from other databases to perform hind-cast models. The project is located in the Norwegian sea, therefore subarctic conditions are considered. Measurements made by Statoil will be used to characterize the sea environment. The majority of the data presented was measured during a meter survey conducted during the period of September 1993 to June 1994 (Petroleum, 2009)

3.4.1 Sea Water Temperatures

Data from the meter survey was complemented with data from two global databases. one from ICES and one from the global ocean hydro graphic data center in US.

Table 3.4.1: Extreme maximum and minimum sea temperatures at location

Depth	Maximum °C	Minimum °C
0	18	4
10	16.5	4.2
20	14.7	4.4
30	14	4.5
40	12.5	4.6
50	11	4.7
60	10.6	4.8
70	10.3	4.9
80	10	5

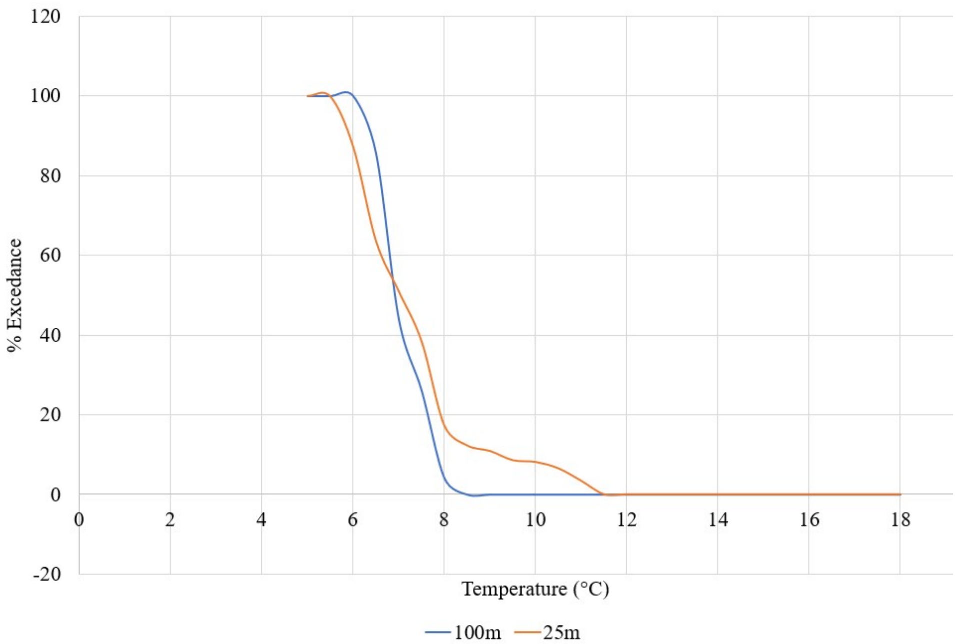


Figure 3.4.1: Percentage Exceedence of Sea Temperatures in the Area

Figure 3.4.1 presents the exceedance of sea temperatures for the upper 100m of the water column. For near-bed sea temperatures, very little data is available, data to a depth of 500m indicates that minimum temperatures have reached 2.5°C. The minimum value at 300m was 5.4°C and at 400m was 4.2°C. It is suggested that a minimum value of +3°C is used for 350m.

As for air temperatures, Extreme temperatures for the 100-year return period level are expected -13°C to the minimum and +23°C Maximum

3.4.2 Currents

Currents were measured over a period of 9 months at depths of 25m, 100m, and 200m below the surface. These measurements were performed near the development and it is suggested that they are representative of the project in question.

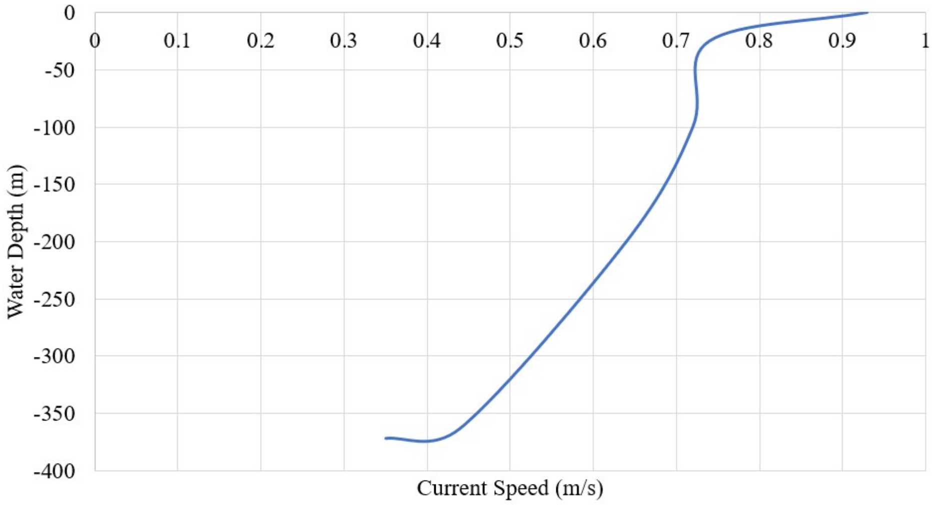


Figure 3.4.2: Total current speed estimates with depth for a range for 100 years return period.

Table 3.4.2: Percentage Exceedence of Sea Temperatures in the Area

Return Period Years	Surface	25m	100m	200m	363m	372m
1	0.7	0.56	0.52	0.49	0.37	0.29
10	0.81	0.65	0.62	0.57	0.41	0.32
50	0.89	0.71	0.69	0.61	0.43	0.34
100	0.93	0.74	0.72	0.64	0.44	0.35

Values presented in Table 3.4.2 can be used as a reference to estimate the external convection coefficients for heat transfer calculations.

3.5 Study Cases

Based on the compiled information about the project, three possible scenarios were constructed in which different solutions can be studied. These cases vary primarily based on the location of the solution and the flow rates that can be accommodated. The placement of the equipment within the system will determine the range of pressures to which it will be subjected. Additionally, the flow rates will significantly influence the sizing or dimensioning of the solution, as they are expected to be key factors in the heat exchange calculations.

To determine the required flow rates, the Equation 3.1 is used, taking into account the internal cross-section area of the pipe. For simplicity, a 10-inch pipeline was chosen as a reference, assuming a target pig velocity of 1 m/s.

$$q = v_{pig} \pi r_i^2 \tag{3.1}$$

Table 3.5.1: Study cases for solutions

Case	ID	Flowrate m3/h	Mass Rate (kg/s)	P inlet (bar)	Tin	Tout	Fluid
1	10	156	43.48	40	13	4	Freshwater
2	10	156	43.48	5	13	4	Freshwater
3	10	43	10	5	13	4	Freshwater

Case 1: The Solution is placed after the pump outlet and it will be connected with the down line hose (Usually 4 in). At this point, the requirements state that there is a velocity of 1m/s to be maintained. Which results in high flow rates in this case, also the pressure requirements could be from 30 to 40 bars at this point.

Case 2: The solution is placed before the pump inlet. In this case, there will be no pressure restrictions and the low-pressure range can be managed, but still is desired to maintain the flow rate of case 1.

Case 3: The heat Solution is placed before the pump inlet. But there is more time available for cooling, In this case, it is desired to study a scenario where the cooling can happen in 24hrs for a volume of 900m^3.

In the 3 cases, the arrangement of the spread or pumping configuration will change. It is important to consider the installation of a subsea pump if it is desired to use seawater as a cooling fluid. It is assumed that this pump is capable of producing the same flow rates as the ones stated in the study cases above.

The main challenges to discuss for each of the proposed solutions include performance/efficiency at high operation pressure, cost-effectiveness, low-pressure drop, reliability, durability, safety and compactness (space-saving).

CHAPTER

FOUR

ALTERNATIVES FOR COOLING WATER

4.1 Subsea Spool Piece With Low Insulation

Spools and Jumpers are essential parts of a subsea production system. They are used for connecting the main components in the system, such as manifolds, wellheads, or riser bases. Subsea Spool pieces have various applications regarding thermal management, like regulating high temperatures in HPHT wells systems.

The proposed solution aims to have a temporal installation of a spool piece near the seabed that connects the hose to the pipeline inlet. The configuration is illustrated in Figure 4.1.1. The material of this piece should allow the most efficient heat transfer rate between the internal fluid and the surroundings as it is flowing. Calculations in this section will frame the design envelope in terms of heat transfer capacity and therefore, the required length to achieve a desired outlet temperature. Sensitivities using different materials, diameters, and thicknesses of the spool piece will be performed. Finally, a discussion of results and operational challenges will be presented.

4.1.1 Materials

The selection of the Spool piece material is of utmost importance when evaluating the feasibility of the cooling method. This choice significantly impacts both the cost and the heat transfer rate between the internal fluid and the surroundings. Compliance with various industry standards, such as DNV-OS-F101, API 17R, 17TR8, ASME B31.4, and B31.8, is crucial for designing reliable subsea spool pieces. Despite being a temporary installation, the Spool piece must withstand the loads generated at high depths and address challenges related to corrosion.

Regarding material selection, Carbon steel is a commonly utilized material for subsea pipeline fabrication, boasting a thermal conductivity of approximately 50 W/m-K. However, copper exhibits a significantly higher thermal conductivity (around 400 W/m-K), although it is less corrosion-resistant. Another material option worth considering is aluminum, which possesses a thermal conductivity of approximately 250 W/m-K. Similar

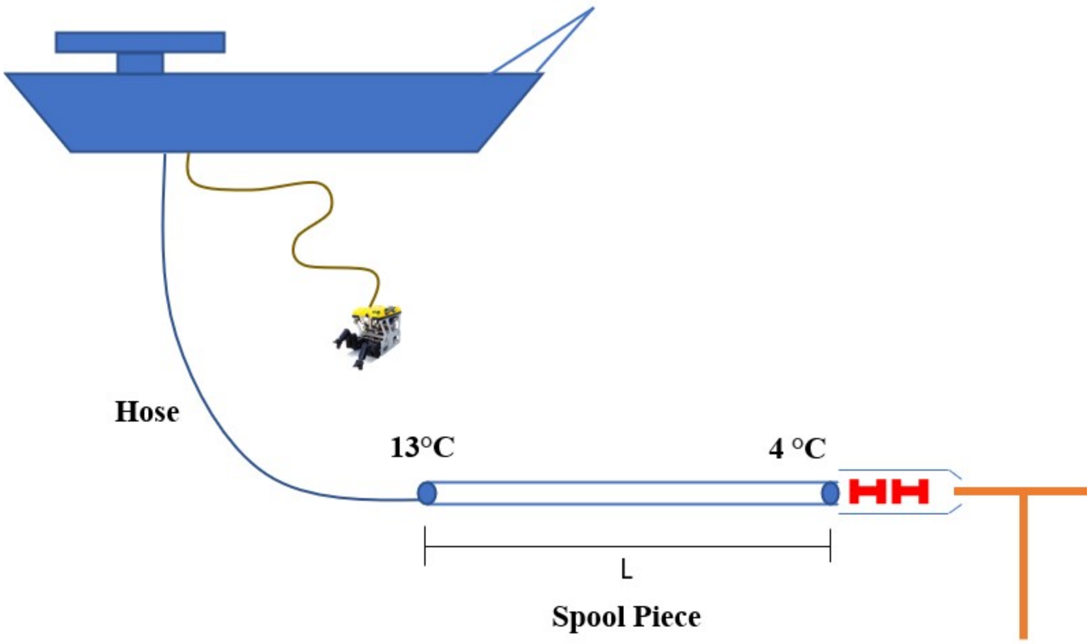


Figure 4.1.1: General Configuration of the Spool Piece Alternative

to copper, aluminum is pricier than carbon steel, but it may be a more suitable choice if high thermal conductivity is a priority.

The material selection for the Spool piece carries substantial implications since the fluid passing through it will subsequently flow into the production or injection line that is intended to be pressure tested, where it must meet stringent quality requirements to operate optimally. Incompatibility between the materials used in the Spool piece and those employed in the downstream line can lead to undesirable conditions such as premature pitting corrosion, posing operational challenges. Therefore, a meticulous selection of compatible materials is essential to ensure the long-term integrity and reliability of the pipeline system that will be delivered to the client.

4.1.2 Heat Transfer Modelling

A thermal analysis will be conducted in order to predict the temperature profile along the pipeline and determine the necessary pipeline length to achieve thermal equilibrium with the surrounding temperature. To facilitate this analysis, a Python code was developed following the outlined process in Figure 2.1.3. The initial base case is modeled using parameters from Table 4.1.1.

The first step involves calculating the Reynolds and Prandtl numbers to determine the nature of the flow. Based on these numbers, different correlations can be employed to calculate the Nusselt number for each flow regime. Subsequently, Equation 2.9 is utilized to estimate the internal convection coefficient, which is a key factor in the heat exchange calculations. Additionally, the external convection coefficient must be determined in order to calculate the overall heat transfer coefficient. The necessary heat to be removed from the system, along with the heat transfer area, is then utilized in Equation (reference) to calculate the required pipeline length to achieve the desired temperature.

An initial simulation was conducted using the design parameters of the spool as outlined in Table 4.1.1. These parameters were derived from the pressure testing case on the 10" flow-line of the reference subsea development discussed in Chapter 3.

The required flow rate was estimated using a volume of the system of 900m3 and considering that it is necessary to maintain a certain flow rate in the pipeline to achieve the desired target pig velocity of 0.5 - 1 m/s. To facilitate this, the first case assumes the use of stainless steel as the material, which possesses well-defined thermal properties suitable for modeling the initial scenario.

Table 4.1.1: Parameters used for modeling

Spool Design Parameters	**Value**
OD	10 in
t	25 mm
Material	Duplex 2205
Thermal Conductivity K_p	19 W/mK
Internal Fluid Properties	
Fluid	Fresh Water
Thermal conductivity Kf_fw	0.6 W/mK
Specific Heat Capacity Cp_fw	4186 J/kgK
Density (ρ_fw)	1000 kg/m^3
Velocity Vf_fw	1 m/s
Flow rate	0.032 m^3/s
Mass rate	32.68 kg/s
Heat Transfer	
Inlet Temperature	13 °C
Seawater Temperature	4 °C

The model results of the based case are presented in Figure 4.1.2 This graph was constructed running several cases of the model with different lengths, in order to predict the output temperatures. It is important to note that extensive academic research has been performed to predict external convection coefficients around cylinders, as this prediction is not part of the scope of the project, sensitivities were run with different assumptions of this coefficient to account for their impact in the heat transfer

The fluid is cooling as it flows along the pipeline and the temperature approaches the sea temperature asymptotically. The main factors driving this change are the high specific heat capacity of the water and the and actual time to cool down could be different due to factors such as turbulence evaluation of energy consumption and cooling capacity relative to flow rate and volume.

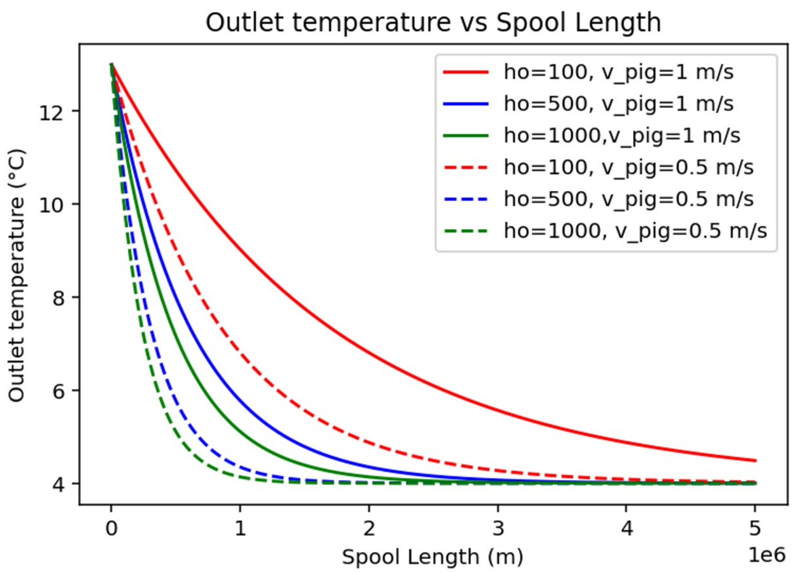


Figure 4.1.2: Outlet temperature vs Spool Length base case

4.1.3 Discussion

This alternative appears to be of interest, particularly due to the company's know-how in spool fabrication and installation, as well as the no need for additional cooling fluids for the heat transfer, nor additional renting costs for extra equipment. The main cost will imply the fabrication of the spool. However, the results demonstrate that the required length to achieve the desired cooling is unrealistic, from the base case it is understood the need of 1e+06 m of pipeline to achieve such a low temperature. Although this value can be reduced by modifying several factors, being the flow rate one of the most important, none of the simulated circumstances make it feasible to achieve a spool piece shorter than 80m, which represents the vessel's overall length.

The implementation of this particular solution would require additional operations in the scope of work, including heavy lifting operations to temporarily install and remove the spool piece after the campaign is completed. Consequently, this work would extend the overall operational time, potentially extending it to several days if met conditions do not allow the heavy lifting and lowering of the spool piece in the splash zone. This will also include additional working time for the remotely operated vehicle (ROV) in the overall cost.

While this alternative may be viable for producing small temperature changes, it falls short in terms of achieving the required temperature differential with the required flow rate. Decreasing the flow rate is unrealistic, due to operational requirements that suggest maintaining a certain pig velocity. In terms of energy consumption, this alternative does not require additional power beyond what is already utilized by the original pumping spread.

It is evident from the heat transfer results, that as the water cools down and approaches the temperature of the cooling medium, the temperature difference decreases, and the rate of heat transfer slows down. Making it especially difficult to achieve the cooling of such a large volume and flow-rates using seawater as a natural cooling medium. Therefore, the study of different technologies such as heat exchangers or refrigeration systems is conducted in the next sections.

4.2 Heat Exchanger in Vessel

This alternative beholds the installation of a single heat exchanger equipment in the vessel deck. The dimension and type of equipment will depend on the operating parameters, fluids involved and flow conditions. Therefore, For each of the study cases proposed in chapter 3, heat transfer calculations will be performed. For the cooling fluid used in the heat exchanger, there are two different possibilities. One is to use seawater from the sea bottom, which will require the use of a submersible pump that brings cold water from a certain depth. Another possibility is to make use of a refrigerant fluid as a cooling medium or a mixture of fluids. This fluid will be loaded onshore and will eliminate the need for the subsea pump. The 3 different configurations using seawater as a cooling medium are illustrated in Figures 4.2.1, 4.2.5 and 4.2.6 and as stated in the system description, the pressure ranges that will be used for dimension will depend on the location of the heat exchanger in the system.

4.2.1 Heat Transfer Calculations

For the design of the heat exchanger, plate heat exchangers and tube-and-shell heat exchangers are of interest due to their availability, efficiency, and big working ranges.

A spreadsheet was prepared to perform heat transfer calculations and providers were contacted in order to dimension the heat exchangers including materials, design, and plate thickness parameters. The inlet temperatures of the freshwater and the coolant fluid will be assumed. It is necessary to calculate the outlet temperature of the cooling fluid by using Equation 2.13. Internal and external convection coefficients are calculated internally in the software selecting the configuration type of hat exchanger and material, as well as giving the flowrate as an input. The Energy required for the cooling is then calculated with Equation 2.13. It is important to calculate the temperatures for the overall heat transfer coefficient using the LMTD method. Therefore Equation 2.14 is used and then Equation 2.11 will account for the total thermoresistances of the system. With all of these parameters, the provider sizing software recommends the existing solution in their portfolio.

The program will be tested through simulations of different use cases. This includes changing the temperature program the cooling fluid and its composition of it as altering the flow rates and pressures.

Case 1: As mentioned, this case will involve a high-pressure system since the heat exchanger will be the last piece of equipment connected to the down-line hose before the fluid travels to the pipeline. Figure 4.2.1 illustrates a simple overview of the spread system with pressures and flow rates, the system must be able to carry the necessary pressure to overcome the pre-aired filled pipeline pressure and to enter the fluids into the system. The hot fluid is assumed to be freshwater and two different cold fluids will be analyzed, Saltwater and MEG. The resume of the properties used and results on the dimension of the heat exchanger are presented in Table 4.2.1 Assumptions on the flow rate of the cold fluid (Seawater) must be made since the dimension of the pump isn't part

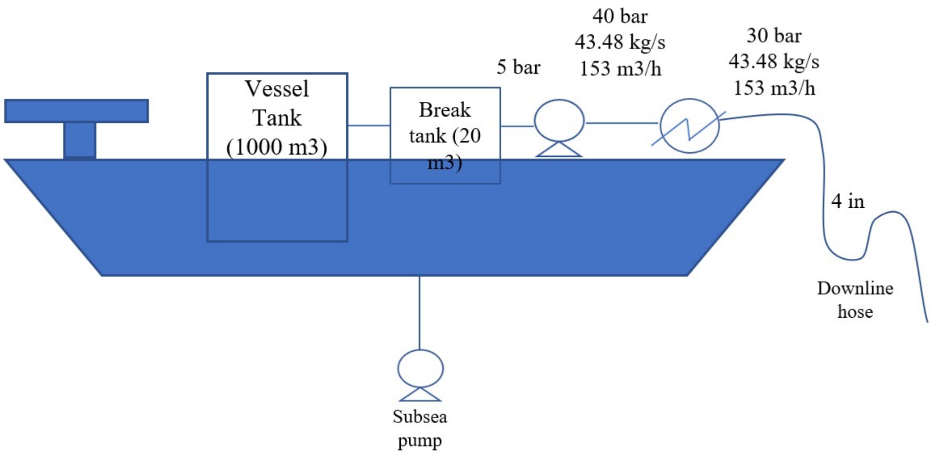


Figure 4.2.1: Simplified Spread Configuration in the Vessel for Case 1 (High Pressure)

of the scope of the study. For initial calculations, the same flow rate as the cold fluid is assumed, and two different cases with MEG at different temperatures were studied.

Table 4.2.1: Dimensions of Heat Exchanger for Case 1 using Saltwater and MEG 30% using two different temperatures as cold fluids.

Cooling Fluid	Seawater	MEG1	MEG2
Cold fluid inlet T	4	-3	1
Hot fluid outlet T	5	5	6
Cold fluid flowrate (kg/s)	43.5	43.5	43.5
LMTD (K)	0.7	6.3	4.4
Design Pressure (bar)	27	40	40
Number of plates	256	268	286
Number of passes	2	1	1
Plate Material	Titanium	Alloy 316L	Alloy 316L
Plate Thickness (mm)	0.8	0.7	0.7
Operating weight (Kg)	10200	1020	1020
Volume (m3)	15.3	1.4	1.4
Dimensions LxWxH	3350x1500x3035	1810x570x1340	1810x570x1340

For this condition, the selection of available heat exchangers is limited due to the pressure and type of fluid requirements. The use of seawater in the system creates the necessity of using corrosion-resistant materials like titanium, which can increase substantially the cost, the type of heat exchangers suitable for handling this range of pressure has to be welded which makes maintenance challenging. Additionally, Figure 4.2.2 shows the temperature program for both fluids in each case. There is a significant limitation on the achievable final temperature with a flow rate and seawater temperature of 4 degrees Celsius. To reach the desired freshwater temperature, a seawater temperature as low as 2 Celsius

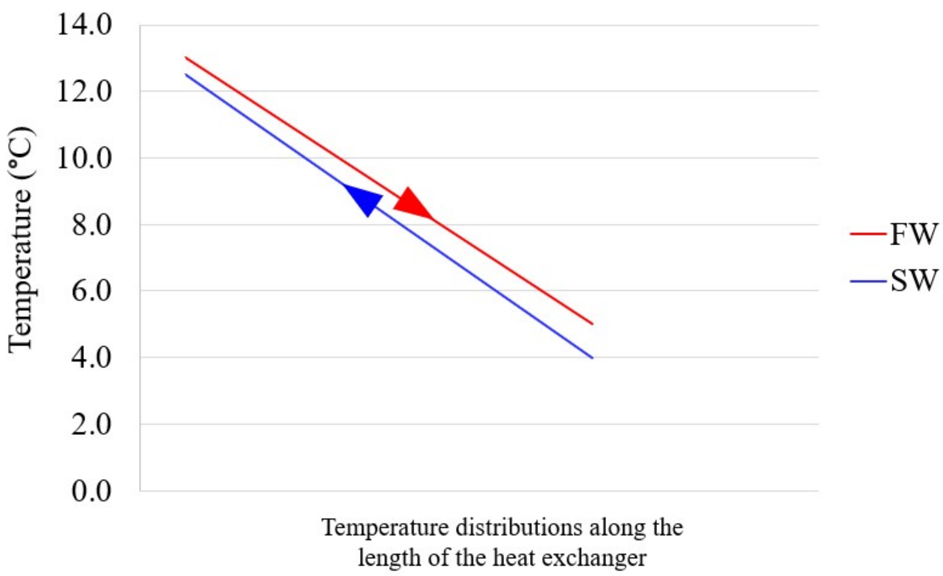


Figure 4.2.2: Temperature distributions in the Heat Exchanger using Seawater as a Cold Fluid

would be required, which is unattainable. 4.2.3 shows the simulations made with by using a 30% MEG (Monoethylene glycol) solution as the cooling medium. Incorporating a cold inlet fluid at -3 °C will allow to reach the target, but with MEG at +1 °C a final temperature of 6 degrees of the freshwater can be reached. In both cases, the proposed heat exchanger solution remains the same and for this option the cold fluid should be sourced from onshore.

Figure 4.2.4 illustrates the resulting temperature program when the flow rate of the cold fluid is decreased by 50% compared to the flow rate of the hot fluid. The logarithmic mean temperature difference (LMTD) starts to decrease at lower flow rates. Eventually, it reaches a point where the two lines intersect, and the desired temperature program becomes unachievable. However, in the case depicted in this Figure, it is still possible to reach a temperature of 5 degrees. To achieve this temperature, the required temperature of the MEG mixture must be -10°C.

Case 2: This case places the heat exchanger previous to the pumping system, therefore the unit will work under low pressure. The flow rate remains constant as in case 1, so the LMTD calculations will remain similar to Figure 4.2.2 and 4.2.3 , showing the limitations of using seawater as a cooling fluid.

The low pressure ranges result in thinner plates of the heat exchanger which results in smaller and lightweight unit. Titanium plates will still be required when working with seawater. Alloy 316L plates can be used for applications that don't involve seawater and use another cooling media instead. The biggest challenge in this application remains to be able to get seawater at this temperature but also to get the MEG at -3 °C and maintain the same flow rate as the hot fluid inlet. The main advantage of this application will be the require type of heat exchanger, since is not welded it will be less expensive it will allow for easy maintenance.

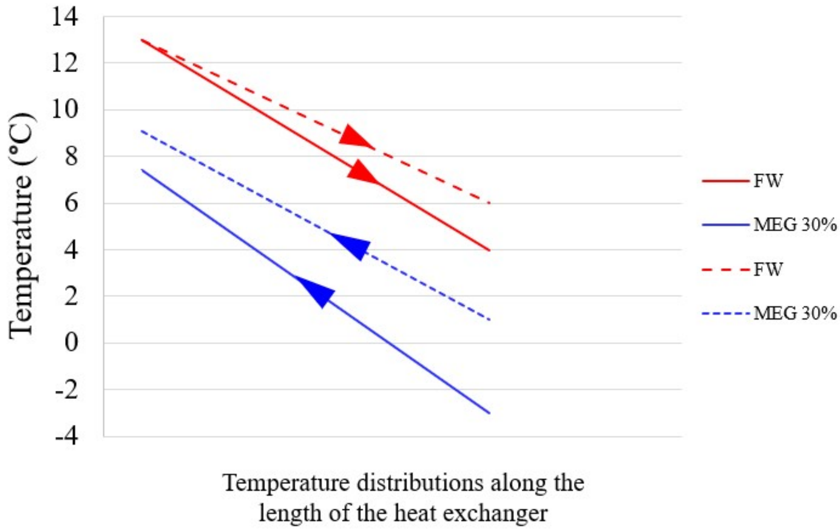


Figure 4.2.3: Temperature distributions in the Heat Exchanger using a 30% Glycol mixture as a Cold Fluid. Two cases using different temperatures in the inlet of the cold fluid.

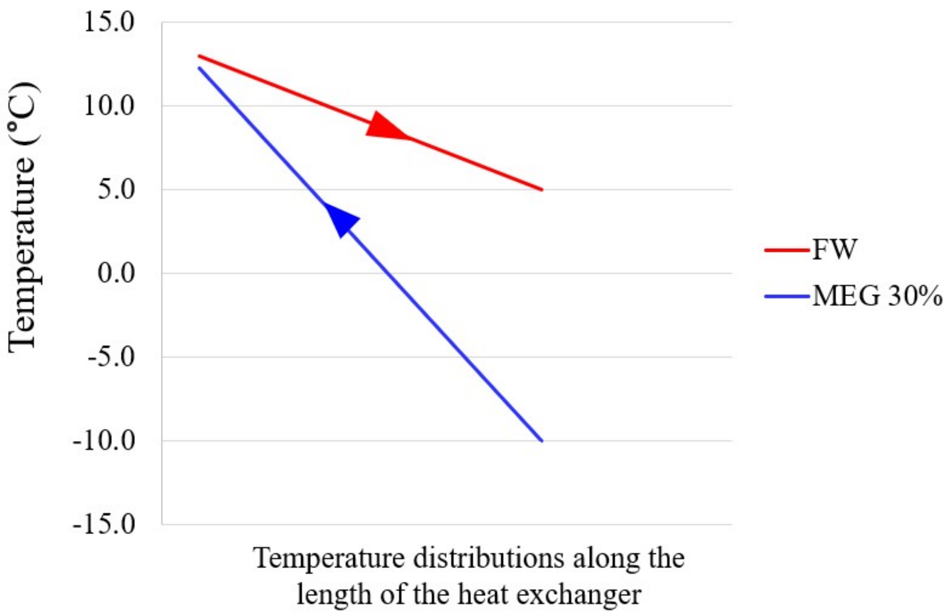


Figure 4.2.4: Temperature distributions in the Heat Exchanger using a 30% Glycol mixture as a cold fluid and decreasing the flow rate by 50%

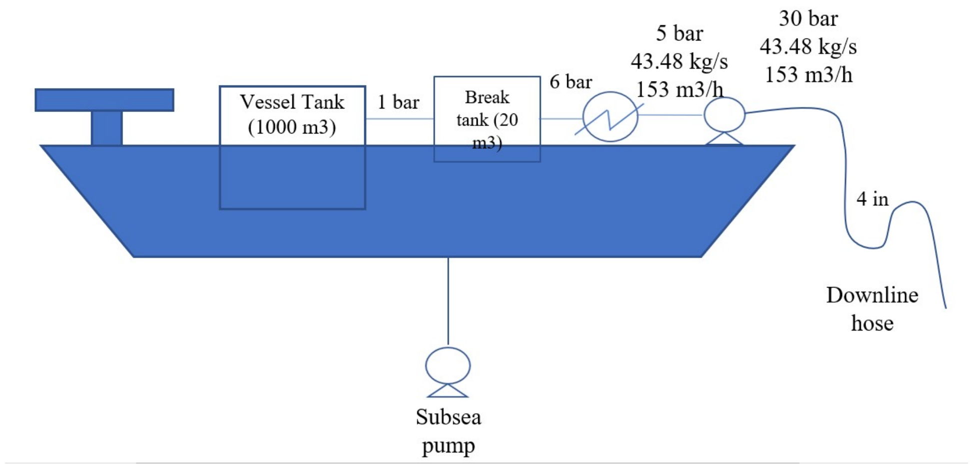


Figure 4.2.5: General Spread configuration for Case 2 (Low Pressure)

Table 4.2.2: Dimensions of Heat Exchanger for Case 2 using Saltwater and MEG 30% as cold fluids.

Cooling Fluid	Seawater	MEG
Cold fluid inlet T °C	4	-3
Hot fluid outlet T °C	5	4
Cold fluid flowrate	43.5	43.5
LMTD	0.7	6.3
Design Pressure (bar)	5	5
Number of plates	471	116
Number of passes	1	1
Plate Material	Titanium	Alloy 316L
Plate Thickness (mm)	0.5	0.4
Operating weight (Kg)	3850	980
Volume (m3)	9.3	2.2
Dimensions LxWxH	3390x900x3040	2100x635x1650

Case 3: The last case involves a lower pressure system as in case 2, but with a decreased flow rate, the cold freshwater produced and storage in a break tank before a pump takes the fluid to higher up pressure and transports it to the pipeline as shown in 4.2.6. Results on the dimension of a unit with these specifications are listed below

As the pressure remains the same as in case 2, the plate thickness is expected to be the same, but the flow rate will affect the number of plates necessary to provide the desired cooling of fluids. The dimensions and weight of the heat exchanger models are substantially reduced, making it very convenient to place them in any vessel or to fit in any spread arrangement.

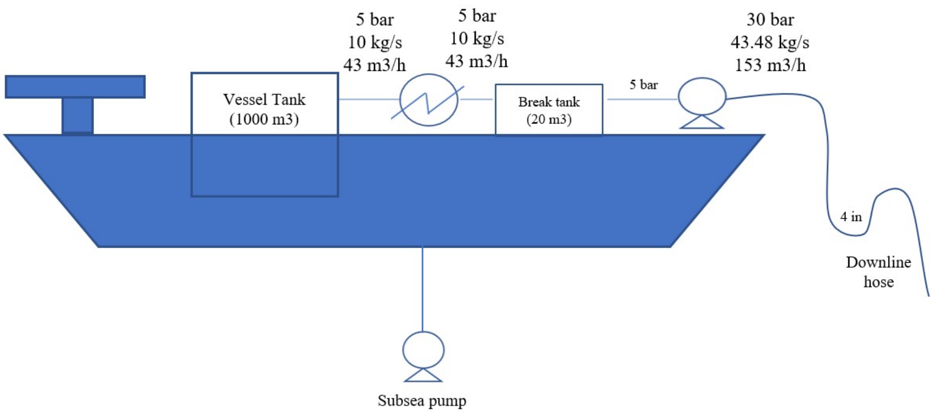


Figure 4.2.6: General Spread configuration for Case 2 (Low Pressure)

Table 4.2.3: Dimensions of Heat Exchanger for Case 3 using Saltwater and MEG 30% as cold fluids.

Cooling Fluid	Seawater	MEG
Cold fluid inlet T	4	1
Hot fluid outlet T	5	6
Cold fluid flowrate	10.4	10.4
LMTD	0.7	4.5
Design Pressure (bar)	5	5
Number of plates	177	97
Number of passes	2	1
Plate Material	Titanium	Alloy 316L
Plate Thickness (mm)	0.5	0.5
Operating weight (Kg)	1010	220
Volume (m3)	15.3	1.4
Dimensions LxWxH	2100x635x1650	890x320x140

4.2.2 Discussion

This is a desired system to install for cooling water, due to the extensive research, technology development, and amount of providers that offer heat exchangers in the market. Nevertheless, the heat transfer calculations performed indicate that using seawater pumped from near the surface is not enough to achieve the desired temperature within the acceptable time and flow rate for each of the studied cases. As for the MEG, additional studies are needed to determine if its truly possible to purchase such higher volumes of MEG at 1 or -3°C. As general conclusion regarding flow-rate requirements, it is possible to achieve one or two degrees above the target temperature maintaining the same flow rate as the hot fluid inlet, that is already substantially high, this is another limit ant for implementing this solution.

Working under high-pressure ranges will result in thicker plates, bigger and heavier units, and will affect the price. The dimensions of the heat exchanger are substantially reduced when working in low-pressure ranges which is desirable when accounting for the reduced available deck space in this type of project. As for the price, quotation of equipment available. This Making alternatives from cases 2 and 3 more attractive.

According to the simulations, if the requirements of flow rate and temperature of fluids are possible to reach, a sort of solution exists in the market and is possible to purchase. This includes the following heat exchanger models: T10-EW Alloy or TL25B for case 1, TL10P Alloy or TL15B TI for case 2, and T6-B Alloy or TL 10B TI for case 3.(Alfa Laval, 2023) A quotation of the equipment will be used in the economic evaluation of the system. Estimates for case 1 round the 1.9 mill NOK for heat exchangers with titanium materials. A reduction of 50% of this price could be expected when working with alloys and a reduction of 15 to 20% of the price of case 1 can be expected when working for heat exchangers that work in low pressure ranges.

The possibility of installation of the equipment onshore could be studied as a possibility in order to keep the system running multiple days ahead without incurring additional costs from the vessel, this will decrease the size of the equipment and the cost will. Nevertheless, for onshore solutions, tanks to store such a volume of water and cooling fluid can increase the cost of the project.

4.3 Water Chiller

The main working principle of a chiller is a typical refrigeration system. The chiller is designed to remove heat from a process, using a refrigerant that circulates through a compressor, condenser, expansion valve, and evaporator circuit as illustrated in Figure 4.3.1 An Industrial water chiller or glycol chiller is of interest due to their capacity of cooling large volumes of water. The concept is utilized in many industries, like breweries, food processing, medicine, and fishing. Where precise outlet water temperature is required.

Refrigeration is a common challenge for fishing vessels. While smaller vessels typically rely on ice for fish storage, larger fishing vessels utilize refrigerated seawater (RSW) systems. In this context, a review of the state-of-the-art of this technology is presented. RSW chiller units are specifically designed for marine applications and are capable of handling large amounts of water. These compact units effectively improve the preservation of catch and prevent the spread of diseases in large volumes.

In a chiller, the coolant is constantly circulating in a closed loop, and helps to extract excess heat from a process, dissipating it afterward in a refrigeration system or heat exchanger. The cooling capacity of a chiller is calculated in tons or watts and it indicates the amount of heat that could be removed per unit of time. Calculations on the sizing of the chiller will be performed in the following section, and according to the results, providers will be consulted and technical requirements will be evaluated.

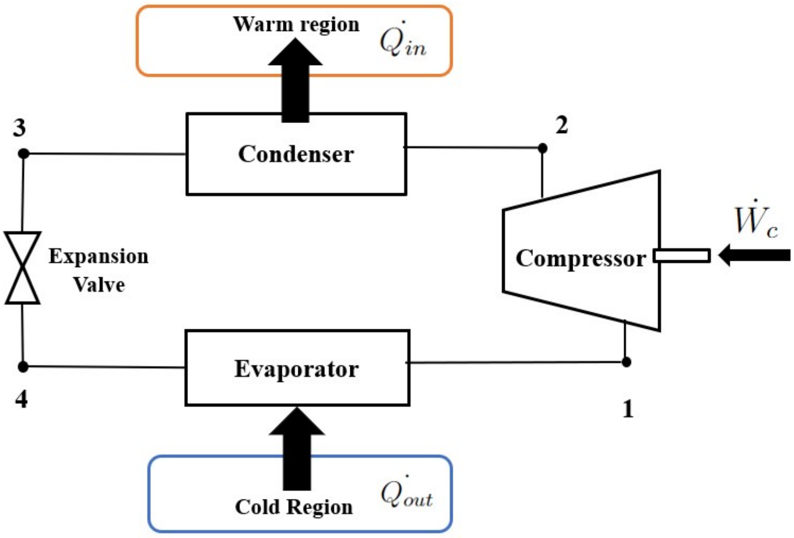


Figure 4.3.1: Schematic of a refrigeration system cycle. (Moran et al., 2014)

4.3.1 Sizing of the Chiller

The main parameter used to conduct a search of a chiller unit is the cooling capacity of the unit, which is mainly determined by the volume of fluid, the temperature program and the flow rate. As it has been proposed for this study, is important to study different scenarios where the available time for cooling the fluid varies. It is desirable for the fluids to be cooled at the same flow rate that the pressure fluid is being pumped in the pipeline, but other spread configurations allow to have more time for this specific activity. And as studied with other solutions proposed above, this will have a big impact on the type and size of equipment.

$$Capacity = \dot{m}Cp_{fluid}\Delta T \tag{4.1}$$

The cooling capacity of the unit is calculated using Equation 4.1 this value will be compared with the available equipment in the supplier's catalog. Currently, different companies in Norway dedicated to marine industry offer a variety of models and sizes of chillers, focused mainly on marine solutions for the fishing farming business. An oversize of 20% is usually considered for these applications.

Table 4.3.1 contains the results for the sizing of a chiller given the parameters of the reference project. It is noted that for lower flow rates, the size of the chiller decreases, which can have several impacts of the financial evaluation of the solution. It is ideal to size the chiller under worst-case scenarios, if it can work under those conditions, then it will work with any other parameters it may face, so initially, it is desired to verify if there is an available model for the highest flow-rate, which for the reference project will be 150 m^3/h. For this flow-rate, a chiller of 1884 kW is desired.

Calculations are based on freshwater without corrosion additives. The capacity of the chiller is increasing proportional to the available time for the cooling the fluid as illustrated in Figure 4.3.3. But it is notice that no big changes in the capacity occur when trying to

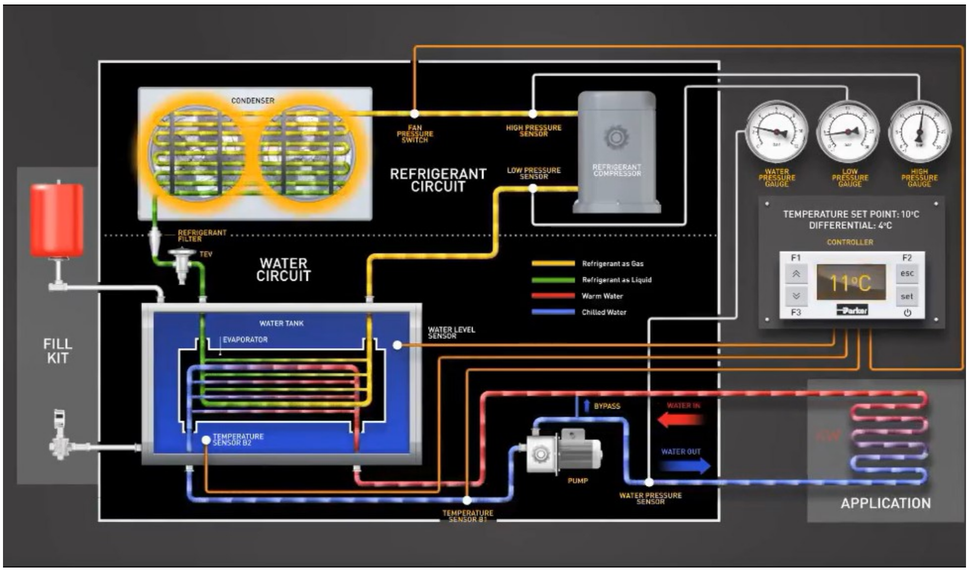


Figure 4.3.2: Animation of a Chiller System (Parker Hannifin Corporation, 2020)

Table 4.3.1: Chiller Sizing for pure Freshwater

Parameters					
Volume m^3	900				
T water in °C	13				
T water out °C	4				
Chiller Sizing					
Available days for cooling	3	2	1	0.5	0.25
Flowrate m^3/h	12.5	18.75	37.5	75	150
Massrate kg/s	3.5	5.2	10.4	20.8	41.7
Capacity kW	131	196	392	785	1570
Size kW	157	235	471	942	1884

cool different fluids. Hence, this type of equipment provides versatility for different kinds of projects, where the pressure test fluid is different in each.

One of the main advantages of working with chillers is the presence of a sensor to measure temperature data and connection to the expansion valve position to increase or decrease the flux of refrigerant in the system.

4.3.2 Available Equipment

The different providers that were contacted for quotations of this kind of equipment, offer a wide range of sizes in the chiller equipment from 50 to 2500kW capacity. Which falls in the range of the estimated capacity of the chiller in this application. Offshore conditions

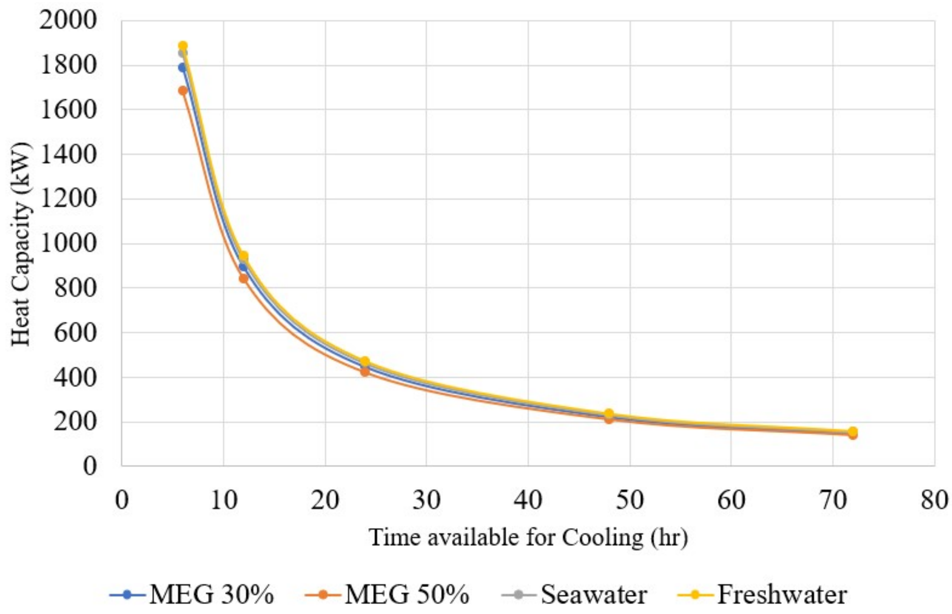


Figure 4.3.3: Chiller Cooling capacity vs available time for cooling for different fluids

of 3/400V/50Hz for power supply were assumed. Two different chiller configurations will be proposed as follows:

Twin Chiller: A Marine water chiller unit with twin compressors has a wide capacity range from 180 kW to 1376 kW. It has a compact type screw compressor with capacity regulation, dry expansion shell, and tube evaporator, that could be used either for fresh or saltwater applications.

The solution is a factory assembled packages including internal piping and electric cabling. The unit works with a low refrigerant charge of R-134a which meet the environmental class requirement.

In this type of unit there will be two separate refrigerant circuits which will give 2x100% redundancy, the main advantage is that it is possible to cool down the fluid in half the time by running the two circuits at the same time. There will be also the need for cooling water pumps for each circuit. Chillers in this range of capacity including control panel, starter panel documentation and assembling have prices starting from 3.5 mill NOK

Single chiller unit A conventional RSW chiller is the primary unit of the system, it cools the seawater to temperatures near or slightly below the freezing point of water, typically between 0°C and 2°C (32°F and 36°F).

This equipment is delivered completely on a base frame for quick and simple installation. The recommended refrigerant to use is Ammonia (NH3). The units are fabricated with a titanium spray evaporator, meaning that in a required case, the equipment is able to function and cool down seawater, which has also been used in previous projects for pressure testing pipelines.

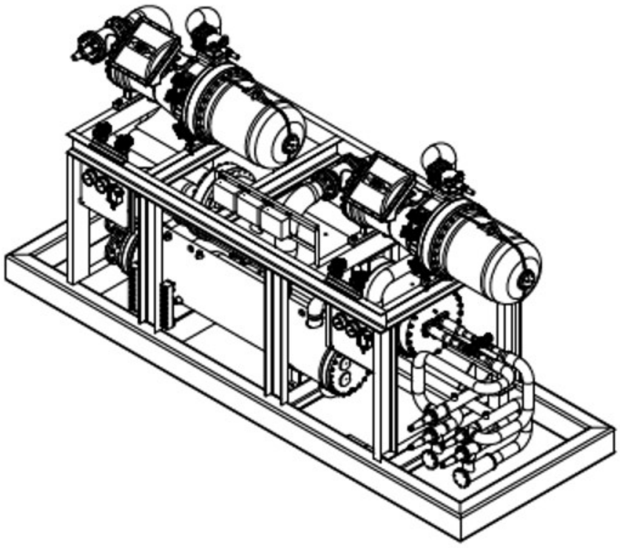


Figure 4.3.4: Compact design of a chiller (Teknotherm, 2023)

4.3.3 Discussion

An RSW system or chiller enables to control of the cooling temperature in an efficient and long-lasting, no formation of fooling due to a closed system. It is possible to make module arrangements for this equipment meaning the installation of more than one unit to improve the efficiency.

This is a very efficient system to cool seawater, due low risk of freezing when working at near 0 temperatures. But working with pure freshwater has more challenges when getting close to freezing temperatures.

Refrigerants such as ammonia (R-744) or Co2 (R-744) offer advantages in the compression system as efficiency on volumes, higher cooling capacities, low operation cost, low noise and vibration levels, and environmental advantages. As for the ammonia charge, a capacity of 120kg of refrigerant is estimated.

4.4 Mobile Desalination Unit

This solution aims to produce the water in the vessel directly using a desalinization system. Sea water with a temperature close to the pipeline surroundings will be pumped to the vessel deck and passed through a mobile unit for water treatment.

The proposed unit for this solution was found in the market, and is a fully automated Reverse Osmosis mobile unit, able to be fast deployable and installed in the vessel. It is extensively used in marine and offshore applications.

Reverse osmosis works by forcing water through a special plastic membrane sheet to remove compounds such as salt, organic compounds, and microorganisms. As water enters in a pressure vessel it flows over the membrane surface as it moves from one end to the other. These pressure vessels can be piled and configured in series to produce large amounts of water 80% of the feed water is recovered and recycled water

4.4.1 Discussion

Table 4.4.1 presents the general features of a desalinization unit capable of producing 500 m^3 of freshwater per day. This unit is capable of working at a minimum temperature of 5 °C. But the outlet temperature for the freshwater is limited to the temperature of the pumped seawater since no heat transfer process will be involved. Not capable of reaching the desired temperature but capable of working under low ranges close to the target. A limitation of this alternative would be the deck space. Even though is a movable unit, analysis of previous project deck layouts, fitting a container of such dimensions might not be achievable.

According to client requirements, water treatment for hydro testing and preservation indicates a maximum content of chlorides of 200 mg/l and PH from 6.5 to 9.5., this complies with the quality of produced water by a desalinization unit studied. Nevertheless, a filtration unit must be installed to filter the freshwater to 50µm prior to injection in the pipeline.

Table 4.4.1: Design Features of Desalinization unit. (HaternboerWater, 2017)

Capacity	500 m^3/d
Installed Power	88 kW, 400/415 Volt (50 Hz) 440/480/690 Volt
Feed Water	46 m^3/h
Dimension	6058L x 2438 (W) x2891 (H) - 20 ft container
Weight	6500 kg
Weight in Operation	9500 kg
Feed water, TDS design	35000 ppm
Maximum Pressure	70 bar
Design Temperature	5 - 25 °C
Quality product Water	$<$ 200 mg/l
Design Recovery	max 50%
Max Pressure @ product water	2 bar
Max Pressure @ brine	3 bar
Material HD manifold	Duplex SS 1.4462
Material low pressure pipping	PVC-U Pn16

An installation of this equipment does not present any limitation regarding design pressure. Since the produced freshwater will be released in the vessel tank

Although it is a non-conventional alternative, this type of system can be of interest due to its broad experience and implementation in the maritime industry. The main advantages include the compactness of the unit, easy installation, possibility of remote operation. But even though the mobility of the unit, the dimensions of the container are still large for a reference vessel presented in Table 3.3.1

4.5 Subsea Heat Exchanger

Research on subsea cooling processes has been developed for offshore field developments since seawater can be used as an effective coolant. And currents affecting the system could increase the heat exchange rate. Two patents developed in Norway will be presented, which were revised to gain more understanding of the processes involved and technological advances on this matter.

4.5.1 Passive parallel tube cooler

A passive cooler is known because it uses no external means for the circulation the of seawater. It is the simplest form of heat exchanger Since it is convenient to use the cooling capacity of the water on the sea bottom the heat exchanger aims to expose the fluid-carrying tubes to seawater. In order to increase the heat transfer area and decrease the pressure drop, the flow is divided into several parallel tubes.

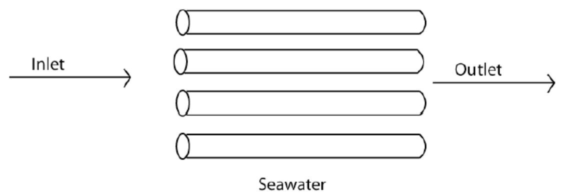


Figure 4.5.1: Patent of a Parallel tube subsea cooler (Sten-halvorsen Vidar, 2014)

The efficiency of the heat exchanger can be modified increasing the contact area by the addition of fine tubes.

This is a simple and robust design. The main disadvantage of this heat exchanger is the control of the outlet temperature. and as for subsea application, when the heat exchanger is exposed to sea currents, the heat exchange rate can change and be difficult to trace. If the facility is reliant on a specific temperature. This type of exchanger may present some challenges.

4.5.2 Subsea Cooler with propeller

This invention has an inlet and outlet for a cooled fluid in the form of a coil enclosed in a duct that will be exposed to flowing seawater generated by a propeller and a rotable actuator as shown in Figure 4.5.2

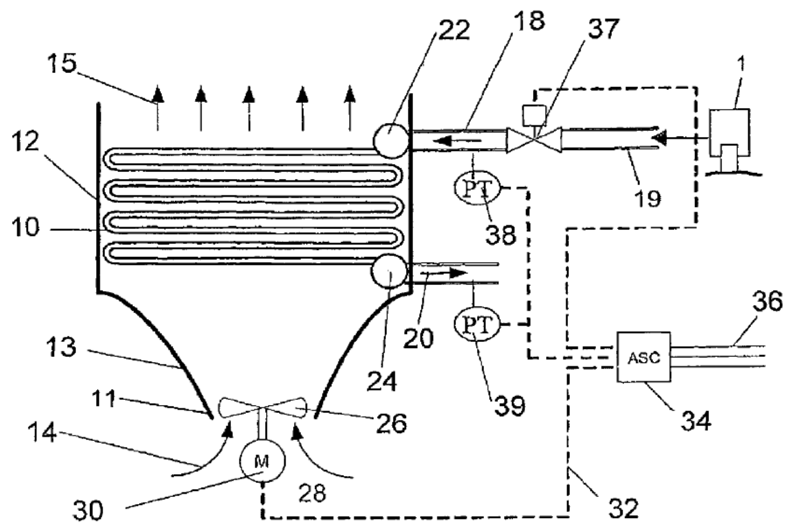


Figure 4.5.2: Patent of a subsea heat exchanger with a cooling propeller (Sten-halvorsen Vidar, 2014)

The propeller creates the desired flow of seawater past the coils positioned in the seawater. The cooling unit may comprise a controller, which is connected to different parts in relation to each other to adjust and achieve the desired cooling of the fluid.

There are different proposed solutions for the actuator, such as an electric motor, a power cable extending from a location or vessel, or a battery pack attached to the cooling unit. One of the main advantages is there is no need of rotatory equipment such as fans, motors, or gearboxes. This ensures that little-to-no maintenance is required

The alternatives presented above are promising regarding efficiency and OPEX, nevertheless, they seem to be more suitable for long-term solutions rather than a temporary solution to install for a short offshore campaign. More research must be done to model the heat transfer capacity of units like these ones and the possible sizing of them. Also, subsea cooling appears to be more effective for applications where the differential of temperature between the internal and external fluid is higher.

CHAPTER

FIVE

FINANCIAL OVERVIEW AND FURTHER DISCUSSION

Of all the solutions discussed in the previous chapter, the most efficient solution appears to be the installation of a water chiller on the vessel deck. This solution is deemed effective and capable of handling large quantities of water efficiently, while allowing control over the water's outcome temperature. Although there are existing heat exchanger equipment options that are more suitable in terms of deck space, weight, and configuration, and these options may be less expensive than a complete refrigeration mobile unit, it is challenging to achieve the necessary flowrates and effective heat transfer at the required temperature using those equipments.

This section will study the economic impact of installing a chiller unit on the vessel by comparing the initial capital expenditure (CAPEX) or investment cost of the unit with the cost savings resulting from the reduction in operation time due to the optimization of the pressure test. Discussions on the assumed values and other costs to be considered will also be conducted.

The economic evaluation of this solution is based on the concepts of cost-effectiveness and cost-saving. An estimated cost for the chiller unit will be presented, derived from interviews with providers, and compared against the cost savings achieved by reducing the vessel's operation time through the implementation of the solution.

For the initial analysis, the highest quotation for the equipment was used, which included a high-capacity chiller delivered inside a container for easy mobility between the vessel and shore. This equipment is estimated to cost 5 million NOK from a Norwegian contractor. Assuming also that will be one of the highest quotations compared with providers located in other countries where material and labour cost are less.

At this point, it is difficult to predict the maintenance costs of the equipment accurately, but it is expected that the unit will not run continuously throughout the year. Some preventive maintenance will be required before each project, as well as unloading and reloading of the ammonia charge. This introduces an additional uncertainty.

The price of ammonia is also uncertain and can vary over time due to factors such as supply or demand dynamics, energy costs, natural gas prices, geopolitical events, regulatory changes, and regional locations. However, according to consulted prices from

S&P Global commodity insights, the price fluctuates between 400 and 700 USD/mt or per 1000 kg (S&P Global, 2023). An average charge of 120 kg of ammonia per offshore campaign is estimated, hence this parameter represents a relatively small percentage of the investment compared to the initial cost of the chiller unit. Therefore, it will be neglected in these calculations

It is also expected that only one unit of this equipment will be necessary for the entire business unit in Norway. Due to the complexity of subsea projects, considerable time is expected between each of them, and the overlapping of campaigns or the use of multiple vessels at a time is not a common occurrence. This means that a single movable unit could be reused in each of the upcoming projects, adding value to the company and making offshore campaigns more efficient

The assumed cost of the vessel was estimated using the average cost from previous projects. The usual cost for vessel operations must consider mobilization and demobilization, which is not included in this analysis, since it is a fixed cost that is already considered regardless the duration of the campaign.

Regarding the reduction in operation time, it is important to mention that the duration of every project varies, as well as the installed equipment and thermal conditions. Therefore, the analysis will be conducted based on a range of 1 to 5 saved days per campaign. However, as shown in Figure 1.0.3, there are projects that can even have waiting times of 7 to 10 days.

With all these assumptions in place, the cost of the chiller unit was compared with the saved days in operation due to the cooling of the water and the optimization of the pressure test time.

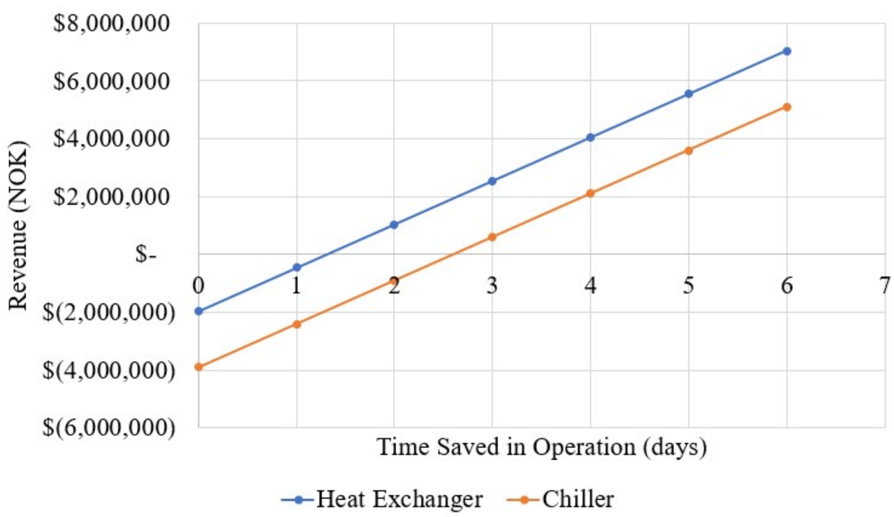


Figure 5.0.1: Money saved from the implemented solution vs saved time of the vessel

As shown in Figure 5.0.1, the implementation of this equipment becomes more profitable as the estimated number of waiting days in the project increases. Even with just a single project that enables optimization of the operation by 3 days, the equipment with the highest quotation could be fully paid.

It is important to note that the solution not only brings economic and operational benefits to the company but also improves the company's CO2 emissions KPI. Reducing the vessel's operation days substantially decreases the company's carbon footprint and aligns with the company's environmental and net-zero goals.

While the cost of a single heat exchanger unit can still vary depending on its size and specific design, it is generally more affordable compared to a chiller system. Based on equipment quotations obtained from thermal solution providers specializing in heat exchangers, the cost-effectiveness ratio was used to compare the economic benefits of choosing a cheaper alternative. This relation compares the cost savings or cost reduction achieved by the solution to the cost of implementing it. A higher ratio indicates a more cost-effective solution, as shown in Figure 5.0.2. The cost-effectiveness ratio substantially increases for less complex solutions such as a single heat exchanger

Is important to note that even though the heat exchanger alternative could be beneficial at first sight, many factors influence the final temperature of the water, and in some cases, the target is not even achievable. Meaning that the reduced time in the offshore campaign could be less.

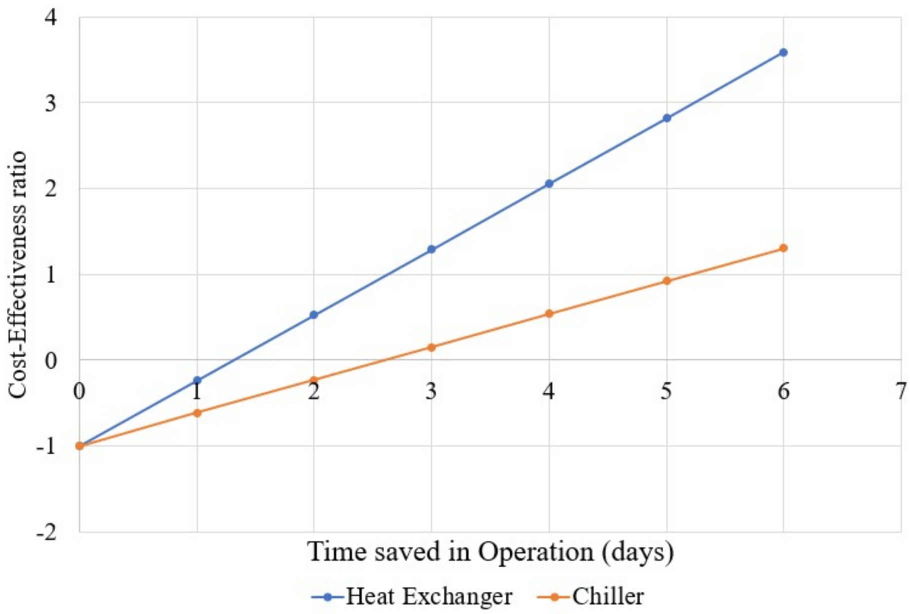


Figure 5.0.2: Cost-effectiveness ratio for 3 options vs saved days of operation

Due to their complexity and higher capacity, chiller systems are generally more expensive than single heat exchanger units. The price of the chiller system increases since it is composed of many individual components such as a compressor, evaporator, and heat exchanger. Still, for each of the solutions and range of prices analyzed, it is profitable to install the solution as long as the operation could be optimized by at least 1 to 3 days in each project.

Based on the results obtained from this study, the recommended next steps include a detailed design of the spread using the proposed technology. This design will determine the final pressure and temperature values that the system will operate under, and it must be verified that the suggested equipment has the capacity to meet the required temperature program. With this, the revision of the proposed design and P&ID by the provider, purchase of the equipment, installation, testing, and implementation of the solution. Further research on alternative and innovative subsea cooling methods and their poential impact on such applications could also be conducted. An assessment of different equipment providers should be carried out to find the most suitable one, considering legal agreements, company location, shipping of the unit, and availability, among other factors. Additionally, studying the possibility and potential impact of renting the unit instead of purchasing it is recommended.

CHAPTER

SIX

CONCLUSION

The temporary installation of a Subsea Spool piece near the seabed to cool down fluid during its flow towards the pipeline inlet termination presents challenges in achieving desired temperatures within a reasonable length. This is primarily attributed to the low efficiency of heat transfer when the temperature differential is minimal, as well as the necessity for high flow rates to meet the company's requirements of flooding the lines. Exploring alternative configurations that balance the need for high flow rates with efficient cooling could be beneficial.

It is theoretically impossible to attain the desired temperature of freshwater using seawater as a cooling fluid under the given time frame since the minimum achievable temperature of it using a subsea pump is expected to be higher than 4 degrees. However, the use of refrigerant fluid at lower temperatures, particularly MEG with its low freezing point, can enable the reduction of the cooling fluid's flow rate. Nevertheless, this presents logistical challenges for the company to initially obtain such fluid at the required temperature and if this is the case, it is recommended to explore the mixing of the fluid directly with the freshwater when water-MEG mixtures are required to be used as a pressure test fluid. If the company intends to install a heat exchanger solution with seawater as the cooling medium, it should accept a 1 or 2-degree increase in the freshwater final temperature, which would still significantly reduce waiting times for stabilization.

The studies on solutions for cooling water in other industries lead to the evaluation of the installation of a mobile chiller system unit, leading to the conclusion that it is recommended to install such a system on board due to various reasons: this unit allows work under temperatures near the freezing point, they can be designed with a wide variety of refrigerants and to handle big pressure ranges and high flow rates. Ammonia is recommended as a refrigerant due to its low environmental impact and high efficiency, Considering the company's existing agreements or contracts MEG is also a refrigerant of interest. It is also important to prioritize redundancy in the system design for reliability and operational continuity offshore.

The installation of a reverse osmosis desalination mobile unit was proposed to produce the freshwater directly in the vessel. This alternative has a different approach and differs from a heat transfer application, but instead is limited to the minimum achievable temperature of the seawater that will be pumped from the sea bottom and the flow rate of the pump. Also, the strict quality specifications of water for hydro testing and flooding pipelines might increase the necessity for additional filtering units. Further research on the average price of this type of unit and the cost of the replacement of the membranes must be done to be comparable with other alternatives.

An economic analysis was conducted to compare the cost of the unit against the savings in time and money resulting from the reduction in offshore campaigns. This analysis revealed the potential to recover the entire investment in the equipment during the first project. Furthermore, purchasing the entire unit would incur minor additional operating expenses (OPEX) for mobilization, installation, and maintenance, which could be covered in future projects. A single heat exchanger configuration is the most economically rational to invest in, but the further investigation must be made to dimension possible achievable temperatures and prices of working with a deep subsea water pump to use seawater as a cooling fluid.

The implementation of a system that not only reduces the time of the vessel on site but also makes use of natural refrigerants such as ammonia, not only brings economic benefits to the company but substantially reduces carbon emissions and aligns with the net zero targets philosophy of the company and in general oil and gas industry.

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