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Evaluation of Process Cooling in Subsea Separation, Boosting and Injection Systems (SSBI)

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MASTER THESIS

for

student Emil Gyllenhammar

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Evaluation of active cooling in subsea separation, boosting and injection systems
Vurdering av aktiv kjøling i undervanns separasjons- pumpe- og injeksjonssystemer

Background and objective

The recent enabling of subsea compression has enabled more advanced subsea processing. As the process systems being placed on the seabed become more complex, the need for system and component cooling arises. Cooling in subsea processing has up to now only been done with passive coolers based on free convection heat exchange with surrounding sea water. Such coolers may not offer the controllability needed for more advanced process systems. Coolers based on active/forced convection heat exchange are widely utilized, but has yet not been applied in subsea applications. Subsea separation, boosting and injection systems include the minimum of components needed for any advanced subsea processing system, which makes it a good candidate for evaluation.

Little information is available on the use of active cooling techniques in subsea processing, as such technologies have not yet been qualified for subsea use. The goal of this study is to evaluate the use of active heat exchanger systems in subsea processing, with focus on subsea separation, boosting and injection systems (SSBI), by identifying the process needs, evaluating cooling system design, and performing detailed heat exchanger thermal design.

The following tasks are to be considered:

1. Present subsea process systems where cooling is required, with focus on subsea separation, boosting and injection. Describe on the different requirements of these process systems, and how these requirements will affect the choice of cooling technology.
2. Evaluate active heat exchanger technologies applicable for a selected process from task 1. Evaluate the design of the heat exchanger system, including variation of stream parameters, heat exchanger configuration and type as well as controllability. A discussion on whether direct or indirect cooling is to be used shall be performed.
3. Perform thermal-hydraulic design of one or more of the heat exchangers presented in task 2. A case with representative field data will be used for the process and heat transfer calculations.

4. Suggestions for further work in the area of active heat exchangers for subsea use shall be presented.

Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

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Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report.

Pursuant to "Regulations concerning the supplementary provisions to the technology study program/Master of Science" at NTNU §20, the Department reserves the permission to utilize all the results and data for teaching and research purposes as well as in future publications.

The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. Based on an agreement with the supervisor, the final report and other material and documents may be given to the supervisor in digital format.

Department of Energy and Process Engineering, 16. January 2012

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Abstract

The next generation of subsea process systems will combine the subsea gas compression technology currently under qualification with the previously developed subsea processing technologies, including separation, multiphase pumping and produced water re-injection. These systems will benefit from process cooling. This paper is an evaluation of the use of process cooling in subsea separation, boosting and injection (SSBI) systems including compression. Fouling is the biggest uncertainty, and potentially the biggest problem, in the design and operation of process cooling for SSBI systems when reliability, size, weight and controllability are considered as the most important design parameters. The room for optimization towards fouling reduction in the process cooling was identified to be in the process system design, in the cooling arrangement, in the heat exchanger selection and in the heat exchanger design. In each of these steps the optimization potential was identified and discussed. A case study was performed in which a direct cooling system using a printed circuit heat exchanger was found to be the most compact solution. The rate of fouling will set the reliability and maintainability of a heat exchanger installed under water. The fouling rate, and subsequently the cleaning strategy will change the design of the heat exchanger. The available information on fouling rate in SSBI systems is not complete. To develop a complete picture of fouling in subsea heat exchangers it is suggested that similar heat exchanger technology already field proven is studied for the collection of detailed operation experience and data. This infromation is useful for the design of subsea process coolers to develop the most compact, reliable and controllable solution.

Sammendrag

Neste generasjons undervannsteknologi for prosessering vil kombinere undervannskompresjon med allerede kvalifisert teknologi som separasjon, flerfasepumping og re-injeksjon av produsert vann. Et slikt system vil dra nytte av prosesskjøling. Denne oppgaven er en evaluering av bruken av prosesskjøling i undervannssystemer for kombinert separasjon, flerfasepumping, kompresjon og vanninjisering, på engelsk forkortet med SSBI. Beleggdannelse er den største usikkerheten og utfordringen i design og operasjon av prosesskjøling for SSBI-systemer når pålitelighet, størrelse, vekt og kontrollerbarhet anses som de viktigste designparameterne. Rom for optimalisering mot reduksjonen av beleggdannelse ble identifisert til å være i prosess-systemdesign, kjølearrangementet, valg av varmeveksler og i design av varmeveksleren. I hvert av disse stegene ble optimaliseringspotensialet identifisert og diskutert. En casestudie ble gjennomført og direkte kjøling ved bruk av en kompaktveksler av typen «Printed Circuit» ble funnet å være den mest kompakte løsningen. Beleggdannelsesraten vil bestemme påliteligheten og vedlikeholdsvennligheten av en varmeveksler installert under vann. Beleggdannelsesraten, og deretter rengjøringsstrategien, vil endre utformingen av varmeveksleren. Den tilgjengelige informasjonen om beleggdannelsesraten i SSBI-systemer er ikke komplett. For å danne et helhetlig bilde av beleggdannelse i varmeveksler under vann er det foreslått at lignende varmevekslere allerede felttestet studeres for innsamling av detaljert data og erfaring fra operasjon. Denne infromasjonen er nyttig i designet av prosess-kjølere for bruk under vann for å kunne utvikle den mest kompakte, pålitelige og kontrollerbare løsningen.

Contents

List of figures

List of tables

Nomenclature

Active cooling – cooling using forced convection

Boosting – Increasing pressure for increased recovery

CAPEX – Capital Expenditure

Direct cooling – cooling against seawater direcly

Indirect cooling – cooling against seawater indirectly

MEG – Mono Ethylene Glycol

Multiphase – A mixture containing more than one phase

Multipass – several tube passes in series in a shell and tube heat exchanger

OPEX – Operating Expenditure

OHTC – overall heat transfer coefficient

Passive cooling – cooling using natural convection

Subsea – Beneath the water surface

Subsea Water Reinjection – Injection of produced water into a subsurface well

SSBI – Subsea separation boosting and injection

Topside – The part of a platform that is placed on top of load bearing structure

R - resistance

T – Temperature

°C – Degrees Celsius

q'' – Heat flux

q – heat transfer rate

U – overall heat transfer coefficient

A - area

ΔT_m – mean temperature difference

m – mass flow rate

C_p – mass heat capacity

ΔT – temperature difference

Introduction

In the summer of 2011 the final decision was made by the partners of the Åsgard field to invest in the technology for a subsea compression module in order to enhance recovery of the reservoir. After separating the gas and the liquids, the gas will be compressed, and the liquids pumped, into a multiphase pipeline which leads to the Åsgard B platform. The compressor module will be the world's first installed under water and represents a milestone in the development of subsea technology.

With compression on the brink of being qualified for underwater use, it is possible to combine the subsea processing components offered today in customized combinations to answer specific field development needs. The next generation of subsea process systems will combine the gas compression technology with the previously developed subsea processing technologies, including separation, multiphase pumping and produced water re-injection. While these technologies constitute simple process systems they are the core components of the process systems that in the future may replace the oil and gas platforms we know today.

The process cooler developed for underwater use in the Åsgard project has not yet been field proven and does not have a sophisticated design; this type of cooler has been designed for compression systems, with anti-surge cooling in focus, and offers no controllability.

For other subsea process system designs the need for cooling may be different from that of subsea compression systems.

Scope of work and limitations

The scope of this paper is to evaluate the use of process cooling in subsea separation, boosting and injection (SSBI) systems including compression. SSBI systems with compression represent the basis for the next generation of subsea process systems. Process cooling in such a system is desirable for increased performance.

In this paper the use of a process cooler as an anti-surge cooler for the compressor will not be evaluated, and it is suggested that the design of an anti-surge cooler is to be evaluated separately from the evaluation of a process cooler.

The evaluation of pressure drop in heat exchangers has not been emphasized in this paper. In a SSBI system the increased pressure drop on the process side would lead to increased compressor work, and increased pressure drop on the seawater side would lead to increased cooling pump work. To not come up with irrational designs the maximum allowable process side pressure drop was set to 1 Bar and the maximum seawater side pressure drop was set to 5 Bar.

Method

The study of process cooling in SSBI systems has been conducted using an onion model inspired by courses in process heat integration at NTNU. The onion model is used to visualize the concept of process cooling design by dividing each step into a different layer, see Figure 1.

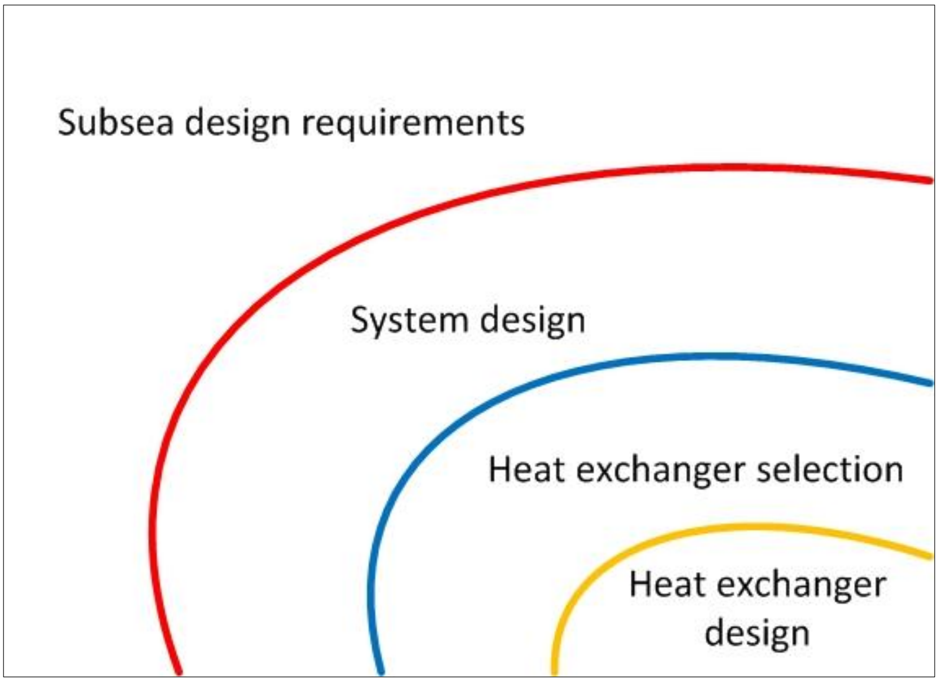

Figure 1: The process cooling study approach

The first step, represented by the outermost layer, is to define the general requirements for subsea process systems and equipment. The final step, represented by the innermost layer, is to determine the details in heat exchanger design which depends on the results from all of the preceding steps. This paper covers each of these five steps in order.

1 Subsea design requirements

Section 1.1 of this chapter is a short presentation of subsea process systems and components, and leads up to Section 1.1.3 where the SSBI system that is the basis for the evaluation in this paper is presented. Section 1.2 presents the design requirements for subsea equipment. Section 1.3 is a summary of the critical subsea design requirements identified in the previous sections, and will be the red thread through this paper.

1.1 Subsea processing

Subsea processing is a term referring to any treatment of the wellstream performed under water, except the injection of chemicals, such as hydrate and corrosion inhibitors [1]. Process technology currently available for use under water includes:

- Separation (2-phase and 3-phase)
- Pumping (single and multiphase)
- Gas compression
- Passive process cooling
- Produced water re-injection and raw seawater injection into wells

Subsea processing provides the ability to produce at lower reservoir pressures than what is possible with a surface production facility and thus increases reservoir recovery. Other benefits include reduced cost and HSE risks. More elaboration on the implementation of subsea processing in field developments and the benefits of this can be found in [1]

1.1.1 Subsea boosting

Subsea boosting is a term used to describe the process of increasing pressure by the use of pumps, compressors or both at the sea bottom. Subsea boosting has been used to increase the pressure of a wellstream when the reservoir pressure is below the minimum needed to maintain the desired production rate. Traditionally this concept has been applied in the oil and gas industry by installing a boosting module at the platform upstream of the processing module [2], to reduce the backpressure of the wellstream and increase the pressure to the required processing pressure. It can easily be shown that installing boosting at the sea bottom allows the reservoir pressure to fall even lower that what is possible with a module located at the platform. The benefits of subsea boosting include:

- Increased production rate
- Increased reservoir recovery
- Lower pipeline pressure drop between reservoir and the platform for a given flow rate and pipeline size

1.1.1.1 Motivation for boosting

The reservoir pressure of a field will be at the very top when the production starts and falls from day one. The production is constrained by the minimum pressure allowed at the platform inlet. If the inlet pressure is allowed to fall below the minimum value he platform process will start to operate outside its design condition. This can potentially harm or destroy equipment, and cause the process to produce a product that is deviating from the specifications. For a gas field the minimum allowable platform production pressure, during the maximum production rate, is reached at the end of the production plateau. At this point the flow rate will have to be reduced in order to maintain the required production pressure. Production is shut down it is no longer economically sustainable, or infeasible due to marginal flow rates. By implementing boosting when the production plateau is at its end, the plateau can be extended, allowing the platform to produce at the same rate as before with even lower reservoir pressures.

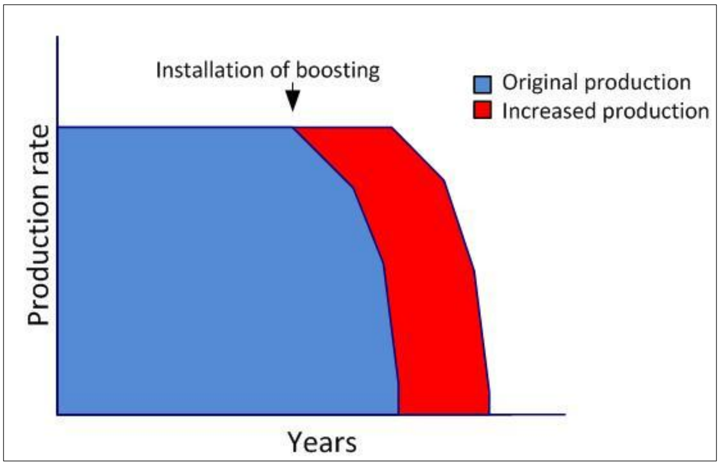

Figure 2: An example of how the plateau production can be extended with the use of compression

1.1.1.2 The recent enabling of subsea compression

The Åsgard field in the North Sea will be the first in the world to have subsea compression installed. This project has qualified many new technologies; of witch one is a subsea gas compressor that can handle small amounts of liquids in the gas [3]. The purpose of the system is to increase the recovery of the reservoir and the main process components include:

- Passive inlet/anti-surge cooler
- Gas/Liquid separator
- Wet gas compressor
- Multiphase pump
- Passive outlet cooler

1.1.2 Subsea separation and water injection

Subsea separation is the process of separating the wellstream under water. Both 2- and 3-phase separation has been qualified for use under water [4]. The water separated out from the wellstream is pumped back into the reservoir or another sub surface well and is referred to as water injection. Costly processing assuring the right water quality would be needed if the water was to be released into the sea, due to its environmentally harmful contents. Such processing is typically needed for all the water produced at a platform. If the produced water is injected back into the reservoir subsea, the platform will have less water to process. This leads to a cost reduction and can effectively debottleneck oil and gas production when water handling capacity is limiting the maximum production rate. Separation and injection of water also reduced the liquid content of the hydrocarbon pipelines to the platform, and by that increases the production rate.

The benefit of subsea separation if used alone is:

- Enables the use of separate gas and liquid pipelines to the platform, or to two different platforms

The benefits of subsea separation combined with water injection include:

- Increased production rate due to reduced liquid content in the pipeline
- Increased recovery due to less pressure drop in the pipeline
- Reduced water production at the platform due to subsea water injection

The complication of water injection is that a hydrate inhibitor can never continuously be used upstream of the water injection point in the process. This is because the hydrate inhibitor will dissolve in the water phase and follow the water that is injected into the reservoir. This would be a waste of hydrate inhibitor. A hydrate inhibitor such as Mono Ethylene Glycol (MEG) will typically have to be mixed with water at a weight percentage in the order of 40-70% [1] to suppress the hydrate formation temperatures sufficiently. The MEG mixed in with the injection water would never be recovered, and would require large amounts of new expensive MEG to be supplied continuously, which is considered both very expensive and infeasible.

When including subsea separation and water injection with subsea boosting, the system is referred to as Subsea Separation Boosting and Injection (SSBI). Qualification of this system was done by Statoil in a pilot project at the Troll field in Norway, before a full scale SSBI system later was implemented at the Tordis field, also in Norway [5] [6]. Both of these systems process liquid dominated wellstreams. Gas compression is not needed since the gas fraction is within what the subsea multiphase pumps can handle. This has limited the use of SSBI to liquid dominated wellstreams. The benefits of SSBI include all the benefits of separation water injection and boosting alone.

1.1.3 Next generation subsea process systems

A basic need in any oil and gas processing facility is the ability to separate all phases and increase pressure of each single phase. The well stream will have to be processed in order to be transported to shore, which is much of the reason oil and gas platforms exist in the first place. Even the simplest forms of oil and gas processing, as that performed on offshore production facilities, will lead to loss of pressure. Export pumps and compressors are needed to increase pressure after processing to transport the products to shore in pipelines.

Compression allows the SSBI system to not only be applied to liquid dominated wellstreams, but also gas dominated wellstreams and anything in between. When including compression with SSBI, one has the core components of an offshore process facility and the fundamental structure of the subsea process systems that in the future potentially can replace the surface production facilities: a vision of the oil company Statoil [4].

SSBI system design with compression includes a minimum of the following process technologies:

- Separation
- Compression with anti-surge cooler
- Single/Multiphase Pumping
- Water Injection

The goal of this SSBI system is one or more of the following:

- Maximum reservoir recovery
- Maximum water removal (water conditioning)
- Function as an oil and gas hub for multiple reservoirs; separating and redirecting the two phases to different platforms
- Reach a specific temperature and pressure for separation, and possibly gas processing such as dehydration in the future

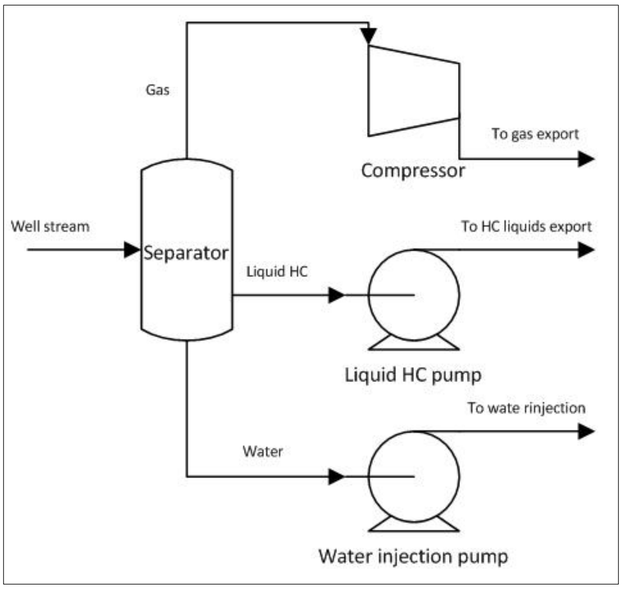

Figure 3: The basic system design of SSBI

1.1.4 The implementation of process cooling

The objective of introducing process cooling to the SSBI system is not set by a process requirement, and the SSBI system will function without any process cooling. The SSBI system performance will however substantially improve with the introduction of process cooling, much because of the improved compressor efficiency, but also because of increased water removal.

The benefits of process cooling in SSBI systems with compression are:

Reduced work: cooling prior to compression reduces the temperature and volume of the gas, thus the compressor work.

Water condensation: Cooling decreases the gas maximum water content and leads to increased water production. This water can in turn be injected subsea and further reduce water production at the platform facility, given that a separation sequence allowing this water to be recovered is chosen, see section The process side3.1.

Reduced material requirements: equipment and pipelines downstream of the compressor can have lower design temperatures due to reduced compressor outlet temperature with reduced compressor inlet temperature.

The coolers in the Åsgard project and the Ormen Lange project are tubular coolers in natural convection heat exchange with the surrounding seawater, so called passive coolers. The cooler in the Ormen Lange project is an anti-surge cooler [7], and the cooler in the Åsgard project is designed as an anti-surge cooler, but will be placed in the process so that it performs continuous process cooling as-well, due to the benefits of this [3]. Both these coolers are implemented in subsea boosting system, where there is no water injection and MEG is used continuously for hydrate inhibition. Both of these coolers have been designed with anti-surge as the main focus shaping the design.

No detailed evaluation of the use of process cooling for subsea process systems is found in the literature, and little of the evaluation leading to the choice of the passive cooling technology for the subsea compression systems can be found. The reason passive cooling has been chosen is most likely its simplicity. These compressor systems are not dependent on process cooling and the precision and the process cooling performance is not of any major concern, the anti-surge cooling performance is however heavily emphasized.

A lot of information is found on the design of heat exchangers, and a lot of information is found on subsea equipment. This paper will combine this information in an attempt to generically evaluate the use of process cooling for SSBI systems.

1.2 Subsea equipment

Subsea equipment is placed on the sea bottom in remote areas, and up to several thousand meters under the sea surface. The equipment will have to work close to flawless for longer periods as it is very hard to reach. Reliability, availability and maintainability are properties of a component or a system that affects its ability to perform its required function. The definitions of these terms in ISO 20815 are:

> *“**Reliability:** ability of an item to perform a required function under given conditions for a given time interval”*

> *“**Availability:** ability of an item to be in a state to perform a required function under given conditions at a given instant of time, or in average over a given time interval, assuming that the required external resources are provided.“*

> *“**Maintainability:** (general) ability of an item under given conditions of use, to be retained in, or restored to, a state in which it can perform a required function, when maintenance is performed under given conditions and using stated procedures and resources.”*

1.2.1 Maintainability

When placing equipment on the sea bottom, it is natural to think that equipment size and weight is of no concern, but this is not the case. If subsea equipment fails and cannot be repaired, the equipment will have to be replaced. The common subsea equipment redundancy of oil companies operating in the North Sea is to have an identical spare stored onshore [8]. When the equipment installed under water fails, a vessel will pick up the spare, travel to the site, lift up the failed equipment and immediately install the spare. The spare is then brought back onshore where it can be repaired and maintained. Downtime, and thus the maintainability of the subsea system, is directly dependent on the time it takes to find an appropriate vessel to this job.

To minimize downtime, the retractable equipment will have to be of a weight and size that is within the lifting capacity of a large number of service vessels operating in the region of the field. The size limitation for subsea equipment is simply based on the available lifting capacity in the region. Table 1 shows the lifting capacity and response time of intervention vessels operating in the North Sea. [8]

Table 1: Intervention vessel response time and lifting capacity in the North Sea

Intervention vessel type	Mobilization time	Max lifting weight
Service	1 to 3 days	50 T
Construction	6 days	150 T
Heavy construction	9 to21 days	300 T
Heavy lift	30 days	1300 T
Super heavy lift	60 days	1500 T

Equipment that is crucial to the process and potentially needs repair during its life time should be within the lifting capacity of the service vessels. These vessels are built for serving the subsea industry with the service of installation, repair and retrieval of subsea x-mas trees (valve assemblies) and have a maximum size limitation for lifting according to this. The maximum size of equipment that can be lifted with the service type intervention vessels is 6m x 6m x 8m.

The maintainability of a subsea component itself may be low, since the component is repaired onshore, without affecting production availability once replaced. Maintainability of the component will however affect the time and cost of repair when the component is brought to a workshop onshore.

1.2.2 Reliability and availability

Reliability and availability of subsea equipment will determine the number of interventions that will have to be performed during the lifetime of the equipment. As these operations are very expensive, it is beneficial to spend more money on increasing the reliability of the equipment to reduce the number of interventions needed [8]. Introducing familiar equipment to a new environment, such as introducing traditional topside process equipment subsea, will always increase risk and reduced reliability [9]. In the subsea industry however, this type of risk is usually minimized through extensive technology qualification which ensures satisfactory component reliability.

The availability of the subsea process system will be directly affected by the ability of the equipment to operate satisfactory without any major needs for maintenance over long periods of time, when the redundancy philosophy is to have the spare unit onshore. A two year operation time without interventions is considered the minimum time interval for the reliability of subsea equipment. [8]. Equipment modules needing the largest types of offshore cranes for installation will have to stay on the sea bottom until decommissioning, as interventions with such vessels are both extremely expensive and very time consuming, see Table 1.

1.2.3 Ambient conditions

Seawater has a lower freezing point than fresh water, and does not have the density peak at 4°C, but continuously increases density down to the freezing point. Depending on depth and salinity the freezing point of saltwater is around is around -1 to -2 °C. [10] The seawater temperature at the bottom of the North Sea approaches this temperature, and subsea projects are known operate with subzero design temperatures [11].

The seawater ambient pressure at the sea bottom is the hydrostatic seawater pressure at the given depth. This is pressure in Bar is roughly the same as the water depth divided by 10, shown in equation (1).

$$p_{absolute} = \rho g h + p_{atm} \approx \frac{0{,}1\ Bar}{m} h + 1\ Bar \qquad (1)$$

1.2.4 Internal design pressure

The design pressure of subsea equipment is, according to ISO 13628-1, the wellhead shut-in pressure, even if the operational pressure may be far below. For some equipment the hydrostatic water pressure can be subtracted from the design pressure, such as for pressure vessels where the gage pressure is crucial for the integrity. Subsea equipment is close to the wellhead, and the design pressure is based on the degree of isolation between the wellhead and the equipment. Proper isolation can be added in the form of High Integrity Pressure Protection System (HIPPS); a fast closing valve system that guarantees a downstream design pressure lower than the wellhead shut in pressure. HIPPS are used upstream of long flow lines or risers to reduce the design pressure, which allows for lower wall thickness and saved material costs [12].

1.3 Summary of subsea design requirements

SSBI systems increase the recovery of reservoirs, improve flow conditions in flow lines and reduce the amount of produced water at the platform. SSBI systems can be used with multiphase pumps, or multiphase pumps and compression. When using compression in SSBI systems, these will benefit from cooling through saved work and potentially increased water recovery for injection. The goal of the SSBI system will affect the requirements to control in the cooling system.

Subsea process equipment in the North Sea that potentially could be in need of maintenance or replacement cannot be of a size larger than 6x6x8 meters and a weight of 50T due to limitations in lifting capacity of service intervention vessels.

Subsea process equipment requires very high reliability, and will have to be designed to operate problem free for a minimum period of two years, to minimize the number of costly interventions needed during its lifetime.

The most important design criteria for a subsea cooler can then be summed up as:

1. Reliability
2. Size
3. Controllability

The design pressure for subsea process equipment is the well-head shut in pressure. The ambient design temperature can be below zero degrees, and the ambient design pressure in bar is approximately the depth divided by ten.

2 Heat transfer theory and equipment

This chapter goes through some of the theory needed to understand the design of cooling systems.

2.1 Basic heat transfer

An extension of Newton's Law of Cooling relates the heat transfer rate q to the mean temperature difference ΔT_m between a hot and a cold stream separated by a heat transfer surface of area A, with an overall heat transfer coefficient U [13] This expression assumes that the heat transfer coefficients is constant over the given temperature range.

$$q = U\ A\ \Delta T_m \tag{2}$$

The heat transfer rate can also be calculated through enthalpy change of either the hot or the cold stream. For a single phase stream, not going through a phase change and with constant specific heat capacity the enthalpy change is calculated as follows:

$$q = dh = mC_p\Delta T \tag{3}$$

A goal of heat transfer calculations can be to find the required heat transfer area of a heat exchanger for a specified heat transfer rate. In order to do this the overall heat transfer coefficient and the temperature difference must be found.

2.1.1 Overall Heat Transfer coefficient

The overall heat transfer coefficient in a heat exchanger is the inverse of the total thermal resistance to heat transfer between the streams.

$$U = \frac{1}{R_{tot}} \tag{4}$$

The total thermal resistance when the resistances are in a series is the sum:

$$R''_{tot} = R''_{internal} + R''_{fouling,i} + R''_{wall} + R''_{fouling,e} + R''_{external} \tag{5}$$

The internal and external thermal resistance is the inverse of the convection heat transfer coefficient h, for each side respectively. For a flat surface this resistance is:

$$R''_{external} = \frac{1}{(h''_{external})} \tag{6}$$

To use the external surface as reference, the internal resistance must be corrected for the area ration between internal and external area.

$$R''_{internal} = \frac{1}{(h''_{internal})} \times \frac{A_{external}}{A_{internal}} \tag{7}$$

The resistance through the wall can be calculated using the material thermal conductivity and wall thickness, and the appropriate equation for the geometry. Fouling on the heat transfer surface reduces the overall heat transfer coefficient, represented by the internal and external fouling resistances $R''_{fouling}$.

2.1.2 Temperature difference

The bulk temperature difference between the hot and the cold stream is the driving force of heat transfer in a heat exchanger. The temperatures along a heat transfer surface can be plotted in a figure, and is referred to as the temperature distribution or temperature profile, shown in Figure 4. The temperature distribution is affected by the flow arrangement, further discussed in section 2.3.1.

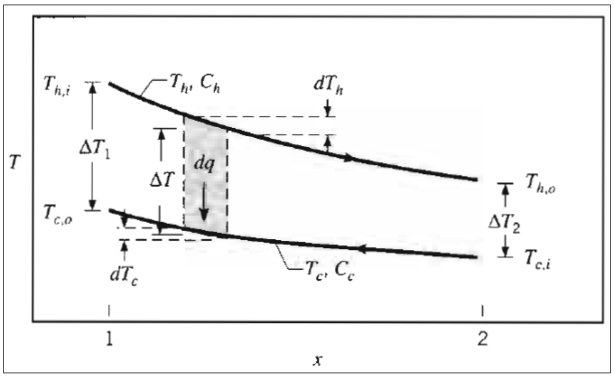

Figure 4: Temperature profile for a counterflow heat exchanger

2.1.2.1 Log mean temperature difference

It can be shown that the mean temperature difference ΔT_m in equation (2) is equal to the logarithmic mean temperature difference of the heat exchanger streams given the following assumptions [13]:

1. No heat loss to the surroundings
2. Axial conduction in the heat exchanger is negligible
3. Potential and kinetic energy changes are negligible
4. The fluid specific heats are constant
5. The overall heat transfer coefficient is constant

When these assumptions are valid, the Logarithmic mean temperature difference can be calculated for the selected to flow arrangement.

2.1.2.2 Temperature difference for special conditions

With some special flow conditions the assumptions behind the LMTD approach are invalid. These cases require the adaption of more complicated solutions.

The overall heat transfer coefficient is highly dependent on Reynolds number and physical properties of the stream, and should only be considered constant for calculations over small temperature ranges. For large temperature ranges the assumption of constant heat capacities becomes less valid as well. One approach to these problems is to calculate the properties at the heat exchanger end points, and use the arithmetic average of these in equation (1) and (2), or simply calculate the properties for the middle temperature of each stream.

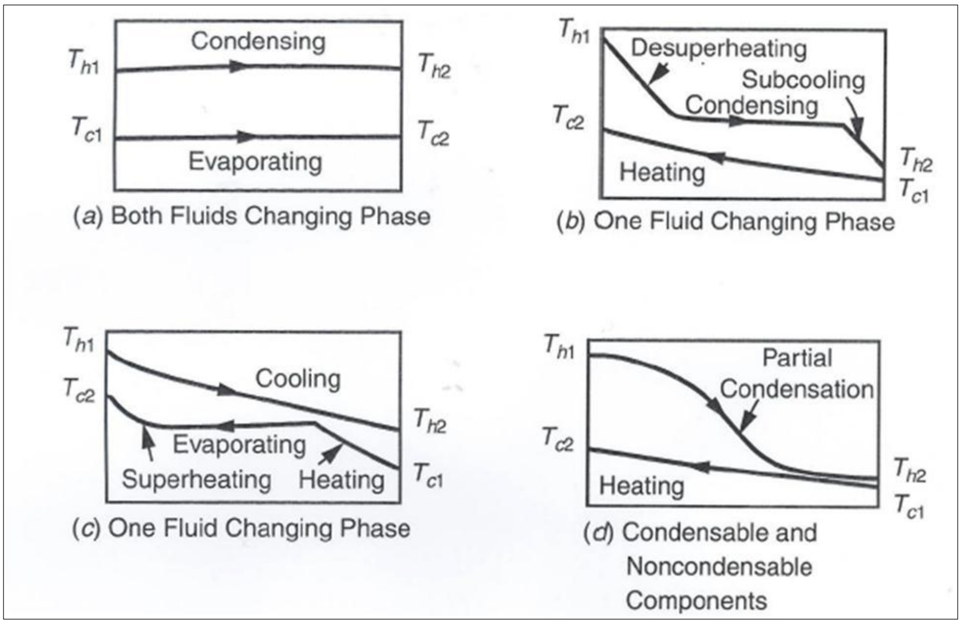

Figure 5: Temperature profiles for special condition streams in a counterflow heat exchanger

The LMTD method is not applicable for large changes in heat capacities and heat transfer coefficients, or problems where more advanced temperature profiles are in play due to phase change in the streams, shown in Figure 5. The heat transfer calculations can then be divided into segments of constant overall heat transfer coefficient and solved numerically or with the finite-difference method [14].

2.1.3 Advanced designs and commercial software

Heat transfer calculations become increasingly difficult when adding multiple phases, multiple component streams and complicated heat transfer geometries. With hydrocarbon gas streams, which contain both condensable and non-condensable gases over the cooling range, the heat transfer calculation model requires complicated modeling of the thermodynamics of the streams combined with the heat transfer calculations.

Commercial software exists for heat exchanger design and rating of the well-established heat exchanger designs, such as the TEMA type shell and tube heat exchangers and plate heat exchangers. Aspentech HTFS and HTRI Exchanger Suite are two software packages suited for such calculations.

Vendors of special heat exchanger designs will have their own models for the design and rating of these, usually kept internally and used for quotations to customers. The uncertainty of heat exchanger design will increase with the lack of model validation through testing and calibration.

If choosing a heat exchanger design that deviates from any of the traditional designs, the heat transfer calculation model will have to be built up with correlations from scratch and preferably validated before used for heat exchanger design and construction.

2.2 Fouling

Fouling is the unwanted accumulation of deposits on the heat transfer surface. The layer of fouling on the heat transfer surface has a lower thermal conductivity than the metal wall, and will reduce the total heat transfer coefficient, as shown with the use of a fouling resistance in equation (5). There is a lot of uncertainty coupled with the prediction of fouling, and the next sections will summarize and explain on the types of fouling, the fundamental fluid mechanics which affects fouling and a short discussion of fouling effect on heat exchanger design.

2.2.1 Classification of fouling

Fouling can be divided into six categories [15]. These six categories of fouling are:

1. **Precipitation**
2. **Particulate**
3. **Chemical reaction**
4. **Corrosion**
5. **Biological**
6. **Freezing or solidification**

While [15] ads the sixth category of solidification, some literature include solidification fouling under precipitation fouling [16]. The six category definition from [15] was preferred due to the emphasis of hydrate formation as a separate category from scaling.

2.2.1.1 Precipitation fouling

Precipitation fouling is often referred to as crystallization or scaling. The most common precipitation fouling is crystallization of salts on the heat transfer surface, due to oversaturation of salts in the stream [17]. Crystallization of both normal solubility salts and inverse solubility salts can cause precipitation fouling to occur both under cooling and heating applications.

2.2.1.2 Particulate fouling

Particulate fouling is the accumulation of solid particles on the heat transfer surface. The solid particles are suspended in the stream due to particle size and stream velocity. A change in stream velocity, as when entering a heat exchanger of larger flow area than an upstream pipe, may form sedimentation on the heat transfer surface if the gravitational settling forces become dominant. The type of particles that form particulate fouling may include anything from sand, mud or biological material to corrosion products, coal or dust. Particulate fouling may work as a catalyst for chemical reaction fouling or solidification fouling [17]. This is due to a growth of nucleation points along the heat transfer surface.

2.2.1.3 Chemical reaction fouling

Chemical reaction fouling occurs from a chemical reaction on the heat transfer surface in which the surface itself is not part of the reaction. Coke is an example of chemical reaction fouling; a crusty deposit of hydrocarbons formed on high temperature surfaces [17].

2.2.1.4 Corrosion

Corrosion fouling is a form of chemical reaction fouling where the heat transfer surface itself is reacting with the stream. Corrosion will decrease the heat transfer surface thickness and produce an extra layer of corrosion product on that surface. The total effect of this is usually very low, but the increased surface roughness may promote other types of fouling [16].

2.2.1.5 Biological fouling

Biological fouling is the accumulation and growth of biological organisms on the heat transfer surface. This type of fouling is divided into microbial and macrobial fouling. Microbial fouling is the formation of small organisms such as algae, bacteria and mould, while macrobial fouling is the formation of larger types of organisms such as clams, mussels and vegetation. Biofouling is typically found in seawater or fresh water systems. Microbial fouling is the main type found in heat exchanger systems, and forms what often is referred to as a biofilm: an even slime layer of biological growth on the surface. Biofouling mainly exists in temperatures between 0 and 90 °C, and thrive in temperatures around 20 to 50 [18]. Open and once through water systems are most prone to biofouling.

2.2.1.6 Freezing or solidification fouling

Solidification fouling can typically come from wax, ice or hydrate formation or any other component that solidifies within the temperature of the heat transfer surface, and can thus accumulate here. This type of fouling is thus temperature dependent, and its accumulation can be predicted from vapour-solid or liquid-solid equilibrium evaluation of the stream components.

2.2.2 Fouling sequence and mechanics

In heat exchangers several types of fouling will usually accumulate simultaneously. Each type of fouling will affect the formation rate of other types of fouling, and it is easy to see that the modeling of fouling becomes a very difficult task, as it is a combination of advanced heat and mass transfer at work. The sequence at which fouling form can however be divided into five categories that can be observed in heat exchangers [19]:

1. **Initiation:** The delayed onset in fouling observed in heat exchangers
2. **Transport:** The mechanisms which transports particles or components from the bulk fluid to the surface
3. **Attachment:** The rate at which the fouling will attach to the surface
4. **Removal:** The rate at which the fouling layer will be removed from the surface
5. **Aging:** the development of the already formed fouling layer over time

These five sequences are related to the velocity boundary layer, the thermal boundary layer, the concentration boundary layer and the forces exerted on a particle on a by a fluid. As the interface to the wall is paramount in both heat transfer and fouling in heat exchangers, understanding the boundary layers will make it easier to understand the sequence of fouling.

2.2.2.1 The laminar velocity boundary layer

The velocity boundary layer is causes by share stress between laminar layers in viscous flow close to a surface. The velocity in the x-direction $u(y)$ will increase with the distance from the surface due to variation in shear stress between the laminar layers, while velocity in the y-direction $v(y)$ is close to zero due to laminar flow. When assuming fully developed conditions along a flat surface, these velocities will not vary in the x-direction. The boundary layer thickness δ is defined as the distance from the wall, where the boundary layer velocity is equal to 99% of the bulk fluid velocity.

$$\delta = \frac{u_x}{u_\infty} = 0{,}99 \tag{8}$$

Given the no-slip-at-wall condition, meaning that the velocity in the x-direction is zero at the wall surface, the shear stress at the wall will be the product of the viscosity and the velocity gradient:

$$\tau_w = \mu \frac{\partial u}{\partial y}\bigg|_{y=0} \tag{9}$$

Increasing the bulk fluid velocity will decrease the boundary layer thickness, and increase the boundary layer velocity gradient $\frac{\partial u}{\partial y}$. From equation (9), a larger velocity gradient leads to higher shear stress at the wall. Even in turbulent flow, a smaller region close to the wall will have laminar flow, and is referred to as the viscous sublayer [13].

2.2.2.2 Thermal Boundary Layer

The thermal boundary layer will exist close to the wall, similar to the way the velocity boundary layer does. The thermal boundary layer thickness is defines as the distance from the surface where the temperature difference to the wall is 99% of the temperature difference between the bulk temperature of the stream and the wall.

$$\delta_T = \frac{T_s - T}{T_s - T_\infty} = 0{,}99 \tag{10}$$

At the heat transfer surface there is no fluid motion, and heat transfer only occurs though conduction [13]. The relation between the thermal boundary layer and the convection heat transfer coefficient may be demonstrated by applying Fourier's law at the wall to find the heat flux, and combine the result with Newton's law of cooling:

$$q" = -k_f \frac{\partial T}{\partial y}\bigg|_{y=0} \tag{11}$$

$$h = \frac{-k_f \frac{\partial T}{\partial y}\Big|_{y=0}}{(T_s - T_\infty)} \tag{12}$$

Increasing the bulk fluid velocity will decrease the thermal boundary layer thickness, and increase the boundary layer temperature gradient. A larger temperature gradient increases the heat flux at the wall, and leads to a higher heat transfer coefficient.

2.2.2.3 The concentration boundary layer

Fouling is a form of mass transfer and will cause a change in concentration C of a species A in the fluid. The concentration boundary layer builds up due to the concentration difference at the surface $C_{A,s}$ and in the bulk fluid $C_{A,\infty}$, with a thickness defined as:

$$\delta_t = \frac{C_{A,s} - C_A}{C_{A,s} - C_{A,\infty}} = 0{,}99 \tag{13}$$

The mass transfer rate at the wall is described in a similar way as heat transfer, with a mass transfer coefficient as a function of the concentration gradient. The mass transfer only occurs through diffusion at the wall interface, while higher up in the concentration boundary layer the mass transfer is both taking place through diffusion and fluid motion. Increasing the bulk fluid velocity will decrease the concentration boundary layer thickness, and increase the boundary layer concentration gradient at the wall. A larger concentration gradient increases the mass transfer rate at the wall [13].

2.2.2.4 Forces exerted on a particle in laminar flow

The boundary layer theories describe the conditions close to the wall which affects fouling. A model for forces exerted on a particle in contact with a surface is however also useful to fully understand the sequence of fouling.

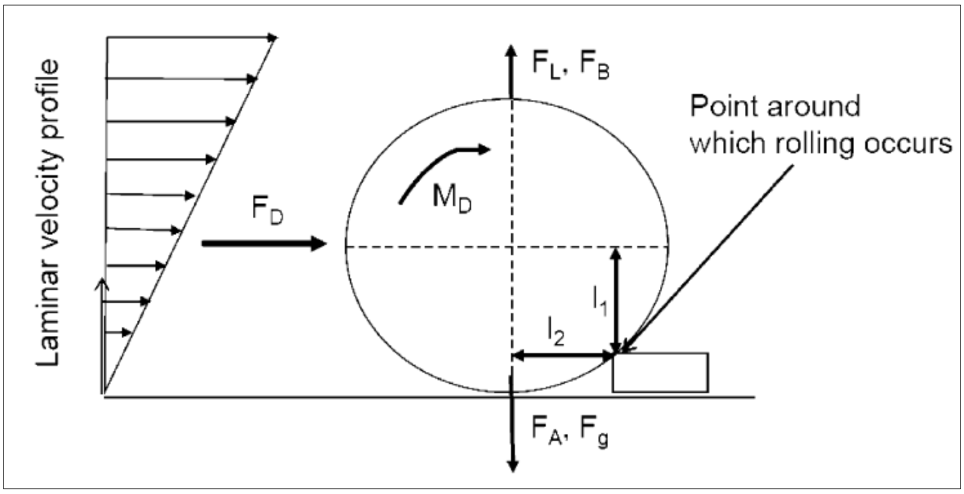

Figure 6: The forces exerted on a particle attached to a wall in laminar flow

Figure 6 shows a particle sticking a surface with in laminar boundary layer region. The forces acting on the particle center of diameter (D) are [20]: Adhesion force (F_A), drag force(F_D), lift force (F_L) gravity (F_g), and buoyancy (F_B). There is also momentum acting on the particle (M_D) due to surface stresses.

Buoyancy and gravity will stay constant for a particle of a constant size. Drag force, lift force and momentum will all increase with increased fluid velocity in the boundary layer. Adhesion forces will have a maximum value for which the particle will stick to the wall and it can be shown that the criteria for particle removal through lifting, sliding or rolling are [20]:

Lifting:

$$F_L + F_B \geq F_A + F_g \tag{14}$$

Sliding:

$$F_D \leq f\left(\left(F_A + F_g\right) - \left(F_L + F_B\right)\right) \tag{15}$$

Rolling:

$$M_D + F_D \cdot l_1 + (F_L + F_B) \cdot l_2 \geq \left(F_A + F_g\right) \cdot l_2 \tag{16}$$

Exact solutions for F_D, F_L, and M_D can be found in [20]. These forces are all linearly dependant on boundary layer velocity at the particle center, viscosity and particle diamater. The most difficult force to quantity in equation (14) to (16) is the adhesion force F_A of the fouling particle to the surface.

2.2.3 The effect of stream parameters on fouling

For a foreign particle to attach to the heat transfer surface this particle must first be transported into the heat exchanger with the bulk stream, pass the laminar viscous sublayer and attach to the wall. As there is no velocity normal to the bulk velocity in the viscous sublayer, the transportation across this layer must be either by gravitational forces, inertia, diffusion, electrophoresis, thermophoresis, or random Brownian motion [15]. When the particle comes in contact with the surface, it will stay there depending on the force balance on the particle as presented in the previous section.

In terms of initiation of fouling and attachment to the wall, the surface properties of the wall play an important role. Increased surface roughness decreases the initiation period for all types of fouling except particulate fouling, which does not seem to have an initiation period [16]. Reducing the surface roughness reduces the amount of nucleation points and reduces the types of fouling dependent on these. Surface coating may also improve fouling resistance. Coating that tends not to stick to the fouling particle will reduce the adhesive force F_A from equation (14) to (16) and lower the velocities needed for particle removal.

Increasing the velocity reduces all boundary layer thicknesses, and increases the removal forces on a particle attached to the heat transfer surface. At the same time the concentration boundary layer will decrease, and fouling requiring constant mass transfer from the bulk stream, such as biofouling, can actually increase with smaller increments of velocity. This is due to increased supply of nutrients to the biofilm through diffusion [18]. H velocities will however in general reduce fouling due to increased removal of particles attached to the wall. Increasing the velocity will also reduce particulate fouling as forces of turbulent diffusion outweigh the gravitational forces that cause settling.

Bulk temperature of the stream does not necessarily say anything about the extent of fouling; it is the difference between the wall temperature and the bulk temperature that matters, and mainly the wall temperature itself. If the wall temperature is within fouling conditions, a portion of the temperature boundary layer will also be within this condition, and a particle may form (i.e. hydrate or salt). When the particle is created it will either be mixed in with the bulk stream and dissolve, or travel to the heat transfer surface and stick as explained earlier in this section. As this type of fouling deposit is temperature dependent, it is also useful to evaluate the temperature effect of the particle when it is attached to the heat transfer surface. The particle itself has a thermal conductivity and affects the temperature gradient. Precipitation or solidification is dependent on nucleation points, and requires a solid particle in the thermal boundary layer to grow on, for initiation. Microscopic particles will always be present, either due to other types of fouling or imperfect filtration. Solidification and precipitation may also form directly on the heat transfer surface [20]. Such formation is dependent on nucleation points on the heat transfer surface, which is provided through surface roughness.

If the stream is heated, the viscosity typically decreases significantly which is reducing the drag force on suspended particles in the stream. This allows for smaller particles to settle due to increased terminal velocities and may form sedimentation.

2.2.4 The development of fouling rate with time

The heat transfer surface roughness and the stream velocity and temperature will affect fouling formation, as presented in section 2.2.3. These variables may be modified in order to minimize fouling, and general guidelines exist for how this should be done in the design of specific types of heat exchangers. But these are not exact design constraints, only guidelines, and do not offer enough certainty to eliminate fouling alone.

The fouling layer thickness will not necessarily develop linearly with time, as the fouling layer affects the stream properties which again affect the fouling rate. The total amount of fouling being added to a surface at any given time is the sum of the attachment rate minus the removal rate.

$$Fouling(t) = Attatchment(t) - Removal(t)$$

The development of a fouling layer is a transient process that is difficult to predict. The development of fouling with time will follow either of four typical patterns, as shown in Figure 7.

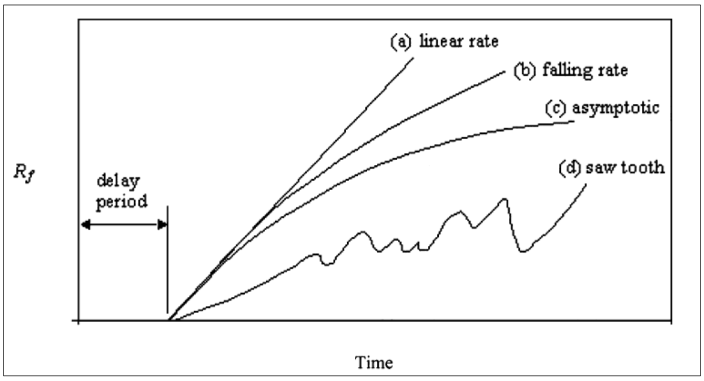

Figure 7: The transient development of fouling resistance

2.2.5 The fouling resistance method and its weaknesses

Fouling reduces the overall heat transfer coefficient as explained in the beginning of this chapter. The area of a heat exchanger will during design have to be increased to compensate for a future reduction of the overall heat transfer coefficient.

Fouling resistance is a specified thermal resistance that can be used in the calculation of the overall heat transfer coefficient, to include the maximum allowable thermal resistance presented by a layer of fouling on the heat transfer surface. By designing the heat exchanger for the situation with the lowest overall heat transfer coefficient, one will end up with a design with an excessive heat transfer area for when the heat transfer coefficient is high, and just enough area when the heat transfer coefficient is low. These two situations are often referred to as clean and dirty; dirty referring to when the heat transfer surface is fouled and the heat transfer coefficient is at its lowest.

The excessive area allows for a fouling layer to develop over a period of time, and still have the heat exchanger operate with the desired heat transfer rate. The heat exchanger will begin to operate with a lower heat transfer rate than what is required when the actual resistance of the fouling layer is equal to the fouling resistance used in the design calculations. At this point in time the heat exchanger must be cleaned.

Fouling resistances for heat exchanger can be found in TEMA [17] for shell and tube heat exchangers, while a variety of fouling resistances for plate heat exchanger can be found in literature [16] [21].The fouling resistance will differ for different types of streams.

The fouling resistance can be seen as a safety margin for fouling that will development during operation. The use of a fouling resistance during design does not take the transient behaviour of fouling into consideration. One can simply not know from the calculation whether the heat exchanger will reach fully fouled conditions within a week, a month or years. The asymptotic or saw tooth type fouling layer developments from Figure 7 can potentially have the heat exchanger operate for a long period of time without the need for cleaning. The falling rate or linear rate fouling will require cleaning sooner.

For a heat exchanger installed for heat recovery, excessive area only means higher performance. For a heat exchanger installed for cooling or heating on the other hand, excessive area could mean overheating or overcooling of the process stream. In order to compensate for this, the heat flux of the heat exchanger must be controlled through the cooling or heating medium. This causes a difference in clean and dirty operation. If the heat exchanger is controlled through flow rate, the flow rate during clean operation will be lower than for dirty operation. If the heat exchanger is designed for control through inlet temperature, the temperature will be different for clean and dirty operation. The use of large fouling resistances can lead the clean operation of the heat exchanger to actually increase the fouling rate due to flow rates and temperatures that accelerate fouling

2.2.6 Fouling reduction

Fouling can be prevented in other means than surface coating or the consideration of the flow velocities and temperatures in the heat exchanger design. Continuous injection of chemical additives can reduce the fouling potential of the stream and extend the operation time between cleaning intervals for the heat exchanger.

The chemicals used will not always be recoverable, meaning the new chemicals will have to be supplied to the stream continuously. This will increase OPEX of the plant. For chemicals that are recoverable, some processing is needed for its recovery, and both CAPEX and OPEX will increase with the use of such systems. If chemical additives sufficiently increase the reliability of the heat exchanger, it will be justifiable to invest in such systems. Chemical additives can be harmful if exposed to nature, and their use will in open systems be restricted by environmental concern and regulatory limits.

Another way to reduce fouling is to process the stream upstream of the heat exchanger to remove the stream components that cause fouling. The simpler form of stream treatment includes coarse mechanical filtration in strainers. The more advanced stream treatments includes processing such as water desalination.

2.2.7 Fouling mitigation

Cleaning techniques for fouling can either be mechanical or chemical. Cleaning will often require the heat exchanger to be taken offline, and removed from plant location. Chemical cleaning involves running a closed loop of strong chemicals through the heat exchanger that will destroy the fouling layer. Mechanical cleaning techniques involve opening the heat exchanger and physically scrape of the fouling layer using appropriate equipment. Mechanical cleaning requires that the heat transfer surface is accessible, which will have to be accommodated in the heat exchanger design.

There are methods for removing fouling in the heat exchangers without shutting down the process or removing the heat exchanger from its process location. These methods are:

- Online mechanical or chemical cleaning techniques
- Increasing/reversing flow
- Increasing/Lowering temperature

Online cleaning techniques include shock dosing of chemicals and installation of mechanical systems using brushes, scrapes etc. An extensive section heat exchanger cleaning techniques can be found in [21].

A simpler way of reducing fouling while having the heat exchanger online is to increase the flow rate for a short period of time, or increase the temperature for a short period of time, alternatively reverse flow. The effect of such techniques will be dependent on the fouling type present.

2.3 Heat exchangers

Heat exchangers can be classified according to their flow arrangement, construction type and type of operation, which are only a few of many ways. Heat exchanger is the description of the heat transfer components itself while other names such as cooler, reboiler and recuperatorare are descriptions of a heat exchanger in a specific type of operation.

2.3.1 Classification according to flow arrangements

The different flow arrangements of heat exchangers will affect the heat exchange effectiveness, and thus its size. Mechanical and hydraulic considerations may favor different arrangements as well. Heat exchanger effectiveness is defined as the actual heat transfer in a heat exchanger divided by the maximum possible heat transfer. The maximum possible heat transfer is the heat transfer that would occur with an infinitely large area

Flow arrangements in a heat exchanger can be divided in three main groups:

- Counterflow
- Parallel-flow
- Crossflow

Most heat exchangers will have flow arrangements that are similar but not identical to these idealized arrangements. Some heat exchanger designs combine different flow arrangements, such as split flow shell and tube heat exchangers.

When talking about heat exchangers one differs between single and multipass streams. A single pass stream flows through the full length of the heat exchanger one single time, while a multipass stream will at the end of the heat exchanger be redirected back a second time through the full length of the heat exchanger. At the end of the heat exchanger a multipass stream will either have to be redirected in a u-tube channel which avoids mixing or in an open type header which allows the streams from different passages to mix before being redistributed into the second flow passage. Header design can be difficult, and multipassing of condensing or two-phase streams should be done with care. The danger of separation in the header, and maldistribution of liquids in the flow channels makes u-tubes the preferred choice for such operations.

The mean temperature difference is larger for counterflow than for parallel-flow heat exchangers, and increasingly so for larger heat exchanger effectiveness [15]

2.3.1.1 Counterflow

Counterflow arrangement is when the hot and the cold stream are flowing in the exact opposite directions along each side of the heat transfer surface.

Counterflow arrangement maximizes the mean temperature difference of the heat transfer surface, and allows for the greatest temperature change in both streams compared to any other flow arrangement. The benefits of counterflow arrangement include:

- The possibility to achieve very high effectiveness
- Less thermal stress since the maximum temperature difference across the wall of the heat transfer area is the lowest in counterflow heat exchangers.

Most commercial heat exchangers not designed for extremely high effectiveness have a different flow arrangement than true counterflow due to the importance of other design factors. []

2.3.1.2 Parallel-flow

Parallel flow arrangement is when the cold and the hot stream are flowing in the same direction along each side of the heat transfer surface.

Some of the benefits of using parallel-flow arrangement include:

- The outlet temperature of the cold stream never exceeds the lowest temperature of the hot stream.
- The variation of the temperature along the heat transfer surface, in the flow direction, will be lower with the parallel-flow arrangement than with the counterflow arrangement.*
- The maximum wall temperature will be lower, and the minimum wall temperature higher, in a parallel-flow heat exchanger compared to that in a counterflow heat exchanger.*

This type of flow arrangement gives the lowest heat transfer effectiveness of the three arrangements. However, for low heat transfer effectiveness, the area penalty of using parallel-flow arrangement instead of counterflow arrangement is small. [15]

2.3.1.3 Cross-flow

Cross-flow heat exchanger arrangement is when the hot and cold stream passes each other in perpendicular directions on each side of the heat transfer surface.

The benefits of crossflow arrangement include:

- Simplified header design in compact type heat exchangers.
- Improved heat transfer coefficient for fluid flow normal through banks of tubes due increased turbulence and the destruction of the boundary layers

The effectiveness of a crossflow heat exchanger is, not very surprisingly, somewhere in between that of a parallel-flow and a counterflow heat exchanger.

2.3.2 Classification according to cooling arrangement

A process cooler using seawater as a heat sink can be classified according to the cooling medium flow mechanism and the allocation of the seawater side [22]. The types of process cooler arrangements are:

- Passive cooling
- Active cooling
    - Direct cooling
    - Indirect cooling

2.3.2.1 Passive cooling

A passive cooler is any cooler that does not have mechanical driven flow on the cold side of the heat exchanger; the flow is driven by natural convection. A subsea passive cooler is a tubular type heat exchanger. The tubes are suspended in the seawater and the process stream flows inside. The hot outer surface of the tubes is in direct contact with the seawater and cools by natural convection. Passive coolers offer no controllability of heat transfer rate and should only be applied to systems with either no temperature constraints or with sufficient margins to the temperature constraints. The heat transfer rate of a passive cooler will vary with the cold side ambient temperature, natural currents, fouling layer thickness and internal flow rate during operation.

2.3.2.2 Active cooling

An active cooler is any heat exchanger installed for cooling, where the cooling medium flow rate or temperature is adjustable. The purpose of adjusting inlet temperature and flow is to control the heat exchanger heat transfer rate. Controlling the heat exchanger heat transfer rate is necessary if the process stream has a target temperature and little room for deviance from this. It is then required to compensate for smaller changes in process stream flow rate and for the reduction of the overall heat transfer coefficient during operation. If the cooling medium is seawater or fresh water that passes through the system one time only, the system is referred to as direct cooling, shown in Figure 8. If the cooling medium is a liquid circulating in a closed loop operating between a hot and a cold heat reservoir, the system is referred to as indirect cooling, shown in Figure 9.

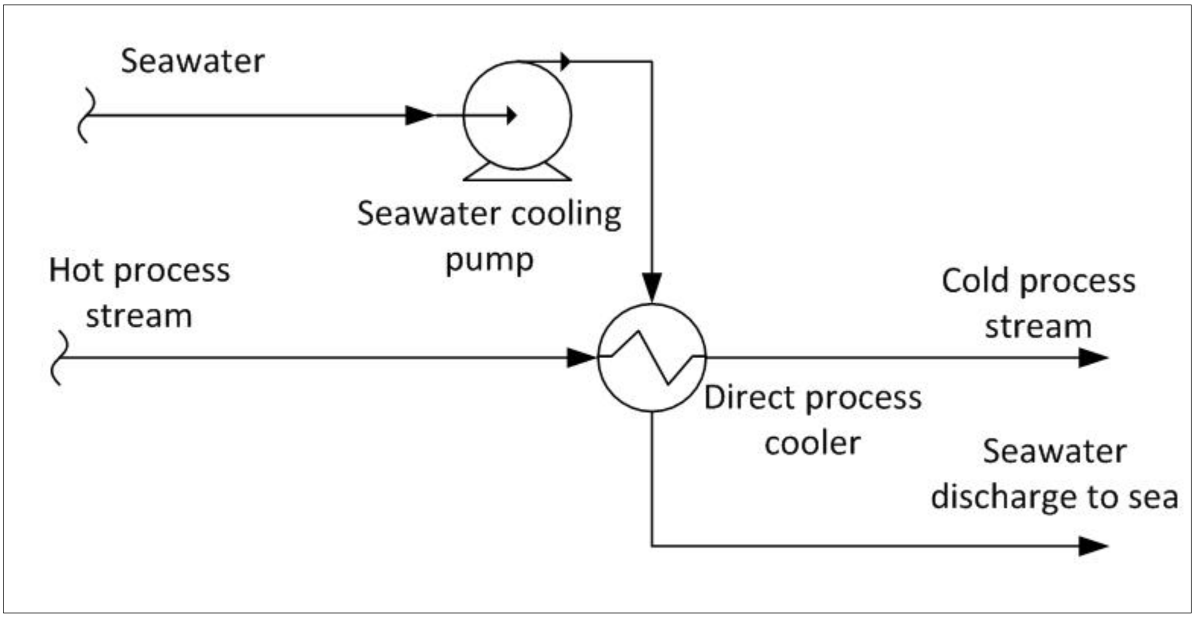

Figure 8: Direct cooling flow schematic

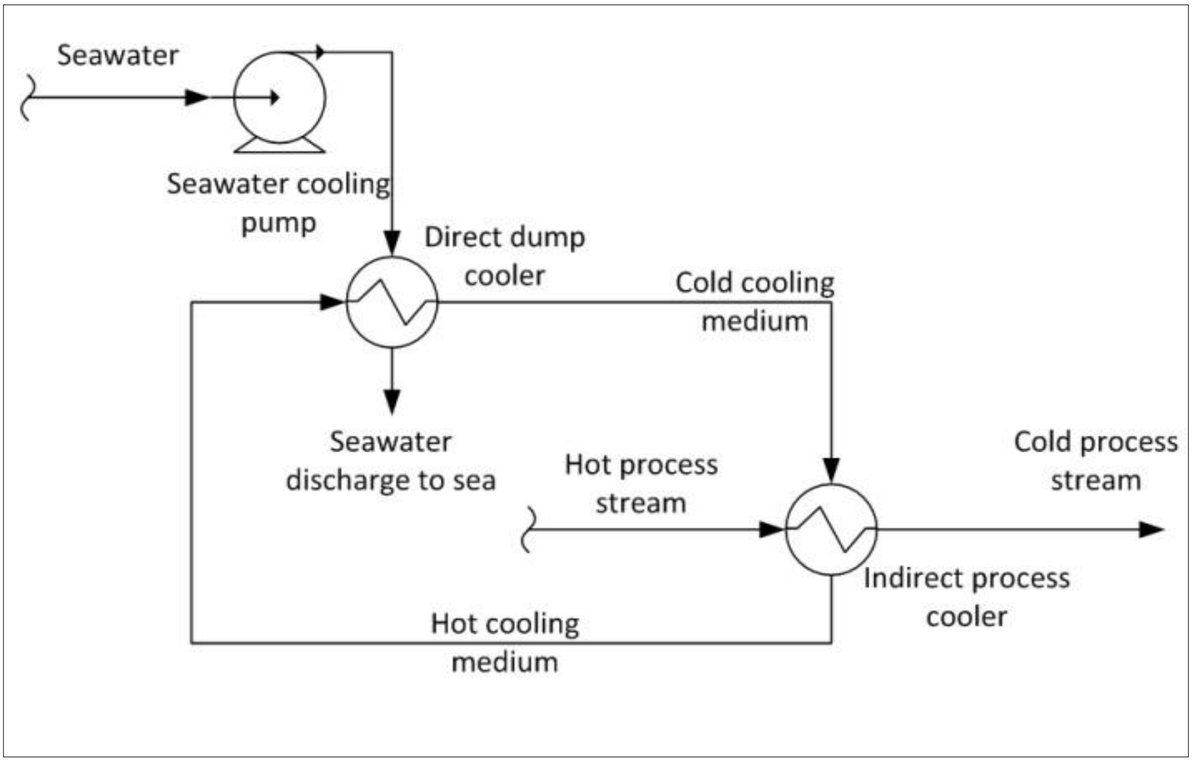

Figure 9: Indirect cooling flow schematic

2.3.3 Classification according to construction

It is possible to classify heat exchangers in three main groups according to construction [15]:

- Tubular heat exchangers
- Plate type heat exchangers
- Regenerative heat exchangers

Regenerative heat exchangers will always have a small degree of leakage between streams. These types of heat exchangers are frequently used in ventilation systems for energy recovery of used air, and should not be used in oil and gas processing where hydrocarbon leakage is a safety issue. Using a regenerative heat exchanger between hazardous and non-hazardous streams is not an option, and this heat exchanger type is therefore excluded from further evaluation. A list of the different types of tubular and plate heat exchangers [15] are listed in Table 2.

Table 2: List of heat exchangers

Tubular heat exchangers include:	Plate type heat exchangers include:
Shell and tube Double pipe Spiral tube Pipe coils	Gasket Welded Brazed Plate-fin Spiral plate Plate coil Diffusion bonded plate fin Printed circuit

Both shell and tube and plate heat exchangers can be improved with the use of extended surfaces such as fins and internal enhancements. For plate heat exchangers and double pipe heat exchangers fins also function as structural reinforcements and usually increases the maximum design pressure.

Shell and tube and tube heat exchangers are the most frequently used heat exchangers in the process industry [15]. These types of heat exchangers are highly customizable to specific process needs and has an extensive track record of operation accompanied by rigid codes and design standards for optimal design. The governing standard for shell and tube design is that of the Tubular Exchanger Manufacturers Association, referred to as TEMA [17]. In addition there is a great deal of support literature on the design of shell and tube heat exchangers.

Plate heat exchangers are the most compact type of heat exchangers. These types of heat exchangers also tend to have less fouling due to much higher wall shear stresses than shell and tube heat exchangers. However the most compact types of plate heat exchangers are more prone to clogging. The plate heat exchangers designed for high pressures are not possible to open and cleaning must be done with chemicals. Shell and tube heat exchangers on the other hand may be designed for high pressure without affecting accessibility for mechanical cleaning.

Shell and tube heat exchangers can be designed for very high pressures. The TEMA defines the shell and tube heat exchangers commercially available. The maximum design pressure of shell and tube heat exchangers depends on size and shape, but can reach very high pressures due to the tubular type of construction.

Plate heat Exchangers are more sensitive to pressure. The commercially available plate heat exchangers with maximum design pressures are listed in Table 3.

Table 3: List of commercially available plate heat exchangers

Heat Exchanger Type	Compactness (m^2/m^3)	Temperature Range (°C)	Maximum Pressure (Bara)
Plate and frame	< 200	-35 to +200	25
Partially welded plate	< 200	-35 to +200	25
Fully welded plate	< 200	-50 to +350	40
Brazed plate	< 200	-195 to +220	30
Platular plate	200	< +700	40
Compablock plate	< 300	< 300	32
Packinox plate	< 300	-200 to +700	300
Spiral	< 200	< 400	25
Brazed plate-fin	800-1500	-190 to +650	90
Diffusion bonded plate-fin	700-800	< 500	> 200
Printed-circuit	200-5000	-200 to +900	>400
Polymer	450	< 150	6
Plate and shell	-	< 400	70

3 System design

A subsea process cooler is transporting heat from the process stream over to the seawater, and can be divided into two sides; the process side and the seawater side. All subsea cooling systems will have a process side and a seawater side. In this chapter these sides are analyzed separately to clearly identify the cause and effect of design problems for each side.

3.1 The process side

There are two possible arrangements of an inlet separator and a cooler in SSBI systems. The process cooler can be placed upstream or downstream of the inlet separator, see Figure 10. The two arrangements have varying process side stream parameters that will affect the heat exchanger design. The seawater side of the heat exchanger will remain unchanged for the two process cooler process locations.

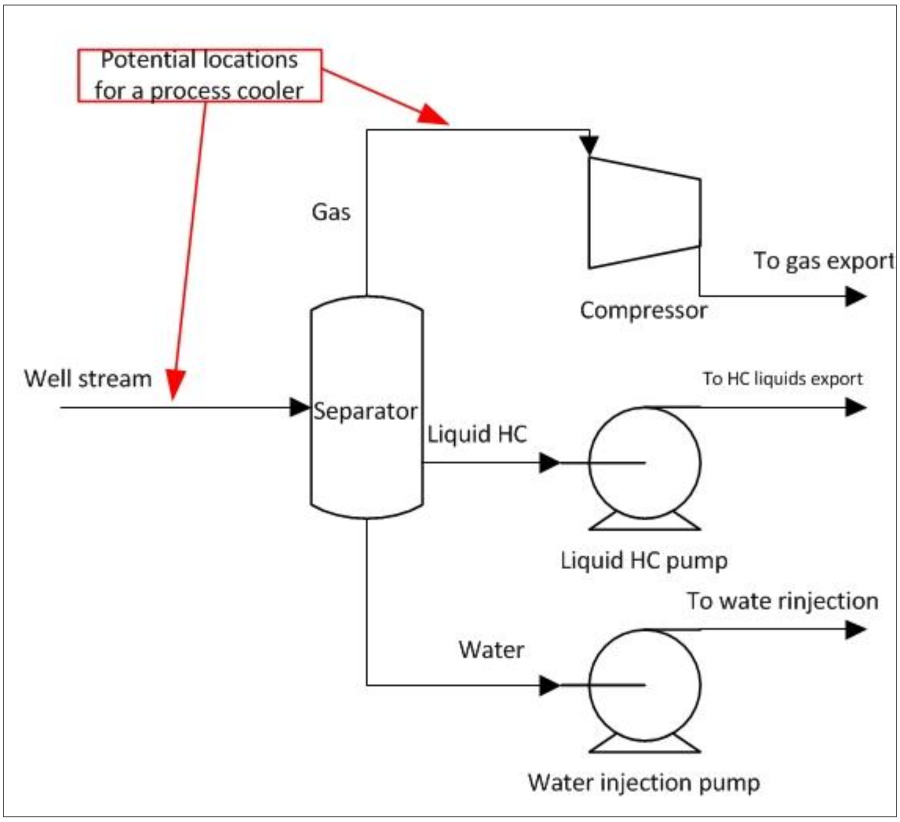

Figure 10: The two different locations for a process cooler

The stream parameters that affect the heat exchanger design and which will vary between these two process locations are:

- Stream composition and phase distribution
- Stream fouling potential
- Stream enthalpy change for the same temperature change (heat transfer rate)

How these stream parameters and the system design is affected by placing the cooling upstream or downstream of an inlet separator is summarized in Table 4

Table 4: The effect of process cooler location on selected parameters

	Upstream Cooler	Downstream Cooler
Stream fouling potential:	High	Low
Flow conditions (in/out):	Multiphase/Multiphase	Single-phase/Multiphase
Heat transfer rate:	Highest	Lowest
Minimum number of separators needed:	1	2
Applicable for liquid dominated systems:	No	Yes
Applicable for gas dominated systems:	Yes	Yes

3.1.1 Stream composition and phase distribution

The wellstream gas will be saturated with hydrocarbons and water, and will thus condense upon cooling. This requires a separator to always be installed between the process cooler and the compressor. Placing the process cooler upstream of the inlet separator fulfills this requirement. Placing the process cooler downstream of the inlet separator, and directly upstream of the compressor will require the installation of a second separator between the process cooler and the compressor, and is the reason the system design is altered between the two process cooler locations, seeFigure 11

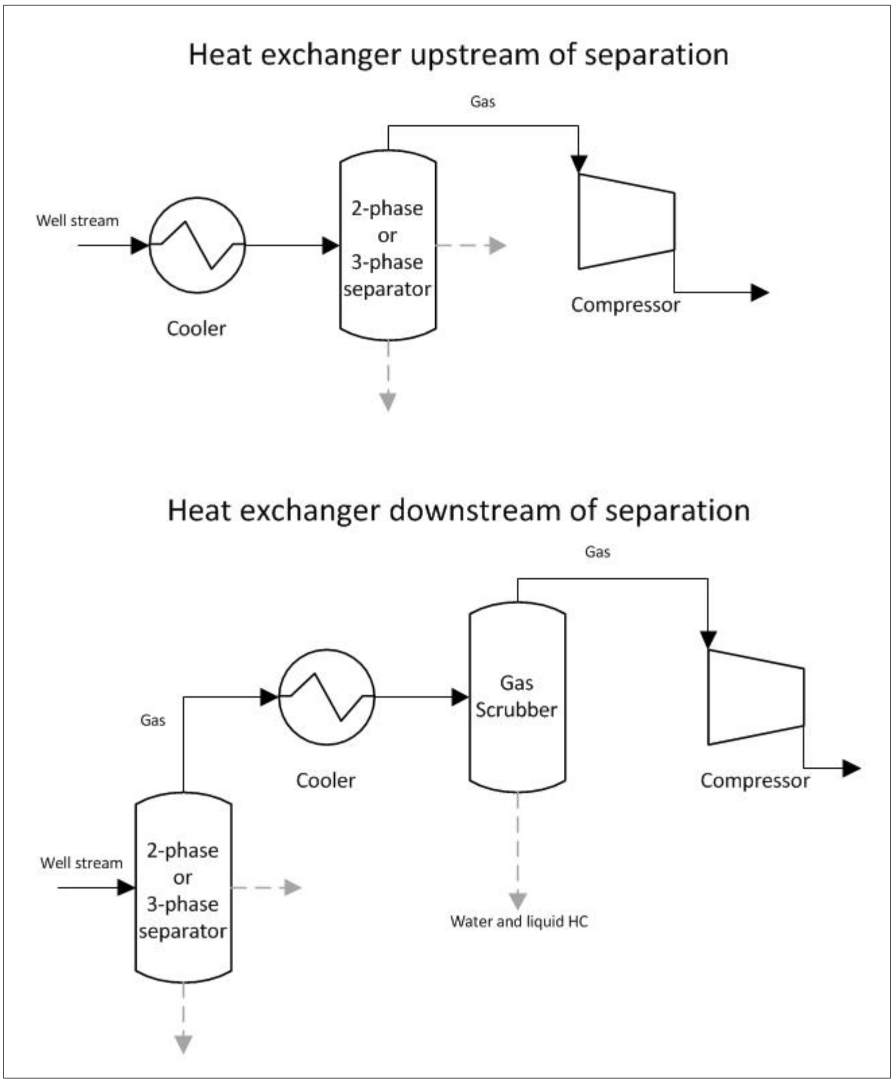

Figure 11: Variations between having the process cooler upstream and downstream of the inlet separator

The multiphase wellstream will always contain some sand. Having the process cooler upstream of an inlet separator requires it to handle such stream impurities but will also reduce the minimum number of components needed in the SSBI system. With two phases in the inlet of a heat exchanger comes the danger of maldistribution of liquid. Since the liquid heat transfer coefficient is much higher than the gas heat transfer coefficient [16], maldistribution could lead to local sub-cooling in the heat exchanger, as well as complicating the overall control of the heat exchanger that is based on the bulk outlet temperature. If the liquid content is high, slugging of the wellstream can be expected, and precision cooling will be difficult due to the drastic dynamic changes in the system.

Having the process cooler downstream of an inlet separator will require an extra separator, but will put less strain on the heat exchanger since sand and slugging problems can be handled in the inlet separator.

The placement of the process cooler and the main goal of the SSBI system will affect the separation sequence in the design of the SSBI system. Separation of gas, liquid hydrocarbons and water in the inlet separator can either be performed in a single single 3-phase separator or a series of two 2-phase separators. The separator between the process cooler and the compressor can also either be a 2-phase or a 3-phase separator. A number of combinations between separation sequence and process cooler location can be chosen. These combinations are not discussed in this thesis, as the choice of the separation sequence does not affect the process stream properties in the process cooler, which is only affected by the placement of the process cooler in the process relative to the inlet separator that is always included in SSBI systems.

If water is to be injected downstream of the process cooler, hydrate inhibitors or other chemicals not allowed in the reservoir cannot be used, as presented in section 1.1.2. A SSBI system design for maximum water removal, with the heat exchanger downstream of separation, and with a series of two phase separation for liquids is shown in Figure 12.

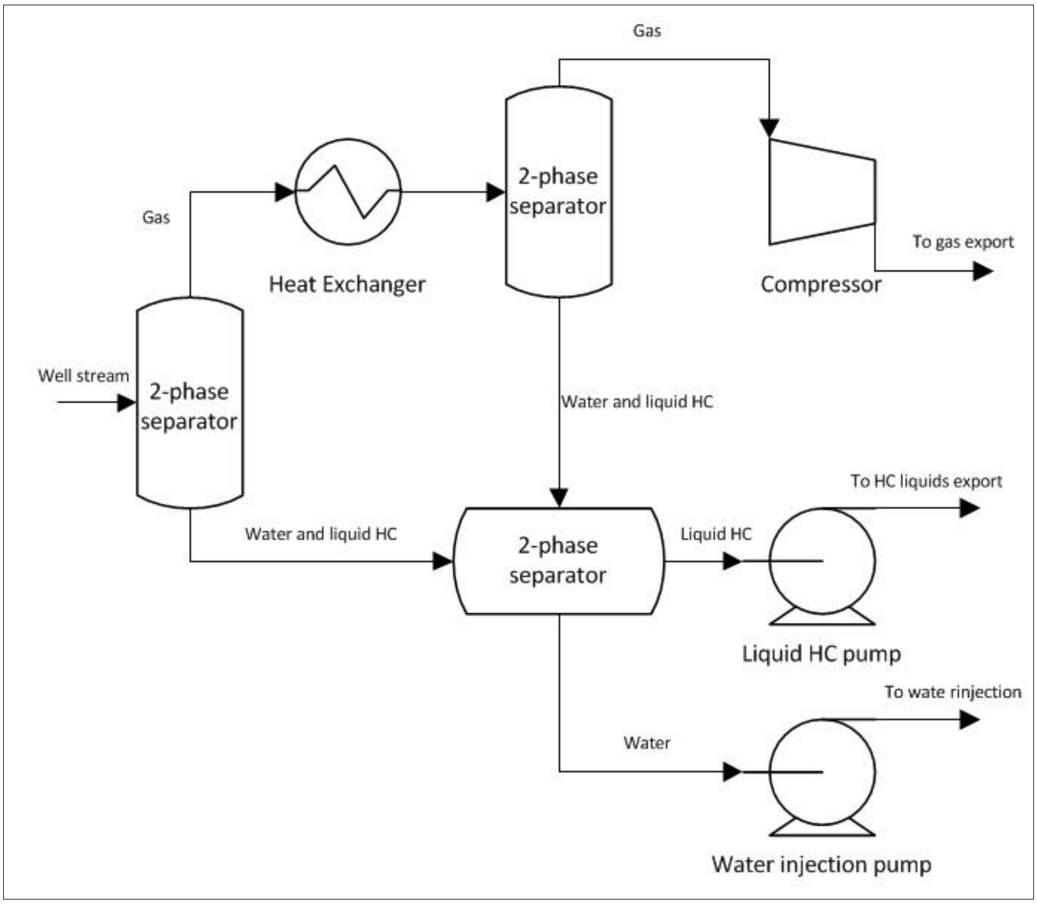

Figure 12: System design for a SSBI system where maximum water removal is the main goal

Separation sequence will affect the system total size, and should be evaluated if total system compactness is of main focus.

3.1.2 Process side fouling

Fouling in a subsea process cooler will be limited to the types listed in Table 5 on the process side of the heat exchanger [15].

Table 5: Process side fouling

Fouling type	Upstream cooler location	Downstream cooler location
Precipitation:	Scale from normal solubility salts in the produced water and the formation water	Scale from normal solubility salts from produced water
Particulate:	Large particles of sand and dirt will form sedimentation	Small particles of sand and dirt suspended in the gas may form sedimentation
Chemical reaction:	No	No
Solidification:	Hydrate formation, wax and asphaltene deposition	Hydrate formation
Corrosion:	CO2 and H2S together with water promotes corrosion	CO2 and H2S together with water promotes corrosion
Biological:	No	No

The difference between having the process cooler upstream and downstream of the inlet separator will be dependent on stream composition and liquid content. With high water contents and a wellstream containing large amounts of oil, the fouling potential upstream of the inlet separator will be much higher than downstream of the inlet separator. The difference between the two process cooler locations will be the least when the wellstream is mostly gas.

All types of fouling will be worsened if the heat exchanger is placed upstream of the inlet separator. Sedimentation and wax formation will be more severe at this location if heavy hydrocarbon liquids are present, which also will include more sand due to higher viscosity.

3.1.3 Process side fouling control

Ways of controlling fouling on the process side are listed in Table 6.

Table 6: Process side fouling

Fouling type	Control method
Precipitation:	Scale inhibitor, temperature control
Particulate:	**Upstream cooler location:** Implementing sand handling in the heat exchanger design **Downstream cooler location:** Using strainers and filter
Solidification:	Hydrate Inhibitor, wax and asphaltene inhibitors, or avoiding formation temperatures
Corrosion:	Selection of corrosion resistant materials, use corrosion inhibitor or surface coating

A complete composition analysis of the wellstream must be done to quantify the extent of corrosion, precipitation and wax formation. Hydrate formation will always be a major threat to hydrocarbon systems with water, and water will always be present in the wellstream [1].

Having the process cooler after the inlet separator will greatly reduce, and possibly eliminate, the extent of wax and asphaltene deposition, which is mainly associated with the cooling of heavy hydrocarbon liquids [23]. Scaling will likely also be reduced by the downstream process cooler location as there will be less produced water. If these fouling mechanisms are low enough, the system could work without the use of scale inhibitor, and without taking wax and asphaltene considerations in the design.

Particulate fouling will be less in a gas stream due to a much lower viscosity than with a liquid stream. This implies that having the heat exchanger downstream of the separator reduces particulate fouling. Particles can be transported with the gas stream if they are small enough or the gas velocity is high enough. The transportation of particles can be controlled with good separator design and strainers. Maximum particle size transported into the heat exchanger can be minimized with the use of strainers and filters. Having strainers upstream of a heat exchanger that is placed upstream of separation will not be possible due to the coarse wellstream.

If operation in the hydrate region is required, hydrate formation is avoided with the use of a hydrate inhibitor. When the use of a hydrate inhibitor is excluded the temperature has to stay above the hydrate formation temperature for safe operation.

Hydrate inhibitors and chemicals not allowed in the water injection reservoir could potentially be used with water injection if a by-pass is installed, see Figure 13. This would however only be possible for shorter periods of time, and the liquids from the heat exchanger will then have to bypass the water injection and be exported together with the hydrocarbon liquids.

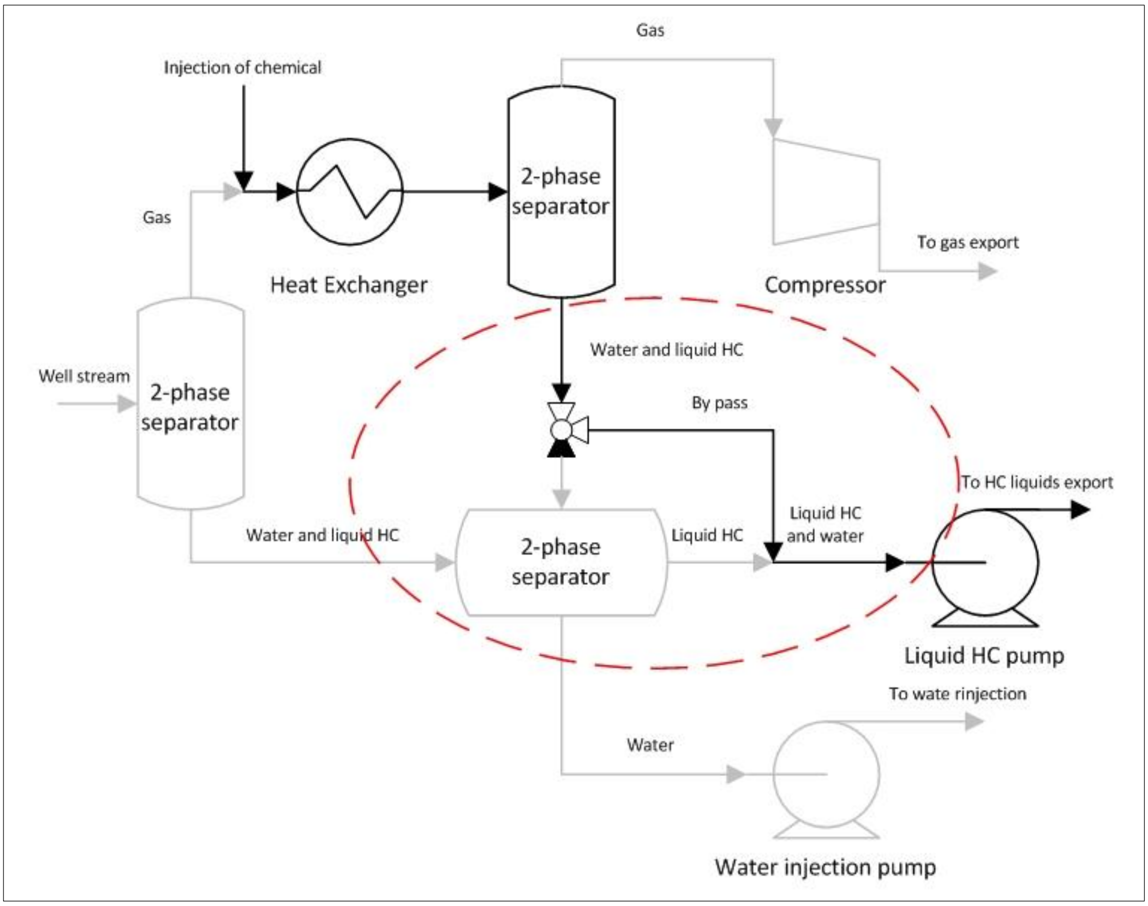

Figure 13: Bypass for use under the injection of chemicals not allowed in the water injection reservoir

3.1.3.1 The importance of the wall temperature: Hydrate formation discussion

When maximum cooling and water removal is wanted in the SSBI system, the process cooler outlet temperature may be close to the hydrate formation temperature. It is then likely that the wall temperature, and a portion of the thermal boundary layer, is within the hydrate formation temperature as discussed in section 2.2.33.1.3.1. This is the background for a discussion in the oil and gas industry on whether hydrates form in systems operating outside the hydrate region, but with low wall temperatures. A thorough and good evaluation of this phenomenon can be found by Nicholas et al.in [20]. Based on the particle dynamics briefly explained in section 2.2.2 and experimentation, Nicholas concluded that hydrates formed in the bulk part of the fluid, including the boundary layer, will not stick to the wall of pipes. It was however observed that hydrates that formed directly on the pipe wall had a stronger adhesive force and the need for a model for the formation and removal forces needed for such formation is not developed.

The potential failure modes of hydrates in the heat exchanger, and how they will affect operation is presented in Table 7.

Table 7: Hydrate formation failure modes

Failure mode	Problems	Counter argument	Potential solutions
Hydrates form on the heat transfer surface	Pressure drop will increase drastically and the heat exchanger will potentially clog	The hydrates will not attach to the heat exchanger surface [XXX]. If they do; a saw tooth fouling resistance will be observed as shown in fig. XXX.	Cycles of temperature raises in the heat exchanger can melt any formed hydrates. The heat transfer surface can be kept above the formation temperature.
Hydrate particles form in the heat exchanger bulk stream and deposits on downstream equipment	Equipment downstream of the heat exchanger such as valves or separators will clog	The small particles of hydrates will enter the bulk fluid and melt.[XXX]	Cycles of temperature raises in the streams can melted any accumulated hydrates. Keep the heat transfer surface temperature outside the hydrate region.

Norsok standard for process design P-100 also addresses the problems with low wall temperatures and hydrate formation:

"The hydrocarbon side skin temperature shall be kept above the hydrate formation temperature or wax appearance point or below any temperature that will cause other fouling (e.g. coking,). " sec. 5.4.1.6 Norsok, P-100

Even though Norsok is the process design standard for installations in the North Sea, including subsea, the oil companies will accept deviations if these can be presented with solid documentation. To conclude on hydrate formation problems in systems with low wall temperatures, a model describing formation, attachment, removal and melting is needed.

Hydrate formation can to some degree be controlled and minimized with the use of inhibitors and correct design, but the only way to guarantee that hydrate formation will not occur is to guarantee a heat transfer surface temperature above the hydrate formation temperature.

3.1.4 Fouling mitigation

There is no difference in the techniques applicable for fouling mitigation between the two process cooler locations.

3.1.5 Enthalpy change

There will be a difference in heat transfer rate between the two process cooler locations. Having the process cooler downstream the inlet separator will only cool the gas part of the wellstream, while having the process cooler upstream of the inlet separator will cool the gas, the liquid hydrocarbons and the water in the wellstream. The difference in heat transfer rate will consequently be directly dependent on the vapor fraction in the inlet separator.

There are no benefits with cooling the liquid, as the benefits of cooling in the SSBI system comes with cooling the gas, as presented in section 1.1.4. Cooling the entire wellstream actually makes the downstream separation of oil and water increasingly difficult due to the increased viscosity.

From a heat transfer rate perspective it is therefore only beneficial to have the heat exchanger upstream of separation in cases where the vapor fraction is high, and the process stream parameters will be similar for the two process cooler locations.

3.2 Seawater side

There are three different ways to use seawater as a heat sink with a subsea process cooler as presented in section 2.3.2. The process cooler can either be a passive cooler, a direct cooler or an indirect cooler. The main differences between these types of cooler arrangements are:

- Fouling control and mitigation
- Controllability
- Number of components

3.2.1 Seawater side fouling

Fouling on the seawater side of the subsea process cooler will be limited to the types listed in Table 8.

Table 8: Seawater side fouling

Fouling type	Description
Precipitation:	Inverse solubility salts in the seawater will form scale on the wall upon heating
Particulate:	Suspended particles in the seawater may form sedimentation
Chemical reaction:	No
Solidification:	No
Corrosion:	Seawater will promote corrosion
Biological:	Biological organisms in the seawater can form biological fouling

The seawater side will accumulate a fouling layer on the outer surface of the tubes in a passive cooler. In a direct cooler, the seawater side is enclosed, and the fouling will develop inside the heat exchanger.

The use of an indirect cooling loop will allocate the seawater side and the process side in two separate heat exchangers: a process cooler and a dump cooler, as shown in Figure 9. The dump cooler will have a cooling medium side and the seawater side, and thus experience the seawater side fouling. The process cooler will have the process side and a cooling medium side. As the cooling medium is a processed and chemically treated medium in a closed loop, it can be assumed that there is no fouling on the heating medium side of both heat exchangers [16].

Seawater fouling is usually quite heavy due to biological growth, and due to the corrosive nature of seawater. There is a lot of literature on the use of seawater in coolers. Seawater is used as a coolant on most oil and gas platforms and many power plants. The cooling water is however collected from relatively small depths for these systems.

It is difficult to find some conclusive documentation on whether biological fouling is a problem on deep waters or not. Scaling can be observed on the previously hot surfaces, but little biological fouling is found, when subsea equipment in the North Sea has been pulled up for repair or demobilization [8]. It could be that the dark conditions on the sea bottom keep the biological fouling to a minimal amount.

3.2.2 Seawater side fouling control

Fouling control on the seawater side will depend on whether the surface is within a confined space as in a direct cooler, or the out in the open as with a passive cooler.

A passive cooler has an open heat transfer surface, and a direct cooler has an enclosed heat transfer surface. Indirect cooling systems can have either an open or an enclosed surface on the seawater side, depending on whether the dump cooler is a passive cooler or a direct cooler.

The use of direct cooling allows treatment, to reduce fouling, of the controlled amount of seawater passing the heat transfer surface. This is not possible with a passive cooler, where essentially the entire ocean is in contact with the heat transfer surface. Figure 14 and Figure 15 shows the difference between an open and enclosed heat transfer surface.

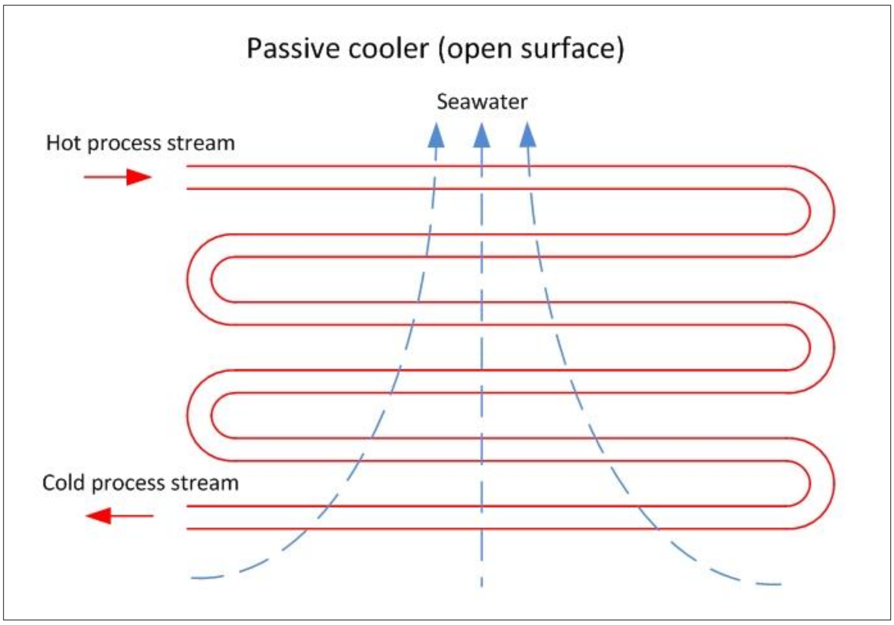

Figure 14: Passive cooler schematic

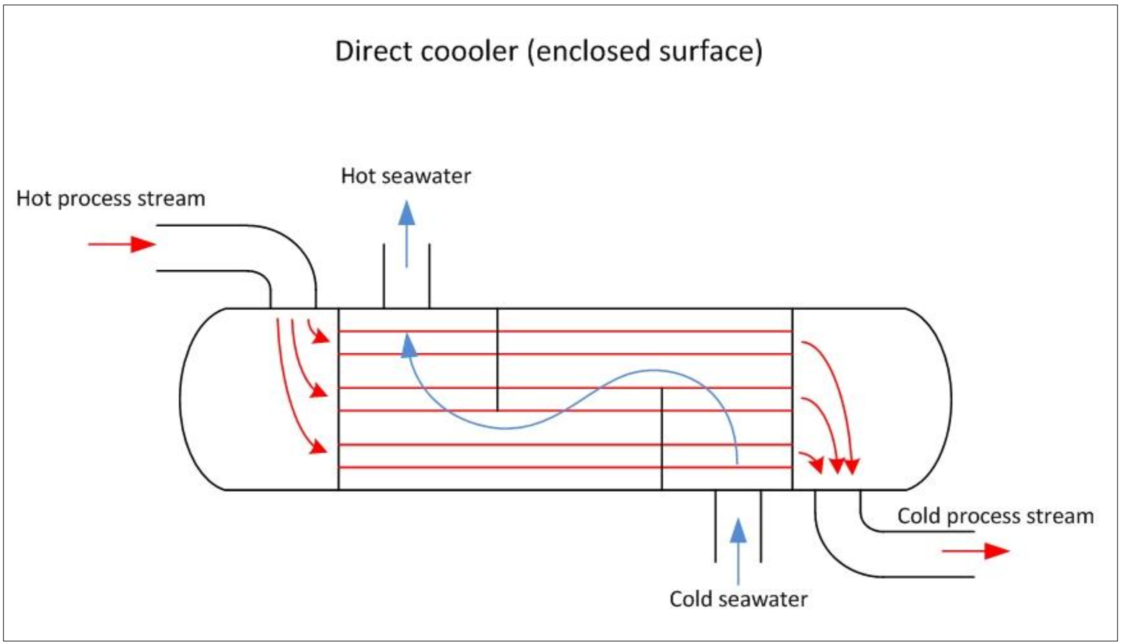

Figure 15: Schematic of direct cooling using a shell and tube heat exchanger

NORSOK states that the surface temperature of seawater heat exchangers should be kept below 60°C to minimize fouling. The general effects of stream temperature and velocity on fouling were discussed in section 2.2.3, and this will further be discussed in the next two sections. The general fouling control mechanisms that can be used for seawater heat exchangers are listed in Table 9.

Table 9: Fouling control

Fouling type	Description
Precipitation:	Keep the surface temperature below 60°C, surface coating, scale inhibitor
Particulate:	Filtration, surface coating
Corrosion:	Surface coating, material selection, sacrificial anode, corrosion inhibitor
Biological:	Biocides, surface coating

3.2.2.1 Open surface

Water treatment is not possible with an open surface, neither is the control of flow rate or surface temperature during operation. All preventive measures to fouling must be done prior to the construction; in the design phase for passive coolers. When the passive cooler is installed, the fouling will occur at its natural rate for the given natural flow and temperature conditions during free convection. The fouling control strategies available for an open surface heat exchanger during design are:

- Surface coating
- Surface temperature correction

As presented in section 2.2.2, the adhesive force of fouling to the surface plays an important role for the attachment rate. Coating of the external heat transfer surface with a material that reduces the adhesive force of biofouling and scaling will reduce the rate of fouling.

The heat exchanger can also be designed so that the surface temperature is within a range that reduces fouling; the penalty of reducing surface temperature is a lower mean temperature difference which must be compensated for with an increase in area. A reduction in surface temperature will also reduce the temperature dependent heat transfer coefficient for natural convection. This reduction will have to be compensated for with a further increase in area.

3.2.2.2 Enclosed surface

A direct cooler has an enclosed surface where a controllable amount of water passes through the heat exchanger.

The heat transfer rate will decide the flow rate and temperature for a heat exchanger in operation. This means that the evaluation of temperature and flow rate inside the heat exchanger must be done in the design phase, to have the heat exchanger operate with the required heat transfer rate within the preferable flow rates and temperatures.

The fouling control strategies available for an enclosed surface heat exchanger during design are:

- Surface coating
- Surface temperature correction
- Stream velocity correction

The relative effects of seawater temperature and flow rate on fouling in seawater heat exchanger is shown in Figure 16 and Figure 17. The data behind these figures is however collected from heat exchangers using seawater as a coolant above water, not subsea.

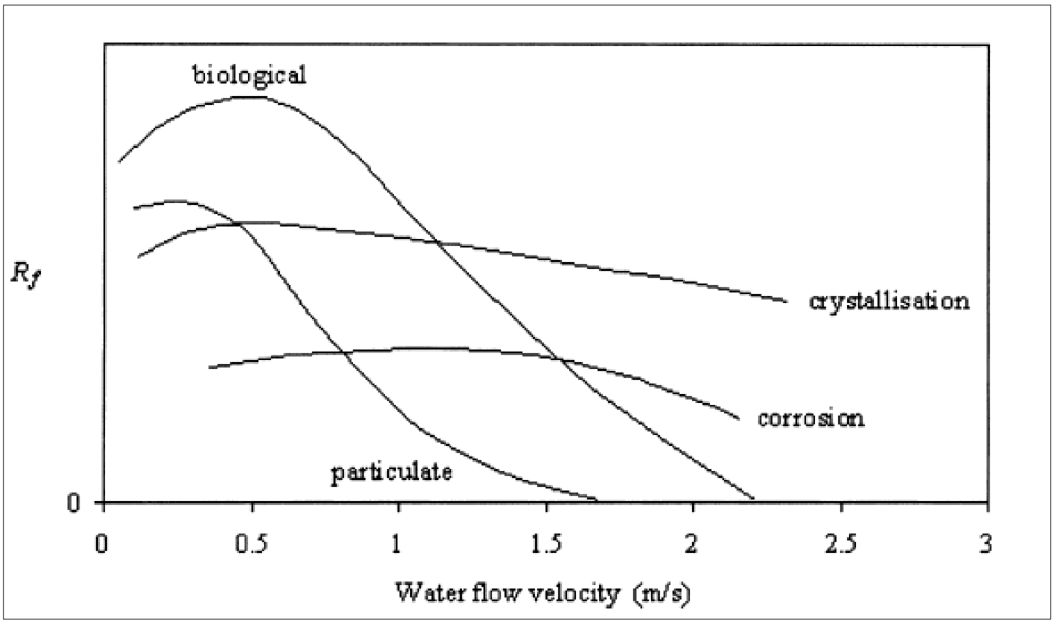

Figure 16: The effect of seawater velocity on relative fouling resistance

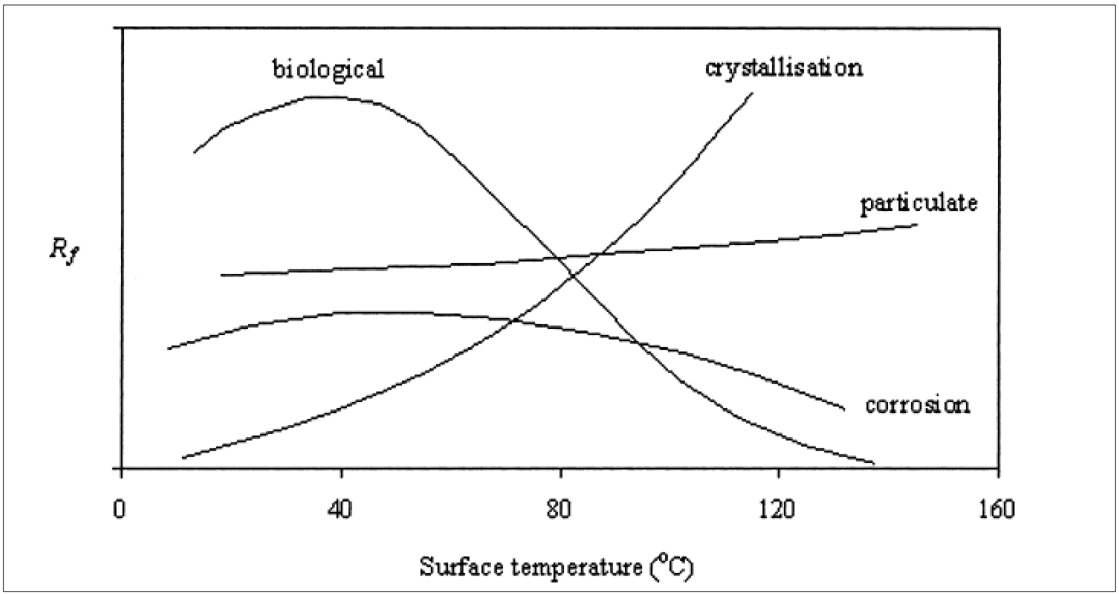

Figure 17: The effect of surface temperature on relative fouling resistance

For an enclosed heat transfer surface, water treatment can help reduce fouling:

The fouling control strategies available for an enclosed surface heat exchanger during operation are:

- Mechanical water treatment
- Chemical water treatment

Mechanical water treatment usually just includes the use of a rough strainer to exclude large particles and animals from the water inlet. If more particle-sensitive heat exchangers are to be used, finer strainers must be used as well.

Chemical treatment, such as the injection of biocides, is limited to nontoxic chemicals, and within regulatory limits for letting these chemicals out in the seawater. Both chlorine and some scale inhibitors can be used in systems using direct cooling with seawater [21]. The injection of the same chemicals can be limited to periodic, high volume injections, such as shock chlorination. Figure 18 visualizes the use of filtration and chemical injection upstream of the ehat exchanger.

A Norwegian company called Seabox has developed a product, with the same name, that is used for killing biological organisms and removing particles subsea. The Seabox does not need a supply of chemicals to operate, only electricity. The usage area for the Seabox is oil reservoirs that benefit from the injection of seawater. Injecting water containing particles and biological material may make the entire reservoir sour or plug the water injection well; the Seabox will prevent this from happening [24]

The Seabox could potentially be used water in a subsea processing facility for water treatment of cooling water, to reduce fouling, and is shown in Figure 19.

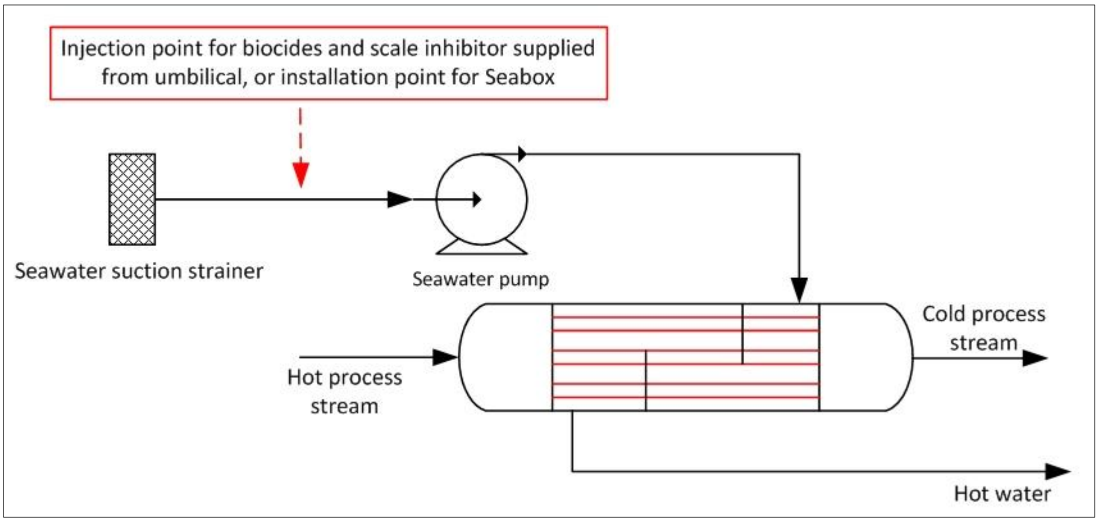

Figure 18: Direct cooling using a shell and tube heat exchanger with upstream filtration and chemical injection

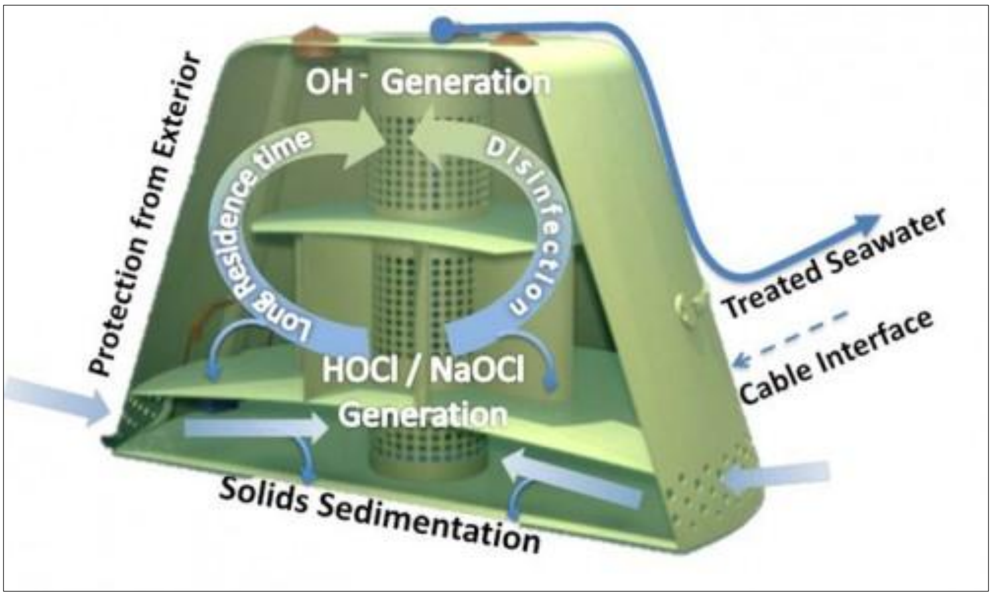

Figure 19: Cross sectional view of the seabox

3.2.3 Heat exchanger controllability

As the process system needs to be started and shut down, as well as operate at reduced process side flow rates from time to time, and the heat exchanger must handle to operate at lower duties. When a fouling resistance is included in the design, the system must be able to compensate for a reduction in heat transfer coefficient due to fouling. The control of a direct or indirect cooler can be done through the adjustment of flow rate or inlet temperature of the cold stream.

The heat exchanger will be designed for the worst conditions, when it needs the largest heat transfer area; maximum heat transfer rate with the maximum fouling resistance. However, the heat exchanger must also be able to operate within the restrictions when there is no fouling resistance, low flow rates on the process side, and far too much heat transfer area available. To compensate for the excess heat transfer area the heat flux must be reduced. The heat flux is the heat transfer rate divided by area and is shown in (17).

$$q'' = U\,\Delta T_m \qquad (17)$$

Flow adjustments will affect both the overall heat transfer coefficient and the mean temperature difference, while adjusting the inlet temperature of the cold stream only affects the mean temperature difference. The heat flux is a function of flow rate and inlet temperature, as shown in (17). The overall heat transfer coefficient is a function the flow velocity and the mean temperature difference is a function of the mass flow rate and the inlet temperatures. The process inlet temperature is fixed, the cold stream inlet temperature is however a free variable.

To control the flow rate of the cooling medium or seawater in a forced convection cooling system, one can use either a variable speed drive pump, or use valves to adjust flow rate in a bypass or recirculation arrangement, as shown in Figure 20.

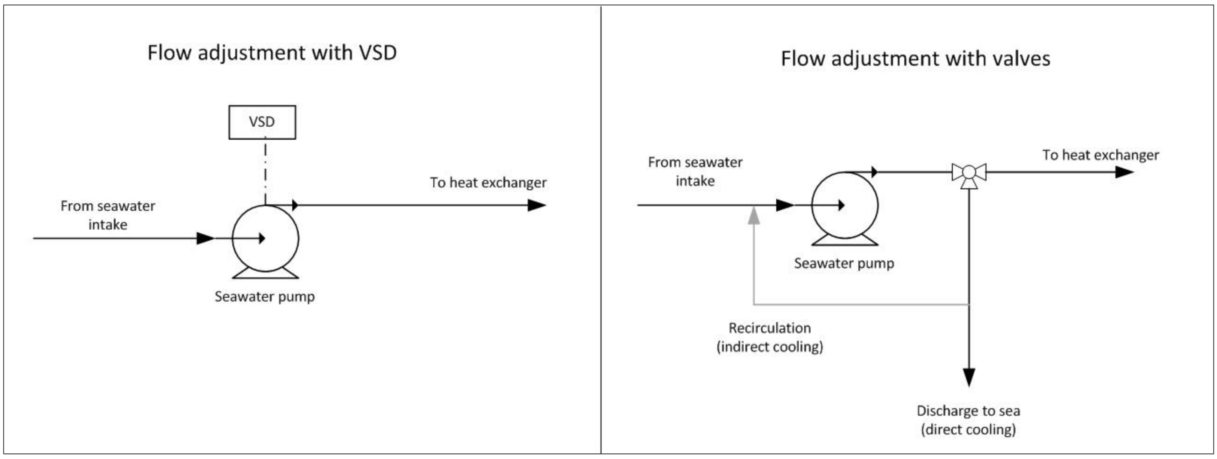

Figure 20: Two ways of having flow control

To adjust inlet temperature on the cold side in a force convection cooling system, one must redirect some of the hot seawater or cooling medium from the outlet of the heat exchanger to the inlet stream, and mix this with the cold cooling medium or seawater. Recirculating liquid from the heat exchanger outlet in a seawater system is dangerous as fouling debris removed by the liquid, ref. section 2.2.2, re-enters the heat exchanger and can potentially plug this. Another strainer should therefore be installed on the recirculation line as shown in Figure 21. In an indirect cooling system this is of no concern due to negligible fouling.

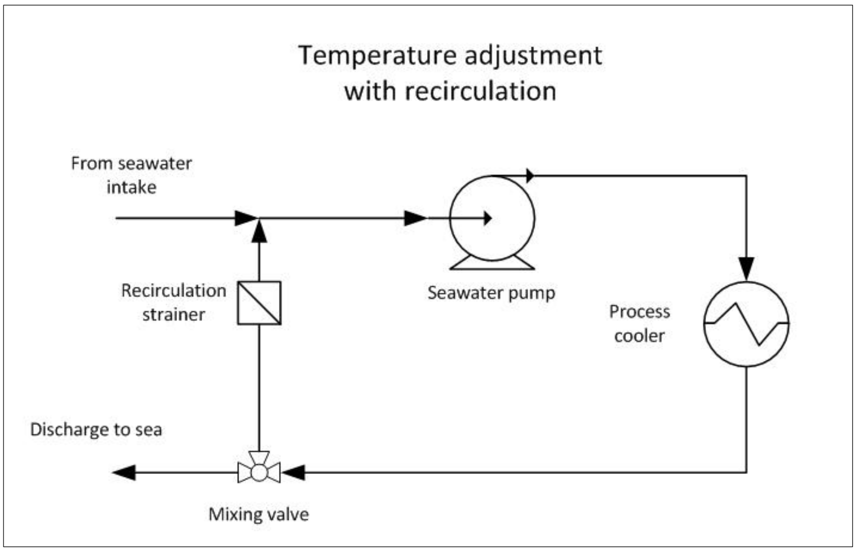

Figure 21: Inlet temperature control of a heat exchanger

Figure 20 and Figure 21 are only meant to show the principals of temperature control and flow control. There are many options in the arrangement of pumps, valves and heat exchangers for control. This has not been studied in detail in the work behind this paper.

The effect of flow adjustment on overall heat transfer coefficient is further discussed in section 5.2.1.

3.2.3.1 Lack of controllability in passive coolers

A passive cooler offers no controllability. The passive cooler must be designed for the maximum heat transfer rate and fouling resistance. For lower flow rates, or clean operation, the heat exchanger will cool the process stream below the target temperature. If deviation from the target temperature is acceptable, one can design the heat exchanger so that it cools less during fouled conditions and too much during clean conditions. This way the target temperature will then be between these two operating points, visualized in Figure 22. It is required that the lowest process stream outlet temperature is within good margin of the hydrate formation temperature, for such a solution to be applicable. The margin is needed to cope with smaller reductions in process side flow. When the process side outlet temperature approaches the hydrate formation temperature, one must either use a hydrate inhibitor or bypass the heat exchanger.

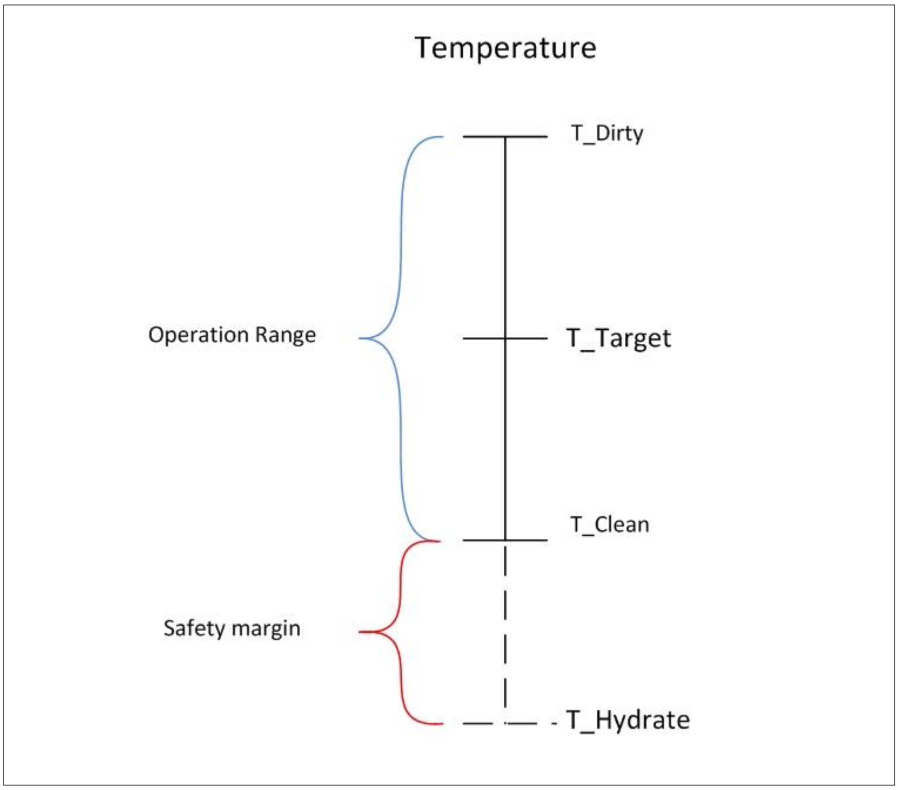

Figure 22: Graphical presentation of the operation range of a passive cooler outside the hydrate region

3.2.3.2 *Direct cooling controllability*

The seawater side wall temperature is not allowed to exceed 60°C, which sets a limit to the minimum heat transfer rate possible to reach with the adjustment of flow rate and inlet temperature of the seawater in a direct cooler.

Figure 23 shows how the temperature at the heat exchanger end points varies with a reduction in the flow rate, or with an increased inlet temperature, of the seawater.

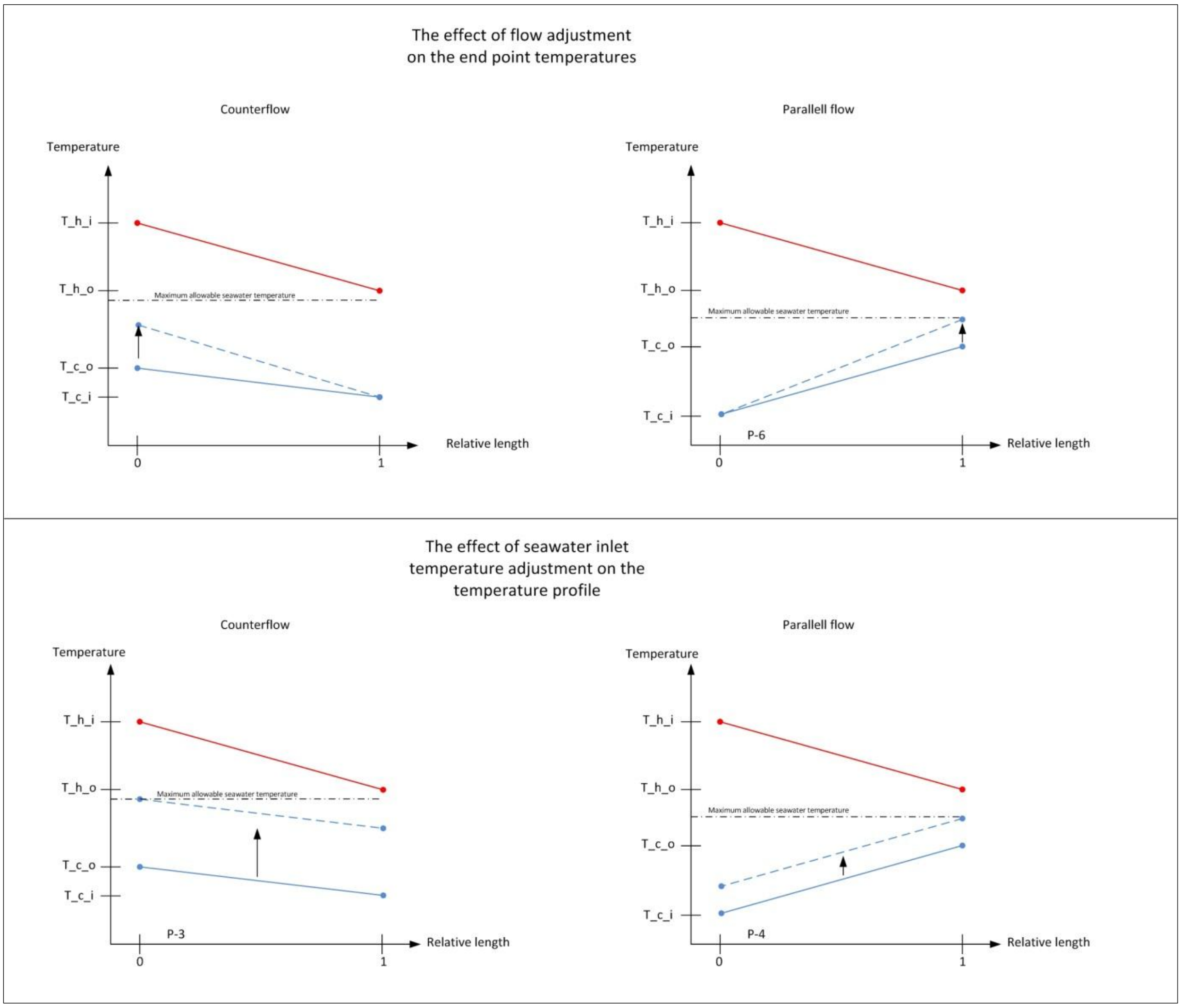

Figure 23: The effect of inlet temperature control on a heat exchanger

The restrictions to the maximum wall temperature on the seawater side of the heat exchanger will constrain the cooling water flow adjustment, and the maximum seawater inlet temperature. If the process stream temperature is far above the maximum seawater side temperature, the adjustment of inlet temperature and flow rate of the seawater will have little effect on the reduction of the heat transfer rate, and one will risk cooling the process stream below the target temperature.

If reducing flow rate and increasing inlet temperature does not sufficiently compensate for the excess area in operation with low heat transfer rate, one must also adjust the heat transfer area. This can be implemented in the design by dividing the heat transfer area into sections which can be turned on and off. In practice this means installing parallel heat exchangers units. Heat exchanger units could also be placed in series, but as this would be problematic due to maldistribution of liquids discussed in section 2.3.1.

The controllability will also be increasingly difficult as the process stream temperatures go above the maximum seawater temperature.

If the process stream inlet temperature is below the maximum seawater temperature of 60°C, the reduction of flow rate or inlet temperature alone will cover all duties down to zero.

If the process stream target temperature is close to the maximum seawater temperature, parallel-flow arrangements will give better controllability, as the maximum cold side temperature in a parallel-flow heat exchanger never will exceed the hot side minimum temperature, presented in section 2.3.1.2.

3.2.3.3 Indirect cooler controllability

Indirect cooling separates the process heat transfer surface from the seawater heat transfer surface and places these in different heat exchanger .The heating medium has no restrictions to temperature, as mentioned in section 3.2.2.

The control of the process cooler in an indirect system will not be limited by a maximum temperature on the cold side, such as with direct process cooling using seawater. The dump cooler will not be limited by the minimum temperature on the hot side, as no hydrate formation or freezing is impossible for the cooling medium.

An indirect cooling system is much more customizable than a direct cooling system due to the indirect cooling medium circuit which can be manipulated and controlled with valve and pump arrangements to achieve the optimal working conditions for each heat exchanger. The controllability of an indirect cooling system can therefore be customized in the design phase for its purpose; rough with a simple system design or very accurate with a more complicated system design.

4 Heat exchanger selection

Once the goal of a subsea process system is identified, the system design is selected and a cooler arrangement chosen one can decide on which type of heat exchanger to use. This chapter screens out applicable heat exchangers for the subsea cooling system based on the findings in the previous sections.

4.1.1 The effect of design pressure on heat exchanger selection

The applicability of heat exchangers for subsea use will first of all depend on the wellhead shut–in pressure, ref. section 1.2.4. Most of the heat exchangers presented in section 2.3.3 will not handle design pressures over 100 Bars. In an oil and gas reservoir, shut in-pressures above this can be expected in most cases [4]. The heat exchangers from section 1.2.4 capable of handling design pressures over 100 Bars are:

- Passive tubular
- Shell and tube
- Diffusion bonded plate-fin
- Printed circuit
- Packinox plate

The Packinox plate heat exchanger is a plate heat exchanger in a pressure chamber. Information was not found on whether such a heat exchanger would work for a stream with a high design pressure and much lower operation pressure, or whether the pressure vessel is pressurized simultaneously and not prior to the heat exchanger being pressurized, to keep the plate heat exchanger from imploding. The Packinox type heat exchanger was excluded from the evaluation, due to lack of available information.

For the indirect cooling arrangement, any type of heat exchanger can be used between the cooling medium and the seawater, as the system can be designed with no pressure difference between these two sides.

The diffusion bonded plate-fin heat exchanger was developed in a collaborative venture between Rolls Royce and Alfa Lava [25]. No vendor of this heat exchanger was found online, and little information about it exists. However, it differs very little from a printed circuit heat exchanger; only the construction technique is slightly different between the two [25]. The printed circuit heat exchanger is offered by the vendor Heatric and has been used in oil and gas processing topside many times before [26] [27]. The diffusion bonded plate-fin heat exchanger is therefore excluded from the evaluation as the printed circuit heat exchanger serves the same purpose.

The applicable heat exchangers for high pressure oil and gas processing subsea evaluated are then:

- Passive tubular
- Shell and tube
- Printed circuit

Table 10 shows the possible combinations of these heat exchanger with the cooling arrangements presented in section 3.2.

Table 10: Applicable heat exchangers, with a minimum design pressure of 100 Bara, for the process cooler

Cooling arrangement	Applicable heat exchangers
Passive	Tubular passive
Direct	Shell and tube Printed circuit
Indirect	**Process cooler:** Shell and tube Printed circuit **Seawater cooler:** Passive and direct systems possible. All heat exchanger types can be used for direct cooling of cooling medium using seawater.

4.1.2 The effect of process location on applicability of heat exchangers

The second most important factor affecting the choice of heat exchanger selection is the location of the cooler in the process, discussed in section 3.1. Having the cooler upstream of separation requires it to handle a stream containing sand and dirt. A printed circuit heat exchanger with its maximum channel size of 3mm is not suited for such operations, and would clog instantaneously. This means that for upstream operation only the tubular heat exchangers can be used, while for downstream operation both the tubular and printed circuit heat exchanger can be used, as sand and dirt is removed from the stream in the inlet separator.

Table 11: Applicable heat exchangers subsea in relation to process cooler location in the SSBI system

Applicable heat exchangers upstream of separation	Applicable heat exchangers downstream of separation
Shell and tube Tubular passive	Tubular passive Shell and tube Printed circuit

4.1.3 The effect of fouling on heat exchanger selection and design

TEMA type shell and tube heat exchangers are highly customizable; there are numerous components in the heat exchanger with a variety of available designs. Most of these design choices depend on thermal efficiency, pressure drop, mechanical integrity and fouling. Fouling plays an important role in the design choices. To make the design choices one needs to know two things about each stream in the cooler:

- Is the fouling rate high?
- Is cleaning needed, and how should the heat exchanger be cleaned?

A shell and tube heat exchanger could be welded into one piece, or built as an assembly. Tubular heat exchangers can be designed with a system that cleans the heat exchanger mechanically during operation [16]. If such a system is not implemented, the heat exchanger must be taken out of service to be cleaned, as presented in section 1.2. A printed circuit heat exchanger is not as customizable as a shell and tube heat exchanger. It is basically a solid block of metal, and has no selectable components other than the plate material channel size and flow arrangements. It can only be cleaned chemically and needs to be taken out of service to be cleaned.

The cleaning techniques available for a heat exchanger will depend on whether the heat exchanger can be disassembled or not. If the heat exchanger can be taken apart, it can also be cleaned mechanically, if not it will have to be cleaned chemically. With a shell and tube heat exchanger one has the choice of making the tube side and the shell side dismountable for mechanical cleaning or not during design. A printed circuit heat exchanger cannot be disassembled. Table 12 lists the cleaning techniques available for the two heat exchanger types:

Table 12: available cleaning strategies for the applicable subsea heat exchangers

Printed circuit cleaning	Shell and tube cleaning
Chemical	Chemical Mechanical Online mechanical cleaning possible

The passive heat exchanger can only be cleaned mechanically and chemically or mechanically on the inside.

4.1.4 Using HIPPS to expand the applicable heat exchanger designs

If a HIPPS module is used upstream of the process module, as mentioned in section 1.2.4, the design pressure can be reduced from the wellhead shut-in pressure and closer to the operating pressure. The gage pressure at which the HIPPS will close is the new design pressure.

If the new design pressure minus the hydrostatic pressure is small, a wider range of the heat exchangers from section 2.3.3 are applicable, as the maximum pressure from this table is the gage pressure. A heat exchanger operating with a maximum absolute pressure of 60 Bara, and an external absolute hydrostatic pressure of 20 Bara, will have an internal gage design pressure of 40 Barg, and a wider range of plate heat exchangers can be used.

It is also worth mentioning that the installation of HIPPS and subsequent reduction of design pressure will reduce the material costs of the heat exchangers, separators and piping in the entire subsea processing plant.

When HIPPS are installed the system design pressure is:

$$Design\ Pressure(Barg) = \ HIPPS\ trip\ pressure\ (Bara) - Hydrostatic\ pressure(Bara)$$

5 Heat exchanger design

This part of the paper is supported by heat exchanger design and simulations from a case study included in the Appendix. The case study includes the heat exchanger design for the cooling system in a SSBI system using a wellstream with a composition, pressure and temperature that is typical for oil and gas fields in the North Sea. The intention is to end up with results that can add information to generic discussions and evaluations so far in this paper. The heat exchangers investigated are shown in Table 13.

Table 13: The heat exchangers investigated in the case study included in the Appendix

Cooling arrangement	Heat exchanger type	Ref.
Passive	Tubular	Appendix B
Direct	Shell and tube	Appendix C
Direct	Printed circuit	Appendix D
Indirect Process cooler Dump cooler	 Printed circuit and Passive tubular	Appendix E
Indirect Process cooler Dump cooler	 Printed circuit and plate Plate	Appendix F

5.1 Flow arrangement

Several different flow arrangements were tested with the shell and tube heat exchanger design in the case study, which can be seen in Table C-3 in Appendix C, and little difference in heat transfer area was found between the flow arrangements.

The heat exchanger operation is limited by the minimum process side temperature and the maximum seawater side temperature, as shown in chapter 3. The problems with low wall temperatures and hydrate formation were shown 3.1.3.1 and the avoidance of scaling was discussed in section 3.2.1.

The hydrate formation curve for the case study is shown in the Appendix 11. This is a typical hydrate formation curve for a methane rich gas; and the hydrate formation temperature will usually be around 17°C at 50 Bara. The seawater maximum temperature is 60°C.

Whether flow arrangement plays an important role or not is dependent on the heat exchanger effectiveness, as presented in section 2.3.1.

5.1.1 For passive coolers

The theory of flow arrangements does not apply directly to a passive cooler, although there are some analogies with natural convection flow across the tube bundle and the placement of the tubes in vertical position or horizontal position. This has not been investigated in this paper.

5.1.2 For direct coolers

High heat exchanger effectiveness cannot be achieved for the direct cooler in the case study, where the constraints set the effectiveness between 0,38 and 0,60 when using seawater as a coolant. The reason for this is that the end point temperatures cannot meet.

Most important of all is the fact that high heat exchanger effectiveness is not desirable in a direct cooler. Higher heat exchanger effectiveness is related to larger heat transfer area and higher temperature on the cold stream outlet. The seawater is dumped directly to the ocean, and is of no value once used; low heat exchanger effectiveness is favored to minimize the heat exchanger size. With low heat exchanger effectiveness, any flow arrangement can be used.

5.1.3 For indirect coolers

The process cooler in an indirect cooling arrangement has only the constraint of hydrate formation temperature, since cooling medium is being used on the cold side instead of seawater. The process cooler can be designed with any heat exchanger effectiveness desired.

The cooling medium temperature out of the process cooler will affect the size of the dump cooler. When the cooling medium only operates between the process cooler and the dump cooler, an optimization on cooling medium temperature versus the total size of the process cooler and the dump cooler should set the cooling medium temperature and heat exchanger effectiveness.

In oppose to a direct cooler, the used cooling medium in an indirect cooler can be of value. The cooling medium can be heat integrated, in which the temperature plays an important role. High temperature cooling medium is of more value for heat integration than low temperature cooling medium. High heat exchanger effectiveness may be desirable from a heat integration point of view, if heat integration is used.

5.2 Thermal resistance

The thermal resistance was presented in section 2.1.1. The inverse of the sum of the resistances will add up to be the overall heat transfer coefficient.

The relative differences of the resistances to heat transfer between the streams passing through a heat exchanger will define the temperature profile across the heat transfer surface, as well as the controllability of the heat exchanger. Cooling arrangements and controllability was presented in section 3.2.3.

The gas heat transfer coefficient will be larger than the liquid side heat transfer coefficient for all cases with forced convection cooling When passive cooling is used the resistances are close to equal on each side of the heat exchanger. Figure 24 shows the resistance distribution in the selected shell a tube heat exchanger used for direct cooling in the case study, and it can be seen that the gas side resistance is a lot larger than any of the other resistances.

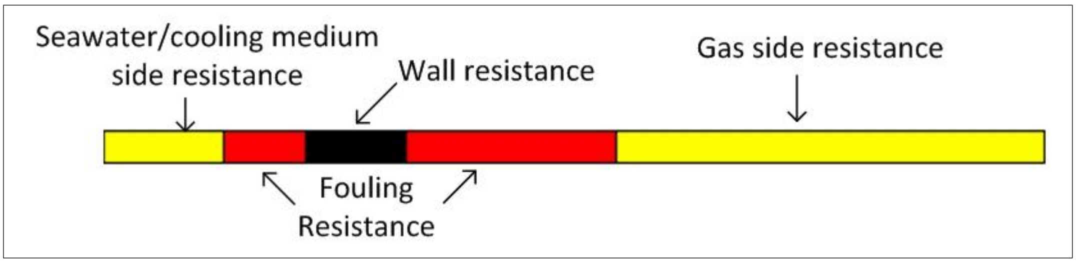

Figure 24: Graphical presentation of the relative resistances in the selected direct cooler from the case study

5.2.1 Effect on controllability

The control of the heat exchanger will be through either temperature or flow control on the seawater side. It was shown in section 3.2.3 how controllability can be achieved either through flow control or through changing the inlet temperature. When changing the flow rate one is changing the mean temperature difference and the overall heat transfer coefficient through changing the resistance on the seawater side. When changing the inlet temperature, one is primarily changing the mean temperature difference and not the overall heat transfer coefficient, when using seawater or a water based cooling medium.

In Appendix C the effect of changing temperature and flow rate on heat flux was checked for the selected shell and tube heat exchanger used for direct cooling. The results can be seen in Appendix C, Figure C-6 to Figure C-9. The results show that controlling the flow has a minimal effect on the heat flux for larger flow rates and large effects on the heat flux for lower flow rates. Changes in the inlet temperature on the other hand, have an almost linear effect on the heat flux.

The reason for this is that the seawater side thermal resistance is minimal compared to the other resistances between the two streams. This can be observed in in Figure 24 where the seawater side resistance is short compared to the total length of the bar representing the total thermal resistance. Adjusting flow rate will have little effect on the total resistance and hence the overall heat transfer coefficient. Even a doubling of seawater side heat transfer coefficient will have a minimal outcome. Adjusting the flow rate does not affect the overall heat transfer coefficient before reaching very low flow rates. At this point the seawater side thermal resistance is large enough to significantly affect the total thermal resistance.

Another reason for the low controllability of heat flux at large flow rates is that the mean temperature difference is so large to begin with, and adjusting the cold side flow rate has minimal effect on the mean temperature difference. The controllability on the mean temperature difference is achieved when a close temperature approach between the hot and the cold stream is reached in the heat exchanger end points. Just like the overall heat transfer coefficient, very low flow rates must be reached before the mean temperature difference is significantly affected by variations in flow.

Adjusting inlet temperature on the other hand will directly and close to linear proportional, affect the mean temperature difference by regardless of initial inlet temperature.

5.2.2 The effect of fouling

The effect of fouling on the heat flux is dependent on the total thermal resistance between the two streams, in the exact same way as the effect of change in flow rate on the heat flux. If the thermal resistance caused by fouling is small compared to the overall thermal resistance, fouling will have little effect on the heat flux. If the fouling resistance is large compared to the total thermal resistance, fouling will significantly affect the heat flux. This trend can be observed in Appendix C, Table C-1, where the overdesign due to fouling is less for lower overall heat transfer coefficients. If looking at the passive process cooler in Appendix E, and the results in Table E-2, it can be seen that the overdesign due to fouling is as little as 18.5% which is much lower than for the forced convection coolers, as a result of the increased total thermal resistance.

5.2.3 The temperature profile

The different thermal resistances will set the wall temperatures of the heat transfer surface. A high resistance on the seawater side will push the wall temperatures closer to the process side bulk temperature. And a large resistance in the heat transfer surface will increase the temperature difference across the wall and push both sides closer to the bulk temperature.

In the process cooler, the seawater side heat transfer coefficient is higher than the process side heat transfer coefficient, due to the fact that the process side is dominated by gas, which generally has a lower heat transfer coefficient than liquids. This means that the wall temperatures will have a tendency to lie closer to the seawater side temperature; this can be observed in Figure C-5.

For the passive cooler the seawater side thermal resistance is almost the same as the thermal resistance on the process side. This means that the temperature profile will lie evenly between the two bulk temperatures, as shown in Figure B-2. The bump observed in the beginning of this curve is due to a change the process side heat transfer coefficient, as the flow regimes transfer from single phase gas flow to annular multiphase flow, as the gas cools and condensates.

For the forced convection dump cooler in Appendix F, section F.1.3, both sides contain liquids and have low thermal resistances. The temperature profile for this heat exchanger is shown in Figure F-2.

5.3 Size

The sizing is interesting in terms of the compactness of the heat transfer surface of the heat exchangers for the different flow arrangements. Both indirect and direct cooling systems will however require components and piping. Sizing of this equipment is beyond the scope of this study, and the size results should not be evaluated as the total system size, but rather the potential of size optimization of the heat exchanger itself with the different arrangements. See Appendix 11 to F for details on the sizing.

Table 14: Overview of the results from sizing of the heat exchangers, collected from Appendix 11 to F

Cooling arrangement	Heat exchanger type	Total heat transfer core volume (m^3)
Passive:	Tubular	**7,58***
Indirect: Process cooler: Dump cooler:	 Printed circuit Passive tubular	**7,55*** (0,46 + 7,09)
Direct	Shell and tube	**0,68**
Indirect: Process cooler: Dump cooler:	 Printed circuit Plate	**0,37** (0,24 + 0,13)
Direct:	Printed circuit	**0,16**

6 Results

The results section is divided in two sections; one section for the results from studying the system design, and one section for the results from studying the heat exchanger design.

6.1 System design

If the wellstream has a high liquid content the heat exchanger should be placed downstream of the inlet separator. If the difference between the two cooler locations is small in terms of stream quality the heat exchanger can be placed upstream of the inlet separator and reduce the number of components in the SSBI system significantly. The selection of the preferable solution must be done on a case by case basis, based on wellstream quality.

The fouling types found to be representing the greatest risk to problem-free operation of the subsea cooler when this is placed downstream of separation is shown in Table 15.

Table 15: Main fouling uncertainties in subsea coolers

Fouling type	Occurrence
Hydrate formation	Process side
Scaling	Seawater side
Biological fouling	Seawater side

Hydrate formation will always occur in hydrocarbon systems with water, as long as the temperature and pressure is right. Hydrate formation is a solidification type fouling, and therefore has an on off occurrence at the hydrate formation temperature. Hydrate formation can be excluded by keeping the process side surface temperature above the hydrate formation temperature. The only way to guarantee that the process side surface temperature is above the hydrate formation temperature at all times is by increasing the cold stream inlet temperature to above the hydrate formation temperature; hydrate formation is then impossible throughout the heat exchanger, and its threat is fully excluded. Increasing the inlet temperature will reduce the mean temperature difference and thus require additional heat transfer area. It is difficult to predict if hydrates will cause problems when operating with low wall temperatures, as there is no complete model for such prediction.

Scaling is a result of inverse solubility salts in seawater, and will always occur on hot surfaces in contact with seawater subsea. The fouling rate is temperature-dependent. Scaling can be minimized by keeping the heat transfer surface temperature below 60°C on the seawater side. If the wall temperature exceeds 60°C, the thermal conductivity of the wall must be reduced, the process side heat transfer coefficient reduced, or preferably the heat transfer coefficient on the seawater side increased. The operation range of the heat exchanger for lower duties will be limited by the maximum value of the cold stream outlet temperature. Scaling can be further reduced with the use of a scale inhibitor if severe.

Biological fouling is the biggest uncertainty in the design of a subsea heat exchanger. The extent of biofouling in subsea equipment with hot surfaces is not sufficiently documented in literature. It is possible that biological fouling is less extensive on deep waters due to the lack of light.

Stream velocity and temperature consideration during design is documented to decrease fouling in seawater systems operating on land. In general the temperature should be kept as low as possible and the stream velocity as high as possible, coherent with the theory of attachment and removal of fouling layers. These are only considerations and not definite rules, and the design is not constrained from these considerations, as with the criteria for avoiding scaling and hydrate formation.

The tradition in heat exchanger design is to design for fouling by the use of a fouling resistance. There is however no way of knowing whether the fouling resistance will build up within weeks, months or years without operational data.

The fouling rate can be reduced with the injection of chemicals. Biocides can be injected to reduce the amount of biofouling on the seawater side. In open systems the injection of biocides is limited by regulatory limits for discharge of the biocides to the sea. The Seabox can be used, as an alternative to biocide injection, to process cooling water subsea to reduce biofouling. With an open heat transfer surface, as with the passive coolers, using chemicals is not possible.

6.2 Heat exchanger design

The importance of flow arrangement in the cooler design is related to heat exchanger effectiveness. For direct coolers the heat exchanger effectiveness should be kept low to minimize the heat exchanger size. This implies that the all flow arrangements can be considered for the heat exchanger, as the difference in heat transfer area between flow arrangements will be small.

For indirect cooling, the desired heat exchanger effectiveness is found through optimization of heat exchanger area in the process cooler and the dump cooler or by heat integration analysis if heat integration is applied. The selection of heat exchanger flow arrangement cannot be chosen on general terms for indirect cooling systems, and will have to be evaluated on a case by case basis. For heat integration in the subsea process plant high heat exchanger effectiveness should be chosen for the process cooler. This will result in a higher temperature of the cooling medium exiting the heat exchanger, which is of more value in heat integration. For high heat exchanger effectiveness the counterflow arrangement is preferred.

The thermal resistance in a heat exchanger determines the effect of changing the mass flow rate and inlet temperature of the cold side. The control of heat flux through mass flow rate control is not very effective. Vast changes in mass flow are needed for small changes in heat flux at larger flow rates. This is due to the very large mean temperature difference in the subsea cooling case and the high thermal resistance on the gas side compared to the seawater side. The heat exchanger could be made more controllable either by lowering the mean temperature difference or by increasing the seawater side thermal resistance, in which both would increase the area of the heat exchanger. As an alternative, it is more desirable to improve the gas side heat transfer coefficient, which would improve both the controllability and area of the heat exchanger. Controlling the temperature on the cold side inlet is the most effective method for controlling heat flux in the subsea case. The inlet temperature has a close to linear proportional relationship with the heat flux.

The effect of fouling on the heat exchanger depends on the magnitude of the fouling resistance compared to the total thermal resistance.

The heat exchanger size is affected by the selection of the heat exchanger and the cooling arrangement, more than other factors. The size of the heat exchanger was compared through the size of the heat exchanger core. Header design was neglected. The use of direct and indirect cooling systems will also include piping, instrumentation strainers, structural support and other equipment that will require some extra space and add weight. The sizes given from the case study are only indicating the opportunities with size optimization of the heat exchanger core itself, with the different cooling arrangements, and it is not an indication of the module total weight and size of a cooling system. The most compact solution was found with direct cooling using a printed circuit heat exchanger. An indirect cooling system using a printed circuit heat exchanger for the process cooler and a plate heat exchanger for the dump cooler was also found to be compact, but will require an extra pump for the seawater heat exchanger. The least compact solution is a passive cooler. The size of a passive cooler is however very dependent on the selected tube pitch and pipe outside diameter and wall thickness.

A simple but a bit larger solution is the indirect cooling using a printed circuit with a passive cooler, this system will be relatively simple, and will protect the printed circuit heat exchanger from seawater side fouling.

7 Discussion

When designing a subsea cooling system with focus on high reliability, it is obvious that fouling plays an important role. To achieve high reliability one must control the risks by identifying failures that may occur and design for the avoidance of these.

The main threat to the reliability of a subsea cooling system will be fouling, and the extent of this problem is difficult to predict.

The only fouling type that universally can be excluded through design is hydrate formation by increasing the inlet temperature of the cold side, and wax formation by placing the heat exchanger downstream of the inlet separator. Seawater side fouling in the process heat exchanger may be excluded with an indirect cooling arrangement, but the issue of seawater side fouling is still not excluded from the system as a whole, only shifted from the process cooler to the dump cooler.

Since it is not possible to fully exclude fouling from the system, it will have to be controlled. It is possible to take precautions against fouling throughout the design of a subsea cooling system as demonstrated in this paper; from system design to heat exchanger detail design. The trade-offs for handling increased fouling, or reduce the fouling rate will, be against the other important design parameters identified in chapter 1; size and controllability. Optimizing the design only with focus on reducing fouling will increase the reliability of the heat exchanger but will also increase the system size and reduce the system total reliability due to the introduction of additional components that may fail. The difficult part is finding the correct balance between the three important design parameters.

It was found little information on the exact extent of fouling for the operation described in this paper. A lot of guidance exists for reducing the fouling rate, but little on information exists on the actual expected rates and the exact effect of design changes on these. Designing against fouling without experience in heat exchanger design is thus very difficult. It is most likely due to the fact that the fouling rates vary too much for general data to be of any value, that this data is not easily available. The concept of subsea process cooling is however a very specific application and it should be possible to collect operational data from the numerous oil and gas platforms under operation to better identify the extent of fouling in a gas cooler early in the process, and in offshore seawater coolers.

The problem with not knowing the fouling rates is that it is problematic to know whether the system should be designed for extreme fouling or barely any fouling at all. In terms of system complexity and size, this is like designing a motorized vehicle without knowing whether it should be tractor or a formula one race car.

The possible degrees of fouling and examples on the design solutions are shown in Table 16, with the relative size and reliability.

Table 16: Degree of fouling, possible design solutions and their relative reliability and size

Degree of fouling	Design	Cooling arrangement and heat exchanger type	System total reliability	System total size
Fouling is not a problem	Design as compact as possible	Direct cooling with printed circuit	High	Low
Fouling is a problem	Design with consideration to stream velocities and temperatures, apply a reasonable fouling resistance.	Direct cooling with printed circuit or shell and tube	High	Low
Fouling is a big problem	Design with consideration to stream velocities and temperatures, apply a reasonable fouling resistance, implement a chemical injection system or water treatment for fouling rate reduction	Indirect cooling with printed circuit process cooler and plate dump cooler	Medium	Medium
Fouling is a major problem, and the required operation time is not possible	Increase redundancy by parallel units or implement a heat exchanger cleaning system subsea	Direct system suing shell and tube, heavy fouling stream on the tube side	Low	High

The method used for cleaning the heat exchanger will also affect the design, and should be clear before the final design is chosen.

A subsea heat exchanger could be cleaned in either of three ways:

1. On the sea bottom with a system that is installed together with the heat exchanger and controlled from land through the control system
2. On the sea bottom with temporarily cleaning equipment deployed and operated from a ship
3. Onshore by first replacing the heat exchanger with a clean spare , and then bring the dirty heat exchanger onshore for cleaning

The emphasis should be put on creating a design with an implemented cleaning system subsea, if fouling is too severe for a two-year operation window between cleaning. This system itself would then have to be extensively qualified for the reliability of the many new subsea components to be assured. If the cleaning will take place onshore, the cleaning strategy is simply based upon the customer's preferred cleaning strategy.

The effect of fouling could be reduced by increasing the total thermal resistance in the heat exchanger, but this would increase the heat exchanger size. If the seawater side is the side with the most severe fouling, an indirect cooling loop using a very large passive cooler as the dump cooler could almost exclude the problem of seawater side fouling. The passive cooler would however need to be very large for this to effectively work. The most compact solution for a subsea cooler is a direct cooler using a printed circuit heat exchanger. Printed circuit heat exchanger can save a lot of space, and increasingly so for higher duties and close temperature approaches.

It is difficult to conclude on whether a passive cooler, a direct cooler or an indirect cooler should be for subsea process cooling. To conclude on the size, weight and reliability of these systems additional work must be done other than the heat exchanger design, to map the total system impact of piping length, pump size, instrumentation and any additional systems needed such as chemical injection.

An indirect cooling loop offers the highest controllability of the cooling arrangements, due to less temperature constraints. Advanced control with shunts, valves and pumps can be used to fine-tune control in an indirect cooling loop. Direct cooling offers high controllability, but within the temperature constraints of the fouling set temperatures. A single passive cooler unit offers no controllability. The main benefits of the cooling arrangements are summarized in Table 17.

Table 17: Comparison of the benefits between cooling arrangements

Passive	Direct	Indirect
-Open heat transfer surface -Simple	-Enclosed heat transfer surface -Compact	-Choice between enclosed or open heat transfer surface -Possibility for heat integration

A passive cooler system is large and simple, but has an open heat transfer surface. An open heat transfer surface is useful if a cleaning strategy where the equipment is cleaned in its installed location is used. Remotely operate vehicles (ROV) can then clean the heat exchanger from the outside, without requiring the heat exchanger to be taken offline.

A direct cooling system offers the benefit of a compact solution and, if seawater side fouling is small, only requires an additional pump. The enclosed heat transfer surface in a direct cooler opens the door for treatment of the cooling water, either with chemicals or filtration.

An indirect cooling loop can have both an open and an enclose heat transfer surface, depending on whether the dump cooler is a forced convection cooler or a passive cooler. The main benefit of an indirect loop is the possibility of using the heat removed from the process stream elsewhere in the subsea process plant for heat integration.

The fact that the subsea cooler will operate with large temperature difference will wipe out some of the finesse of heat exchanger design. Small design changes that in a large duty and close-temperature-approach heat exchanger would have a major effect on size, will for the relatively low duty and large-temperature-difference subsea heat exchanger have little outcome. The positive side of this is that there is more freedom in the design of the heat exchanger itself once the proceeding design decisions on system design and cooling arrangement are frozen.

The biggest opportunity for size optimization in a subsea process cooling system is to clearly map the severity of fouling, so that accurate countermeasures in design can be performed to meet the exact need, instead of almost arbitrarily overdesigning the process cooling system with regards to fouling. This becomes increasingly important if the goal is to minimize size of the process cooler, as the overdesign due to the use of fouling resistances is much higher for compact solutions with high heat transfer coefficients.

One of the things repeated in all literature on heat exchangers is the value of operational experience. Experience from the operation of a similar heat exchanger in a similar process is the best guideline for heat exchanger design.

Subsea process coolers are indeed a new type of technology which lacks qualification. However, other than the marinization of the equipment, the design could be very similar to the heat exchangers designed to operate in oil and gas processing plants offshore. Experience from such operation should be gathered to reduce the uncertainty of subsea cooler design.

8 Conclusion

Fouling is the biggest uncertainty, and potentially the biggest problem, in the design and operation of process cooling for SSBI systems when reliability, size, weight, and controllability is considered as the most important design parameters.

The room for optimization towards fouling reduction for the process cooling was found to be in the process system design, in the cooling arrangement, in the heat exchanger selection, and in the heat exchanger design. In each of these steps the optimization potential was identified and discussed.

The biggest room for size optimization in a subsea process cooling system will be within clearly mapping the severity of fouling. Overdesign due to fouling is large in subsea cooling systems, and the knowledge of fouling rate is incomplete.

The only types of fouling that can be completely excluded through design are hydrate formation and wax formation. All other types of fouling will have to be controlled based on the expected formation rate, and a case by case evaluation.

The rate of fouling will set the reliability and maintainability of a heat exchanger installed under water, through the required frequency of cleaning and the chosen method for cleaning. The fouling rate and subsequently the cleaning strategy for the cooling system will significantly change the design of the heat exchanger. Subsea coolers may potentially require a cleaning system to be implemented under water if the fouling is severe enough, in which the main design focus should be on the implementation of such a system.

Fouling rates should preferably be accurately identified before an optimal cooling system is chosen for process cooling subsea.

To develop a complete picture of fouling in subsea heat exchangers it is suggested that similar heat exchanger technology already field proven is studied for the collection of detailed operation experience and data. This infromation is useful for the design of subsea process coolers to develop the most compact, reliable and controllable solution.

9 Suggestions for further work

For further work it is suggested that a more specific study is done on fouling in existing heat exchangers, in similar operation to that of a subsea process system. Heat exchangers on oil and gas process platforms can be investigated for the collection of data to look for universal design conditions and typical fouling rates. As a minimum the study should result in an identification of the worst case scenario that can be expected with regards to fouling on the process side and the seawater side respectively. Having a worst case scenario would sharpen the design focus of a subsea process cooler.

The data collected for analysis should include:

- Stream operation temperature, pressure and composition data
- Wall temperature and velocity estimations
- Fouling resistance data
- Operation history and time between cleaning intervals
- Fouling type observed, if possible
- Chemical injection philosophy and data
- Water intake depth and seawater temperature (seawater sides only)

There is no point of studying all heat exchangers on a platform arbitrarily; the heat exchanger that is to be studied must have some similarities with the subsea cooler designs, so that the results are comparable for the two. Examples of heat exchangers to study are:

- The feed gas cooler found on gas processing platforms, directly downstream of separation, which will be identical to the downstream cooler location studied in section 3.1
- The seawater heat exchangers found on almost all platforms for cooling of the cooling medium

Another suggestion to further study is to gather data from the use of x-mas trees. X-mas trees are frequently pulled up for repair, and could be used to gather data on seawater fouling from different locations, as several surfaces of the x-mas tree are warm. Data collection could vary from simple observation descriptions to detail collection of wellstream data and estimation of surface temperatures under operation, combined with observation, and possibly measurements of the fouling layer.

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11 APPENDIX

A – Case description

The test case is a small field in the North Sea, where several reservoirs are directed to a SSBI module. The well-stream is a mixture of oil and gas. The oil, gas and water phases are separated and the water re-inject. The hydrocarbon gas and liquids are separately compressed and pumped in relatively short pipelines to the platform. Due to a limit on the use of MEG, continuous hydrate inhibition is not possible in the gas pipeline. The philosophy is to insulate the pipeline and keep the pipeline internal temperature above the hydrate formation temperature for the entire length of the pipeline. To save compressor work and reduce material requirements downstream of the compressor, it is wanted to cool the wellstream. Due to the lack of MEG, the wellstream can however not be cooled too much, and precision on the cooling is needed within the limits of +/- 5 Degrees Celsius. The calculations in the case study were performed using:

- Hysys V7.3 for process simulation
- The HTFS bundle for the design calculation and simulation of heat exchangers.
- Excel for calculation of the natural convection heat transfer coefficient
- An online calculation script by Heatric for sizing of printed circuit heat exchangers

The wellstream has a low vapor fraction and an appropriate system design for such cases, as discussed in section 3.1, was chosen; having the heat exchanger downstream of inlet separation. Figure A-1 shows an overview of the system as it was simulated in Hysys. Table A-1 shows the composition of the wellstream (PT-001), the stream in (PV-001) and the stream out (PT-002) of the heat exchanger. Table A-2 shows the properties of these streams.

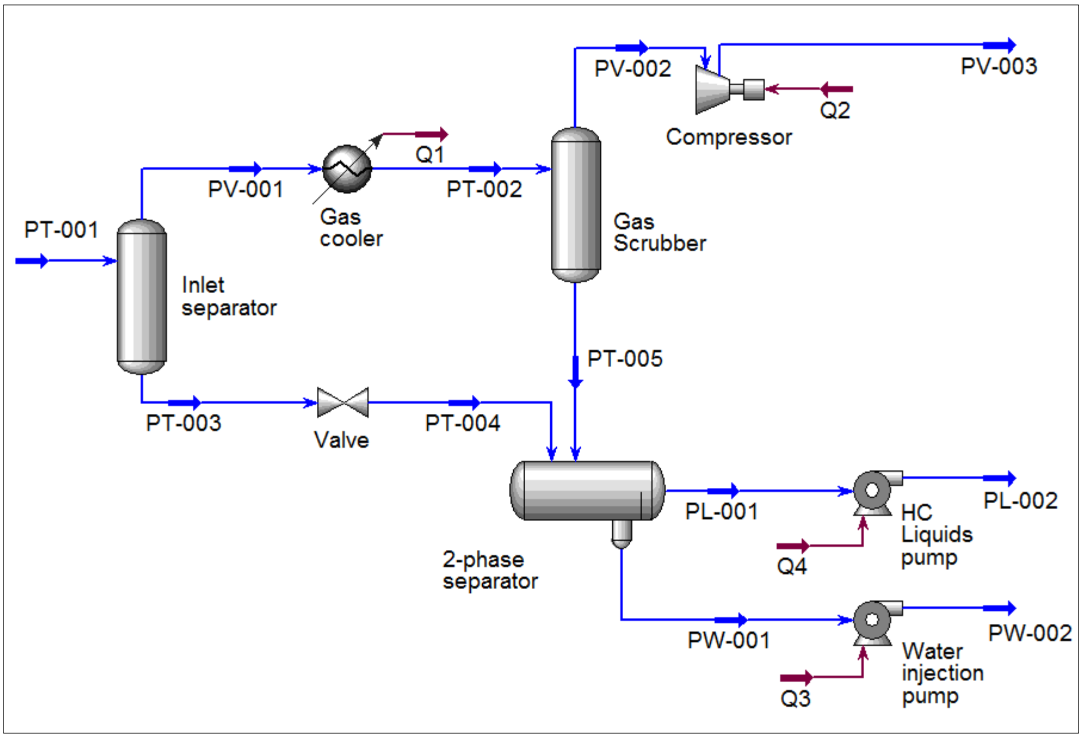

Figure A-1: The system design and simulation PFD from Hysys V.7.1

Table A-1: Composition of the streams

Component	Mole fraction		
	PT-001	PV-001	PT-002
Nitrogen	0,0017	0,003998	0,003998
CO2	0,016505	0,036481	0,036481
Methane	0,34014	0,784644	0,784644
Ethane	0,030811	0,066319	0,066319
Propane	0,023331	0,04474	0,04474
i-Butane	0,003697	0,006126	0,006126
n-Butane	0,010226	0,015813	0,015813
i-Pentane	0,003655	0,004414	0,004414
n-Pentane	0,004886	0,005425	0,005425
n-Hexane	0,005999	0,004242	0,004242
n-Heptane	0,010308	0,00427	0,00427
n-Octane	0,010703	0,002437	0,002437
n-Nonane	0,006441	0,000791	0,000791
H2O	0,486555	0,019543	0,019543
C10-C12G	0,00373	0,000253	0,000253
C13-C14G	0,001939	0,000033	0,000033
C15-C17G	0,002278	0,000003	0,000003
C10-C12F	0,001104	0,000077	0,000077
C13-C15FV	0,000853	0,000011	0,000011
C10-C13FH	0,00038	0,00002	0,00002
C14-C16FH	0,000214	0,000001	0,000001
C10-C13AO	0,006213	0,000332	0,000332
C14-C16AO	0,003564	0,000024	0,000024
C17-C20AO	0,003646	0,000001	0,000001
C15-C16AG	0,000143	0,000001	0,000001
C19-C22AG	0,000145	0	0
C23-C59AG	0,000093	0	0

Table A-2: Property for the streams in Figure A-1

Property	Unit	PT-001	PV-001	PT-002
Vapour Fraction		0,416504	1	0,978224
Temperature	C	95	95	60
Pressure	bar	50	50	49
Act. Vol. Flow	m^3/s	0,934113	0,821895	0,718861
Mass Flow	kg/s	123,8008	32,11168	32,11168

Five different designs were selected for evaluation for the design case, based on the findings in chapter 4. These designs are:

1 Passive cooling using a tubular heat exchanger
2 Direct cooling using a shell and tube heat exchanger
3 Direct cooling using a printed circuit heat exchanger
4 Indirect cooling using a printed circuit heat exchanger on the process side, and a passive heat exchanger on the seawater side
5 Indirect cooling using a printed circuit heat exchanger on the process side, and a plate heat exchanger on the seawater side

A.1.1 Design specifications and constraints

- The wall temperature shall never be within the hydrate formation temperature on the process side. The hydrate formation temperature for the process stream (PV-001 and PT-002) is shown in Figure A-2.
- The seawater side temperature shall not exceed 60°C
- The minimum flow velocity in controllable coolers should be 1 m/s
- Design pressure is 200 Bara on the process side and 6 Bara on the seawater side
- The fouling resistances for tubular exchangers are 0,0018 m^2*K/W on the process side, and 0,0009 m^2*K/W on the seawater side, picked from TEMA [17]
- The seawater depth is 500m
- Design pressure is 200 Barg on the process side, and 10 Barg on the seawater side
- ressure is 50 Bara on the process side and 55bara on the seawater side.
- The maximum pressure drop on the process side of the heat exchanger is 1 Bar

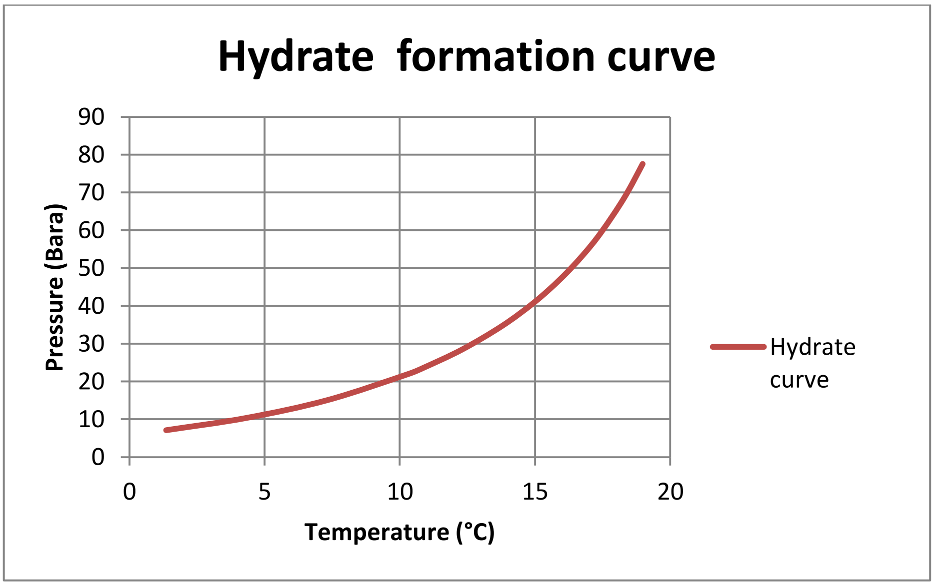

Figure A-2: The hydrate formation curve for the case

B – Passive cooler thermal-hydraulic design

A rough estimate of the size of a tubular passive process cooler was performed. The pipe size was set to 38mm. The wall thickness was set to 4.19mm; calculated with TASC+ according to ASME.

B.1.1 Method

On the seawater side the natural convection heat transfer coefficient was found using equation (18); the Churchil and Chu correlation for horizontal cylinders [13]:

$$\overline{Nu}_D = \left(0{,}6 + \frac{0{,}387 {Ra_D}^{1/6}}{(1 + (0{,}559/\mathrm{Pr})^{9/16})^{8/27}}\right), \quad Ra_D \leq 10^{12} \tag{18}$$

The heat transfer coefficient was calculated in excel, using thermal properties for seawater collected from the HTFS Props software.

The TASC+ software of HTFS was used to calculate the overall heat transfer coefficient, by manually defining the external heat transfer coefficient of the tubes with input from the excel calculation. The bulk seawater temperature was set equal to 0°C for the entire surface by letting the mass flow rate on the seawater side approach infinity. This does not affect the external heat transfer coefficient which is set manually, only the temperature difference. Since the external heat transfer coefficient only can be defined as an average for the operation range in TASC+, an iterative method was used between TASC+ and the excel calculation spread sheet. An average coefficient between 0-90°C was initially used in TASC+. The external surface temperatures along the tubes were calculated by TASC was collected, and a new average external heat transfer coefficient for the correct temperature range was calculated and used in TASC+. This procedure was repeated until the iteration converged.

B.1.2 Results

The results for the external temperature coefficient for 38 mm horizontal tubes submerged in 0°C seawater is shown for the surface temperature range of 0°C to 95°C in Figure B-1.

The overall heat transfer coefficient and other relevant performance data of the simulated heat exchanger are shown in Table B-1. The heat exchanger size and weight are shown in Table B-2.

Figure B-2 shows the wall temperatures of the heat exchanger during clean operation, and it can be seen that temperatures on the process side are above the hydrate formation temperature, and that the temperature on the seawater side is below 60°C.

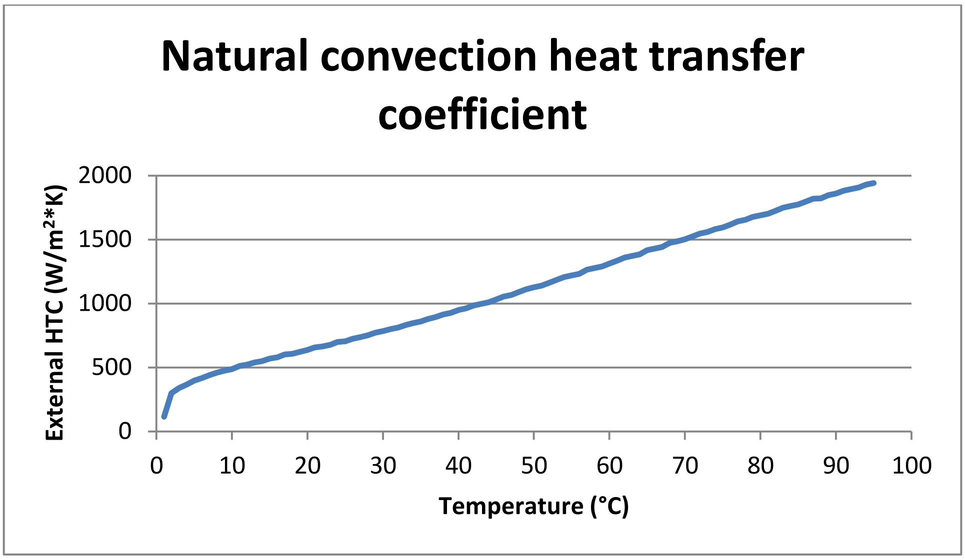

Figure B-1: **External heat transfer coefficient during natural convection versus the surface temperature for 38mm tubes in 0°C seawater.**

Table B-1: **Heat exchanger thermal-hydraulic performance**

Description	Units	Results
OHTC clean / dirty	W/m^2*k	455,7 / 385,9
Effective MTD	°C	76,16
Hot stream temperature out clean / dirty	°C	57,5 / 62
Actual area	m^2	120
Overdesign due to fouling	%	18,5
Number of tubes		115
Tube length	mm	9000
Process side pressure drop	Bar	0,2
Process side velocity	m/s	10,4

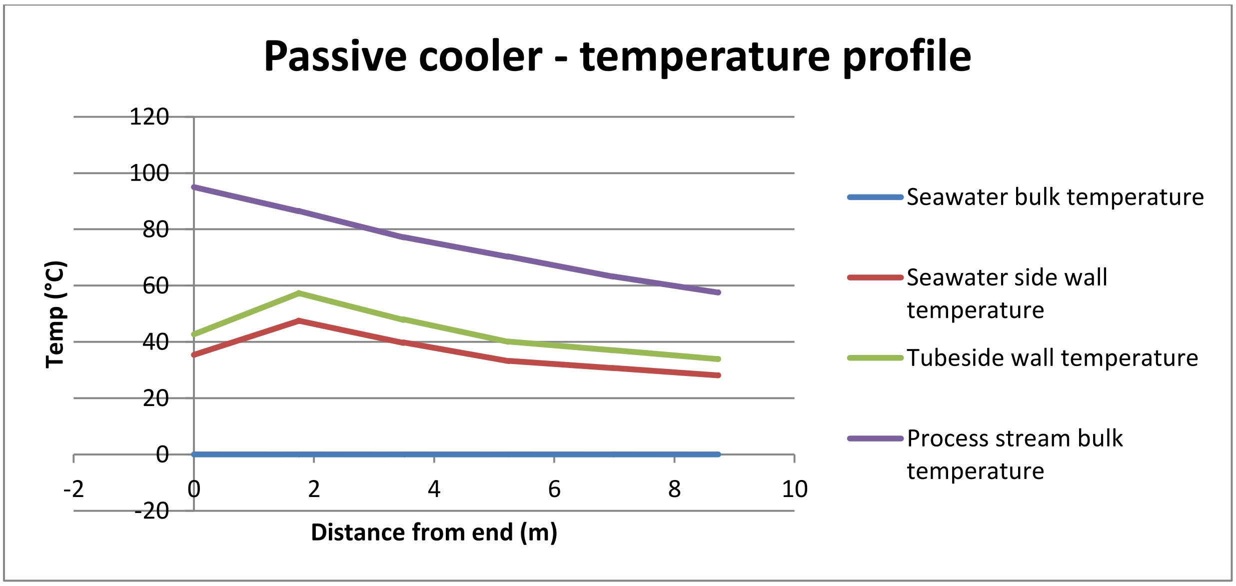

Figure B-2: Temperature profile along the heat exchanger tube length without fouling.

The size of the heat exchanger will depend on both the tube pitch and the tube arrangement. Table B-2 shows the dimensions and weight of the tubes, if the tube length is bent one time, as in a u-tube shell and tube heat exchanger, and a 50mm tube pitch is used in a 90 degrees square tube arrangement.

Table B-2: Heat exchanger size

Description	Units	Results
Tube pitch	mm	50
Tube passes		2
Dimensions	mm	4500 x 1836 x 918
Weight	kg	2,7

B.1.3 Comments:

This calculation method does not take into account the flow and thermal effects of having multiple tubes placed parallel in the vertical plane. The temperature difference between the water and the tubes will decrease as the seawater is heated and travels up the vertical plane due to density difference. Simultaneously this temperature increase will cause a chimney effect when the pipes are placed closely, and more cold water will be drawn into the heat exchanger from the bottom and the sides.

The calculation indicates that the external heat transfer coefficient for natural convection is much lower than that for forced convection, and highly dependent on surface temperature.

The top on the temperature profile around 2m is due to an improved convection heat transfer coefficient inside the tubes, and is most likely cause by the transition from single phase gas flow to multiphase annular flow

C – Direct shell and tube cooler thermal-hydraulic design

A direct cooler using a shell and tube heat exchanger was designed.

C.1.1 Method

The shell and tube design software TASC+ was used to study and design shell and tube heat exchangers. The process stream data was imported in TASC+ from Hysys V7.3.

The TEMA [17] standard defines the components shell and tube heat exchangers, and shows the possible standardized choices in shell types and rear-end and front-end head types; these are shown in Figure C-1 and are the choices available in TASC+.

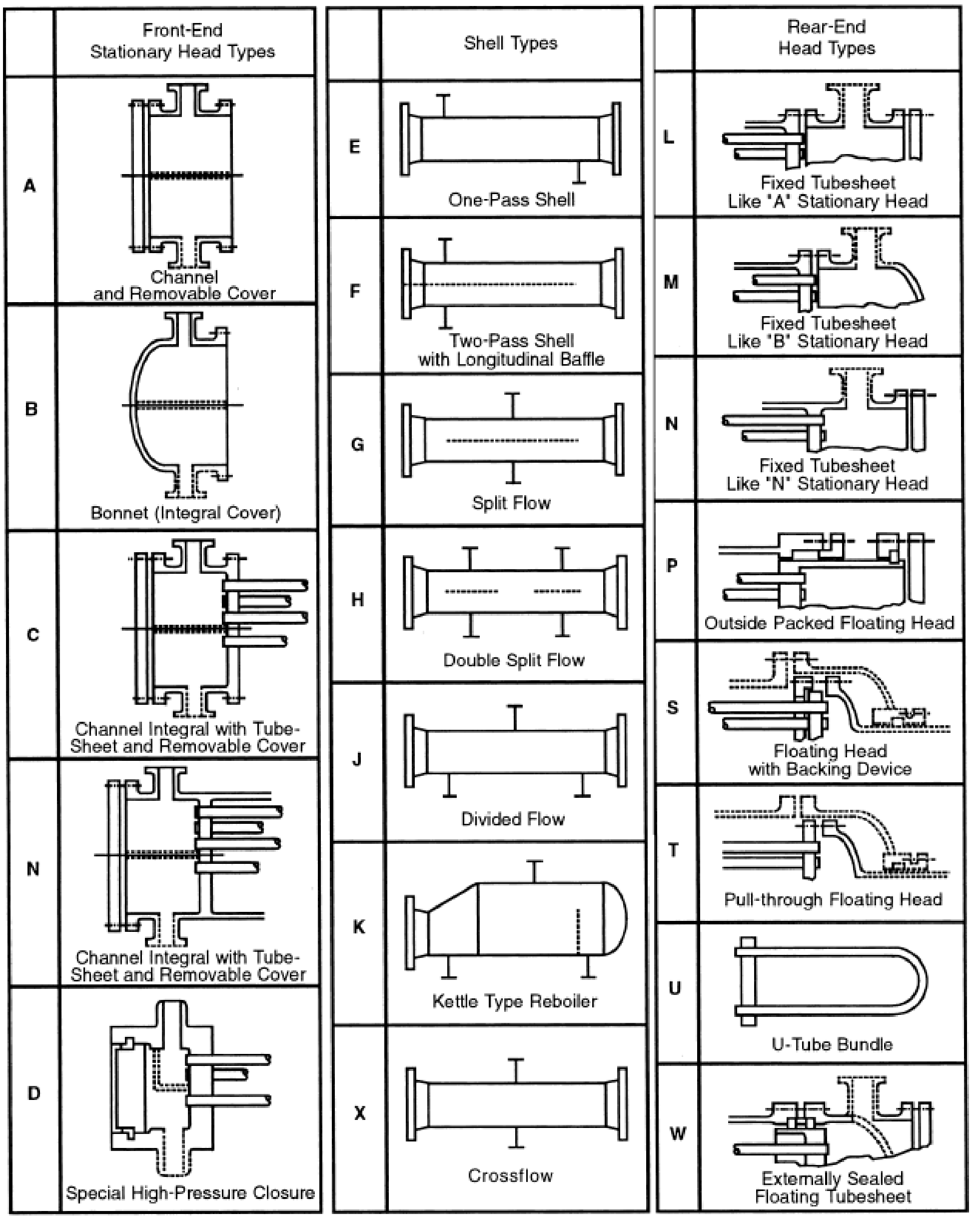

Figure C-1: Standard TEMA type shells and heads.

The various head design fulfils different maintenance, cleaning and thermal expansion requirements. Thermal expansion was not a problem in the design case and due to the lack of information on the required cleaning and maintenance strategy, the simplest type of heads were chosen; the N type heads.

The designs that were evaluated are E and X TEMA shells with counterflow, parallel-flow and multipass flow arrangements. The process stream was allocated both on the tube side and the shell side. The combinations tested are summarized in Table C-1.

Table C-1: The designs investigated for the shell and tube heat exchanger

Shell types	Flow Arrangement	Allocation of process stream
E and X	Counterflow, Parallel-flow, Cross-flow, Multipass	Tube side and Shell side

The flow rate for the seawater pump was set to 45 kg/s; approximately 160 m^3/h. This was not chosen arbitrarily, but after iterative simulations with heat exchangers in HTFS, testing the effect of flow rate and temperature on U-value and mean temperature difference, shown in Figure C-6 to Figure C-9.

The design function in TASC+ lets you define the heat exchanger constraints in your case, and will then suggest a number of designs and pick one based on your optimization goal, which could be minimum area or cost. Area was picked as the optimization goal. When TASC+ finds a thermal-hydraulic design that performs as specified, it is in most cases without fulfilling all of the TEMA or ASME requirements which are checked by the program. The next step is then to go through the list of error messages, and manually try to solve these. One must try to work on the design until the errors are solved. The difficult part is that often new error messages appear, when one is taken away, due to design changes. Table C-2 shows the methods for solving some of the problems with heat exchanger design by changing design parameters in TASC+.

Table C-2: An overview of encountered problems with designing shell and tube heat exchangers in TASC+, and possible solutions to these problems, collected from TASC+ course material.

Adjusted parameter	High pressure drop shell side	High pressure drop tube side	Low coefficient shell side	Low coefficient tube side	Temp. cross	Vibration indication
Baffle type	Double/triple segmental	-	Single segmental	-	-	Double/triple segmental or no tubes in window
Shell type	J or X-type shell	-	E or F type shell	-	E, F or G type shell	J or X type shell
Tube pattern	Rotated square or square	-	Triangular	-	-	Rotated square
Tube diameter	Increase to 1" or 1.25"	Increase to 1" or 1.25"	Decrease to 0,625" or 0,5"	Decrease to 0,625" or 0,5"	-	Increase to 1" or 1.25"
Baffle cut	Use 30% to 40%	-	Use 15% to 20%	-	-	-
Tube pitch	Increase to 1.4 to 1.5 x OD		Limit to TEMA standard spacing	-	-	Increase to 1.4 to 1.5 x OD
Fluid allocation	Switch sides	Switch sides	Switch sides	Switch sides	-	Switch sides
Arrangement	Increase # of exch. in parallel	Increase # of exch. in parallel	Increase # of exch. in series	Increase # of exch. in series	Increase # of exch. in series	Increase # of exch. in parallel
# Tube passes	-	Limit to one tube pass	-	Increase # tube passes	Limit to one tube pass	-
Tube type	Plain	Plain	Externally enhanced	Internally enhanced	-	-

C.1.2 Results

Multipass flow arrangement or heat exchangers in series cannot be used for the tubes when the process stream is on the tube side due to the risk of maldistribution of liquids.

The design chosen is the cross-flow NEN heat exchanger. The results for the designs evaluated are shown in Table C-3. Figure C-2 shows the general arrangement of the selected heat exchanger, and Figure C-4 shows the tubesheet layout.

Table C-3: Results from heat exchanger design in TASC+

Process fluid side	TEMA Type	Flow arrangement	Parallel units	Size m	Volume	Weight	Heat transfer Area	OHTC Dirty/Clean	Overdesign due to fouling
-	-	-	-	m	m^3	10^3 kg	m^2	W/m^2K	-
Counterflow	Tube side	NEN	1	5790 x 660 x 460	1,76	2,7	58,8	957 / 1382	45 %
Parallel-flow	Tube side	NEN	1	5989 x 660 x 460	1,82	2,8	61,8	959 / 1385	44 %
Cross-flow	Tube side	NXU	1	3729 x 834 x 635	1,98	6,1	69,6	815 / 1105	36 %
Multipass	Tube side		Excluded due to potential problems with phase distribution						
Counterflow	Shellside		Feasible design not found within the constraints						
Parallel-flow	Shellside		Feasible design not found within the constraints						
Cross-flow	Shellside	NXU	1	4400 x 892 x 692	2,70	5,9	76	749 / 962	29 %
Multipass	Shellside	NEN	2	4512 x 922 x 723	2,92	9,6	59,8	974 / 1368	42 %

	Units	Process side	Seawater side
Load	kW	-3760	
Flow rate	kg/s	32,117	45
Temp in	°C	95	0
Temp out	°C	60	20,44
Pressure in	bara	50	56
Pressure drop	bar	0,56	0,59
Design pressure	barg	200	10
Design temp	°C	95	95
Fouling resistance	m^2K/kW	0,00018	0,00009
Overdesign due to fouling	+%	44	
Material		Titanium	
OHTC dirty / clean	kW/m^2K	959 / 1385	
LMTD	K	63,32	
Area required	m2	61,8	
Core volume	mm	0,68	
Mass	tonne	2,8	

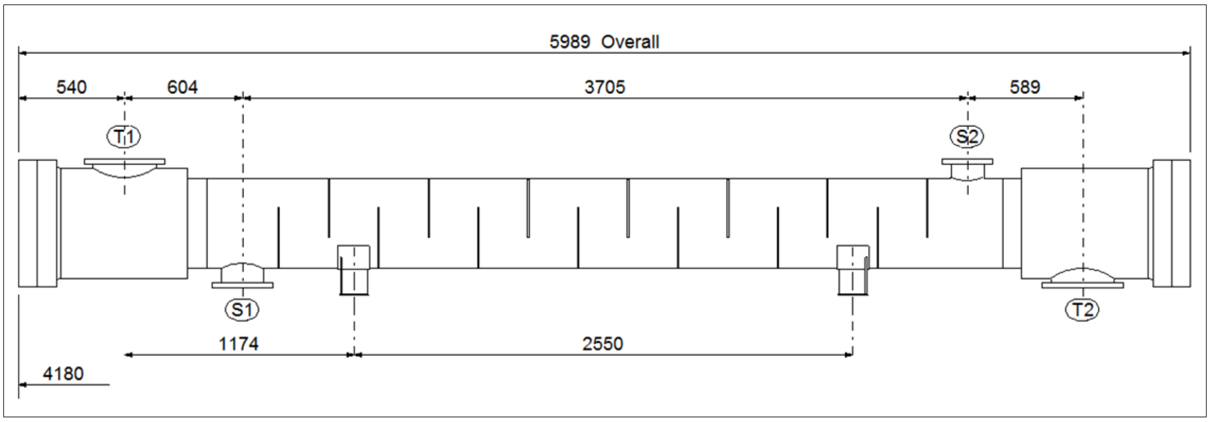

Figure C-2: General arrangement drawing of the selected heat exchanger

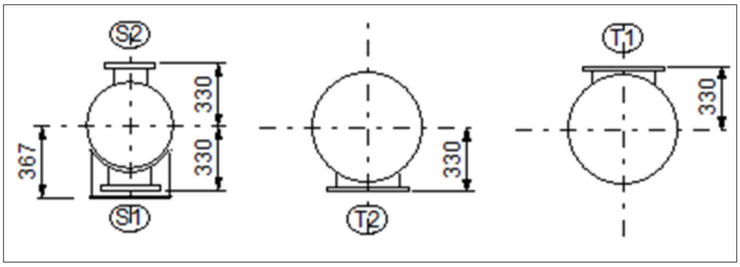

Figure C-3: The selected heat exchanger seen from the side

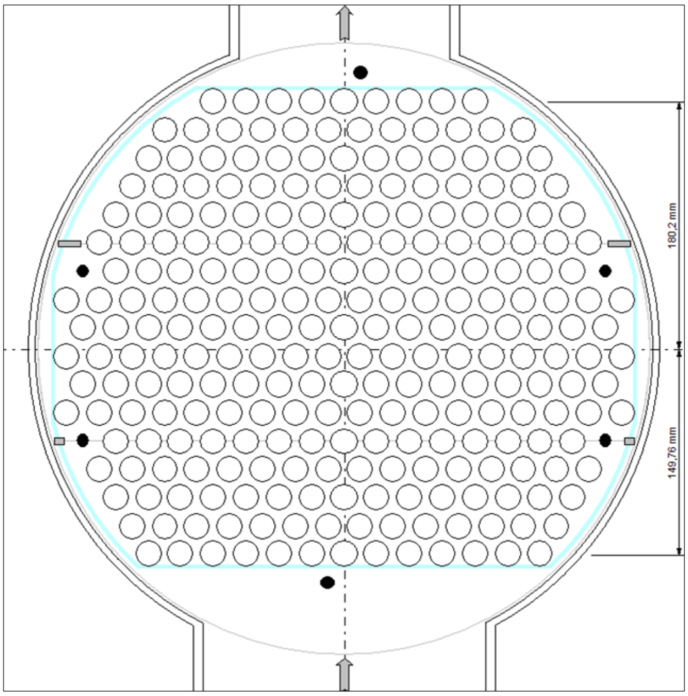

Figure C-4: **Tubesheet layout for the selected heat exchanger**

The temperature profile for the selected heat exchanger without fouling is shown in Figure C-5. It can be seen that both the process side wall temperature is above the hydrate formation temperature, and that the seawater side temperature is far below 60°C. As the Target temperature of the hot stream is 60°C and the flow arrangement is parallel-flow, the cold side temperature can never exceed 60°C, and the wall temperature will thus stay below this temperature for most cases.

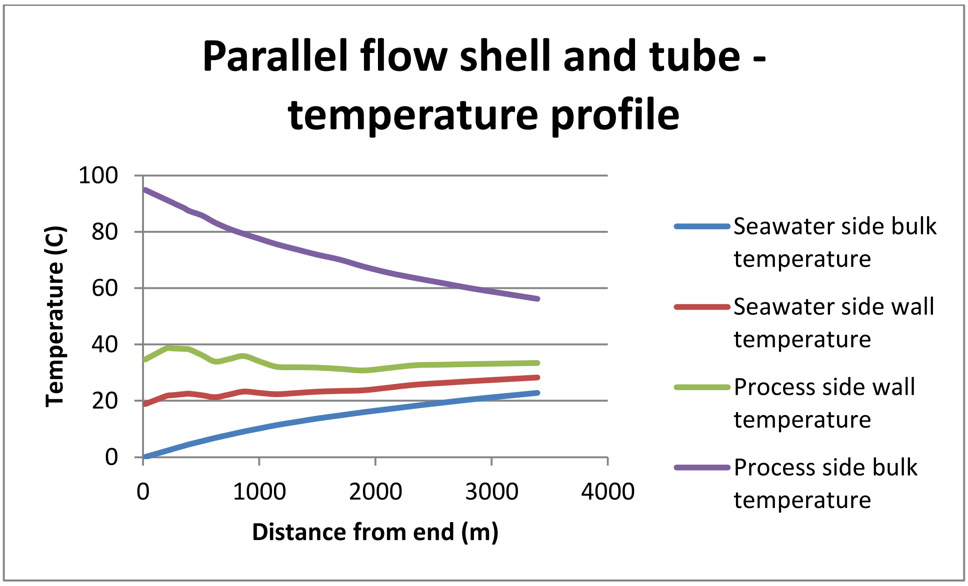

Figure C-5: Temperature profile for the selected heat exchanger

Figure C-6 to Figure C-9 shows how the heat flux is affected by change in flow rates or inlet temperatures for parallel-flow and counterflow arrangements during clean and dirty conditions. The simulations are for the two first NEN heat exchangers in Table C-3. The figures can be used to find the required area for a given flow rate or inlet temperature, or to find the required flow rate or inlet temperature for a given area during design.

As an example using Figure C-6 and the dirty heat transfer coefficient to define the size; if the heat exchanger is designed with a flow rate of 60 kg/s, a heat flux of 62 kW/m^2 can be expected. For the required heat transfer rate of 3760kW to cool the process stream to 60°C, an area of 60,6m^2 is needed.

When the heat exchanger is installed and is clean from fouling, a flow rate of 60kg/s would give a heat flux of 86 kW/m^2, and a total heat transfer rate of 5160kW when the area is 60,6m^2. This heat transfer rate is too high, and the process stream would cool too much. It is therefore necessary to reduce the flow rate until the heat flux of the clean heat exchanger is equal to the heat flux of the fouled heat exchanger. The blue line is the system curve for a clean heat exchanger and the flow rate can be found for the desired heat flux. At the heat flux of 60kW/m^2 the heat exchanger will operate with flow rate of 20kg/s.

This means that a clean heat exchanger requires 66% reduction in flow rate to operate with the design heat flux. The flow rate will then gradually be increased by the operator as the heat exchanger begins to foul over time, to maintain the design heat flux.

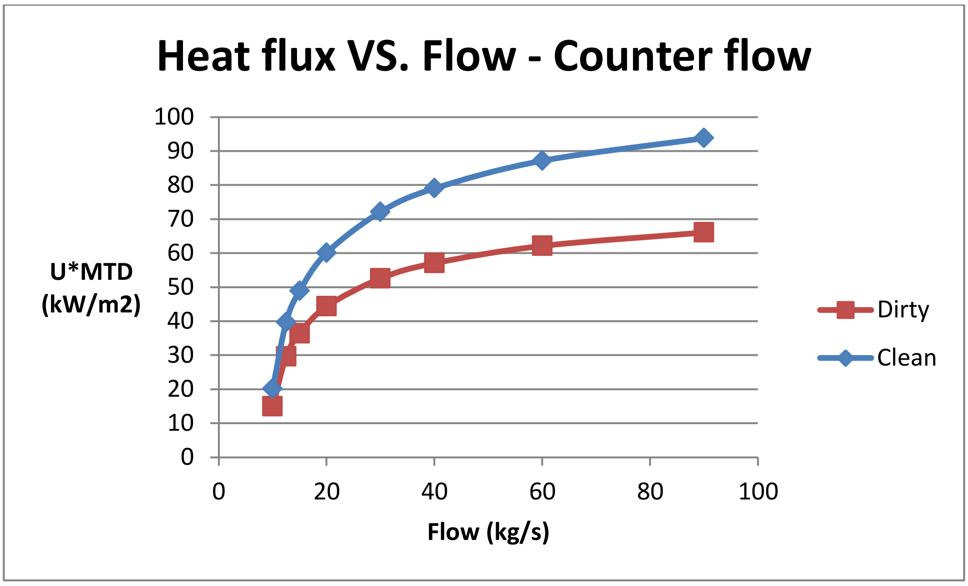

Figure C-6: Heat flux versus flow rate for the counterflow heat exchanger in Table C-3, with seawater on the shell side and 0°C fixed inlet temperature.

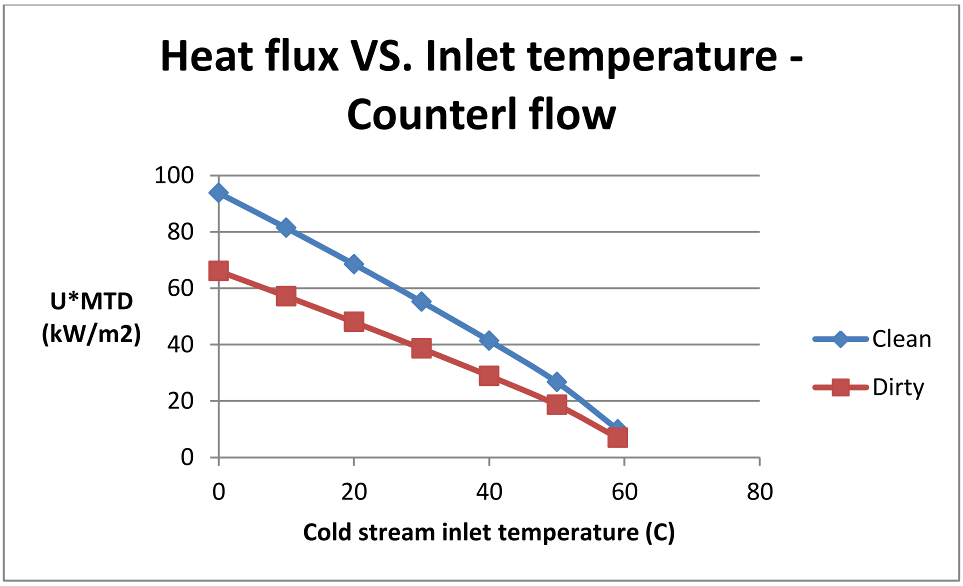

Figure C-7: Heat flux versus inlet temperature for the counterflow heat exchanger in Table C-3, with seawater on the shell side and 45kg/s fixed flow rate.

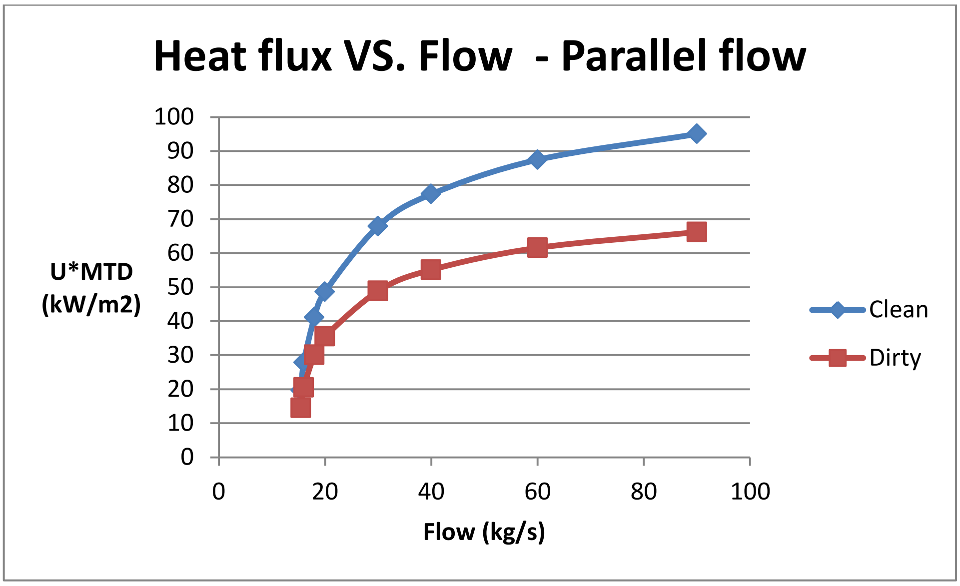

Figure C-8: Heat flux versus flow rate for the parallel-flow heat exchanger in Table C-3, with seawater on the shell side and 0°C fixed inlet temperature.

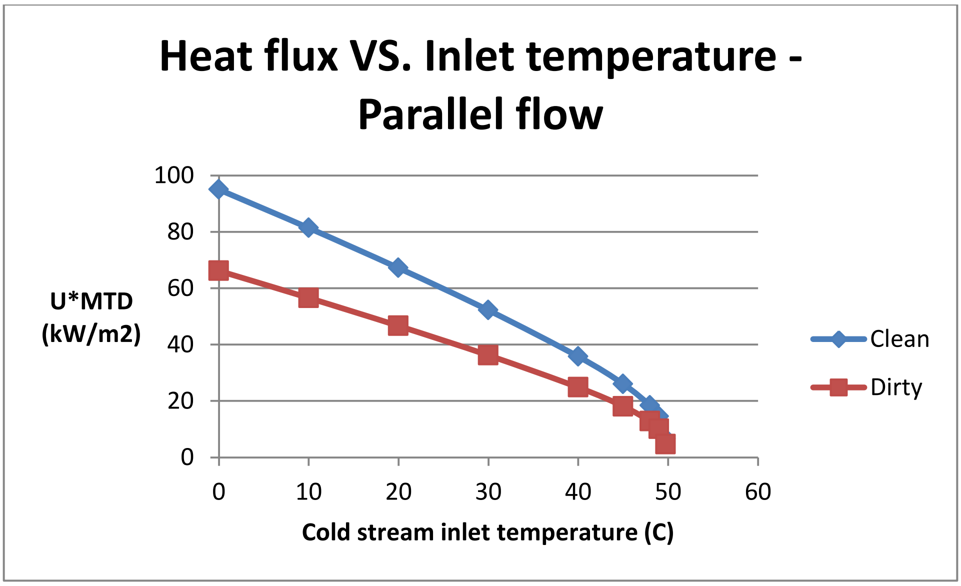

Figure C-9: Heat flux versus inlet temperature for the parallel-flow heat exchanger in Table C-3, with seawater on the shell side and 45kg/s fixed flow rate.

D – Direct printed circuit cooler

D.1.1 Method

An online sizing programme made available for registered users by the vendor of printed circuit heat exchangers Heatric, was used for the printed circuit heat exchanger sizing. This programme is however not very sophisticated, and except from including design pressure and temperature in the sizing. It seems like no more than a basic calculation using (2), and some additional correlations for size. However, as Heatric offers this as a sizing programme for their customers, it is assumed that the sizing at least gives an indication of the range of the heat exchanger size. The size given from the programme is the core dimensions on the heat exchanger, and not the size of the heat exchanger.

D.1.2 Results

Table D-1: Printed circuit heat exchanger sizing

	Units	Process side	Seawater side
Load	kW	-3760	
Flow rate	kg/s	32,117	45
Temp in	°C	95	0
Temp out	°C	60	20
Pressure in	bara	50	56
Pressure drop	bar	1	2
Design pressure	barg	200	10
Design temp	°C	95	95
Overdesign due to fouling	+%	20	
Material		Duplex	
Overall HTC	kW/m^2K	1,8	
LMTD	K	67,2	
Area required	m2	37	
Core dimensions	mm	450 x 600 x 600	
Mass	tonne	1,1	

The heat exchanger size from the sizing is the core dimensions, without heads. Simply from looking at pictures of printed circuit heat exchangers, one can see that the head designs are quite large compared to the heat exchanger core. To make the heat exchanger in Table D-1 more comparable to the design in TASC+, it is assumed for that the heads with nozzles build 350mm out on each side of the heat exchanger core. The flow arrangement will affect the header design in printed circuit heat exchangers [28], and a cross-flow arrangement is chosen. Figure D-1 shows an overview of the heat exchanger with heads and nozzles. The nozzle sizes are the same as for the process side of the shell and tube heat exchanger in section C.

The assumptions of head size were used for all the printed circuit heat exchangers in the case study.

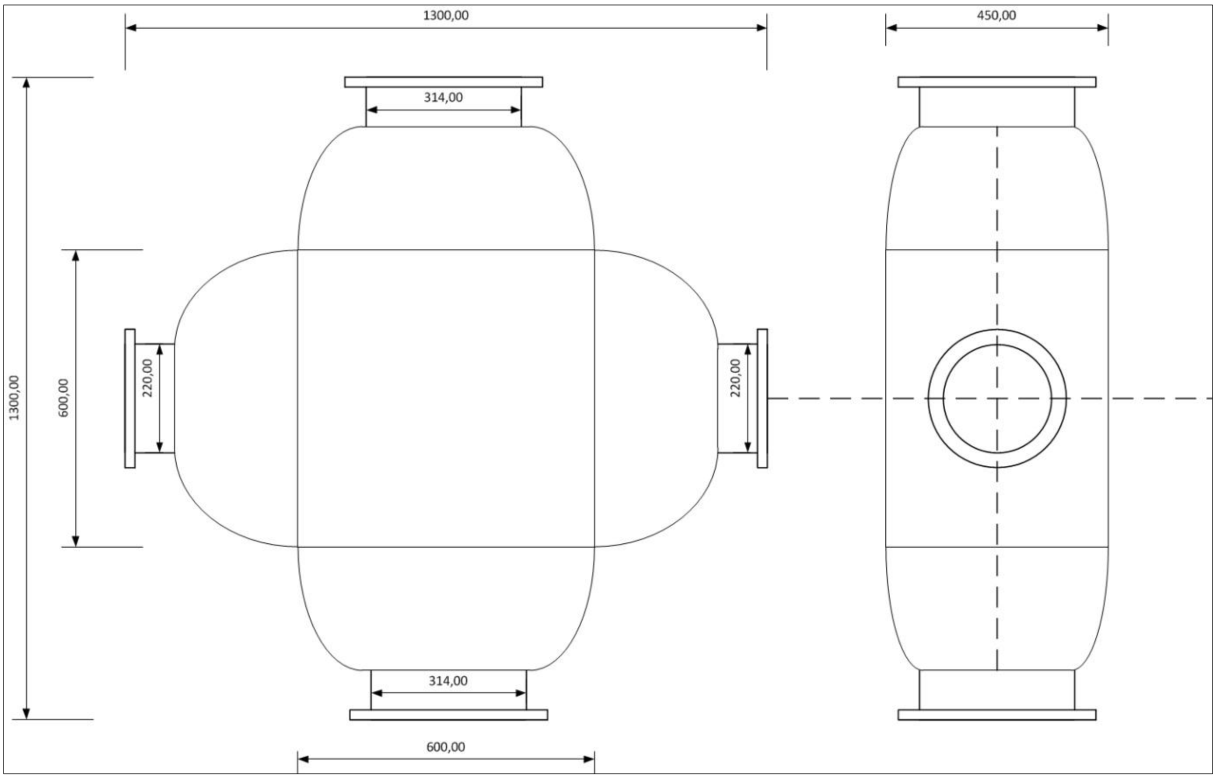

Figure D-1: Direct cooling: printed circuit dimensions, with heads and nozzles

The printed circuit heat exchanger will also require a strainer mounted on the process inlet, which will take up some extra space.

E – Indirect printed circuit cooler with passive dump cooler

The size of an indirect cooling system depends on the heat transfer area required both by the process cooler and the dump cooler. An optimization can be done on the area, but this is time consuming, and of little value without very accurate data. The tendency is however known, and the smallest temperature difference should be in the printed circuit heat exchanger, which has the highest u-Value and surface density. The minimum temperature difference in the printed circuit heat exchanger was set to 20°C. A mix of 20% MEG in water was used as the cooling medium in the simulations. This gives the following operating conditions in the closed cooling medium circuit:

- High temperature: 75°C
- Low temperature: 40°C
- Flow rate: 27 kg/s

E.1.1 Method

The printed circuit heat exchanger was sized in the same way as the printed circuit heat exchanger in section D, and the passive cooler was sized in the same way as the passive cooler in B, but with cooling medium on the hot side instead of the process stream. The same tube size and pitch was used.

E.1.2 Results process cooler

The results for the printed circuit heat exchanger size are shown in Table E-1, and the design with added length to compensate for headers and nozzles are shown in Figure E-1.

Table E-1: Printed circuit heat exchanger sizing

	Units	Process side	Cooling medium side
Load	kW	-3760	
Flow rate	kg/s	32,117	27,1
Temp in	°C	95	40
Temp out	°C	60	75
Pressure in	bara	50	56
Pressure drop	bar	1	2
Design pressure	barg	200	10
Design temp	°C	95	95
Overdesign due to fouling	+%	20	
Material		Duplex	
Overall HTC	kW/m^2K	1,8	
LMTD	K	50	
Area required	m2	129	
Core dimensions	mm	450 x 600 x 1800	
Mass	tonne	3,9	

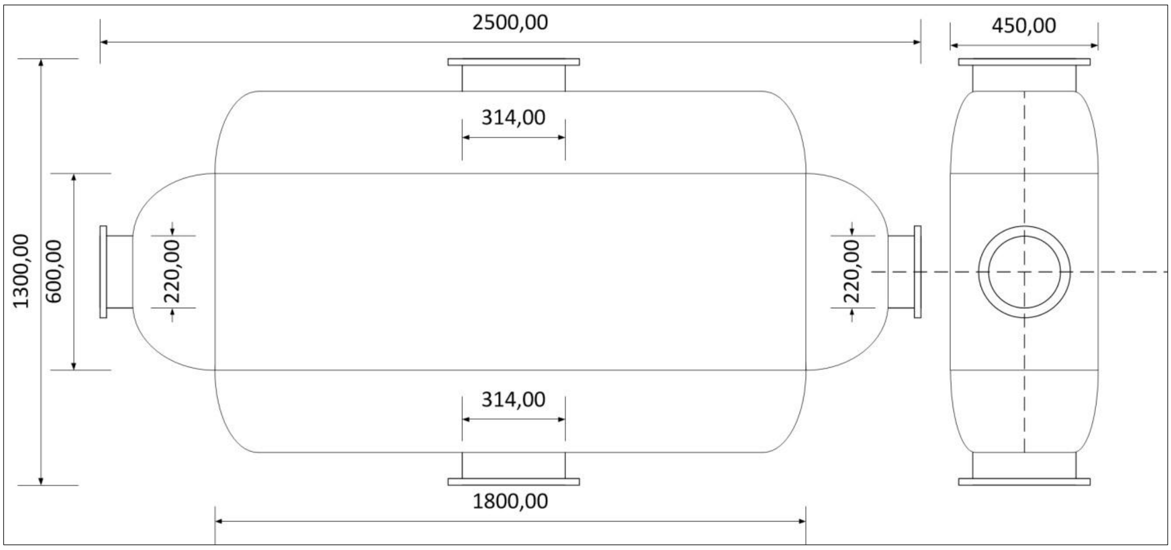

Figure E-1: Indirect cooling: printed circuit dimensions, with heads and nozzles

E.1.3 Results dump cooler

The size of the passive cooler for the indirect loop is shown in table Table E-2. The temperature profile over the passive cooler is shown in Figure E-2.

Table E-2: Passive cooler size

Description	Units	Results
Temp in	°C	75
Temp out	°C	40
Overall HTC clean / dirty	W/m^2K	641 / 606
Effective MTD	°C	55,71
Area needed	m^2	111
Number of tubes		28
Tube length	mm	34 900
Cooling medium side pressure drop	Bar	0,86
Cooling medium side velocity	m/s	1,4
Tube pitch	mm	50
Tube passes		12
Dimensions	mm	2900 x 2212 x 1106
Weight	kg	2,9

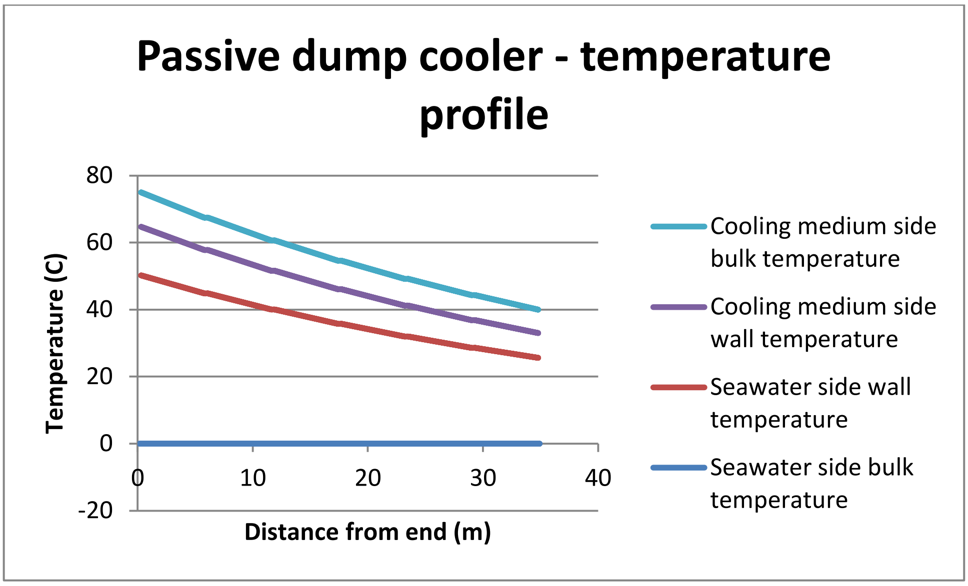

Figure E-2: The temperature profile for the passive dump cooler

F – Indirect printed circuit cooler with plate dump cooler

If a plate heat exchanger is used for the dump cooler design in an indirect cooling system, an extra pump is needed to run seawater through the plate heat exchanger. The overall heat transfer coefficient and heat exchanger surface density are much closer in a plate heat exchanger and a printed circuit heat exchanger, than in a printed circuit heat exchanger and a passive heat exchanger. This means that the temperature difference could be equally divided between the two. Again a mixture of 20% MEG in water was used as cooling medium. A minimum temperature difference of 30°C was set in both the plate heat exchanger and the printed circuit heat exchanger. This gives the following operating conditions in the closed cooling medium circuit:

- Closed circuit high temperature: 65°C
- Closed circuit low temperature: 30°C
- Flow rate: 27 kg/s

F.1.1 Method

The printed circuit heat exchanger was sized in the same way as the printed circuit heat exchanger in section D. The plate heat exchanger was designed using the Plate+ module of HTFS for thermal-hydraulic design.

F.1.2 Results process cooler

The results from the printed circuit heat exchanger sizing are shown in Table F-1. The geometry of the core with headers is shown in Figure F-1.

Table F-1: Printed circuit heat exchanger sizing

	Units	Process side	Cooling medium side
Load	kW	-3760	
Flow rate	kg/s	32,117	26,3
Temp in	°C	95	20
Temp out	°C	60	55
Pressure in	bara	50	56
Pressure drop	bar	1	2
Design pressure	barg	200	10
Design temp	°C	95	95
Overdesign due to fouling	+%	20	
Material		Duplex	
Overall HTC	kW/m^2K	1,8	
LMTD	K	40	
Area required	m2	63	
Core dimensions	mm	450 x 600 x 900	
Mass	tonne	1,9	

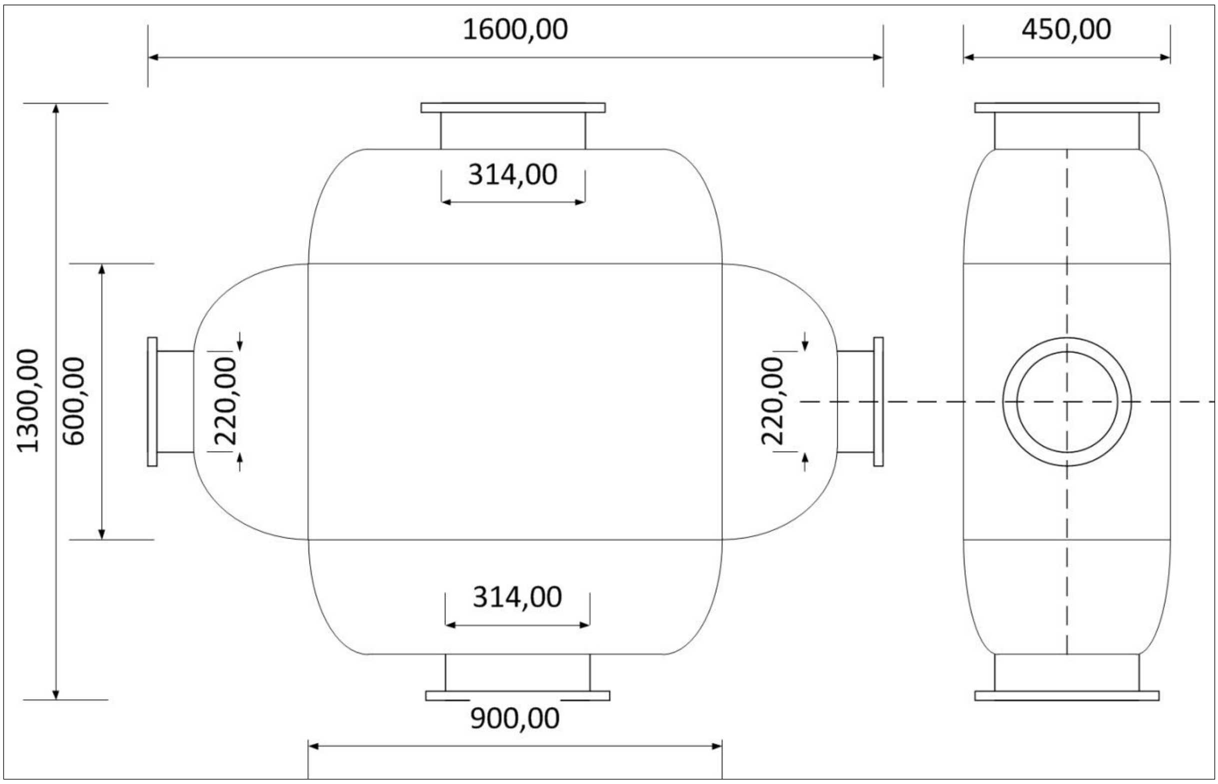

Figure F-1: Indirect cooling with plate dump cooler: printed circuit dimensions, with heads and nozzles

F.1.3 Results dump cooler

The results from HTFS Plate is shown in Table F-2, the temperature profile for this heat exchanger is shown in Figure F-2, and the geometry shown in Figure F-3.

Table F-2: Plate dump cooler design from HTFS Plate+

	Units	Cooling medium	Seawater side
Load	kW	-3760	
Flow rate	kg/s	32,117	26,3
Temp in	°C	55	0
Temp out	°C	20	35
Pressure in	bara	55	56
Pressure drop	bar	1,17	1,10
Design pressure	barg	10	10
Design temp	°C	95	95
Fouling resistance	m^2K/ kW	0	0,00003
Overdesign due to fouling	+%	12,7	
Material		Titanium	
Overall HTC Clean / Dirty	kW/m^2K	4380,8 / 3932,9	
Effective MTD	K	20	
Area required	m2	48,4	
Core dimensions	mm	360 x 380 x 1475	
Mass	tonne	0,388	

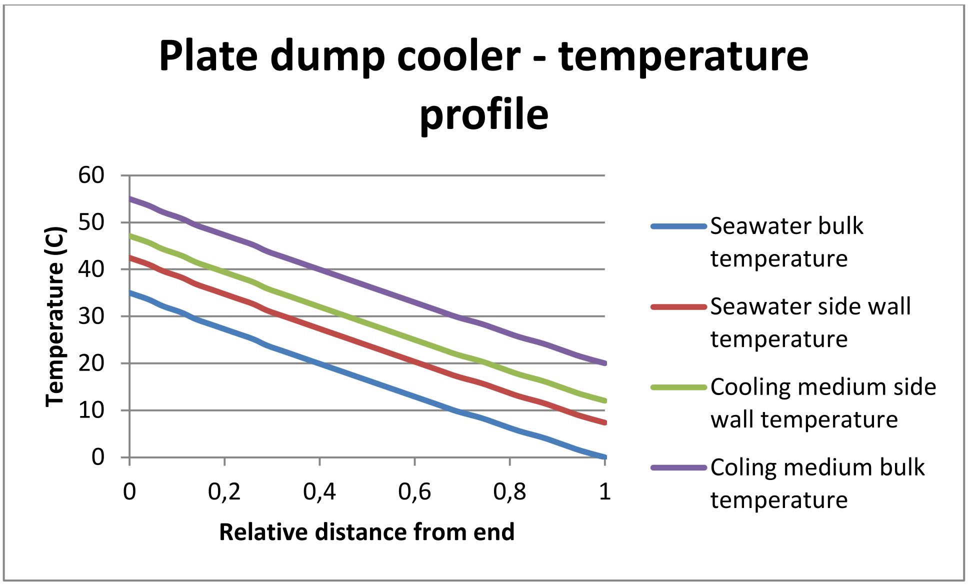

Figure F-2 Temperature profile for the plate dump cooler

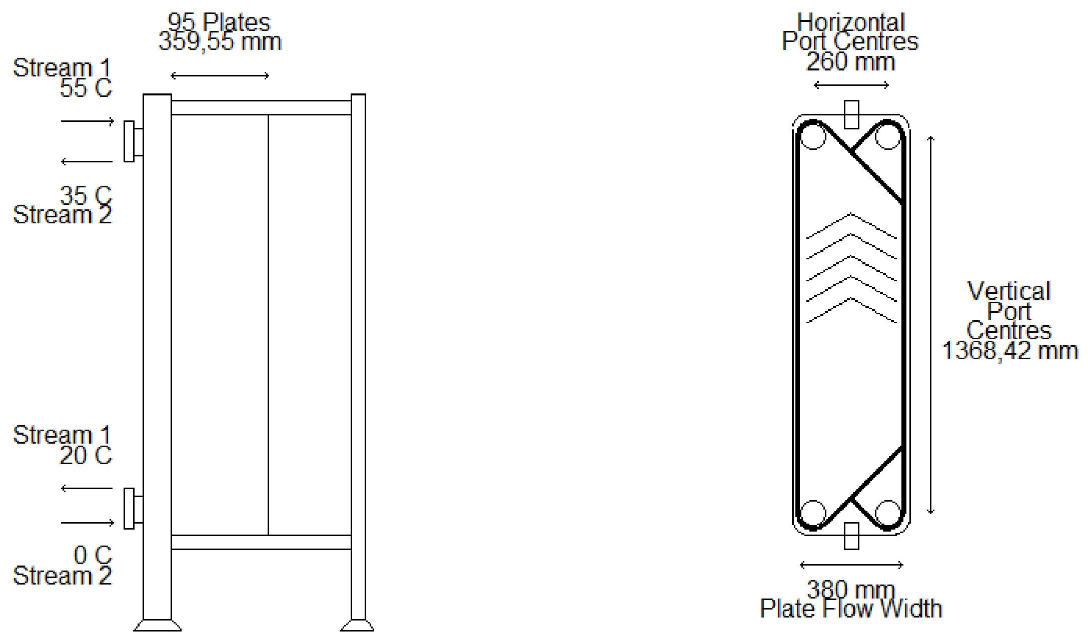

		Plate thickness	0,9 mm
Actual surface area	49,4 m2	Compressed plate pitch	3,83 mm
Number of passes Stream 1 / 2	1 / 1	Area of each plate	,52 m2
Effective channels Stream 1 / 2	47 / 47	Chevron angle (to horizontal)	30
Number of exchangers	1	Material type	Stainless
		Port diameter	100 mm

Figure F-3: Dimensions for the plate dump cooler