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Electrically driven heat pumps for process heat supply in oil and gas production facilities

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Abstract

The global energy demand is increasing together with a larger focus on reducing the global warming influenced by political restrictions and carbon taxes that force to find new ways to avoid emissions. Large amounts of low temperature heat are available in oil and gas facilities and they are not exploited due to lack of waste heat utilization. Available low-grade heat has a temperature range suited as heat sources for heat pumps in industrial processes.

Industrial scale heat pumps challenge the problem using natural refrigerants or hydrocarbons. Evaluating them, heat and power supply from other sources can achieve the goal of minimize the CO_2 emissions from petroleum activities in electrical driven oil and gas processing facilities.

The evaluation of the proposed heat pump system is based on a working fluid selection and overall efficiency analysis in high temperature heat pump applications. A literature review was carried out to find potential heat sources and process heat requirements clarifying heat duties and temperature levels in the facility.

It defines two main cases to evaluate the performance, one for moderate temperature application (150 ℃) and another for high temperature (200 ℃) application. Results as capacities, heat and compressor duties, or temperatures can help to select the correct fluid based on its behaviour in the system. Ammonia represents a good behaviour as working fluid in heat pump application together with water but its huge volume flow can involve excessive equipment dimensions compared to ammonia. Hydrocarbons as pentane or butane are also analysed, obtaining good results. Different system such as multi-stage, cascade or gas phase systems are introduced to analyse different fluid behaviours.

Features regarding complexity and equipment specifications are studied based on the results obtained. Axial or centrifugal compressors would be very suitable for this application. Considering heat exchangers, shell and tube exchangers could be problematic due to two-phase streams requiring welded plate and frame exchangers for closer approaches.

Economic aspects are calculated using electricity costs and heat supply costs compared to a "standard" FPSO with gas turbines as the main driver. CO_2 emissions are also calculated analysing the results of the emissions saved and taxes involved.

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1 INTRODUCTION

1.1 Background

The global energy demand is increasing together with a larger focus on reducing the global warming that enforces industrial production to act more energy efficient and environmentally friendly. Large amounts of low temperature heat are available and they are not exploited due to lack of waste heat utilization. Available low-grade heat has a temperature range suited as heat sources for heat pumps in industrial processes. The application in industrial installations requires a more complex specification and it comprises different ways of heat integration and levels of heat source temperatures.

Political restrictions and carbon taxes are forcing to find new ways to avoid emissions. In 2016, greenhouse gas emissions from petroleum activities corresponded to about 13.8 million tons CO_2 eq. (carbon dioxide equivalent) comprising 82 % emissions from gas turbines [1]. It appears the necessity to replace them, supplying heat and power from other sources, aiming at minimized the CO_2 emissions from petroleum activities. Industrial scale heat pumps challenge the problem using natural refrigerants or hydrocarbons that not will increase global warming or the ozone layer depletion.

Several heat pump systems compatible with high temperature heat recovery in industrial processes exist, but their utilization is not wide spread yet. Evaluating the potential for each respective heat pump system should be possible to determine and rate their relative applicability at different temperature levels. The implementation of industrial scale heat pumps in high temperature operations can help lower operational cost in industrial processes as well as reduce their emissions of substances harmful to the environment.

1.2 Scope

The objective of the Master thesis is to evaluate heat pump options for heat supply to electric driven oil and gas processing facilities, aiming at minimized electric power need and CO_2 emissions, as well as acceptable features regarding complexity and cost, equipment size/weight, and safety.

Based on an initial work in the fall semester, there is a need for more detailed analyses of specific system solutions with heat pumps, including refrigerant selection, process design, heat source selection, and system configuration.

Analyses of changes in electric power input and total CO_2 emissions for relevant installations need to be included, as well as assessment of system complexity, equipment size/weight, safety aspects, and potential use in revamp/retrofit and new-build situations. The analyses may be linked to specific cases with realistic/available heat sources, heating temperature needs, and capacity requirements.

1.3 Work structure

The project consists of a literature review followed by an evaluation of the proposed heat pump system with the working fluid selection for high temperature heat pump applications. The project is divided into 9 chapters including the references.

Following the introduction with the background and scope in Chapter 1, the second chapter provides a brief explanation of the processes in oil and gas facilities according to the heat requirements. It separates the heat requirements in two types of facilities depending on the production: oil + associated gas and gas-condensate field.

Chapter 3 resumes the potential heat sources available in the plant that can be used for the heat pump systems. Chapter 4 explains the basis for the analysis, defining the temperature levels, modelling and equipment specifications.

Chapter 5 includes the working fluid selection, evaluating and analysing the results obtained of different fluid behaviours in the heat pump modelled. It has been divided into pure fluids, cascade system, mixture fluids and gas phase fluids. Chapter 6 describes the heat pump integration in the processes and the description (sizes, capacities…) of the equipment involved.

In Chapter 7, calculations of CO_2 emissions are included together with energy supply costs, analysing the results of the emissions saved and the economic aspects involved in comparison with a common FPSO with gas turbines to supply power and heat.

Discussions are included in Chapter 8 including the conclusions and suggestions for further work, followed by references in Chapter 9.

2 PLANT PROCESS DESCRIPTION AND HEAT REQUIREMENTS

In this chapter, a brief explanation describes the thermal processes that occur in the plant with the heat requirements involved. The aim in this chapter is to evaluate the temperature levels and heat duty demands in the plant, resuming all in Table 2.1 at the end of the Chapter. Sections 2.1 and 2.2 separate the heat duties according to the plant production type: oil-associated gas and gas-condensate production.

2.1 Oil and associated gas production

When an oil with associated gas production characterizes the reservoir, the dominating heat demand can be in the well stream heating or in the associated gas treatment. With cold reservoirs well stream heating is required in the inlet (before entering in the separation train) to keep the optimal temperature for a correct performance in the separation. Figure 2.1 illustrates where the heat requirement is located in the separation train. The first one is located in the inlet just before the first separator and the second requirement between the first and second stage.

2.1.1 Well stream heating

The main objective is to achieve a maximum liquid recovery from the stream, stabilizing the oil hydrocarbon and gas stream after two or three stages. Reservoir temperatures can vary from tens of degrees to temperatures higher than sixty degrees. Depending of the reservoir temperature, the heating required can be higher or lower. The heat vaporizes the lighter hydrocarbons extracted in subsequent separators and the resulting gas is sent to the gas processing train. The heating up is also done to achieve the required vapour pressure specifications of the product stream at the outlet of the final separator. In general, the optimal temperature required in the inlet should be around 50 - 60 ºC avoiding hydrates when depressurizing.

2.1.2 Separation stages

Between the stages, heat is also required to improve the separation performance. In this part, almost all the water has been removed from the stream in the first separator. For this reason, the heat duty required in this part is much lower than for the well stream (referencing the previous case). The heating between stages should keep the optimal temperature, increasing the temperature level until a range of 70 - 80 ºC.

2.1.3 Heat demand examples (basis for analysis cases)

Considering the reference [2] related to the well stream heating and based on a mixing stream, it can extract some useful values. The stream comprises a domination of oil and water cut with associated gas.

The oil production of the plant is around 200,000 bpd with two well streams from the reservoir at temperatures of 30 ºC (Field 1) and 45 ºC (Field 2). Both streams are heated until 60 ºC before the first stage, with a high GOR (1986 STD_m^3/m^3) and flow rates of 7645 m^3/h for the Field 1 and 8814 m^3/h for the Field 2. The heat duty required in the well stream heating for both streams is about 280 MW, which means a significant heat duty. That large heat duty is based on the high water cut, involving the 80 - 85 % of the stream.

The large water amount contained in the stream produces the increase of the heat required. Large water production can occur at end-life production wells where the oil production rate is low. To get a general view of the heat demand, avoiding specific cases (end-life conditions and high water cut) and assuming for the analysis cases a demand around 140 MW, which is the half of the heat demand of the reference mentioned.

As mentioned, the heat is also required between the separation stages. Referring to the same case (oil and water dominated with associated gas), the heat duty among the separation stages is around 17 MW with a flow rate of 1378 Sm^3/h [2]. That value is obtained after removing most of the water contained. It can assume a general heat duty requirement of 20 MW for the analysis, covering the heating demands for 200,000 bpd between the separation stages.
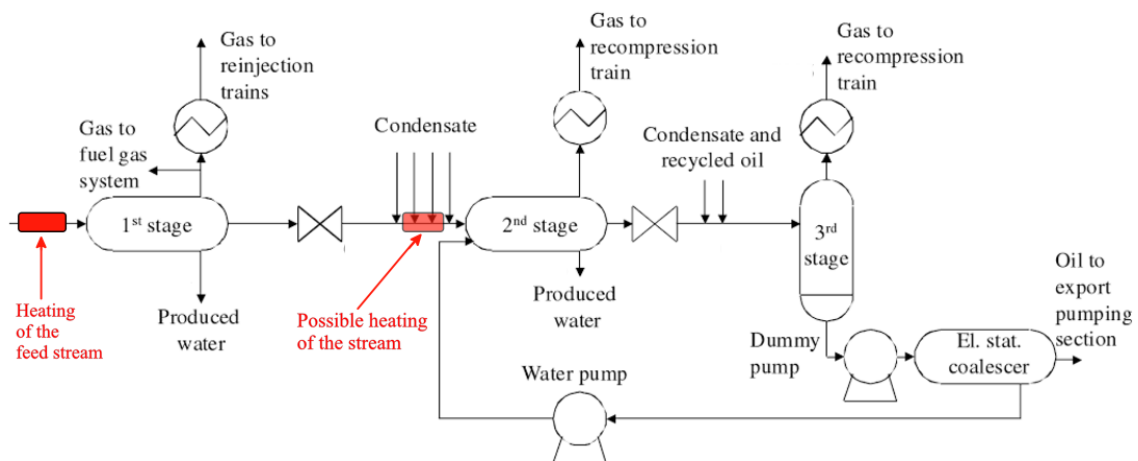
Figure 2.1 Heat requirements in the separation train [25]

2.2 Gas and condensate production

On the contrary, a gas-condensate field requires more heat demand in the gas treatment part than in the oil stabilization. The main heat demands are gas dehydration with TEG regeneration and condensate stabilization.

2.2.1 Gas dehydration treatment

The gas extracted and compressed is dehydrated using the absorption process with Tri-Ethylene Glycol (TEG). Incoming gas (wet gas) is first cooled before it enters in the gas treatment facility. That stream cooling is analysed as a potential heat source in Chapter 3.

During the gas dehydration process with TEG glycol, the water is absorbed by the glycol from the wet gas. The rich glycol is regenerated removing the absorbed water in the regenerator. The regeneration process consists of heating the depressurized rich glycol evaporating the water fraction contained. Considering the process description of the reference [3], the dehydration with TEG occurs at temperatures of 200 ºC avoiding higher temperatures because of potential glycol degradation. The lean glycol is re-used to absorb the water from the stream, completing the loop.

Figure 2.2 illustrates where the heat requirement is located together with the heat recovery. The main heat requirement is in the reboiler of the column while the residual heat of the product is recovery to heat the inlet before the column.

Figure 2.2 Heat supply and heat recovery in the TEG dehydration process [25]

2.2.1.1 Example of heat requirements in the condensate stabilization

Based on an oil field with a gas associate production of 20 MSm^3/day [8] and TEG flow of 119 m^3/day to reach the water dew point of -18 ºC, at 70 bar(a) to meet specifications. The reboiler in the regenerator keeps the process temperature at 200 – 205 ºC requiring 3 MW for the flow mentioned.

The lean glycol stream leaves the regenerator at this temperature and it is cooled using the residual heat to heat the inlet stream (rich glycol) in the regenerator. The rich glycol (with water absorbed) at around 15 – 20 ºC is heated until 100 ºC with this residual heat before it enters the regenerator. The second temperature level increase in the reboiler, from 100 ºC to 200 ºC.

In addition, studying the Snøhvit field, the heat duty required by TEG dehydration process would be around 20 - 30 MW for a gas production of 20 MSm^3/day [9]. In short, the heat requirement for the gas dehydration with TEG can assume a general heat demand of 30 MW to cover the entire heat requirement in the process for the production mentioned.

2.2.2 Condensate stabilization

A condensate stabilization unit is often necessary. All the liquid fractions produced in the process must be stabilized before its export and storage to avoid gas phase separation in pipelines or tanks due to the light components.

In the stabilizer, the light hydrocarbons boil-off from the condensate stabilization. The storage conditions govern the final temperature and pressure of the stabilized condensate. Normally, the vapour pressure determined as Reid Vapour Pressure (RVP) is the main parameter with a value of 0.70 bar(a) in summer or 0.80 bar(a) in winter.

If the feed temperature to the stabilizer increases, more percentage of light components will evaporate from the condensate; thus leaving less amount of volatile component in the product, reducing the RVP of the product and reducing the liquid volume to be sold. Figure 2.3 represents the heat requirements and heat recovery in the condensate stabilization process. As in the previous case, the main requirement is in the reboiler of the regenerator while the residual heat of the product is recovery to heat the inlet before the column.

2.2.2.1 Example of heat requirements in the condensate stabilization

During the stabilization process, the stabilizer works at high temperature levels around 200 ºC [4]. A condensate stabilization process has been modelled with a condensate production of 150 m^3/h (22 000 bpd) similar to the Snøhvit field that contains rich gas with condensate [4]. The feed enters into the stabilizer at 10 - 15 ºC, leaving the product from the bottom at 200 ºC. The RVP at 37.8 ºC is 0.80 bar(a) according to the limit in winter.

The heat duty required by the reboiler to reach this temperature, according to the specification, is 14 MW. Normal heat duties values are around 9 - 10 MW for the same condensate production [3]. Depending on the gas composition, the heat duty in the reboiler will vary but, in general, a heat duty of 10 - 15 MW would be enough to cover the heat requirements in the condensate stabilization for a production of 20,000 bpd.
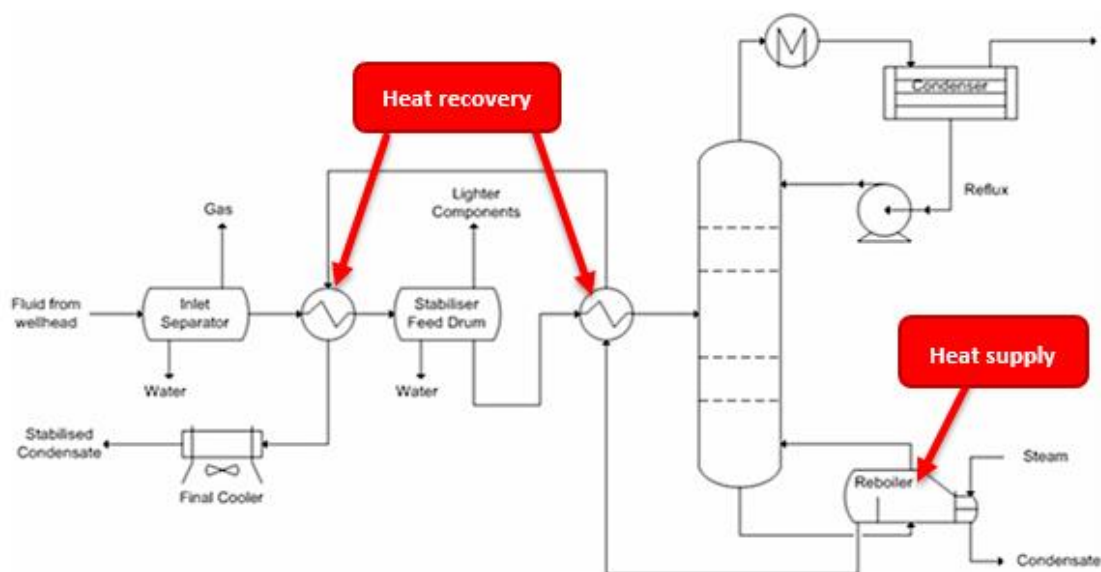
Figure 2.3 Heat requirement and heat recovery in the TEG dehydration process [25]

2.3 Heat requirements summary

Table 2.1 groups the heat duty requirements of both types of production to get an overview for further work. It can identify four main heat consumers: in the oil production, the largest demand is in the well stream heating and in the oil-gas-water separation process, while for the gas production facility the largest is the condensate stabilization and the gas dehydration with TEG-glycol.

Table 2.1 Summary of the process heat requirements

Heat consumers	Oil + associated gas production			Gas + condensate production		
	Oil-Gas-Water Separation			Condensate Stabilization	TEG Gas Dehydration	
	Well stream heating	Heating between separation stages				
Plant production rate	200 000 bpd			22 000 bpd	20 MSm^3/d	
Heat duty (MW)	280	6	11	15 (Reboiler)	30	
Fluid heated	Well stream	Oil stream		Condensate	Rich solvent	
Inlet temperature (°C)	30 - 45	55	65	10 - 15	15 - 20	100 (feed to the regenerator)
Outlet temperature (°C)	55	65	80	180 - 200	100	200

3 POTENTIAL HEAT SOURCES

Normally, the processes mentioned in Chapter 2 require outlet stream cooling in order to meet the product specifications. For this reason, it could be an option use that residual heat from the cooling despite of rejecting it to the seawater. The chapter describes these potential sources with the temperature levels and flow rates available, summarizing all of them in Table 3.1 at the end of the chapter.

3.1 Residual heat from the oil-gas-water separation train

The stabilized oil temperature leaving the separation unit is too high (75 ºC) in order to meet the product specifications and it should cooled down to 25 – 40 ºC. Actually, the residual heat is often used to heat the feed before the first separation stage reducing the temperature by 10 ºC, but the stream has to be cooled again by seawater until its export conditions (50 ºC) because the temperature is still high (illustrated in Figure 3.1). That residual heat rejected to the seawater it can become a potential heat source. The cooling duties related to the oil stabilized stream can reach in total 16 - 20 MW [2], shown in the Table 3.1.

In Table 3.1, the cooling duty is divided in two heat exchangers (HX): first HX is actually used to heat the feed stream and the second HX is used to reject heat to the seawater (illustrated in Figure 3.1). The stabilized oil (potential heat source) would exit the separation unit at 70 - 80 ºC and it should be cooled until 50 - 60 ºC depending on the specifications. The heat source flow rate available in the system is the oil export flow rate, in this case, the flow rate is 1333 Sm^3/h equal to 200 000 bpd of oil production.
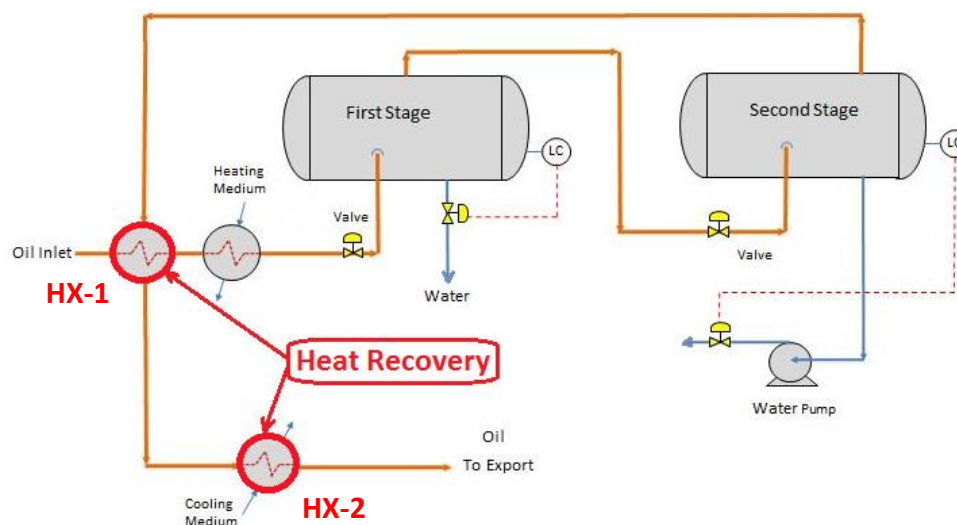
Figure 3.1 Heat recovery in the oil-gas-water separation train (two stages) [25]

3.2 Cooling from the gas re-compression

In the gas recompression, no heat requirement is required but cooling between compression stages is required to reduce compression power and temperature level. The outlet temperature and the compression work in the compressor depend on the stream composition, molecular weight and pressure ratio. Referred to a gas stream of 84 % methane, the temperature reached at the compression outlet is up to 150 ℃, increasing the pressure from 14 bar to 55 bar (first compression stage) [2]. Figure 3.2 illustrates where the potential heat recovery is located in the gas re-compression train.

For another case [5], the temperature reached after the first recompression is up to 130 ℃, increasing the pressure from 20 bar to 60 bar. In short, the range of temperatures can be around 120 – 150 ℃ in the outlet so cooling is required. The energy recovery potential is moderate due to the moderate temperatures and the heat transfer. Considering an example in reference [5], an export compression train with a gas flow rate of 540 t/h, gas production of 15 MSm^3/d, can require a cooling duty of 20 - 25 MW between compression stages. The temperature in the compressor outlet reaches 125 ℃ being cooled down until 100 ℃ before the next compression stage. The gas stream is cooled using a coolant, e.g. seawater which defines the minimum low temperature reachable in the cooling.

In other example from reference [8], for a similar gas production rate, the cooling duty required is higher reaching 60 MW. In this case, the gas stream temperature is reduced from 140 ℃ to a lower temperature 30 ℃ avoiding condensation. To sum up, analysing both examples it may conclude that the gas stream can be a potential heat source with a maximum estimated duty of 60 MW and temperature levels around 120 - 150 ℃ down to approximately 30 ℃.
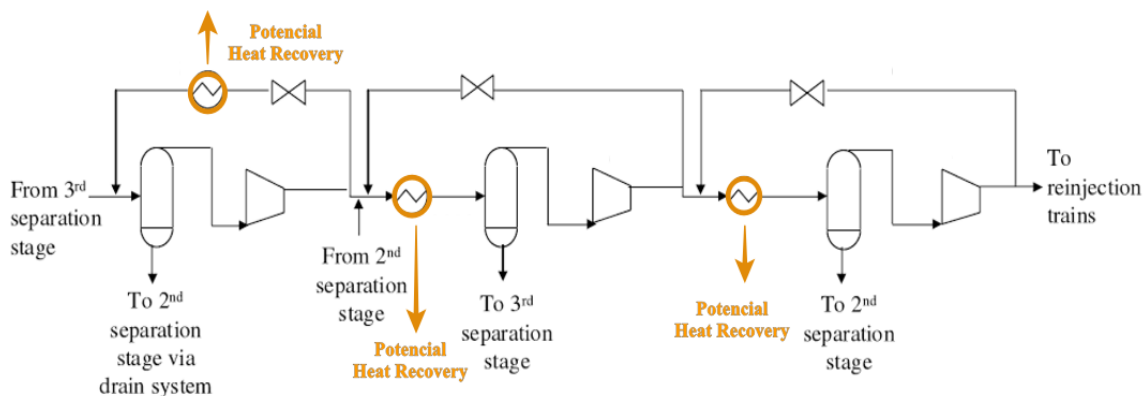
Figure 3.2 Potential heat recovery in the gas re-compression train [25]

3.3 Cooling from the condensate stabilization

In the condensate stabilization, cooling is required to cool down the feed before the process to reduce the gas fraction but also, the condensate product is cooled to meet the specifications for the storage or export.

The feed arrives at the unit at temperatures no higher than 40 ℃ and it should be cooled until temperature level of 15 - 20 ℃. The feed cooling duty can comprise values of 300 MW for a gas feed of 20 MSm^3/d and condensate production of 22 000 bpd, similar gas production as Snøhvit gas and condensate field [4]. This potential heat source is characterised by the low temperature level (maximum level of 40 ℃).

The second cooling requirement in the stabilization is required to cool the condensate product. As explained in Section 2.1.4, the condensate leaves the stabilizer at a temperature level of 200 ℃, too high for the storage specifications. The condensate product is cooled down until temperatures ≤ 60 ℃, depending on the storage/export specifications. The cooling duty in this part can reach values around 10 – 15 MW with a condensate flow rate of 150 m^3/h. In Table 3.1 appears detailed information about it.

3.4 Cooling from the TEG gas dehydration

Like the condensate stabilization, in the dehydration unit the gas feed (wet gas) should be cooled until low temperatures around 20 - 25 ℃. The temperature is decided based on the hydrocarbon phase envelope to ensure that the temperature of the wet gas stream entering the TEG contactor is above the hydrocarbon dew point. Depending on the initial temperature of the gas stream and the flow rate, the cooling duty can comprise tens of megawatts.

The cooling reduces the temperature to 25 - 35 ℃ before the inlet separator. That cooling represents a very low temperature source as a potential heat source for the heat pump system. As mentioned in Section 2.2, the regenerated glycol exits the bottom of the regenerator at around 200 ℃ and it is cooled until temperatures around 50 ℃ to be re-used again. The feed (rich glycol), which enters the regenerator, uses the residual heat from the hot regenerated stream. Considering the gas production rate mentioned in the previous sections with a gas production of 20 MSm^3/h, the cooling duty can represent a range of 30 MW [8]. See Table 3.1 for a detailed data summary.

Table 3.1 Potential heat sources summary

POTENTIAL HEAT SOURCES					
	Oil-Water-Gas Separation Train		Export Gas Compression	Condensate Stabilization	TEG Gas Dehydration
Plant production rate	200 000 bpd (Oil, high GOR)		15 MSm^3/d (Gas, high GOR)	22 000 bpd (Condensate)	20 MSm^3/d (Gas field)
Description	Residual heat in the cooling for oil stabilization (HX-1 / HX-2 see Figure 3.1)		Cooling of the export gas after re-compression stage + intercooling between stages	Cooling of Stabilized Condensate	Cooling of the regenerated TEG glycol
Cooling duty average (MW)	6	10	60	15	30
Stream to be cooled (heat source)	Stabilized oil		Export Gas	Condensate	Lean TEG glycol
From T_{inlet} (ºC)	70 - 80	60 - 70	150 - 130	200	200
To T_{outlet} (ºC)	60 - 70	50 - 60	60 - 30	40 - 60	50
Flow rate (kg/h)	1,140,000	180,000	600,000	100,000	5600

4 BASIS AND MODELLING ANALYSIS

4.1 Temperature glide and case definitions

The temperature glide refers to the temperature change obtained in heat exchangers during heat processes. It occurs by working near to supercritical conditions with pure fluids (releasing its sensible heat), using a mixture of two (or more) substances with different thermal characteristics, or using gas phase substances.

Mixtures of two different compounds in the working fluid composition can allow changes in the system performance. The mixture vaporizes and condenses at gliding temperature and selecting the proper composition and pressure level, the temperature glide can be adapted to the system and the temperature levels available in your process. The Figure 4.1 shows an example of a pure fluid working at subcritical conditions (left picture) where the working fluid evaporates at constant pressure. In the right picture, it appears a gliding temperature because of the supercritical conditions. In the supercritical state, the refrigerant is a compressed gas and the temperature is independent of the pressure. Due to this independency, heat rejection occurs at constant pressure with a reduction in temperature.

We can define two cases depending on the heat requirement seen before: Case I related to a medium temperature requirement and Case II related to a high temperature requirement. In Case I, the maximum temperature level of the heat source is 70 °C, rejecting the heat to the working fluid and being cooled until a minimum temperature defined by the coolant (working fluid). It produces a gliding temperature in the heat source from 70 °C to the minimum temperature.
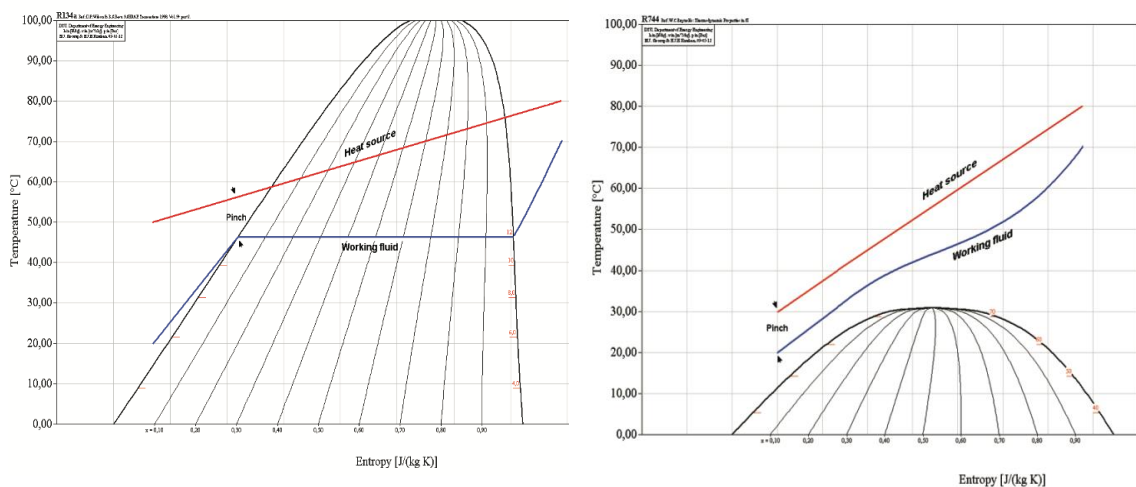
Figure 4.1 Example of transcritical and supercritical cycle in T-s diagram.

A maximum temperature can be approached by the heat exchanger ($\Delta T_{exchanger} \approx 10$ ℃) involving a minimum temperature of 60 ℃. The temperature glide of the heat source is defined from 70 ℃ to 60 ℃.

Considering now the process requirement in Case I, which is defined to 100 ℃, the working fluid has a temperature glide from the compressor outlet temperature to the minimum temperature defined by the process stream or the secondary fluid used in the heat transfer. It involves a temperature glide in the condenser from 150 ℃ (compressor outlet temperature) to 70 ℃ (condenser outlet) defined by the coolant. The compressor outlet temperature is higher than the temperature supplied to the process due to transfer method losses (direct with the working fluid or indirect using a secondary fluid - hot water, hot oil, steam -) and the heat exchanger losses.

In short, the working fluid in the heat pump system should reach a minimum temperature of 150 ℃ to compensate the losses and being able to supply at least 100 ℃ to the process. The system uses a secondary fluid to transfer the heat to the process despite of a direct heat transfer method. This assumption is the most unfavourable situation in the heat transfer involving more heat losses and a total temperature difference of 30 - 50 K depending on the fluid used. See Section 4.4 for assumptions of temperature differences and pressure drops in the equipment.

Referring to Case II, the maximum temperature level of the heat source is 100 ℃, evaporating the working fluid at 90 ℃ and leaving the evaporator at this temperature. The temperature glide of the heat source is defined from 100 ℃ to 90 ℃. The temperature required in the process should be around 200 ℃ and the temperature reached by the working fluid after the compressor should be at least 200 ℃ to cover the demand. As for the Case I, there is a temperature glide in the condenser from the inlet at around 200 ℃ and the outlet at 130 ℃ (defined by the coolant).

A high temperature level of 200 ℃ is more challenging for heat pump systems because of the low temperature sources. The assumption in this case is the most favourable conditions with the direct transfer method (without a secondary fluid) to reduce the temperature difference (ΔT) avoiding large temperature differences and heat transfer losses. Figure 4.2 represents in a T-Q diagram the different cases analysed.
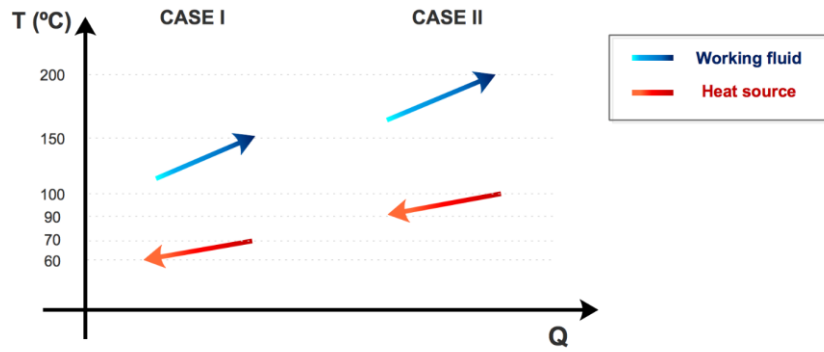
Figure 4.2 Cases representation in T-Q diagram.

4.2 System performance with temperature glide

The coefficient of performance or COP of a heat pump system is a ratio of the useful heating provided to work required. Higher COPs equate to lower operating costs. The maximum reachable COP for a reversible process can be simplified as the following formula:

$$COP = \frac{T_H}{T_H - T_C} \tag{1}$$

The cycle shown in the Figure 4.2 would take place between the cold source at temperature T_C (heat source) and the hot temperature T_H (process requirement) without variable temperature. In the real cycle, the variable temperature exits influencing the simplifications made in the Coefficient of Performance (COP) because of the no isothermal conditions.
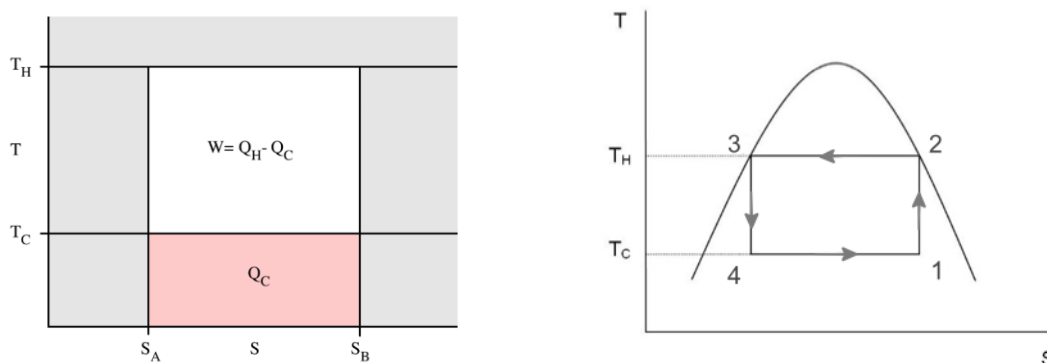
Figure 4.3 Reversed Carnot heat pump cycle representation.

The theoretical COPs obtained for the reversed Carnot cycle in the cases defined in Section 4.1 would be:

- For the Case I, the maximum COP is 4.7 with a maximum temperature of 150 ℃ and minimum temperature of 60 ℃.
- For the Case II, the maximum COP is 4.3 with a maximum temperature of 200 ℃ and minimum temperature of 90 ℃.

By opting for temperature glides, heat exchange can take place with a lower average temperature difference between the heat pump and the hot source/sink. Lorentz Cycle can fit very well with the real effect of the heat pump, illustrated in Figure 5.4. It is also illustrated the Carnot Cycle based on an isothermal heat transfer processes as a comparative measure.
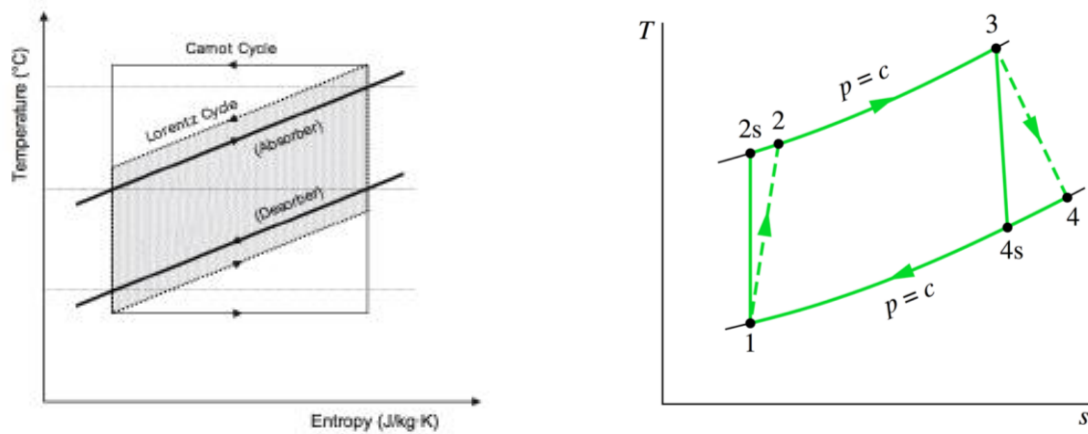
Figure 4.4 Irreversibility Carnot cycle represented in T-s diagram [21].

The temperatures used in equation (1) can be modified to estimate thermodynamic average temperature ($\bar{T}$) of the heat exchanger using the following relation [24]:

$$\bar{T}_{12} = \frac{h_2 - h_1}{s_2 s_1} = \frac{C_p(T_2 T_1)}{C_p \ln\left(\frac{T_2}{T_1}\right)}$$

Cp is constant because the fluid is considered ideal and pressure is constant so a logarithmic term appears changing the equation to:

$$\bar{T}_{12} = \frac{\Delta T_{12}}{\ln(\frac{T_2}{T_1})}$$

ΔT_{12} is the temperature difference between the hot end (T_2) and cold end (T_1) of the heat exchanger. The relation giving the average temperature only applies when the driving forces in the heat exchanger are strong (large temperature difference between the working fluid and the heat source/sink) or small temperature glides.

$$COP_{Glide} = \frac{\bar{T}_H}{\bar{T}_H - \bar{T}_C}$$

T_H and T_C represent the highest heat sink and lowest heat source temperature. Defining $\Delta T_{H,g}$ and $\Delta T_{C,g}$ as the available temperature glides in the hot and cold heat exchanger, it obtains:

$$COP_{Glide} = \frac{\frac{\Delta T_{H,g}}{\ln(\frac{T_H}{T_H - \Delta T_{H,g}})}}{[\frac{\Delta T_{H,g}}{\ln(\frac{T_H}{T_H - \Delta T_{H,g}})} - \frac{\Delta T_{C,g}}{\ln(\frac{T_C + \Delta T_{C,g}}{T_C})}}$$

Considering all of these aspects and compressor efficiencies, pressure losses and temperature variation it can be assumed a general second law efficiency of 75 - 80 %. The second law states that as energy is transferred or transformed and all real engines lose energy (heat) to the environment.

The maximum reachable COP is different now, changing from the theoretical COP. The COP in the Case I can reach a value of 3.8 while for the Case II the maximum COP would be 3.4. These values are based on the machine efficiencies and temperature difference assumptions defined in Section 4.4 allowing to identify the expected performance.

4.3 Modelling basis

As shown in Figure 4.5, the closed vapour compression cycle consists of four components: an expansion valve or expander, a compressor, an evaporator and a condenser or cooler. In vapour compression systems, heat from the refrigerated object is absorbed by evaporating working fluid or refrigerant.

Then in the condenser, the working fluid rejects the heat (condensation) to the heat consumer at a higher temperature level than it was absorbed in the evaporation. The simulation model for the process is developed with UNISIM software. Peng-Robinson equations of state are used in the basis for a better accuracy in the calculations in two-phase area.
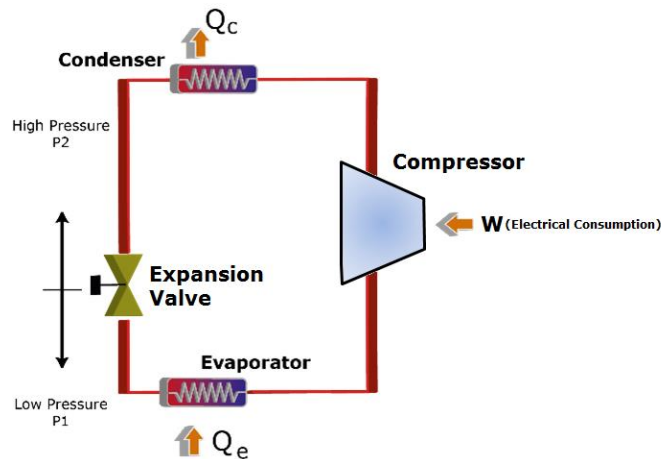
Figure 4.5 Schematic heat pump representation [Araner]

4.3.1 Multi-stage systems

In contrast with single-stage systems, multi-stage systems consist of several (two or three) compression stages with intercooling among them. It improves the efficiency, avoiding high discharge temperatures and big pressure ratios. The intercooler among stages limit the discharge temperatures improving the efficiency in the next compression. The cooler helps to reduce the power consumption of the compressor because it reduces the performance losses due to the heat produced in the compression.

Figure 4.6 shows a two-stage compression heat pump system modelled with UNISIM. In the condenser there is a difference in comparison with the evaporator. In the heat exchanger, it is not defined any coolant stream (see Figure 4.6) to be able to fix the heat duty value of the heat exchanger.
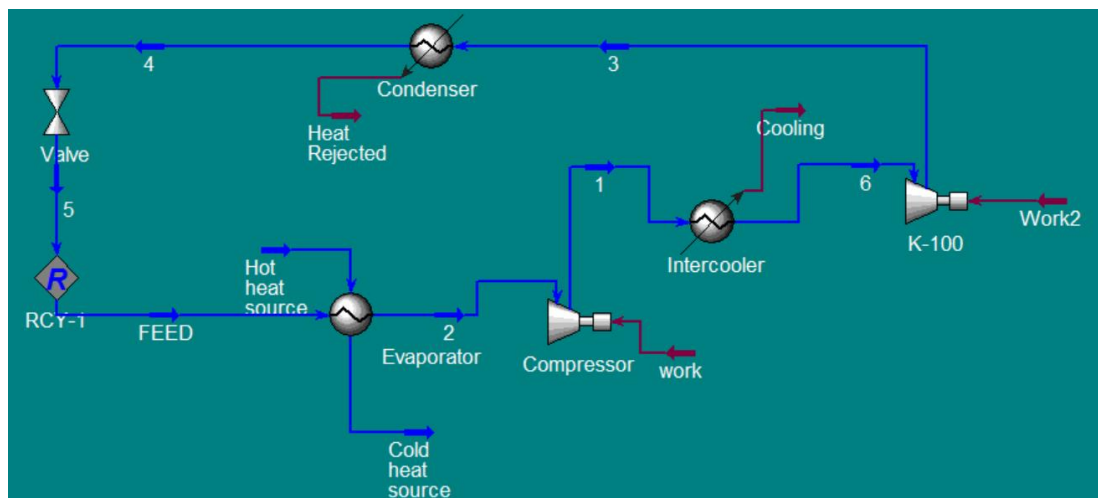
Figure 4.6 Two-stage heat pump system

4.3.2 Cascade system

Other possible system is the cascade system consisting of two subsystems using different refrigerants performing separate cycles with an individual control of each one. The refrigerants may be chosen with convenient properties according to the good suitability with the heat process requirements. The disadvantage of the cascade system is its higher work consumption and it has an irreversible loss due to heat transfer in cascade condenser. The heat transfer loss is dependent on the operating conditions and it can reduce the coefficient of performance significantly.
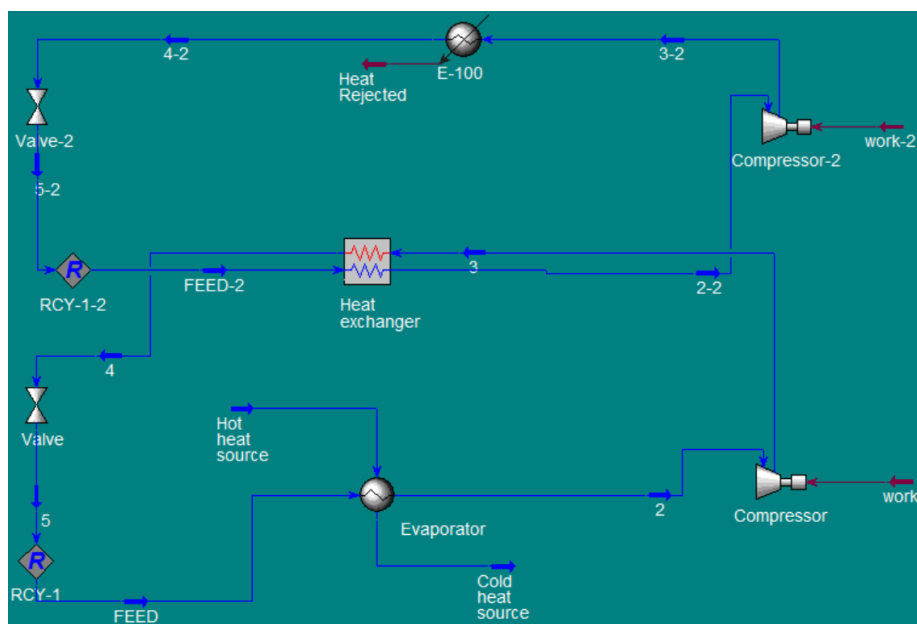
Figure 4.7 Cascade heat pump system

4.3.3 Gas phase system

Gas (mixture) fluids may also be a good option working as reversed Brayton cycles. It is possible to improve the efficiency of the cycle using vapours close to the critical point condition, but not exclusively using CO_2. Gas phase cycles implies a difference in the system. The turbines replace the valve to expand the fluid.

In the evaporator and in the condenser, there are no evaporation of the liquid and condensation of the vapour because the vapour fraction is always equal to one (gas phase). The gas is cooled in condenser, adding a recuperator in the condenser outlet (still hot) to use that residual heat to re-heat the stream before enters in the compressor. Figure 4.8 illustrates the system modelled in Unisim.
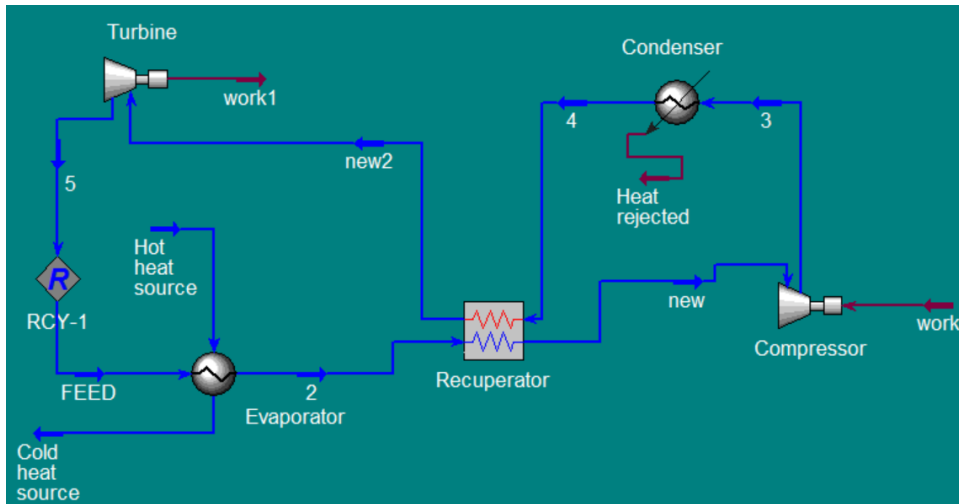
Figure 4.8 Gas phase heat pump system

4.4 Equipment specifications and assumptions

Pressure drops across various equipment and efficiencies of pumps and compressors have a considerable impact on the process flow, so pressure drops have been defined to check realistic COPs. These sub-sections explain the parameters of various equipment and the efficiencies that are used in Unisim simulations.

4.4.1 Compressor specifications

The compressor type selected, according to the compressor selection chart shown in Section 6.2.1, is the centrifugal compressor because of the good performance with the volume flow rates and pressure ratios required. Polytropic efficiency is defined to a reasonable value of 80 % in Unisim software [7].

4.4.2 Heat exchanger specifications

In Unisim to define a heat exchanger, the most common is the shell and tube type. With this type, it can appear temperature cross or minimum temperature approach errors. To avoid problems some exceptions can appear using LNG exchangers. It may be needed heat exchangers such as plate/fin types or printed circuit types, to facilitate closer temperature approaches and temperature crossing (see Section 6.2.2 for the heat exchanger description).

The theoretical size estimation to specify the heat exchanger requires the calculation of some parameters such as minimum heat transfer area, volume or the length (depending on the heat exchanger) or the transfer coefficient per area as shown below:

$$U = \cfrac{1}{\cfrac{r_{ext}}{r_{int}} \cfrac{1}{h_{in}} \cfrac{r_{ext} \, ln \frac{r_{ext}}{r_{in}}}{k} \cfrac{1}{h_{ext}}} \cfrac{W}{m2K}$$

After the transfer coefficient calculation, heat duty exchanged and the temperature difference, it can define the heat transfer area required:

$$Q = UA\Delta T_m \quad \rightarrow \quad Area = \frac{Q}{U\Delta T_m} \, \text{m}^2$$

Unisim software is able to define the product UA for the heat exchanger specification. The parameters calculated with the software are similar, more precise and complete than the theoretical methods because of the calculation method.

4.4.3 Temperature assumptions

In the evaporator, the outlet temperature is defined while the software due to the expansion in the valve (defining the pressure difference or final pressure) defines temperature of the working fluid in the evaporator inlet.

For the Case I, the temperature of the working fluid in the evaporator outlet is 60 ℃ while for the Case II is 90 ℃, according to the temperature level of the heat source. The software, because of the compressor outlet (pressure defined), calculates the inlet temperature of the working fluid in the condenser. The outlet temperature of the working fluid in the condenser is defined to a value equal to 70 ℃ for the Case I and 130 ℃ for the Case II.

4.4.4 Pressure assumptions

Considering the pressure drop in the evaporator and in the condenser are sufficient to account for pressure drop across the equipment and piping. A pressure drop of one bar is fixed in the condenser and in the evaporator to evaluate the fluid behaviour in real conditions.

In the valve, the outlet pressure (or pressure difference) is defined according with the vapour pressure at the outlet temperature defined in the evaporator. In the compressor, it defines the outlet pressure and it coincides with the vapour pressure at the outlet temperature in the condenser counting the pressure drop of one bar. With the expansion valve, no work is applied. Ideal processes with isentropic expansion assume equal working fluid enthalpies at the inlet and outlet of the valve. That is not feasible in real applications and some energy is lost to the environment.

4.4.5 Capacities

The heat duty required in the process has been defined according to the Section 3.1. As mentioned, in the condenser the heat duty can be fixed without defining the coolant streams. For the Case I, the heat duty defined is 150 MW while for the case II, the heat duty is 50 MW.

Fixing the heat duty value on the condenser, the software calculates the working fluid flow required in the cycle. The software also calculates the heat source flow rate required in the evaporator to evaporate the working fluid and calculating the heat duty of the evaporator that varies on the working fluid selected. The heat source is only defined by the inlet and outlet temperature.

In the evaporator, the heat sink streams are not defined. Only the heat duty has been fixed calculating the heat flow for the heat sink. Unisim, defining the heat sink streams (coolant stream) with the pressure and temperature inlet, can calculate the sink flow rate.

5 WORKING FLUID SELECTION: PROPERTIES AND PROCESS REVIEW

The working fluids that could use for the heat pump application are some hydrocarbons available in the plant such as ethane, propane, butane, pentane or mixtures. In addition, other interesting fluids are analysed for this application too, such as ammonia, water or nitrogen, which are natural working fluids.

Table 5.1 shows and compares some refrigerant properties such as critical pressure and temperature, latent heat, and volumetric capacities. Furthermore, the higher latent heat of the fluid, the lower circulation rate required involving smaller pipe diameters.

Properties as the latent heat of evaporation or the volumetric heating capacity can be useful know the heat amount required to evaporate the fluid influencing the gas phase volume. Volumetric heating capacity represents the heat duty rejected in the condenser in (kJ) per working fluid in the compressor inlet (m³). Water has the lowest heating capacity per volume involving low heat rejected in the condenser because of the high volume in comparison with ammonia that involves the largest heating capacity.

The suction volume flows and pressure ratios define the compressor size. As shown in Table 5.1 (References: [11] [12] [13]), the largest suction volume flow rate is for water, which is 23 times higher than ammonia (the smallest). Ammonia represents the smallest volume flow rate reducing the pipe diameters, compressor sizing and the equipment in comparison with water. Other properties such as low refrigerant viscosity, high thermal conductivity, non-flammable or to be harmless in case of leakage are also important to evaluate the suitability.

	T_{crit} (ºC)	P_{crit} (bar)	Global Warming Potential	Latent heat of evaporation at NBP (kJ/kg)	Suction volume flows Case II (m³/h)	Volumetric heating capacity Case II (kJ/m³)
Butane (C_4H_{10})	152	38	Low (3)	386	25,230	7708
Pentane (C_5H_{12})	197	34	Low (5)	366	65,630	3466
Ammonia (NH_3)	132	113	0	1369	7,193	35985
Water (H_2O)	374	220	0	2257	172,300	1315

Table 5.1 Refrigerant properties for vapour compression cycles.

5.1 Pure fluids

5.1.1 Propane

The critical temperature and pressure of propane are 97 ℃ and 43 bar, respectively. The Figure 5.1 represents the transcritical compression cycle in the P-h and T-s diagram. It has been represented considering 150 ℃ as the highest temperature reached after the compression (1→2) in Case I at a pressure of 120 bar. It can be cooled until 60 ℃, depending of the cold stream temperature (coolant) that enters in the condenser, absorbing the heat from the propane. After the expansion until 20 bar, the propane would evaporate at 60 ℃. In short, propane would not be suitable as working fluid in the heat pump system because of the low critical temperature.
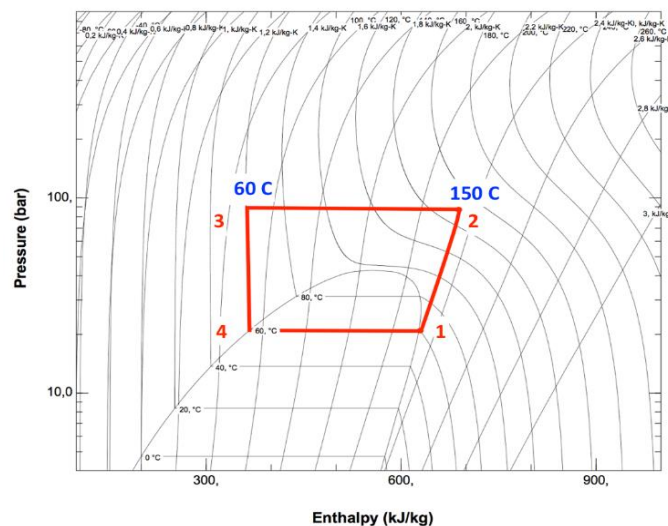
Figure 5.1 Propane P-h diagram

5.1.2 Butane

N-butane (R600) and iso-butane (R600a) have similar performances at low temperature conditions. Butane has relevant thermodynamic properties, such as a high critical temperature ($T_{critical}$ =152 ℃) that allows to achieve high temperatures after the compression stage and with moderate pressures makes the butane as a potential working fluid.

Figure 5.2 illustrates the butane cycles in the P-h diagram for the Case I and Case II. The cycle consists of a gas compression (1→2) following by a cooling (depending of the cold stream) at this pressure. Then, the next step is the expansion (3→4) until low pressure reducing the temperature and the fluid evaporation (4→1) completing the cycle.
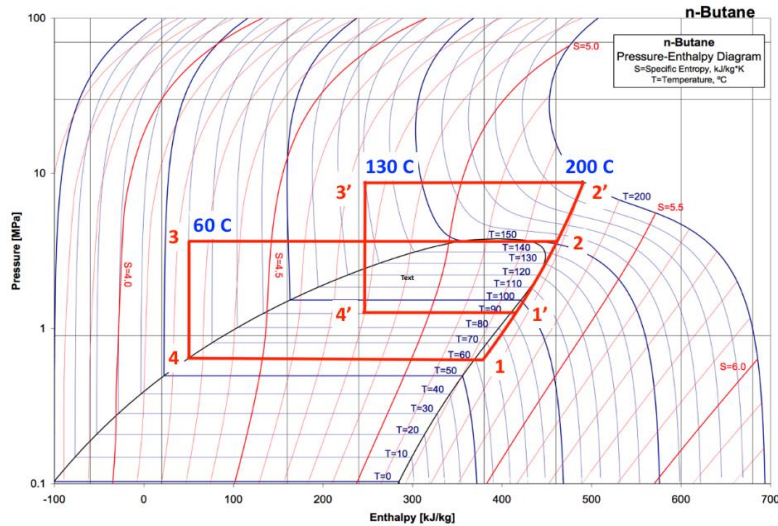
Figure 5.2 Butane P-h diagram

The single-stage used to evaluate butane is a no suitable option because of the excessive compressor pressure ratios reached up to 9.5 for Case I and 8.5 for Case II. It would be more suitable a two-stage system (adding two compressor) without intercooling between them because the discharge temperatures are not so high. Converting the single-stage to a two-stage system, the COP efficiency improves for Case I up to 4 while for Case II it reduces until 3 but pressure ratios are now in the normal limits no higher than 4, which is more reasonable.

Looking at Table 5.2, the two-stage system results are shown. Volumetric suction flows are useful to size the compressors for each case. They have been reduced in comparison with the single-stage. Appendix A shows more detailed data can be analysed such as heat source rates, pressures, duties and the comparison with the single-stage.

Table 5.2 Butane two-stage system results

	Case I	Case II
Q Evaporator (MW)	117	36
Compressor Work (MW)	23 + 14	10 + 6
Volumetric suction flow in the first /second compressor (Act_m³/h)	114,100 / 36,080	25,230 / 6,405
Heat Source mass flow (kg/s)	2608	762
Pressure Ratios	3 / 2	3 / 2.5
COP	**4.0**	**3.0**

5.1.3 Pentane

Pentane (R601) has properties very similar to those of butane and hexane. It has a higher critical temperature (197 ºC) than butane and a moderate critical pressure. It becomes a suitable working fluid to achieve high temperatures. This alkane is sometimes a component fraction in feed but may have to be imported to the plant. As in the previous fluid, the pressure ratios reached by single-stage system are so high for the compressors so the single-stage system changes to a two-stage system.

By analysing the UNISIM results, the COP obtained in Case I with the two-stage system is equal to 3.3. The total compression work required is 46 MW with a volume flow rate 439,600 m^3/h in the first compressor. In the evaporator, the heat duty required is 105 MW with a heat source flow of 2412 kg/s. The maximum pressure ratio reached is 4 which is much lower than in the single-stage.

In Case II, the COP obtained is 3.1 with a total compression work required of 15 MW according to the efficiencies defined in Section 4.3.1. In the evaporator, the heat duty required is 34 MW with a heat source flow of 775 kg/s. Appendix B shows more data about the simulation results.

Table 5.3 Pentane two-stage system results

	Case I	Case II
Q $_{Evaporator}$ (MW)	105	34
Compressor Work (MW)	22 + 24	7 + 8
Volumetric suction flow in the first /second compressor (Act_m^3/h)	439,600 / 114,500	65,630 / 22,930
Heat Source mass flow (kg/s)	2412	775
Pressure Ratios	4 / 4	3 / 4
COP	**3.3**	**3.1**

5.1.4 Ammonia

Ammonia has a high critical temperature lower than pentane but very suitable for the heating application. Important aspects of the ammonia are the good heat transfer properties due to the low viscosity, the high conductivity and high latent heat involving less circulation rate. On the contrary, ammonia is highly corrosive to zinc, copper and their alloys and it is not usually available in the facility unlike hydrocarbon fractions (butane, pentane…) so it appears the necessity to import ammonia to the plant.

Ammonia has a strong odor, being easily recognizable in the surrounding air with a minimum concentration of 45 ppm and from the safety point of view, the Immediately Dangerous to Life or Health Concentration is 300 ppm. Table 5.4 shows the data obtained for the ammonia single-stage cycle. The COP obtained in Case I is equal to 3.9 with a compressor work of 40 MW. The compressor reaches a high discharge temperature with a pressure ratio of 5. For the Case II, the COP is equal to 3.8 with a compressor work of 15 MW and more reasonable pressure ratio of 3.

Table 5.4 Ammonia single-stage system results

	Case I	Case II
Q $_{Evaporator}$ (MW)	112	37
Compressor Work (MW)	39	15
Volumetric suction flow in the compressor (Act_m^3/h)	23,800	5,974
Pressure Ratio	5	3
COP	**3.9**	**3.8**

The problem related to the ammonia single stage is the excessive discharge temperature at the compressor outlet in Case influenced by the high isentropic index. In order to improve the efficiency and to avoid these temperatures, the cycle can be modified adding two compression stages with intercooling between them. Figure 5.3 illustrates the work reduction in the cycle as the lower discharge temperature reached after each compression stage in Case I.
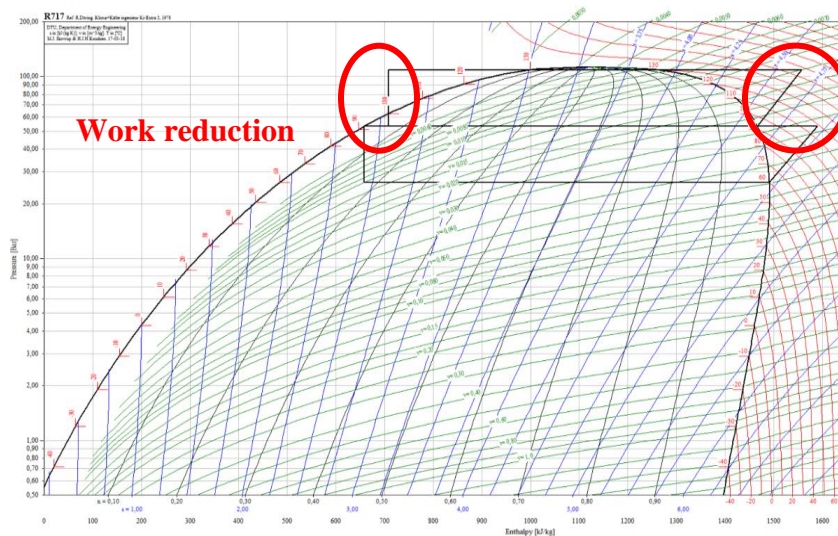
Figure 5.3 Work reduction for the ammonia two-stage system in the p-h diagram

Considering the two-stage cycle and looking the main data at Table 5.5, it can be seen that the efficiency improves in both cases. In Case I, the COP increases up to 4.2 while in Case II, it reaches a value of 3.9. In comparison with the single-stage system, the evaporator heat duty increases together with the heat source flow and the compression work it can be explained by the intercooling between compression stages. Pressure ratios have decreased from 4 to 2 in both cases. Appendix C shows more data.

Table 5.5 Ammonia two-stage system results

	Case I	Case II
$Q_{Evaporator}$ (MW)	132	49
Compressor Work (MW)	21 + 20	8 + 10
Volumetric suction flow in the first /second compressor (Act_m^3/h)	27,920 / 11,130	7,193 / 3,069
Heat Source mass flow (kg/s)	3050	1110
Pressure Ratios	2 / 2	2 / 2
COP	**4.2**	**3.9**

5.1.5 Water

Water has a very high critical pressure and temperature (374 ºC and 220 bar), becoming very suitable for heating applications. Water has a high volumetric flow but also the highest latent heat influences the specific volume in conjunction with the refrigerant flow rate. It is not flammable, toxic and nor miscible with oil. Water is cheap, non-corrosive and it is available in the facility so the water import would not be required. Table 5.6 shows the main data obtained. The COP obtained for the Case I is equal to 3.3 with a compressor work of 45 MW and a large suction flow equal to 1,252,000 m^3/h. For the Case II, the COP is equal to 3.1 with a compression work of 16 MW and a flow rate of 137,200 m^3/h. Appendix D shows more information about the simulation

The cycle works at sub-atmospheric pressures in the valve outlet reaching pressures of 0.2 bar in Case I and 0.7 bar in Case II. It becomes a challenge for the valve, evaporator and compressor. Refrigerants preferably should have a minimum operating pressure above one atm to avoid sub atmospheric pressure inside the circuit (air and moisture influx) and to limit the vapour volume that need to be handle by piping.

Table 5.6 Water single-stage system results

	Case I	Case II
$Q_{Evaporator}$ (MW)	105	34
Compressor Work (MW)	45	16
Volumetric suction flow in the compressor (Act_m^3/h)	1,252,000	137,200
Pressure Ratio	24	20
COP	**3.3**	**3.1**

Some problems appear related to the single stage such as excessive discharge temperatures in the compressor outlet and big pressure ratios. In order to improve the efficiency, avoiding that excessive temperature and reducing the pressure ratios, the cycle can be modified adding three stages with intercooling among the stages. The main data are shown in Table 5.7. The COP obtained with the system is now equal to 4.2 for the Case I while for the Case II it is equal to 3.7. The efficiency has increased in comparison with the single-stage. Besides, the pressure ratios have been significantly reduced. On the contrary, the working fluid flow, compression work and heat source flow have increased.

Table 5.7 Water three-stage system results

	Case I	Case II
$Q_{Evaporator}$ (MW)	131	43
Compressor Work 1 (MW)	17	6.5
Actual volume suction flow in the first compressor (m^3/h)	1,566,000	172,300
Pressure Ratios (Compressor 1/2/3)	3.5 / 3 / 2	3.5 / 2 / 2.5
COP	**4.2**	**3.7**

5.1.6 Pure fluids results

The results show that the best performance results are for ammonia with the highest COP and the minimum volume flow in the compressor inlet (the largest volume section of the system because of the gas phase). Besides, ammonia requires the largest heat source flow implying the largest heat absorption by the fluid in comparison with the others.

It is important that the fluid absorbs as much as possible heat from the heat source and afterwards reject it in the condenser. The high-pressure conditions in the ammonia cycle (up to 200 bar in Case II) would involve large compressor dimensions.

As seen in Table 5.8, the second highest COP is for water but unlike with ammonia, water has the largest volume flows involving big pipe diameters and equipment. Water volume flows in the first compressor are 56 times higher than ammonia in Case I and 24 times higher in Case II. Besides, water also requires high heat source demand in comparison with butane or pentane. A challenging aspect mentioned before is the sub-atmospheric pressure conditions of the cycle.

Referring now to the hydrocarbons, butane has a better performance than pentane in both cases. Butane has lower volume flows (3 - 4 times less than pentane) and it requires higher heat source demands to evaporate the working fluid. Pentane has the lowest COP in both cases. The illustration of Figure 5.4 shows the efficiency in the single compression stage. For water, ammonia and butane, the data obtained are no realistic because of the single stage, being water not so good at high temperatures and ammonia reaching theoretical COP values. As seen above, the system required for these fluids is two stages because of the pressure ratios as shown the COP values in Table 5.8. It should be remarked the efficiency values decrease for Case II (high temperature case) in all the fluids because it is more challenging due to the heat pump limitations.

Considering compression works, they are very similar among the different fluids. The compression work for water is slightly higher than in the others because of the large volume suction flow. The main parameter that influences the compressor size is the volume flow together with pressure conditions, as seen in the results; water system would have the biggest compressor sizes.

Table 5.8 Pure fluids results summary

Pure Fluids (two-stages)	Volume suction flow in the first compressor (Act_m³/h)		Heat source flow rate (kg/s)		COP	
	Case I	Case II	Case I	Case II	Case I	Case II
Butane	114,100	25,230	2608	762	3.5	3.2
Pentane	439,600	65,630	2412	775	3.3	3.1
Ammonia	27,920	7,193	3050	1110	4.2	3.9
Water	1,566,000	172,300	3023	978	4.2	3.7
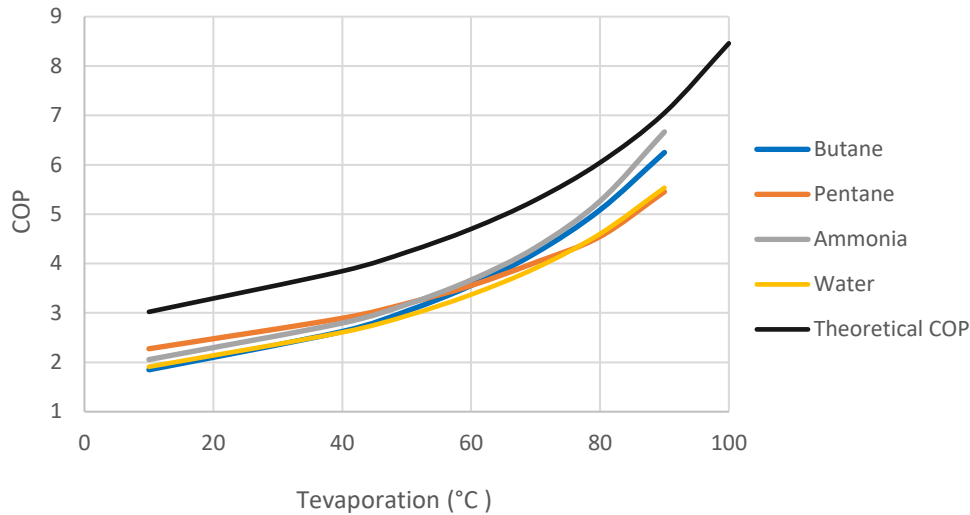
Figure 5.4 COP results for pure fluids in the single-stage system (Case I)

5.2 Cascade systems

5.2.1 Ammonia - Butane cascade system

In this case, the system consists of two cycles: the upper cycle works with ammonia (higher temperatures) and the lower cycle with butane (lower temperatures). The reason for that configuration is the better performance of the ammonia at higher temperatures than butane, seen in the results of Section 5.1. With a low heat source, it can reach higher temperatures in the evaporation of the second fluid, improving the total efficiency of the cycle; on the contrary, it appears some losses because of the intermediate heat exchanger between cycles (lower and upper).

Table 5.9 shows the data obtained with Unisim. The improvement is more marked for the Case I reaching a COP equal to 4.0 than for Case II that is lower. Compared to the individual performance of butane or ammonia, the total performance improves combining both in the cascade system.

The heat duty in the evaporator is a bit higher in both cases than using the pure fluids. It involves higher heat source flows requiring larger heat sources in comparison with using the fluids separately. The total volume suction flow of working fluid (pentane + butane) is much lower for the Case II in the cascade system than using ammonia or butane separately. Appendix E shows more details about the simulation.

Table 5.9 Results for cascade system with ammonia and butane

	Case I	Case II
Q Evaporator (MW)	113	35
Pressure Ratio 1	3	3
Pressure Ratio 2	2	2
Actual volume suction flow in the butane compressor (m^3/h)	133,000	44,560
Actual volume suction flow in the ammonia compressor (m^3/h)	36,290	10,880
COP	**4.0**	**3.4**

5.3 Mixture fluids

The effort led to the performance test of the mixture in the heat pump system evaluating the mixtures because the correct mixture selection is an optimization problem. The optimization defines the optimal compound composition in the mixture. In this case, the aim is not the optimization. As seen in Section 4.1, mixtures can be easily adapted to the system and the temperature level available due to the gliding temperature. Two mixtures have been evaluated with the system: propane/butane and butane/pentane. Appendix F and Appendix G include more data and temperature glide curves in the heat exchanger.

5.3.1 Propane - Butane mixture

Propane can increase the latent heat of the mixture while butane keeps high the critical temperature. The working fluid composition is 20 % propane and 80 % butane. The cycle has a COP equal to 3.6 for the Case I. In comparison with pure butane (see Section 5.1.5), it involves a better efficiency due to the compressor work reduction (reducing the pressure ratio until 5 in Case I). The mixture reduces the volume flow up to 16% lower increase in comparison with butane.

In Case II, the COP obtained for the mixture is 2.9, which is lower than using pure butane. The mixture performance is worse at higher temperature scenarios and the propane critical temperature reduces the achievable temperatures can explain it. Compression work and evaporation heat duty are similar than in the pure butane.

5.3.2 Pentane - Butane mixtures

A mixture of pentane and butane will provide a higher molecular weight to the working fluid, which is beneficial to pressure raised in the compressors. Varying the pentane fraction in the working fluid compositions it can be evaluated the performance.

Defining a working fluid composition of 20% pentane and 80% butane, the COP improves up to 3.7 compared to use pure butane (3.5) or pentane (3.3). In Case II, the COP obtained is 3.3 compared to using pure butane or pentane, which are 3.2 or 3.1 respectively (see Sections 5.1.2 and 5.1.3). Considering the volume flow rates, the mixture has lower volume rates than using the fluids separately involving 72 % lower than pure pentane in Case I and 55 % lower in Case II. Besides, heat source flows are higher for the mixture than for the pure fluids, requiring larger heat sources involving larger heat absorptions.

Increasing the pentane fraction up to 50 % of the composition, the COP obtained for the Case I is 3.5, lower than in Case II which is 3.1 and high-pressure ratios are obtained for the single compression stage. Modifying the cycle to a two-stage compression system, the COP improves a little bit achieving a COP value of 3.6 in Case I and a value of 3.3 in Case II in comparison with the single stage.

As mentioned, the efficiency tends to reduce as well as the pentane fraction increases in the composition. It can be explained because of the worse pentane performance than butane, as seen in the pure fluids results (Section 5.1.6).

5.3.3 Mixture fluids results

Extracting some important data in the Table 5.10 and analysing the coefficient of performance it clarifies that the better performance is for the mixture with pentane and butane. Propane - butane mixture has the lowest efficiency in both cases and it can justify for the low critical temperature of the propane. The COP is much lower when the temperature level is higher as it occurs in Case II. On the contrary, propane reduces the volume flow up to 23 % lower in the compressor inlet (Case I) in comparison with pentane mixture).

Considering pentane, if the pentane fraction increases, the efficiency reduces and pressure ratio increases requiring two-stage compression system to improve the efficiency. Figure 5.5 shows the COP variation in a single-stage compression system of different mixtures related to the evaporation temperature and comparing with the theoretical coefficient.

As illustrated in Figure 5.5, pentane-butane mixture has a better performance at low and high temperatures while the propane-butane mixture is suitable for medium temperatures. The mixture with the highest pentane concentration (50 %) involves larger compression works that is logical because of the lowest efficiency obtained. On the contrary, propane has the lowest compression work, as illustrated in Figure 5.8, because of the lower volume flow in comparison with pentane mixture.

Table 5.10 Mixture fluid results

Mixture Fluids	Volumetric suction flow in the compressor (Act_m^3/h)		Heat source flow rate (kg/s)		COP	
	Case I	Case II	Case I	Case II	Case I	Case II
20/80 Propane-Butane	95,970	21,840	2551	750	3.6	2.9
20/80 Pentane-Butane	124,800	29,570	2534	790	3.7	3.3
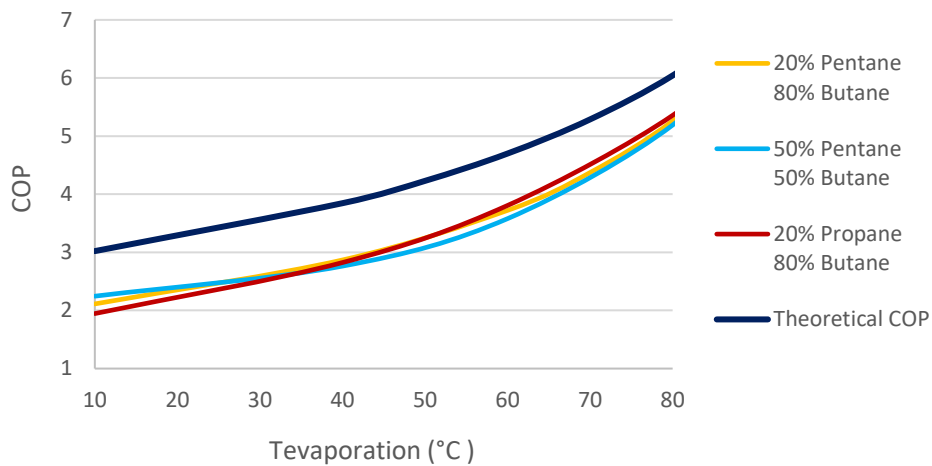
Figure 5.5 Single stage results for mixture fluids in the Case I
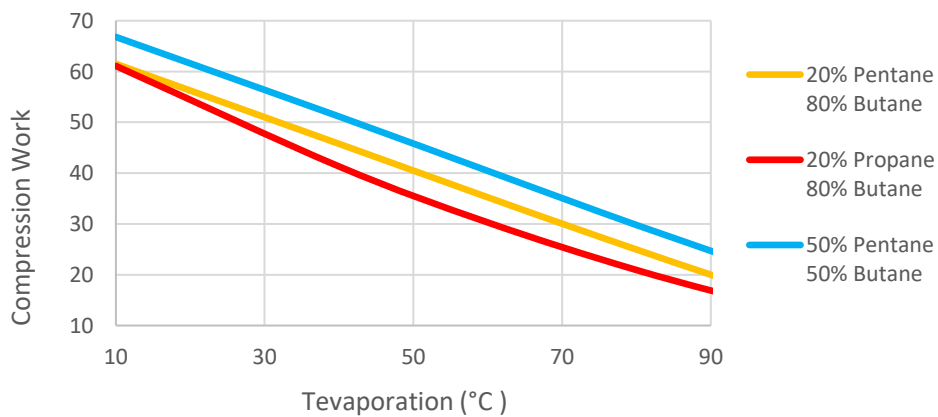
Figure 5.6 Compression work results for mixture fluids in the Case I

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5.4 Gas phase fluids

5.4.1 Nitrogen

The highest COP reached in the single stage compression cycle is 1.8 in Case I and 1.4 for the Case II. Trying to improve these COPs and considering the temperature in the condenser outlet is still high, a recuperator can be added (as illustrated in Section 4.3.3). As Table 5.11 shows, the COP obtained with the cycle + recuperator is a bit higher but also low. For the Case I, the COP is now equal to 1.9 while for the Case II is 1.7. The working volume rate has increased from the cycle without recuperator while the heat source flow is very similar. More details as the flowsheet diagram or temperature glide in the heat exchanger and recuperator are shown in Appendix H.

Table 5.11 Results of nitrogen gas system with recuperator

	Case I + Recuperator	Case II + Recuperator
$Q_{condenser}$ (MW)	150	50
$Q_{Evaporator}$ (MW)	71	21
Compressor Work (MW)	147	54
Actual volume suction flow in the compressor (m³/h)	99,330	22,020
Compression Ratio	2	2
COP	1.9	1.7

6 HEAT PUMP INTEGRATION AND SYSTEM SOLUTIONS

6.1 Heat pump integration

Aspects concerning the practical feasibility of the heat pumps such as space and size limitations. Removing the actual large gas turbines would release some area suitable for the heat pump. Perhaps, one logical place would be to place the heat pump where the gas turbines are currently located (barge) because there is already an infrastructure for the heating system.

Figure 6.1 shows one possible heat pump integration in the process with a secondary fluid (hot oil or steam). The heat pump would use the residual heat "hot source" from the processes mentioned before and it would upgrade the temperature level delivering to the process requirements (blue dashed lines) by a secondary fluid.

In this example, three temperature levels define the demand (high, medium and low temperature), two of them being supplied by the heat pumps. The blue dashed lines represent the heat duty delivered to the process stream from the heat pump. The dotted blue lines between heat pumps represent one heat pump with lower temperature level can support the other one with a higher temperature level and heat to reach higher temperatures. That heat pump support allows to reduce the dependence of the heat pump on high temperature sources. An electric heater or a fuel furnace can cover all the demands would support the high-temperature demand not covered by the heat pump.
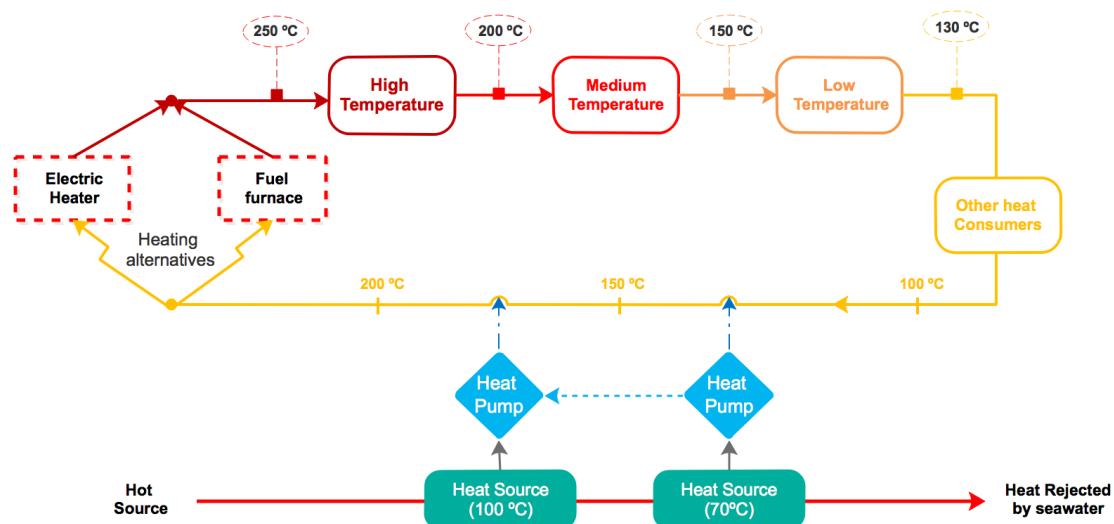
Figure 6.1 Possible heat pump integration layout with a secondary fluid (indirect).

Considering all the possible solutions for heat pump integration with the process, Figure 6.2 shows the integration of the heat pump system in the gas recompression process. In the scheme, the gas stream after the recompression is used as heat source, being cooled and evaporating the working fluid of the heat pump system in the evaporator. The working fluid rejects the heat absorbed from the gas stream in the condenser. This heat is rejected to other process demand in the plant.
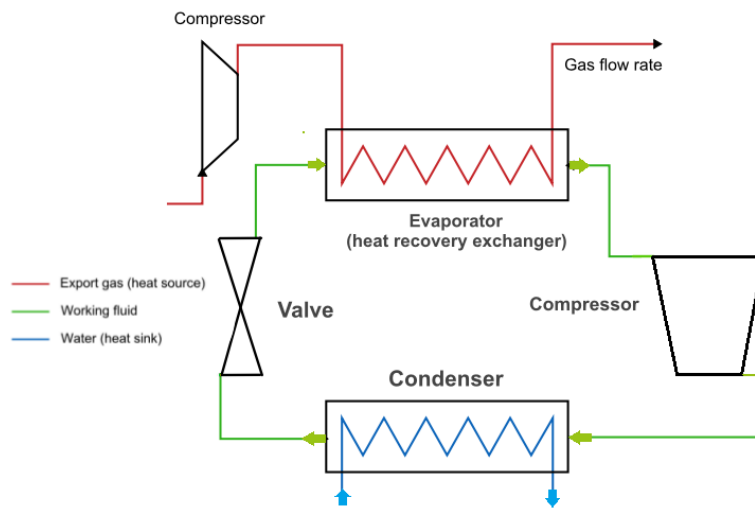
Figure 6.2 Possible heat pump integration layout for gas recompression

The last heat pump integration is part of the TEG gas dehydration process and it is illustrated in Figure 6.3. The working fluid is evaporated with the residual heat of the outlet stream from the regenerator. The working fluid is then superheated with the water vapour that leaves the regenerator and after the fluid compression, the heat is rejected. The working fluid rejects the heat absorbed to the process stream in the condenser-reboiler.
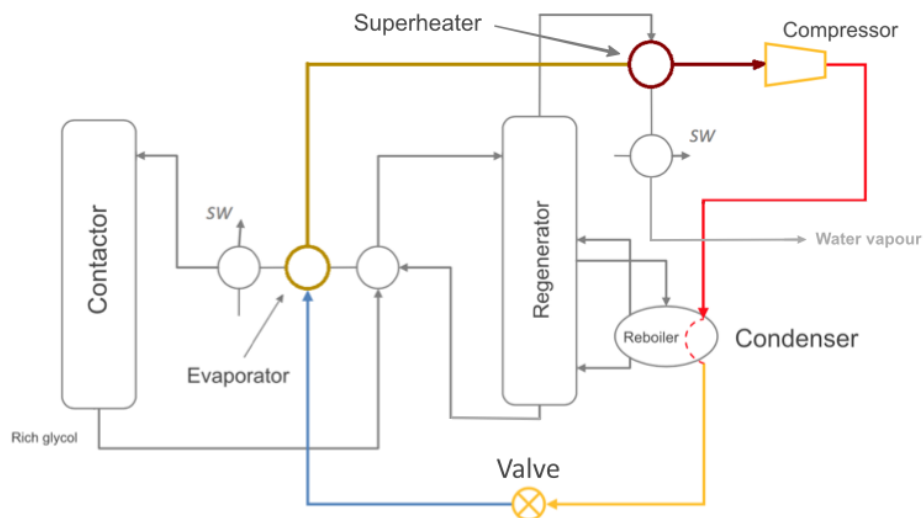
Figure 6.3 Possible heat pump integration layout for TEG gas dehydration

6.2 Main equipment description

This section describes the equipment required in the heat pump systems according to the working fluids and different cases explained in the previous sections. The main equipment consists of compressors, heat exchangers, expanders and piping.

6.2.1 Compressor description

Compressors increase the pressure (and accordingly the temperature) of the working fluid vapour exiting the evaporation stage in a heat pump. Industrial application requires relatively large compressor systems, with high initial costs, due to increased heat load requirements accompanied by larger working fluid flow rates.

General recommendation for an application of the compressor requires centrifugal, screw compressors or axial for large capacities. If high compression ratios are required, centrifugal compressors may be the best option while if very large volumes are required (300,000 – 500,000 m^3/h), axial compressors may be a good option [22].

Depending on the available energy source, the compressor is driven using steam turbines, gas turbines or electric motors. Considering electric driven facilities, the compressor is driven by an electric motor.

One of the main aspects to define a compressor is the capacity together with the suction/discharge pressures and temperatures. The capacity is the amount of gas moved per unit of time as volumetric capacity in m^3/h or mass capacity kg/h.

6.2.1.1 Compressor types

The compressor comprises two main types: **dynamic** and **positive displacement compressor**. Figure 6.4 shows the suitability of the compressors for different capacities in m^3/h and isentropic head data. The axial, centrifugal and screw compressors are more adaptable for large capacities than other compressors.

Axial compressors work in low-moderate pressures no higher than 30 – 40 bar while centrifugal can work at higher pressures. In short, axial, centrifugal and screw compressors may be suitable for the current heat pump application. Table 6.4 resumes the limitations for the different compressors representing the minimum capacity required for both cases with dashed lines to identify where the cases are.

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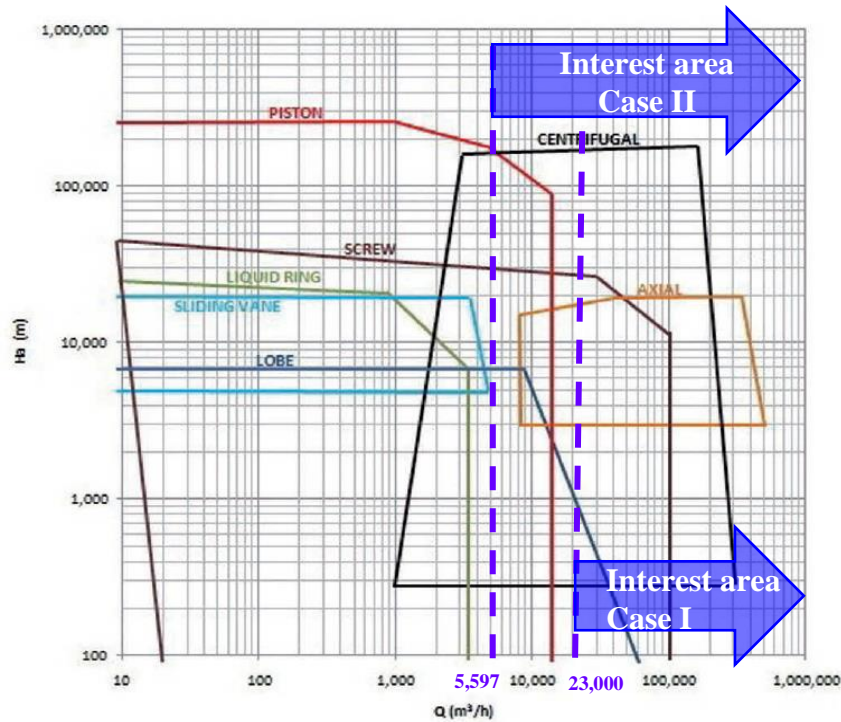
Figure 6.4 Compressor type selection chart [22]

6.2.1.1.1 Dynamic displacement compressors

Dynamic compressors speed up the gas fluid to a high velocity and then decelerate the gas flow with a diffusor, reducing the velocity and causing a pressure increase. The dynamic displacement group includes axial and centrifugal compressors, which are illustrated in the Figure 6.5.

Axial compressors are designed for high volume flow, relatively low-moderate pressure applications (<30 bar) and have high efficiency. An external fabricated horizontally split casing holds an inner stator blade carrier. Centrifugal compressors are turbomachines with a continuously flowing refrigerant stream. Centrifugal compressors are available in sizes up to volume flow of 500,000 m^3/h (Limitation given by General Electric). Depending on the type of construction, the discharge pressure can reach hundreds of bars.

6.2.1.1.2 Positive displacement compressors

Positive displacement compressors increase working fluid pressure by confinement and reduction of its gas volume. This group comprises reciprocating piston, rotary screw and screw compressors (see Figure 6.6). Rotary screw compressors draw gas into a void created as two helical rotors mesh. Once the rotors pass by the inlet port, the cavity decreases in size for the remainder of the rotation, compressing the gas.

Screw compressors can handle more than 30 bar using cast steel casings but it is not commonly used due to capital cost and availability. Current systems can limit the output gas temperature to 250 °C [16]. Screw compressors are also capable of compressing practically all gases. Most positive displacement compressor types are available in oil-free or oil-less designs where no lubricant is injected into the air. Oil-free and oil-less machines typically have 10 - 20 % lower efficiency and require more maintenance.
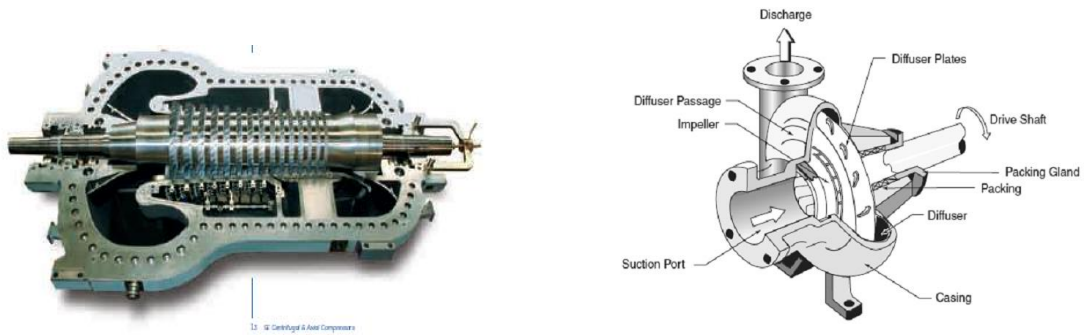
Figure 6.5 Axial compressor (left) and centrifugal compressor (right). [18]

Figure 6.6. Rotary screw compressor element. [Atlas Copco 16]

Table 6.1 Compressor limitations summary

	Type	Capacity (m³/h)	Discharge temperature max. (°C)	Discharge pressure max. (bar)
Dynamic displacement compressor	Axial	500,000 – 1,300,000 [1]	Up to 350 [1]	25 [3]
	Centrifugal	300,000 - 500,000 [3]	200 – 250 [2]	600 [3]
Positive displacement compressor	Screw	100,000 [4]	200 – 225 [2]	30 [2]

[1] Limitation given by Siemens   [3] Limitation given by General Electric
[2] Limitation given by GBH Enterprises   [4] Limitation given by World Pumps

6.2.1.2 Compressor sizing, weights and other results

The main aspects that primarily affect to the dimensions of the compressor are maximum working pressure, capacity and power consumption. In this section, it resumes the results obtained with Unisim for the different working fluid systems and cases (Medium Temperature and High Temperature). To size a compressor capacity, it should take into account possible changes in the planned fluid consumption data, and later incremental expansion of compressed fluid needs.

Figure 6.7 shows the results for the pure and mixture fluids in Case I. For the pure fluids, the lowest volumetric capacity is for the ammonia cycle. It comprises an actual volume flow of 27920 m^3/h in the two-compression stage system. On the contrary, water involves the largest volume flow in the compressor reaching an actual value of 1,566,000 m^3/h (Case I) in the multi-stage compression system.
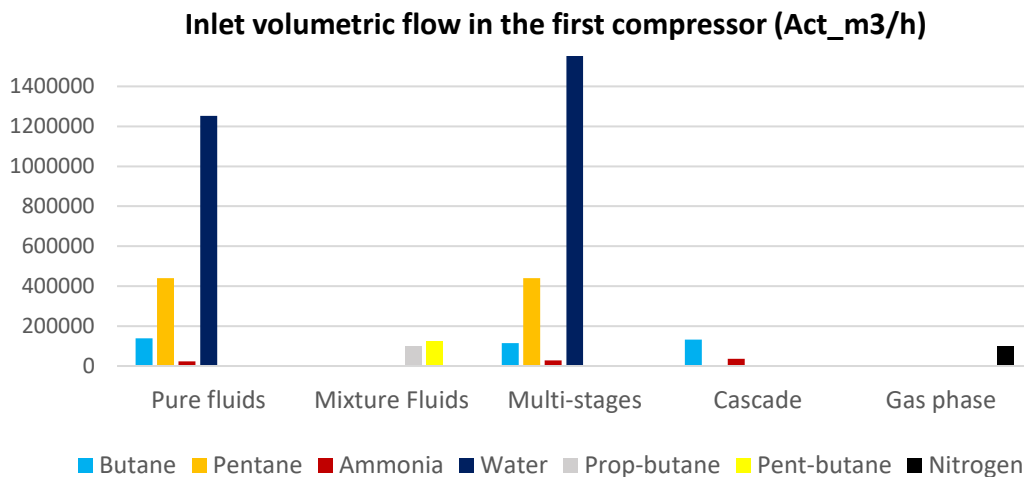
Figure 6.7 Volumetric flow results in the first compressor inlet (Case I)

Figure 6.8 shows the same data as the previous figure but related to Case II. For the pure fluids, ammonia has a same behaviour involving the lowest volume flow in the inlet of the first compressor. It involves a value of 7139 m^3/h. Water also involves the largest volume flow as in Case II reaching an actual value of 172,300 m^3/h in the multi-stage system.

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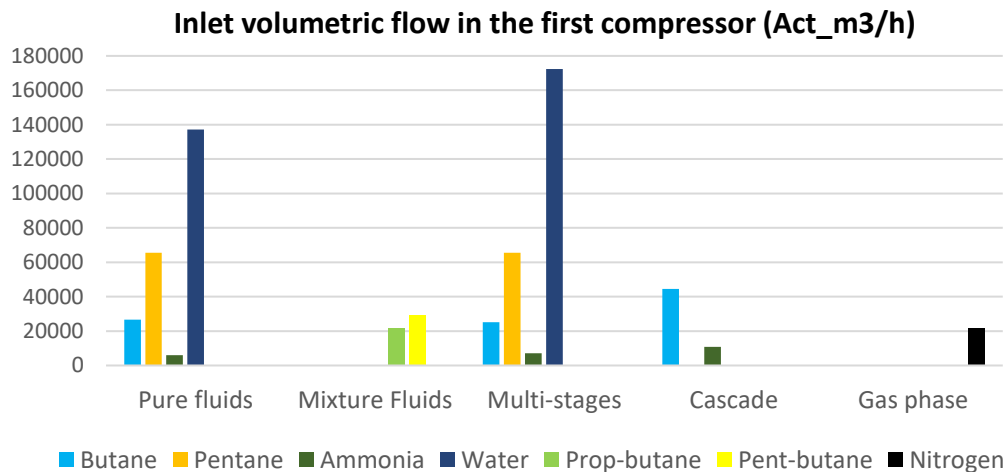
Figure 6.8 Volumetric flow results in the first compressor inlet (Case II)

As shown above, the water cycle works with the largest volume suction flows (in the first compressor) and it would involve the largest compressors sizes and pipe sections. Adding stages reduce the pressure ratios and work per compressor improving the efficiency and the selection and costs of the compressor.

Table 6.2 illustrates different data as the compressor duty, volume suction flows and pressure conditions. In comparison with water, ammonia has much higher discharge pressures but on the contrary, ammonia involves much lower volume suction flows. Ammonia represents the smallest volume flow in the multi-stage cycle too with a better performance than in the single-stage, reducing the pressure ratio and reducing the work per compressor.

Table 6.2 Compressor duties, capacities and pressure ratio results

Multi-stage systems	Compressor duty (MW)		Actual volume suction flow (m³/h)		Pressure ratios		Suction / Discharge pressure (bar)	
	Case I	Case II	Case I	Case II	Case I	Case II	Case I	Case II
Water (Three-stage system)								
Compressor 1	17	7	1,566,000	172,300	3.5	3.5	0.2 / 0.7	0.7 / 2.5
Compressor 2	15	5	480,800	52,730	3	2	0.7 / 2	2.5 / 6
Compressor 3	13	5	180,700	23,280	2	2.5	2 / 4.8	6 / 14
Ammonia (two-stage system)								
Compressor 1	21	8	27,920	7,139	2	2	23 / 53	50 / 91
Compressor 2	20	18	11,130	3,046	2	2	53 / 121	91 / 200

Referring to the cascade systems, the combination of two fluids implies a better performance reducing the compression work (see Table 6.3). Discharge pressure is lower in comparison with pure ammonia cycle. The volume flows in the combination of the two cycles are smaller than using the fluids individually, reducing the dimensions of the compressor and the weight.

Table 6.3 Results of cascade system

Cascade System	Compressor duty (MW)		Volumetric suction flow (Act_m³/h)		Pressure Ratios		Suction / Discharge pressure (bar)	
	Case I	Case II	Case I	Case II	Case I	Case II	Case I	Case II
Butane (lower)	22	7	133,000	44,560	3	3	4 / 13	4 / 25
Ammonia (upper)	15	7.5	36,290	10,880	2	2	20 / 35	30 / 50

Finally, Table 6.4 shows the compressor duties, volume suction flow and pressure conditions for the nitrogen as gas phase system:

Table 6.4 Results of gas phase system

Gas phase	Compressor duty (MW)		Volumetric suction flow (Act_m³/h)		Pressure Ratios		Suction / Discharge pressure (bar)	
	Case I	Case II	Case I	Case II	Case I	Case II	Case I	Case II
Nitrogen	147	54	99,330	22,020	2	2	59 / 111	119 / 202

Looking at the Figure 6.7, the capacities obtained for the Case I are only in the range of axial and centrifugal compressors, while for the Case II, it can be noted that the flow capacities are in the suitable range of piston, screw, centrifugal and axial compressors (taking into account the pressure limits). Water is the most challenging capacity because it is above the limits represented in the Figure 6.4.

Depending on the manufacturer these limitations can vary, Siemens supply axial compressors with higher limits (see Table 6.1). The main manufacturers chosen are General Electric Atlas Copco or Siemens because of the limitations in the volumetric capacity and discharge pressure. To avoid higher volume flows than the manufacturer specifications, it can be used parallel compressors dividing the flow in two streams but increasing the cost of a second machine.

6.2.2 Heat exchanger description

The optimum design and placement of the heat exchangers depends on some parameters such as the volume flow through the heat exchanger, saturation temperature of the working fluid and the heat transfer efficiency. Temperature cross problems or closer temperature approaches can appear, requiring heat exchangers that facilitate this. Shell and tube exchangers are sometimes problematic so it appears the necessity to apply other type such as plate-frame or plate-fin type and printed circuit type.

6.2.2.1 Shell-and-tube exchangers

Shell and tube exchangers are the most popular but also the more problematic type when the working fluid condenses/evaporates within the tubes because it affects the transfer abilities. This type of exchanger consists of a cylindrical shell with two tube plates welded onto its ends, shown in the Figure 6.9.

The gas-liquid mixture formed complicates the control of the heat transfer effects and optimize performance. Superheating the evaporating working fluid (to ensure dry compression) is also problematic, requiring some margin in operation. Pressure losses through the thin and long tubes may also be significant. Depending on the manufacturer, the design can comprise pressures up to 320 bar and temperatures up to 600 °C [15].
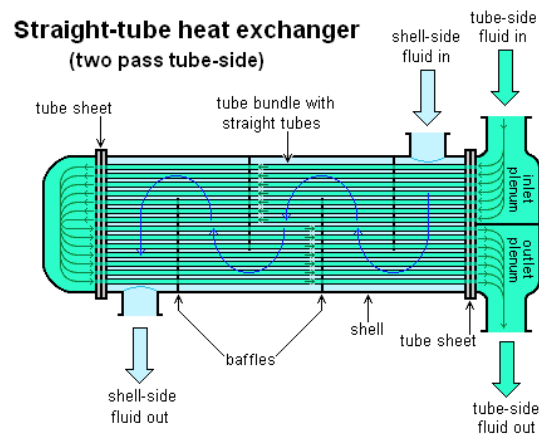
Figure 6.9 Shell and tube heat exchanger [15]

6.2.2.2 Plate-and-frame exchangers

Plate-and-frame heat exchanger can be a compact solution for large duties. It has the benefits as a compact and flexible design (easily adapted to the heat pump designed performance suitable for two-phase streams), excellent heat-transfer abilities and very limited maintenance.

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The media in the heat transfer is lead into the plate pack through portholes at the corners and it distributes into the passages between the plates by the arrangement of sealing welds, shown in the Figure 6.10.

The plate heat exchangers are suitable for the majority of relatively uncomplicated heat transfer jobs using different fluids as media. The maximum design temperature is 500 °C and a pressure up to 50 bar (Limitations given for Alfa Laval) excluding use in some of cases.
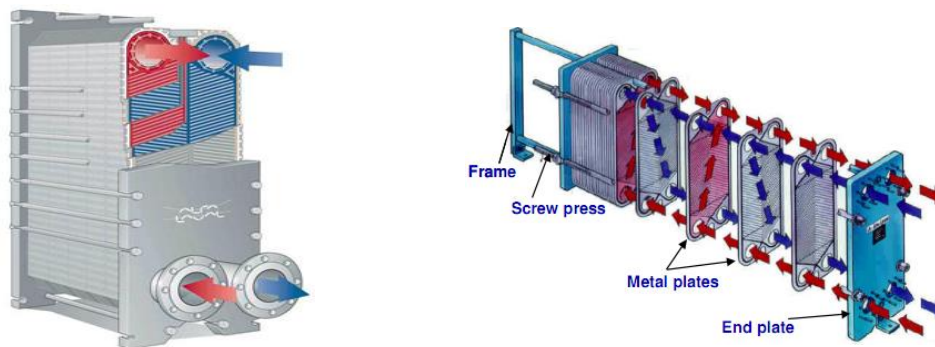
Figure 6.10 Plate-and-frame heat exchanger scheme [17]

6.2.2.3 Plate-and-fin exchangers

Corrugated metal fins are placed between flat plates and the structure is joined together by brazing. The concept is shown in the Figure 6.11. The fins have the dual purpose of holding the plates together, thus containing pressure, and of forming a secondary (fin) surface for heat transfer. At the plates edges are bars, which contain each fluid within the space between adjacent plates.

The heights of corrugations and bars may vary between plates. For a liquid stream, it normally uses a low height corrugation, matching high heat transfer coefficient with lesser surface area while for a low-pressure stream we can use a high corrugation height, matching low coefficient with higher surface area but also giving larger through area to achieve lower pressure drop.

Stainless steel units are currently limited to 50 bar design pressure and temperatures up to 750°C while the aluminium are limited to 100 bar and 70 °C. Above this temperature, a change of header material will allow operation to 120°C with reduced design pressures (Limitations given by IMI Marston Ltd.).
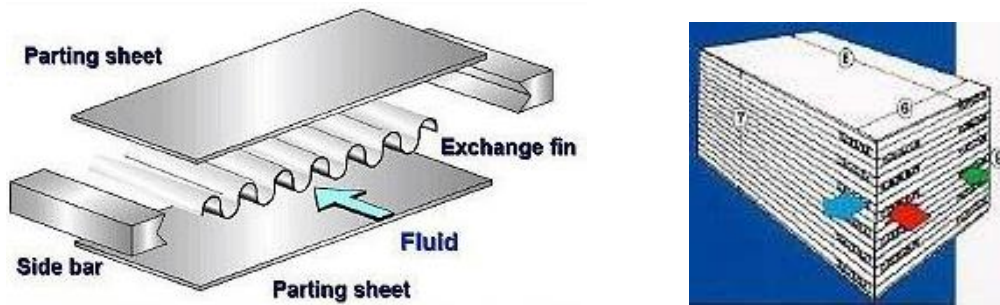
Figure 6.11 Plate-and-fin heat exchanger scheme (Shenzhen SAE).

6.2.2.4 Welded plate-and-shell exchangers (Tubular)

When it requires demanding duties, such as high pressure and high temperature applications, welded plate-and-shell heat exchangers can avoid these kind of problems. The design allows using the exchanger with liquids, gases and two-phase mixtures and it characterizes a small footprint and lightweight minimizing installation, operating and maintenance costs. Depending on the manufacturer, the maximum pressure reached can be 100 bar and at temperatures up to 450 °C (Alfa Lava manufacturer), being very suitable for the heat pump application.
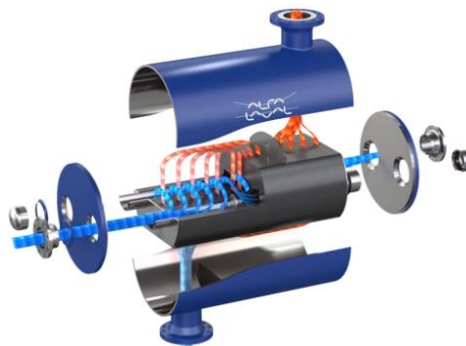
Figure 6.12 Welded plate-and-shell heat exchanger scheme [18]

6.2.2.5 Heat exchanger sizing, weights and other results

Shell ant tube are comparatively cheaper than compact exchangers. However, the process parameters can result in extreme dimensions because of the low LMTD so printed circuit would have to be preferred choice.

As seen, plate-and-fin or welded plate-and-shell heat exchangers can be a good option because of the two-phase mixtures and the closed temperature approaches being excluded for ammonia use due to the pressure limit up to 100 bar. Plate and frame will be also excluded also in some cases because of a lower pressure limit.

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To design the exchanger, volume capacities, pressure and temperature limits affect specifications, involving different sizes, materials and weights. Two heat exchangers are required in the cycle: the evaporator and the condenser. In the cascade systems, an intermediate heat exchanger is required to exchange the heat between cycles.

In the evaporator, the maximum temperature reached is 100°C according to the Case II, while for the condenser the maximum inlet temperature is 200°C in the Case II. It can define two design temperatures closer to 100 °C and 200 °C to avoid oversizing. A good parameter to analyse the heat transfer performance of the exchanger is the UA factor, which is the product of the overall heat transfer area (U) and the heat transfer area (A), see equipment specifications in Section 4.4.

The larger UA factor, the larger is the size of the equipment. Figure 6.13 illustrates the UA results obtained in the evaporator for the Case I while Figure 6.14 represents the results obtained in the Case II. In the analysed cases, water has the largest UA value because of the huge water volume flow in that case increasing the minimum heat transfer area required by the heat exchanger.

As mentioned, volumetric flow through the heat exchanger is also important to size heat exchangers. Figure 6.15 and Figure 6.16 illustrates the different volume flow rates in the evaporator inlet for the cases analysed. Nitrogen has the largest flow in the evaporator inlet because of the gas phase. The second largest volume is for the mixture of propane-butane and pentane-butane. As seen, mixtures has low values of UA implying a worse heat transfer and involving higher volume flows.
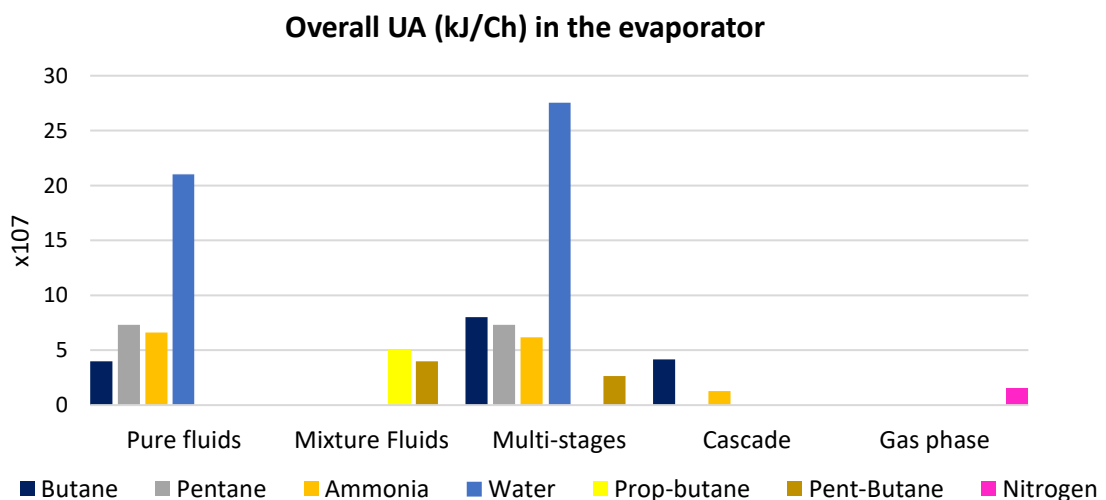
Figure 6.13 Overall UA results obtained in Case I
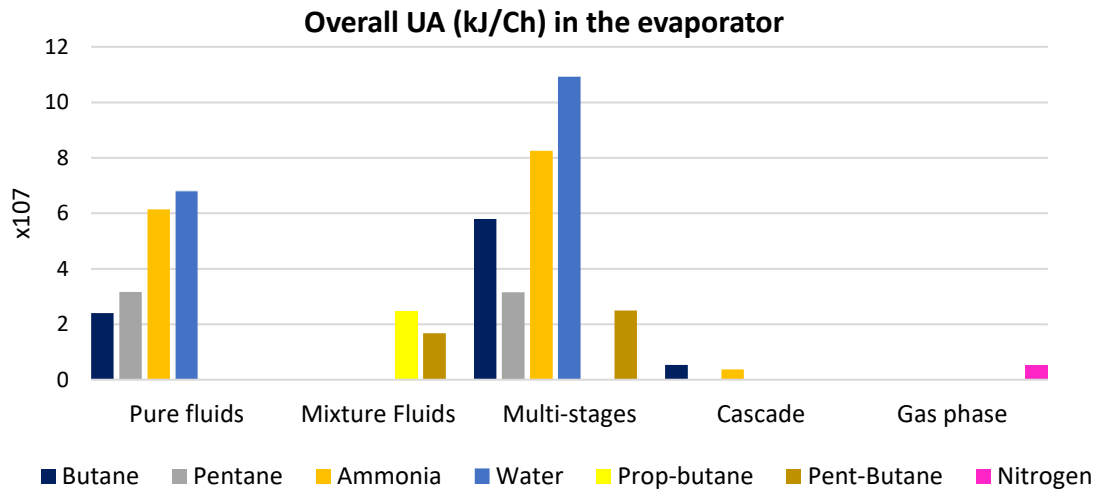
Figure 6.14 Overall UA results obtained in Case II
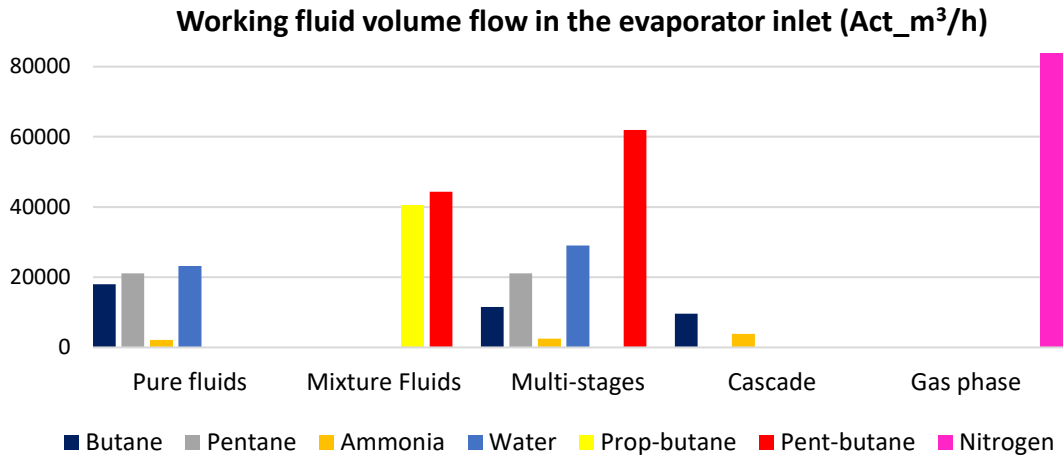
Figure 6.15 Evaporator inlet volume flow results (Case I)
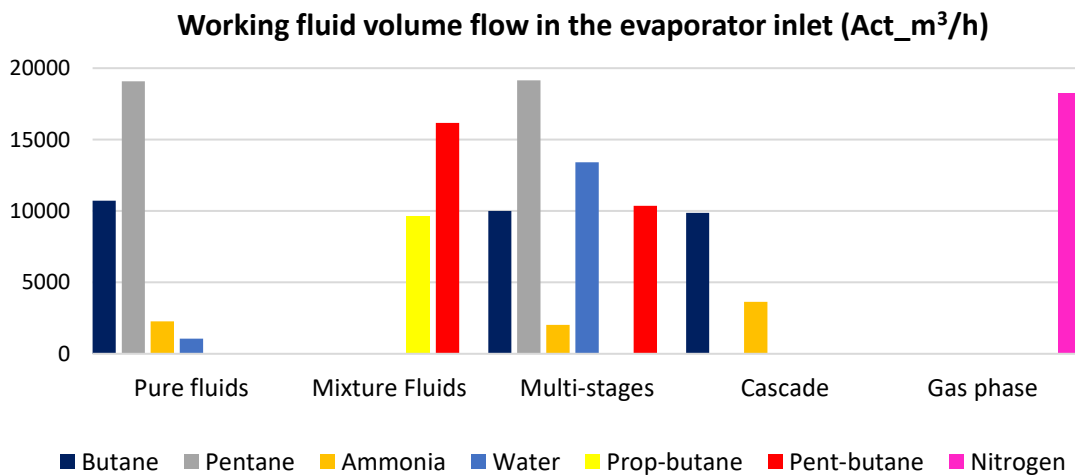
Figure 6.16 Evaporator inlet volume flow results (Case II)

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As for the condenser, the design is very similar to the evaporator but the difference is the heat duty is already defined for the condenser (see Chapter 4). The condenser usually has a vent for removal of non-condensable gas because they decrease the heat transfer rate. To size the condenser, volume flow rates are used. Figure 6.17 and Figure 6.18 illustrate different volume flow rates in the condenser inlet for the cases analysed. As seen in Section 6.2.1 water involves the largest volume suction flow in the compressor involving the largest volume flow in the evaporator inlet for both cases. The second largest flow is for nitrogen reaching similar water flow values in Case II.

Working fluid volume flow in the condenser inlet (Act_m³/h)
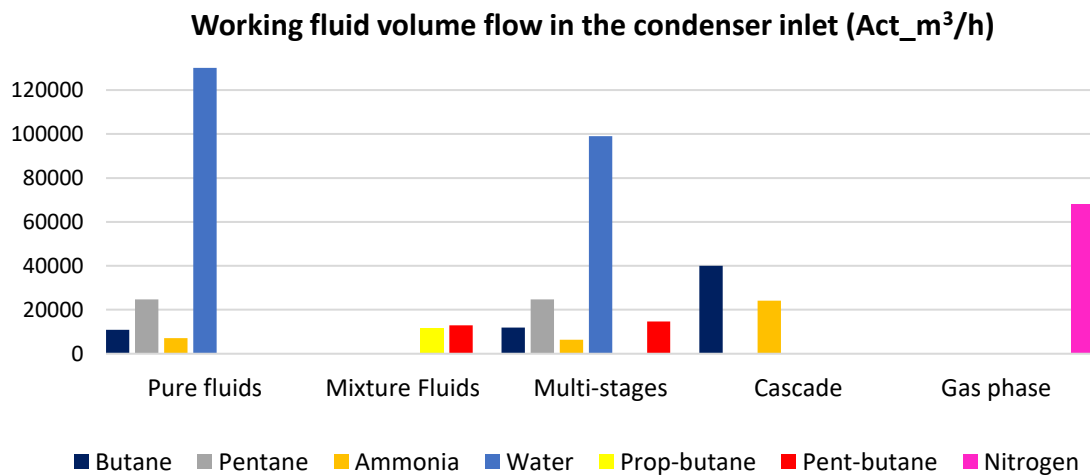
Figure 6.17 Condenser inlet flow results (Case I)

Working fluid volume flow in the condenser inlet (Act_m³/h)
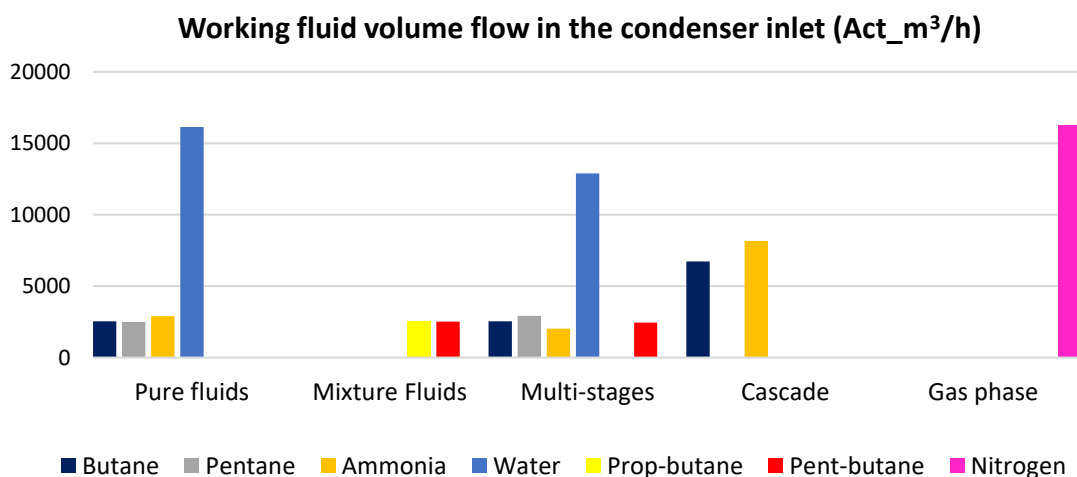
Figure 6.18 Condenser inlet flow results (Case II)

6.2.3 Expansion valve

Expansion valves are components used to decrease the working fluid pressure after heat release to the designated heat sink. Its mechanical configuration is greatly simplified compared to the other components in heat pump systems, as its sole purpose is to restrict the working fluid's flow rate as it enters the evaporation stage. When the cycle works in gas phase, the gas is not condensed and an expander (turbine) is required to reduce the pressure of the gas fluid. With the expander, the pressure reduction produces work in the turbine.

To size the valve, it is important to know the working pressure and temperature for the rating according to the ASME B16.34 Standard Class. After that, knowing the volume flow and the section of the pipe it defines the pipe flanges and flanged fittings (normally according to the ASME B16.5 Standard).

The rating showed in the Table 6.5 is based on Carbon Steel as the material valve, which is very common and suitable for this application (see Appendix I for the valve rating). The maximum working temperature is defined because of the condenser outlet. For the Case I, the maximum working temperature is 70 °C while for the Case II is equal to 130 °C. Moreover, the cascade system requires two valves, one for the lower cycle and one for the upper cycle included in Table 6.5.

Table 6.5 Rating of the valves

		Max. Working Pressure (bar)		CLASS	
		Case I	Case II	Case I	Case II
Pure Fluids	Butane	38	95	300	900
	Pentane	16	37	150	300
	Ammonia	120	145	900	1500
	Water	5	14	150	150
Mixture Fluids	20/80 Propane - Butane	43	103	300	900
	20/80 Pentane - Butane	32	75	300	600
	50/50 Pentane - Butane	35	60	300	400
Cascade System	Ammonia (upper cycle)	35	50	300	400
	Butane (lower cycle)	13	25	300	300
Gas phase	Nitrogen	111	202	900	1500

6.2.4 Piping

In order to design the pipes, it differences the monophasic (only liquid) and two-phase flow lines. It should define the maximum velocity in the gas and liquid lines taking the most dominating phase for the calculations when it is a two-phase flow line.

Considering the volume flow rates obtained in the different parts of the heat pump system, it defined the largest flow rate is located in the compressor inlet due to the gas phase. The larger flow, the larger pipe diameter required so, in that part of the heat pump system the pipe will have the biggest sections. On the contrary, the minimum section is in the condenser outlet because the liquid phase and it is still pressurized.

Analysing the different flow volumes of the working fluids in Chapter 5, it clarified that the largest flow is for the water involving the biggest pipe section, while ammonia represents the smallest volume flow comprising the minimum pipe section.

After defining the nominal diameter, it selects the schedule of the pipe considering the maximum working temperature and pressure conditions for the material selected (Carbon Steel). Defining the schedule, it obtains the outside diameter and the wall thickness of the pipe. The Appendix J shows the rating table for Carbon Steel pipes, defining the schedule with the nominal diameter and working conditions.

Considering the allowable pressure drop, it would calculate pipe lengths and weights. Pipe section will therefore have to be dimensioned for the optimal ratio of pressure drop and U-value (size / cost of the heat exchanger). Volume flows are highly variable through the condenser or evaporator, particularly in the working fluid side because of the condensation/evaporation.

The pipe length is unknown, so some calculations cannot be completed. It should be considered the minimum pipe lengths, lowest weight and the most compact equipment because of the offshore plants targets. Low heat losses from the pipes require insulation, particularly where the working fluid has the highest temperature, increasing the weight and the cost of them.

7 ENERGY AND EMISSIONS ANALYSIS IN A REAL CASE

The power and heat demand in FPSOs and oil and gas production facilities are normally covered using gas turbines to produce power and with waste heat recovery units that cover the heat demand. A base line scenario is defined consisting of a "standard" FPSO with gas turbines to evaluate the impact of using fuel gas as the main driver.

As this report focuses on electric driven facilities that cover the power demand with electricity from the grid or local stations other scenarios appear. The increase of the electrical grid availability and the emission reduction policies has increased the interest of providing new plant projects with power from it. Depending on the desired ratio between electrical grid and gas turbine usage, the emissions are calculated consequently.

Considering the heat supply, it appears three scenarios to cover the heat demand in electrical driven facilities: using gas fired heaters, using direct electric heaters (resistance heating), or using heat pumps.

With the following equation, the CO_2 emissions based on the power loading can be calculated, using a normal plant availability value of 350 days, a gas turbine efficiency. The stoichiometric CO_2 formation factor is extracted from [27]:

$$\dot{m}_{CO2} \left[\frac{kg}{year}\right] = \Phi \frac{E_{tot}}{\eta_{GT}} \alpha$$

$$E_{tot} \left[\frac{kWh}{year}\right] = L_{tot} \, [kW] \; x \; 365 \frac{days}{year} \; x \; 24 \frac{h}{day}$$

Where:

Φ is the CO_2 formation factor that is equal to 0.18 kg CO_2/kWh for gas natural and 0.25 kg CO_2/kWh for fuel oil [27].

L_{tot} is the total power loading of the plant.

E_{tot} is the total energy over one year in kWh/year.

α is the availability of the plant (0.95).

η_{GT} is the gas turbine efficiency, which is 40 % according to General Electric.

For 2017, the tax rate for natural gas is 445 NOK per ton of CO_2 in Norway [1]. The combination of the carbon tax and the emissions trading system means that companies on the Norwegian shelf pay up to 500 NOK per tonne for their CO_2 emissions, which is higher than in other sectors in Norway and very high compared with carbon prices in other countries.

7.1 Base line scenario

The base line consists of a FPSO with an oil and associated gas reservoir and an oil production of 200000 bpd, similar to the case mentioned in Section 2.1.3 of the reference [2]. Now, in the simulation the water cut has been changed to a lower value (< 5 % of the stream) than in the case seen in Section 2.1 (around 80 % of the stream).

The reservoir can be characterized by an early production well because of the low water cut, while in the case seen in Section 2.1.3, it was characterized as an end-life production well. Gas turbines and a waste heat recovery unit carry out the heat and power supply.

Changing the water cut in the simulation, the power consumption obtained is now 270 MW (GOR 1961 m^3/m^3), while the heat demand is around 80 MW (much lower than in the case seen in Section 2.1.3 because no heat is used to heat the water excess). Depending on the turbine specification, one megawatt electric (MWe) produces 1.35 megawatt thermal (MWt) with an efficiency around 35 - 40 % (given by General Electric).

Considering the Brent oil price is 50 $/ bbl [26] and the oil production is 200,000 bpd, the annual economic income from oil sales would be 29,500 million NOK. Referring now to the power and heat supply costs with gas turbines, the natural gas price in the industrial activity is 135 NOK/MWh involving a total annual cost in energy production of 306 million NOK. Normally, the value is considered to be almost zero until quite a few years of production, when the injection pressure becomes high. The discounted price of future gas sales will be low today (8 % p.a.) resulting a price of 63 NOK/MWh during the plateau conditions. Considering 10 years of plateau production before gas export starts, it involves an annual total cost of 143 million NOK.

Considering CO_2 emissions, the production of 270 MWe involves emissions because the gas turbine would produce 350 MWt in exhaust gases, which is more than enough to cover the heat demand. Replacing the values in the equation seen above, it obtains an amount of 1 million tons per year for the natural gas. That value is realistic because in comparison with Melkøya facility produces 920000 ton/year [25] with an installed power production of 215 MW and a production capacity of 4.1 MTPA. In conclusion, 1 million tons of CO_2 using natural gas would be released. The total CO_2 tax amount for this case would be 505 million NOK for the natural gas as fuel. The margin obtained counting energy costs and emissions is shown in Table 7.1. The costs can reach up to a 4 % (incl. CO_2 taxes) of the total oil sales for the natural gas after the plateau conditions.

Table 7.1 Economic analysis base line scenario

		Million NOK		
Oil sales income		29,500		
Energy costs	Power	0 (considering gas price almost 0)	143 (Plateau conditions)	306
	Heat			
CO₂ emissions taxes		505	505	505
Benefit		28995	28852	28689

7.2 Scenario A: Heat covered by fuel fired heaters

The procedure followed is the same as seen in the previous one. The power demand is 270 MW covered by electricity (electrical driven facility) while the heat demand of 80 MW is covered by fuel fired heaters.

Talking about the power and heat supply costs, it assumes an electricity price in the industrial activity of 400 NOK/MWh [28]. Depending on the energy (thermal, renewable or hydropower production) supplied to the grid the price can vary. The power supply cost (for 270 MW) with electricity from the grid would be 907 million NOK per year. The heat supply cost (80 MW) burning gas natural in the heaters would be 42 million NOK during the plateau conditions. The CO_2 emissions associated with the gas natural combustion would be 300 million tons.

Finally, the margin obtained between the oil sale income minus energy costs and taxes would be 28251 million NOK during 10 years of plateau conditions (no gas sales). Assuming the fuel gas cost almost zero, the benefit will be 28593 million NOK. After the plateau conditions, the heat supply cost would be 91 million NOK obtaining a margin of 28202 million NOK. During 10 years of plateau conditions, the benefit is 601 million NOK lower than using gas natural for all the energy requirements (heat and power) after that, the margin is 487 million NOK lower than using natural gas for power supply and exhaust gases for heat supply.

Table 7.2 Economic analysis scenario A: fuel fired heaters

		Million NOK		
Oil sales income		29,500		
Energy costs	Power	907		
	Heat	0 (considering gas price almost 0)	42 (Plateau conditions)	91
CO₂ emissions taxes		300	300	300
Benefit		28593	28251	28202

7.3 Scenario B: Heat covered by electric heaters

As the previous case, the power demand is covered by electricity from the grid. The heat demand in this case is also covered by electricity using electric heaters. The total energy demand would be the sum of both which is equal to 350 MW. The cost involved in heat and power supply would be 1176 million NOK per year.

96 % of the electric energy in Norway is from hydropower production [28] assuming that, this case does not involve emissions released over the power production. Other option would be import electricity from a local station (e.g. combined cycle) involving the emissions in the power generation or it could have carbon capture and storage compensating with "negative" CO_2 emissions.

As shown in Table 7.3, the margin obtained counting the heat and power supply costs is 28324 million NOK. During the plateau production, the benefit of electric heaters is lower (73 million NOK) than using fuel fired heaters. After the plateau production, the benefit is higher for electric heaters than fuel fired heaters, involving 122 million NOK more. Compared with the base case, the costs are higher with electric energy supply than using gas turbines in all the cases, reaching costs up to 671 million NOK more than gas turbines (considering gas price almost zero). After 10 years of plateau conditions, electric driven facilities have a cost of 365 million NOK more than the base case.

Table 7.3 Economic analysis scenario B: electric heaters

		Million NOK
Oil sales income		29,500
Energy costs	Power	907
	Heat	269
CO_2 emissions taxes		-
Benefit		28324

7.4 Scenario C: Heat covered by heat pumps

In this scenario, the power demand is also covered with electricity while the heat pump system would cover the heat demand. As seen in Section 5.1, ammonia would have a good performance in the system as a working fluid with an average COP of 4 (See sub-section 5.1.4). Applying ammonia cycle with two compression stages (defined in Section 4.1 of modelling basis) with the real heat demand seen above equal to 80 MW, it would require approximately a total power consumption by the compressors of 25 MW.

The total power supply cost for the compressors in the heat pump system would be 84 million NOK. Besides, the cost to cover the power demand (270 MW) with electricity from the grid would be 907 million NOK. It assumes no CO_2 emissions realised because of hydropower production from the grid involving no emission taxes.

Additional heat may be required using fuel furnaces because of the heat pump limitations, involving emissions.

As seen in Table 7.4, the heat supply cost is much lower than in the previous cases. Furthermore, with the heat pump system only the compressor requires a power demand. The heat supply cost is up to 83 % lower (emission tax included) than using gas turbines in the plateau production and 93 % after these conditions. In comparison with fuel-fired heaters, the cost is up to 72% lower in plateau conditions and 79 % in normal production. In electrical driven facilities, the heat supply cost is 69 % lower using heat pumps than electric heaters.

Table 7.4 Economic analysis scenario C: heat pump system

		Million NOK
Oil sales income		29,500
Energy costs	Power	907
	Heat	84
CO_2 emissions taxes		-
Benefit		28509

7.5 Comparison

The margin obtained in the last scenario using heat pumps would be 28,509 million NOK comprising the energy costs the 3.4 % of the total oil sales. The maximum benefit obtained among the scenarios is 28995 million NOK related to the base scenario due to the plateau conditions that involves the fuel gas price almost zero. However, after that plateau period, the scenario with heat pump after 10 years becomes more profitable.

The total heat and power costs (emission taxes) in the base line scenario would be 1.7 % of the total oil sales (plateau conditions gas cost almost zero) and 2.7 % in normal conditions. Considering the fuel fired heater scenario with electric power supply, the total heat and power costs would be 4.1 % for the plateau conditions and 4.4 % in normal production of the total oil sales. For heat pump system, the costs would consist of 3.3 % of the total oil sales.

Besides, the benefit in the electrical driven case with heat pump is up to 307 million NOK more than partly electrified (only power supply and fuel fired heaters for heat supply) and 185 million NOK more than using electric heaters to cover the heat demand.

Based on the International Energy Agency, the cost of an industrial heat pump can be a value close to 6000 NOK/kW (for 150kW). If we considered a pay-back time of 3 years, the heat pump cannot cost more than 450 million NOK involving a cost of 5600 NOK/kW heating capacity. It may seem a challenge but it has to be considered an economy of scale in this case; the larger the heating capacity of the equipment (heating capacity), the lower the cost will be.

Figure 7.1 represents the different costs for heat and power supply, together with the CO_2 taxes for the natural gas. As seen, the base line scenario has the lowest cost because of low cost fuel gas in plateau conditions. The exhaust gases produced in the power production with gas turbines cover the heat demand so there is not cost involved for heat supply. Scenarios A, B and C have the same power supply cost because it comprises an electric driven case for power supply. The lowest heat supply cost is for scenario C due to the heat pump system because the only cost is the compressor power. CO_2 emissions released due to fuel gas are illustrated for the base line scenario with gas turbines and for the fuel fired heaters scenario. Heat pump scenario could involve some emissions if additional heat is required using fuel furnaces.
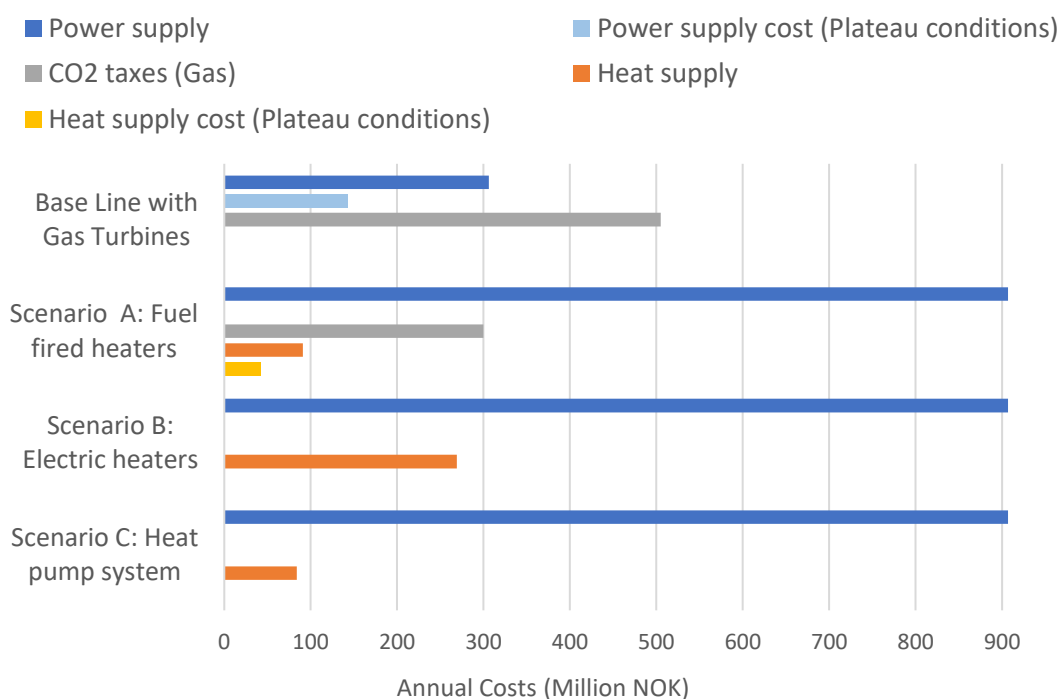
Figure 7.1 Annual costs in different scenarios

7.6 Shutdown problems, heat supply problems.

If there is a shutdown or malfunction in the heat pump system, the risk of no heat supply can stop the plant operation. Different alternatives can be implemented to reduce the risk or fight with that situation.

One option is auxiliary fuel furnaces. They can turn on when the failure appears in the plant (e.g. lack of heat source) supplying heat to the heat pump. Fuel furnaces can also replace the function of the heat pump (if it completely fails) heating directly the process stream which requires to be heated.

Another option can be use parallel redundancy by placing units in parallel can manage the situation (depending on the failure). The n+1 configuration is a safeguard to ensure the system is always available. Sometimes that configuration is too expensive to be implemented as the compressors. Compressors redundancy is not often, because compressors are big, expensive and many auxiliary systems are involved, duplicating them too. Referring to heat exchangers, redundancy is not usual too but more reasonable than compressors.

If two heat pump systems are installed (one system for the Case I and the other for Case II) and the failure appears in one of them, the available heat pump can support the other one. It would supply heat avoiding or compensating the complete shutdown.

8 DISCUSSION AND CONCLUSIONS

8.1 Discussion

Available heat supply at high temperatures is more challenging due to the temperature limitation of the heat pumps or the associated components. Material improvements and equipment studies can conduct to achieve the goal, handling these operational restrictions. The process characteristics justify two case studies with low temperature heat sources. Case I reflects the medium temperature supply requirement around 100 ºC while Case II reflects the challenges of the high temperature supply at 200 ºC.

A simple simulation is developed to evaluate the working fluid suitability for the heat pump, extracting energy efficiencies and capacities data. The evaluations are based on heat and power requirements data from general literature of production plants, providing a good scenario for the heat pump analysis.

Chapter 5 introduces the working fluid selection with the potential applicability in high temperature industrial heat pumps. Different fluids were evaluated in two cases (medium and high temperature) for the possible application. Initiated by a preliminary literature review based on the heat process requirements and potential heat sources available, and finalized by an equipment specification and economic analysis.

Four main solutions are available: multi-stage heat pumps, cascade heat pumps, mixtures or gas phase heat pumps. Single stage compression systems are not suitable because of the high-pressure ratios so two-stage systems are deemed most applicable. Cascade and multi-stage systems have the best performance reaching high COPs. On the contrary, they involve more space and weight requirements than single-stage systems because of the machine numbers.

In cascade systems, two cycles are involved, duplicating the equipment required whereas for multi-stage systems the compressor number increases according to the stages required. Cascade system improves the performance than using butane as pure fluid but on the contrary, it reduces the efficiency in comparison with ammonia. The system efficiency still good comprises values between butane and ammonia. Summarizing the mixtures (with no optimization aim), major concentration of butane improves the system efficiency, as it is logical butane has a better performance than pentane. Using two stages with mixtures, the efficiency improves as for pure fluids.

The last system carried out was the gas phase heat pump, using nitrogen as working fluid obtaining low efficiencies. A little modification appears on the system, replacing the valve by a turbine. It analyses some results obtained for the proper operation as volume flow rates, temperatures and duties for compressors and heat exchangers. Equipment dimensions, footprint and weights are important aspects related to these simulation results. Finally, an economic analysis has been carried out calculating CO_2 emissions amount, carbon taxes and energy supply costs. Among these supply alternatives, the electrical driven facility with heat pump system shows a profitable future for normal production facilities because during plateau conditions the gas price is so low, becoming the gas turbines application more profitable.

8.2 Conclusions

Industrial heat pumps can effectively supply medium grade heat energy (100 - 200 ℃) recovering residual heat from low temperature sources. Considering the fluids analysed, ammonia reaches the maximum COP resulting in a value of 4.2 for Case I and 3.9 for Case II. Good points of ammonia are the lowest volume flow, reducing the equipment and pipe sizing, and the maximum heat source flow involved, that means ammonia absorbs the maximum heat amount in comparison with the others involving a compressor work reduction because of the low volume flow. Water has also high COPs but it appears the problem of excessive volume flows (ammonia is 98 % lower volume flow) involving huge and heavy equipment.

The heat pump integration would be challenging due to aspects as space limitations, sizes and weights. By contrast, the gas turbine removal would release some area suitable for the heat pump system, especially for compressors and heat exchangers that are the biggest equipment. The current heat scheme may have to be re-designed using as much as possible the existing infrastructure.

The most suitable compressors would be axial for water because of the huge volume suction flows up to 1,566,00 m^3/h and low discharge pressures up to 14 bar. Centrifugal compressors are suitable for the rest of fluids because lower flow rates and higher discharge pressure are required. In principle, screw compressors do not fit with the pressure and flow requirements. Two-stages allows a better implementation; the first stage requires a higher volume suction flow with a lower pressure discharge allowing one type of compressor (e.g. axial compressor) in comparison with the second stage requires lower flows but higher discharge pressures (e.g. centrifugal compressors).

60

Shell and tube heat exchanges involves problematic aspects in the performance because of the two-phase regimen and temperature cross or small temperature approaches. The most suitable heat exchangers would be frame, printed or fin plate type.

Considering expansion valves and piping, there are not so detailed specification because of the unknown lengths and pressure drops. Only rating can be done, defining the temperature and pressure design based on the maximum working conditions.

From the economic and emissions view, using heat pump system to supply heat in electrical driven facilities it is not profitable in comparison with a standard FPSO with gas turbines to supply heat and power. The costs (energy + emission taxes) comprises 3.3% of the total oil sales for an electrical driven facility with HP while for a "standard" FPSO, the costs comprise 1.7 % of the total sales during the plateau production. After that, the costs rise up to 3 % of the oil sales (influenced by oil and gas market price).

In comparison with the other scenarios, heat pumps would be more profitable. For the electric heater scenario, the energy cost reaches 4 % of the oil sales and for fuel fired heater scenario, the total energy cost is 4.2 % of the oil sales. Assuming could have carbon capture and storage installation or a renewable energy production for the power supply of the compressor heat pump, the emissions would be zero. Heat pumps may save between 1 to 1.5 million CO_2 tons or 500 to 700 million NOK in comparison with a standard FPSO (depending on the fuel used).

8.3 Suggestions for a further work

An important part of this thesis is the technical review, where the industrial scenarios, promising working fluids and market available components are identified. Industrial heat pumps for high temperature applications are not a complex task but it requires a further study for more detailed data as piping lengths, weights and minimum areas required for the equipment. From the technical point, the material limitations are covered because as seen compressors are able to work at these operation conditions.

Shell-and-tube heat exchangers are typically employed, but other designs are available and should be investigated. Plate and fin or printed exchangers are among the most prominent alternatives. Other aspects as the real feasibility need to be evaluated according to dimensions for the system implementation in the plant. Other suggestion would be the heat source evaluation to check the real availability of heat sources in the plant according to heat requirements for the heat pump system.

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APPENDIX A – BUTANE PROCESS DATA

Case I. Flowsheet diagram

Condenser		
Duty	150,0	MW
Feed Temperature	155,4	C
Product Temperature	70,00	C

Compressor2		
Power	1,541e+004	kW
Capacity (act feed vol flow)	3,513e+004	ACT_m3/h
Feed Pressure	15,22	bar
Product Pressure	38,00	bar
Product Temperature	155,4	C

FEED		
Temperature	50,48	C
Pressure	5,000	bar
Mass Flow	384,0	kg/s
Comp Mole Frac (H2O)	0,0000	

Evaporator		
Duty	117,6	MW
Shell Side Feed Mass Flow	2711	kg/s
Tube Inlet Temperature	50,48	C
Tube Outlet Temperature	60,00	C
Shell Inlet Temperature	70,00	C
Shell Outlet Temperature	60,00	C

Compressor		
Duty	27,95	MW
Feed Temperature	60,00	C
Feed Pressure	4,000	bar
Product Temperature	111,7	C
Product Pressure	15,22	bar

Temperature glide in the evaporator

Case II. Flowsheet diagram

E-100		
Duty	50,00	MW
Feed Temperature	199,1	C
Product Temperature	130,0	C

K-100		
Power	1,120e+004	kW
Capacity (act feed vol flow)	1,326e+004	ACT_m3/h
Feed Pressure	20,00	bar
Product Pressure	86,00	bar
Product Temperature	199,1	C

FEED		
Temperature	10,17	C
Pressure	1,486	bar
Mass Flow	201,5	kg/s
Comp Mole Frac (H2O)	0,0000	

Evaporator		
Duty	40,22	MW
Shell Side Feed Mass Flow	230,7	kg/s
Tube Inlet Temperature	10,17	C
Tube Outlet Temperature	90,00	C
Shell Inlet Temperature	100,0	C
Shell Outlet Temperature	60,01	C

Compressor		
Duty	42,21	MW
Feed Temperature	90,00	C
Feed Pressure	1,000	bar
Product Temperature	199,1	C
Product Pressure	20,00	bar

Temperature glide in the evaporator

APPENDIX B – PENTANE PROCESS DATA

Case I. Flowsheet diagram

K-100		
Power	2,177e+004	kW
Capacity (act feed vol flow)	4,396e+005	ACT_m3/h
Feed Pressure	1,000	bar
Product Pressure	4,000	bar
Product Temperature	98,87	C

Compressor		
Duty	23,68	MW
Feed Temperature	98,87	C
Feed Pressure	4,000	bar
Product Pressure	16,00	bar
Product Temperature	149,9	C

Evaporator		
Duty	104,6	MW
Shell Side Feed Mass Flow	2412	kg/s
Tube Inlet Temperature	57,79	C
Tube Outlet Temperature	60,00	C
Shell Inlet Temperature	70,00	C
Shell Outlet Temperature	60,00	C

E-100		
Duty	150,0	MW
Feed Temperature	149,9	C
Product Temperature	70,00	C

FEED		
Temperature	57,79	C
Pressure	2,000	bar
Mass Flow	328,3	kg/s
Comp Mole Frac (n-Pentane)	1,0000	

Temperature glide in the evaporator

Case II. Flowsheet diagram

Compressor		
Duty	7,817	MW
Feed Temperature	122,8	C
Feed Pressure	9,500	bar
Product Pressure	35,00	bar
Product Temperature	198,9	C

Condenser		
Duty	50,00	MW
Feed Temperature	198,9	C
Product Temperature	130,0	C

Evaporator		
Duty	33,96	MW
Shell Side Feed Mass Flow	774,9	kg/s
Tube Inlet Temperature	88,27	C
Tube Outlet Temperature	90,00	C
Shell Inlet Temperature	100,0	C
Shell Outlet Temperature	90,01	C

FEED		
Temperature	88,27	C
Pressure	4,500	bar
Mass Flow	167,5	kg/s
Comp Mole Frac (n-Pentane)	1,0000	

K-100		
Power	8253	kW
Capacity (act feed vol flow)	6,563e+004	ACT_m3/h
Feed Pressure	3,500	bar
Product Pressure	9,500	bar
Product Temperature	122,8	C

Temperature glide in the evaporator

APPENDIX C – AMMONIA PROCESS DATA

Case I. Flowsheet diagram

E-100		
Duty	150,0	MW
Feed Temperature	189,1	C
Product Temperature	70,00	C

K-100		
Power	2,000e+004	kW
Capacity (act feed vol flow)	1,113e+004	ACT_m3/h
Feed Pressure	53,00	bar
Product Pressure	121,0	bar
Product Temperature	189,1	C

FEED		
Temperature	56,71	C
Pressure	24,00	bar
Mass Flow	133,2	kg/s
Comp Mole Frac (H2O)	0,0000	

Evaporator		
Duty	132,2	MW
Shell Side Feed Mass Flow	3050	kg/s
Tube Inlet Temperature	56,71	C
Tube Outlet Temperature	55,01	C
Shell Inlet Temperature	70,00	C
Shell Outlet Temperature	60,00	C

Compressor		
Duty	20,91	MW
Feed Temperature	55,01	C
Feed Pressure	23,00	bar
Product Temperature	145,2	C
Product Pressure	53,00	bar

Temperature glide in the evaporator

Case II. Flowsheet diagram

Temperature glide in the evaporator

APPENDIX D – WATER PROCESS DATA

Case I. Flowsheet diagram

E-100		
Duty	150,0	MW
Feed Temperature	241,1	C
Product Temperature	70,00	C

K-101		
Power	1,289e+004	kW
Capacity (act feed vol flow)	1,807e+005	ACT_m3/h
Feed Pressure	2,000	bar
Product Pressure	4,760	bar
Product Temperature	241,1	C

K-100		
Power	1,489e+004	kW
Capacity (act feed vol flow)	4,808e+005	ACT_m3/h
Feed Pressure	0,7000	bar
Product Pressure	2,000	bar
Product Temperature	229,0	C

FEED		
Temperature	59,97	C
Pressure	0,1980	bar
Mass Flow	56,14	kg/s
Comp Mole Frac (H2O)	1,0000	

Evaporator		
Duty	131,1	MW
Shell Side Feed Mass Flow	3023	kg/s
Tube Inlet Temperature	59,97	C
Tube Outlet Temperature	60,00	C
Shell Inlet Temperature	70,00	C
Shell Outlet Temperature	60,00	C

Compressor		
Duty	16,56	MW
Feed Temperature	60,00	C
Feed Pressure	0,1980	bar
Product Temperature	214,8	C
Product Pressure	0,7000	bar

Temperature glide in the evaporator

Case II. Flowsheet diagram

E-100		
Duty	50,00	MW
Feed Temperature	287,3	C
Product Temperature	130,0	C

K-101		
Power	4844	kW
Capacity (act feed vol flow)	2,328e+004	ACT_m3/h
Feed Pressure	6,000	bar
Product Pressure	14,00	bar
Product Temperature	287,3	C

K-100		
Power	4750	kW
Capacity (act feed vol flow)	5,273e+004	ACT_m3/h
Feed Pressure	2,500	bar
Product Pressure	6,000	bar
Product Temperature	251,6	C

FEED		
Temperature	90,00	C
Pressure	0,7016	bar
Mass Flow	20,17	kg/s
Comp Mole Frac (H2O)	1,0000	

Evaporator		
Duty	42,84	MW
Shell Side Feed Mass Flow	977,7	kg/s
Tube Inlet Temperature	90,00	C
Tube Outlet Temperature	90,00	C
Shell Inlet Temperature	100,0	C
Shell Outlet Temperature	90,01	C

Compressor		
Duty	6,491	MW
Feed Temperature	90,00	C
Feed Pressure	0,7016	bar
Product Temperature	258,2	C
Product Pressure	2,500	bar

Temperature glide in the evaporator

APPENDIX E – CASCADE PROCESS DATA

Case I. Flowsheet diagram

Condenser		
Duty	150,0	MW
Feed Temperature	163,1	C
Product Temperature	70,00	C

Ammonia Compressor		
Power	1,522e+004	kW
Capacity (act feed vol flow)	3,629e+004	ACT_m3/h
Feed Pressure	20,00	bar
Product Pressure	35,00	bar
Product Temperature	163,1	C

FEED		
Temperature	50,48	C
Pressure	5,000	bar
Mass Flow	339,1	kg/s
Comp Mole Frac (Ammonia)	0,0000	
Comp Mole Frac (n-Butane)	1,0000	

Butane Evaporator		
Duty	112,9	MW
Shell Side Feed Mass Flow	2604	kg/s
Tube Inlet Temperature	50,48	C
Tube Outlet Temperature	60,00	C
Shell Inlet Temperature	70,00	C
Shell Outlet Temperature	60,01	C

Butane Compressor		
Duty	21,83	MW
Feed Temperature	60,00	C
Feed Pressure	4,000	bar
Product Pressure	13,00	bar
Product Temperature	104,8	C

Temperature glide in butane evaporator (left) and in ammonia evaporator (right)

Case II. Flowsheet diagram

Temperature glide in butane evaporator (left) and in ammonia evaporator (right)

APPENDIX F – PROPANE BUTANE MIXTURE

Case I. Flowsheet diagram

Condenser		
Duty	150,0	MW
Feed Temperature	154,0	C
Product Temperature	90,00	C

Compressor		
Duty	39,38	MW
Feed Temperature	60,00	C
Feed Pressure	7,481	bar
Product Pressure	43,00	bar
Product Temperature	154,0	C

FEED		
Temperature	50,21	C
Pressure	6,481	bar
Mass Flow	470,9	kg/s
Comp Mole Frac (Propane)	0,2000	
Comp Mole Frac (n-Butane)	0,8000	

Evaporator		
Duty	110,6	MW
Shell Side Feed Mass Flow	2551	kg/s
Tube Inlet Temperature	50,21	C
Tube Outlet Temperature	60,00	C
Shell Inlet Temperature	70,00	C
Shell Outlet Temperature	60,00	C

Temperature glide in the evaporator

Case II. Flowsheet diagram

Condenser		
Duty	50,00	MW
Feed Temperature	199,8	C
Product Temperature	130,0	C

Compressor		
Duty	17,14	MW
Feed Temperature	90,00	C
Feed Pressure	14,00	bar
Product Pressure	103,0	bar
Product Temperature	199,8	C

FEED		
Temperature	88,18	C
Pressure	15,00	bar
Mass Flow	204,8	kg/s
Comp Mole Frac (Propane)	0,2000	
Comp Mole Frac (n-Butane)	0,8000	

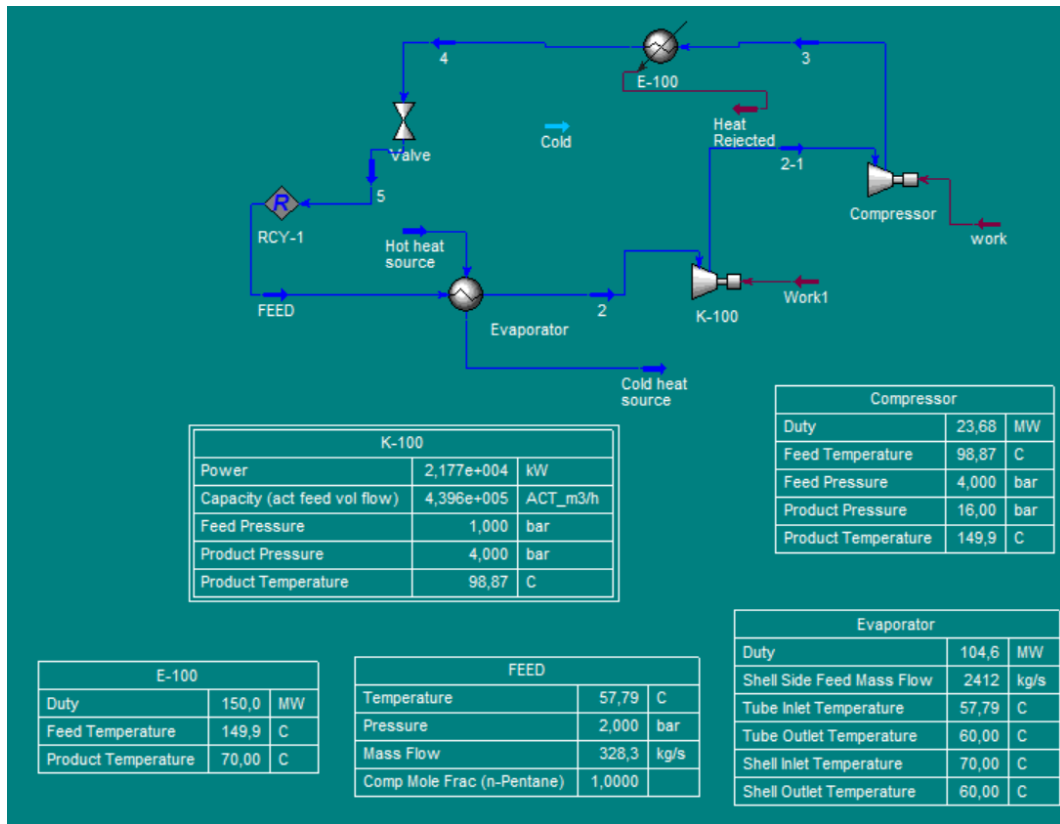| Evaporator		
Duty	32,86	MW
Shell Side Feed Mass Flow	748,8	kg/s
Tube Inlet Temperature	88,18	C
Tube Outlet Temperature	90,00	C
Shell Inlet Temperature	100,0	C
Shell Outlet Temperature	90,00	C

Temperature glide in the evaporator

APPENDIX G – PENTANE - BUTANE PROCESS DATA

Case I. Flowsheet diagram

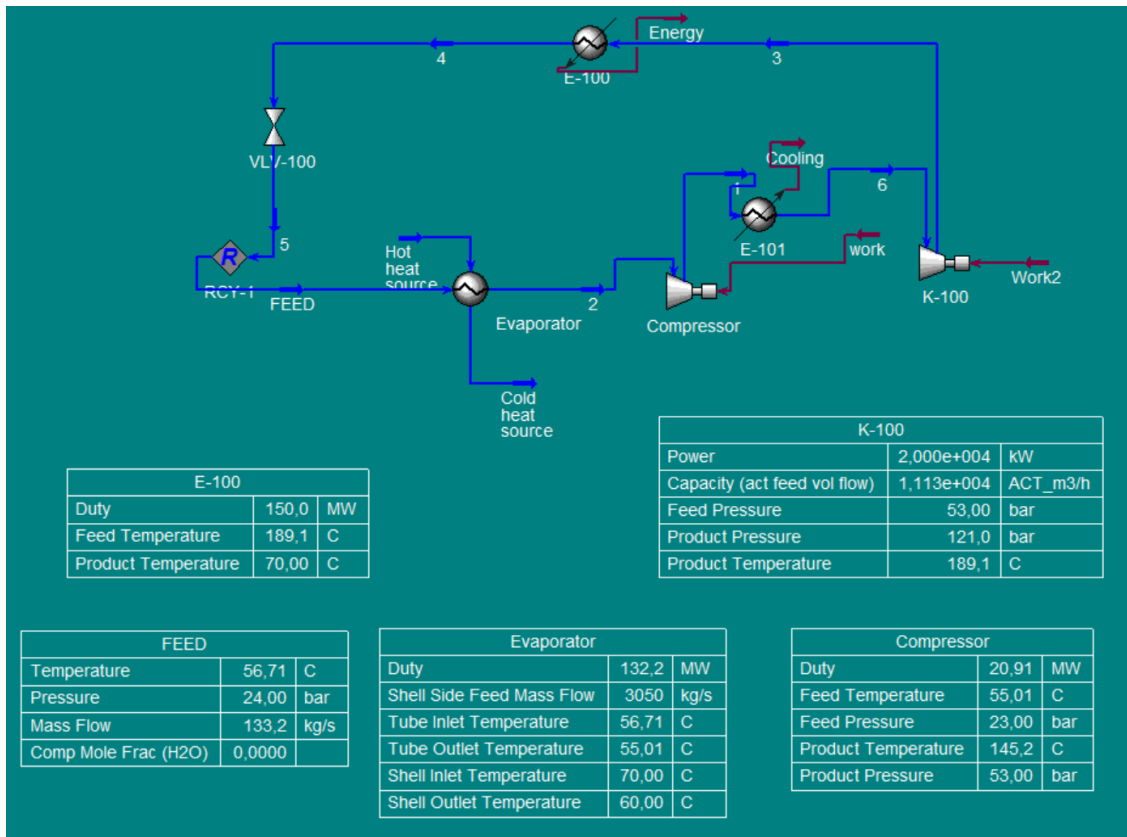| Condenser		
Duty	150,0	MW
Feed Temperature	152,3	C
Product Temperature	80,00	C

Compressor		
Duty	40,24	MW
Feed Temperature	60,00	C
Feed Pressure	4,617	bar
Product Pressure	31,54	bar
Product Temperature	152,3	C

FEED		
Temperature	45,34	C
Pressure	3,615	bar
Mass Flow	395,9	kg/s
Comp Mole Frac (n-Butane)	0,8000	
Comp Mole Frac (n-Pentane)	0,2000	

Evaporator		
Duty	109,7	MW
Shell Side Feed Mass Flow	2534	kg/s
Tube Inlet Temperature	45,34	C
Tube Outlet Temperature	60,00	C
Shell Inlet Temperature	70,00	C
Shell Outlet Temperature	60,01	C

Temperature glide in the evaporator

Case II. Flowsheet diagram

Condenser		
Duty	50,00	MW
Feed Temperature	200,1	C
Product Temperature	130,0	C

Compressor		
Duty	15,35	MW
Feed Temperature	90,00	C
Feed Pressure	9,563	bar
Product Pressure	74,37	bar
Product Temperature	200,1	C

FEED		
Temperature	81,34	C
Pressure	8,563	bar
Mass Flow	195,9	kg/s
Comp Mole Frac (n-Butane)	0,8000	
Comp Mole Frac (n-Pentane)	0,2000	

Evaporator		
Duty	34,65	MW
Shell Side Feed Mass Flow	789,5	kg/s
Tube Inlet Temperature	81,34	C
Tube Outlet Temperature	90,00	C
Shell Inlet Temperature	100,0	C
Shell Outlet Temperature	90,00	C

Temperature glide in the evaporator

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APPENDIX H – NITROGEN PROCESS DATA

Case I. Flowsheet diagram

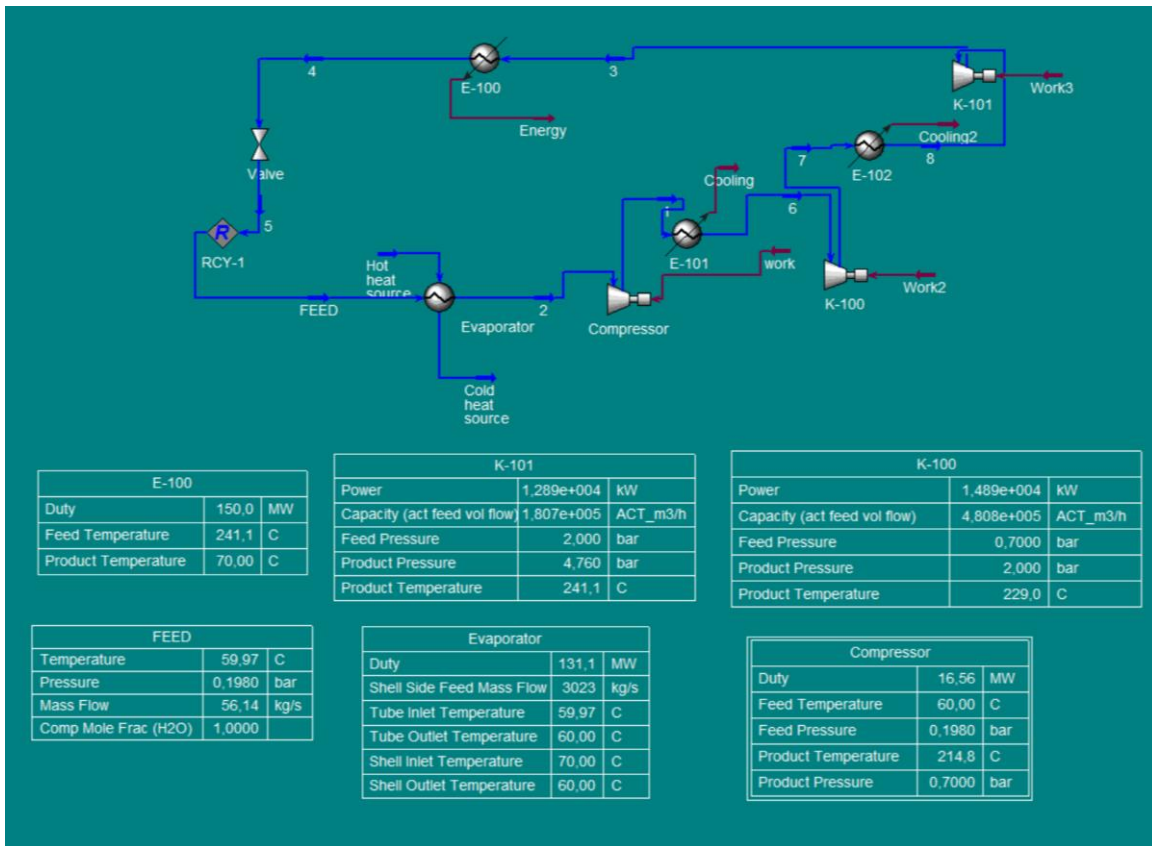| FEED		
Temperature	22,05	C
Pressure	60,00	bar
Mass Flow	1621	kg/s
Comp Mole Frac (n-Butane)	0,0000	
Comp Mole Frac (n-Pentane)	0,0000	

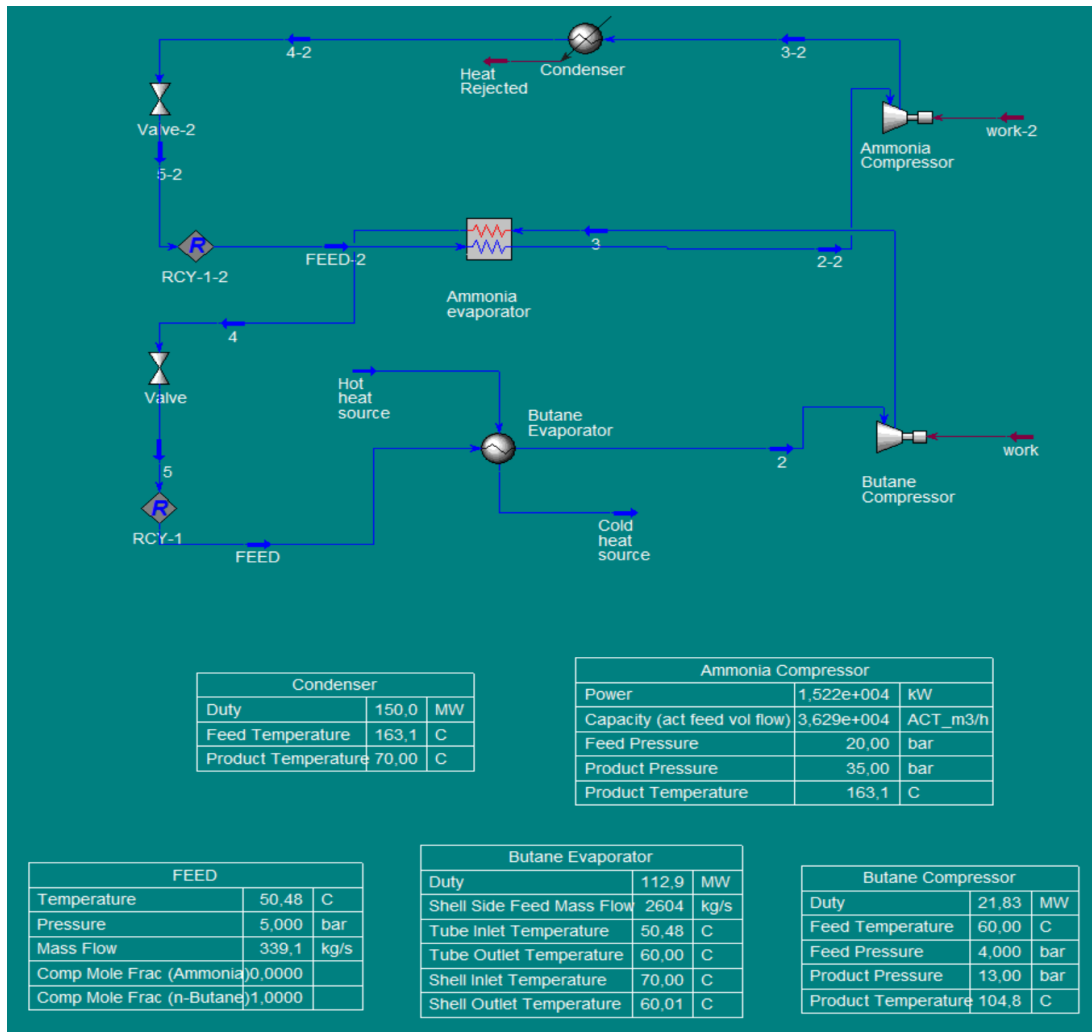| Evaporator		
Duty	70,37	MW
Shell Side Feed Mass Flow	1623	kg/s
Tube Inlet Temperature	22,05	C
Tube Outlet Temperature	60,00	C
Shell Inlet Temperature	70,00	C
Shell Outlet Temperature	60,00	C

Compressor		
Duty	147,5	MW
Feed Temperature	65,00	C
Feed Pressure	59,00	bar
Product Pressure	111,5	bar
Product Temperature	150,0	C

Condenser		
Duty	150,0	MW
Feed Temperature	150,0	C
Product Temperature	70,00	C

Temperature glide in the evaporator (left) and in the recuperator (right)

Case II. Flowsheet diagram

FEED		
Temperature	60,41	C
Pressure	120,0	bar
Mass Flow	604,8	kg/s
Comp Mole Frac (n-Butane)	0,0000	
Comp Mole Frac (n-Pentane)	0,0000	

Evaporator		
Duty	21,24	MW
Shell Side Feed Mass Flow	484,2	kg/s
Tube Inlet Temperature	60,41	C
Tube Outlet Temperature	90,00	C
Shell Inlet Temperature	100,0	C
Shell Outlet Temperature	90,00	C

Compressor		
Duty	54,21	MW
Feed Temperature	120,0	C
Feed Pressure	119,0	bar
Product Pressure	202,3	bar
Product Temperature	200,0	C

E-100		
Duty	50,00	MW
Feed Temperature	200,0	C
Product Temperature	130,0	C

Temperature glide in the evaporator (left) and in the recuperator (right)

APPENDIX I – VALVE RATING

Rating table based on maximum pressure and temperature working conditions. Selected the valve material and defined the maximum working conditions in the valve, it obtains the class of the valve.

RATING Pres.-Temp
acc. ANSI B.16.34- 1996 (STANDARD CLASS)

Case I Case II   Pres. (bar-g)

CLASS	Temp. (°C)	-29/38	50	100	150	200	250	300	350	375	400	425	450	475	500	525	550	575	600
150	A105N/WCB	20.0	19.2	17.7	15.8	14.0	12.1	10.2	8.4	7.4	6.5	5.6							
	F12/WC6	20.0	19.2	17.7	15.8	14.0	12.1	10.2	8.4	7.4	6.5	5.6	4.7	3.7	2.8	1.9	1.3	1.3	1.3
	F22/WC9	20.0	19.2	17.7	15.8	14.0	12.1	10.2	8.4	7.4	6.5	5.6	4.7	3.7	2.8	1.9	1.3	1.3	1.3
	F316/CF8M	19.0	18.4	16.2	14.8	13.7	12.1	10.2	8.4	7.4	6.5	5.6	4.7	3.7	2.8	1.9	1.3	1.3	1.3
	F316L	15.9	15.3	13.2	12.0	11.0	10.2	9.7	8.4	7.4	6.5	5.6	4.7						
	LF2	20.0	19.2	17.7	15.8	14.0	12.1	10.2	8.4										
300	A105N/WCB	51.1	50.1	46.4	45.2	43.8	41.7	38.7	37.0	36.5	34.5	28.8							
	F12/WC6	51.7	51.1	48.8	46.4	45.5	44.5	42.2	40.2	38.8	36.6	35.1	33.8	31.7	27.8	20.3	12.8	8.5	5.9
	F22/WC9	51.7	51.2	49.0	46.6	44.8	44.2	42.4	40.2	38.8	36.6	35.1	33.8	31.7	27.8	21.9	16.4	11.7	7.6
	F316/CF8M	49.6	48.1	42.2	38.5	35.7	33.4	31.6	30.4	29.7	29.1	28.7	28.1	27.4	26.8	25.8	25.0	24.1	21.4
	F316L	41.4	40.0	34.5	31.2	28.7	26.7	25.5	24.0	23.6	23.2	22.7	22.3						
	LF2	51.1	50.1	46.4	45.2	43.8	41.7	38.7	37.0										
400	A105N/WC8	68.1	66.8	61.8	60.3	58.4	55.6	51.6	49.3	48.6	46.0	38.3							
	F12/WC6	69.0	68.2	65.0	61.8	60.6	59.3	56.6	53.6	51.7	48.8	46.8	45.1	42.2	37.1	27.0	17.0	11.3	7.8
	F22/WC9	69.0	68.3	65.4	62.2	59.8	59.0	56.6	53.6	51.7	48.8	46.8	45.1	42.2	37.1	29.2	21.8	15.6	10.2
	F316/CF8M	66.1	64.1	56.3	51.3	47.6	44.5	42.1	40.5	39.6	38.8	38.2	37.5	36.5	35.7	34.4	33.3	32.1	28.5
	F316L	54.8	53.3	46.0	41.6	38.3	35.6	34.0	32.0	31.5	30.9	30.3	29.7						
	LF2	68.1	66.8	61.8	60.3	58.4	55.6	51.6	49.3										
600	A105N/WCB	102.1	100.2	92.8	90.5	87.6	83.4	77.5	73.9	72.9	69.0	57.5							
	F12/WC6	103.4	102.3	97.4	92.7	91.0	88.9	84.9	80.5	77.6	73.2	70.2	67.6	63.3	55.6	40.5	25.5	17.0	11.8
	F22/WC9	103.4	102.4	98.1	93.3	89.7	88.4	84.9	80.5	77.6	73.2	70.2	67.6	63.3	55.6	43.8	32.7	23.4	15.3
	F316/CF8M	99.3	96.3	84.4	77.0	71.3	66.8	63.3	60.8	59.4	58.2	57.3	56.2	54.7	53.7	52.6	49.9	48.2	42.9
	F316L	82.7	79.9	69.0	62.5	57.4	53.4	50.5	48.1	47.2	46.3	45.4	44.5						
	LF2	102.1	100.2	92.8	90.5	87.6	83.4	77.5	73.9										
900	A105N/WCB	153.2	150.2	139.1	135.7	131.5	125.2	116.2	110.9	109.4	103.5	86.3							
	F12/WC6	155.1	153.4	146.3	139.1	136.4	133.4	127.3	120.7	116.4	109.8	105.3	101.4	95.0	83.4	60.8	38.3	25.5	17.6
	F22/WC9	155.1	153.6	147.1	139.9	134.5	132.7	127.3	120.7	116.4	109.8	105.3	101.4	95.0	83.4	65.8	49.1	35.1	22.9
	F316/CF8M	148.9	144.4	126.6	115.5	107.0	100.2	94.9	91.3	89.1	87.3	86.0	84.2	82.1	80.5	78.9	74.9	72.3	63.3
	F316L	124.0	119.8	103.5	93.8	86.1	80.1	75.8	72.2	70.8	69.5	68.1	66.1						
	LF2	153.2	150.2	139.1	135.7	131.5	125.2	116.2	110.9										
1500	A105N/WCB	255.3	250.4	231.9	226.1	219.1	208.6	193.7	184.8	182.3	172.5	143.8							
	F12/WC6	258.6	255.7	243.8	231.9	227.4	222.3	212.1	201.2	194.0	182.9	175.5	169.0	158.3	139.0	101.3	63.8	42.5	29.4
	F22/WC9	258.6	256.0	245.2	233.2	224.2	221.1	212.1	201.2	194.0	182.9	175.5	169.0	158.3	139.0	109.6	81.8	58.5	38.2
	F316/CF8M	248.1	240.6	211.0	192.5	178.4	166.9	158.1	152.1	148.1	145.6	143.3	140.4	136.8	134.1	131.5	124.8	120.5	107.2
	F316L	206.8	199.8	172.4	156.1	143.5	133.5	126.2	120.2	118.0	115.8	113.5	111.3						
	LF2	255.3	250.4	231.9	226.1	219.1	208.6	193.7	184.8										

APPENDIX J – PIPING RATING

Rating table based on maximum pressure and temperature working conditions. Selected the nominal size and diameter of the pipe and defining the material together with the maximum working conditions, it obtains the schedule of the pipe.

CARBON STEEL GRADE B PIPES - ASTM A53M, A106M, API 5L, Seamless

Nominal Size (DN)	(NPS)	Outside Diameter (mm)	Schedule		Wall Thickness (mm)	-29 to +38 / 137.9	204 / 137.9	260 / 130.3	343 / 117.2	371 / 113.8	399 / 89.6	427 / 74.5
250	10	273.1		20	6.35	5.7	5.7	5.4	4.8	4.7	3.7	3.1
				30	7.8	7.0	7.0	6.6	6.0	5.8	4.6	3.8
			STD	40	9.27	8.4	8.4	7.9	7.1	6.9	5.5	4.5
			XS	60	12.7	11.6	11.6	11.0	9.9	9.6	7.5	6.3
				80	15.09	13.9	13.9	13.1	11.8	11.4	9.0	7.5
				100	18.26	16.9	16.9	16.0	14.4	14.0	11.0	9.1
				120	21.44	20.0	20.0	18.9	17.0	16.5	13.0	10.8
			XXS	140	25.4	24.0	24.0	22.7	20.4	19.8	15.6	13.0
				160	28.58	27.3	27.3	25.8	23.2	22.5	17.7	14.7
300	12	323.9		20	6.35	4.8	4.8	4.5	4.1	4.0	3.1	2.6
				30	8.38	6.4	6.4	6.0	5.4	5.2	4.1	3.4
			STD		9.53	7.2	7.2	6.9	6.2	6.0	4.7	3.9
				40	10.31	7.9	7.9	7.4	6.7	6.5	5.1	4.2
			XS		12.7	9.7	9.7	9.2	8.3	8.0	6.3	5.3
				60	14.27	11.0	11.0	10.4	9.3	9.0	7.1	5.9
				80	17.48	13.5	13.5	12.8	11.5	11.2	8.8	7.3
				100	21.44	16.8	16.8	15.8	14.2	13.8	10.9	9.0
			XXS	120	25.4	20.0	20.0	18.9	17.0	16.5	13.0	10.8
				140	28.58	22.7	22.7	21.4	19.3	18.7	14.8	12.3
				160	33.32	26.8	26.8	25.3	22.7	22.1	17.4	14.4
350	14	355.6		10	6.35	4.4	4.4	4.1	3.7	3.6	2.8	2.4
				20	7.92	5.5	5.5	5.2	4.6	4.5	3.5	2.9
			STD	30	9.53	6.6	6.6	6.2	5.6	5.4	4.3	3.6
				40	11.13	7.7	7.7	7.3	6.6	6.4	5.0	4.2
			XS		12.7	8.8	8.8	8.4	7.5	7.3	5.7	4.8
				60	15.09	10.6	10.6	10.0	9.0	8.7	6.9	5.7
				80	19.05	13.4	13.4	12.7	11.4	11.1	8.7	7.3
				100	23.83	17.0	17.0	16.0	14.4	14.0	11.0	9.2
				120	27.79	20.0	20.0	18.9	17.0	16.5	13.0	10.8
				140	31.75	23.0	23.0	21.7	19.5	19.0	14.9	12.4
				160	35.71	26.1	26.1	24.6	22.2	21.5	16.9	14.1
400	16	406.4		10	6.35	3.8	3.8	3.6	3.2	3.1	2.5	2.1
				20	7.92	4.8	4.8	4.5	4.1	3.9	3.1	2.6
			STD	30	9.53	5.8	5.8	5.4	4.9	4.7	3.7	3.1
			XS	40	12.7	7.7	7.7	7.3	6.6	6.4	5.0	4.2
				60	16.66	10.2	10.2	9.6	8.7	8.4	6.6	5.5
				80	21.44	13.2	13.2	12.5	11.2	10.9	8.6	7.1
				100	26.19	16.3	16.3	15.4	13.8	13.4	10.6	8.8
				120	30.96	19.4	19.4	18.4	16.5	16.0	12.6	10.5
				140	36.53	23.1	23.1	21.9	19.7	19.1	15.0	12.5
				160	40.49	25.8	25.8	24.4	22.0	21.3	16.8	14.0