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Evaluation and Selection of the Precooling Stage for LNG Processes

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Evaluation and selection of the pre-cooling stage for LNG processes

Evaluering av pre-kjøling for LNG prosesser

Background and objective.

Natural Gas liquefaction technologies are of great importance nowadays, where day after day the gas demand is increasing and the piping systems are not flexible to fit the required capacity and destinations. Nowadays, about 30 LNG plants are in operation and more than 16 new projects are being developed around the world.

LNG technologies involve different selection of equipment (i.e. heat exchangers, compressors, etc) and multiple process definitions (i.e. type of refrigerant, pressure levels, temperature approach, etc). Only the liquefaction process can represent between 40 and 57% of the total investment of a LNG value chain, and the major costs in this area are related to compressors/drivers and heat exchangers. Therefore for any new plant, selection of the correct liquefaction technology and associated equipment is very influential in reducing cost and increasing project viability.

During the last decade, an important amount of work has been focused on the design of LNG processes. Most of the developed liquefaction processes include a first stage that is well known as precooling stage, where the natural gas is cooled down to a temperature that depending on the precooling technology varies from -30 to -50 °C. In LNG processes the refrigerants are based on light hydrocarbons and can be divided on two groups: pure and mixed refrigerants. The advantages and disadvantages of considering a mixed refrigerant or pure refrigerant cycle in the precooling stage are not well understood.

The main objective of this project work will be to analyze and study the selection of the precooling stage in LNG technologies from the conceptual design. A comparison between the refrigerant types at the precooling stage will be also considered in order to determine the advantages or/and disadvantages of the different configurations.

The following questions should be considered in the project work:

1. Review the state of the art of LNG technologies focused on efficiencies, capacities and costs.
2. Define a case study considering a particular escenario for evaluating LNG technologies.
3. Evaluation of the traditional LNG technologies in a specific case and analizying the effect of a new configuration and refrigerant type on the precooling stage.

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Department of Energy and Process Engineering, 27th January 2012

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Abstract

As the worldwide energy consumption continues to grow, natural gas and especially LNG are expected to keep contributing significantly with this growth. More than 95% of the installed LNG facilities use a precooling cycle as the first stage of the liquefaction process. In this work, a technical comparison between different precooling cycles for LNG processes is carried out through computational simulations using Aspen HYSYS®. The aim is to provide future project developments with a clear idea of the technical advantages/disadvantages involved in the selection of the process for the precooling cycle in LNG processes.

The precooling circuit is treated as a stand-alone cycle first and then implemented in an entire liquefaction process; the propane precooled mixed refrigerant (C_3MR) and the mixed fluid cascade (MFC®) processes are used for this purpose. The parameters studied are essentially coefficient of performance (β), heat exchanger UA value, compressor power, suction volumetric flow and pressure ratio. Two cases, cold (6 °C) and warm (25 °C) climate conditions are considered for each study.

A three stage propane precooled process was found to be the most energetically efficient among the studied cases, even better than a two stage mixed refrigerant process (C_2/C_3) for both climate conditions; however, the performance in terms of energy consumption is not the only parameter taken into account and therefore a selection chart is provided. Under warm climate conditions a propane precooling circuit showed to be the most recommended process. For cold climates, however, a two stage mixed refrigerant cycle reaching ca. -50 °C is the preferred alternative, since in this case the low ambient temperature gives the propane precooled process a low share in the entire cycle. Other cases, such as a single stage mixed refrigerant cycle and a mixed refrigerant including n-Butane are taken into account.

Based on the obtained results, a new, highly efficient configuration for natural gas liquefaction has been suggested, it is to be implemented in relatively warm climate conditions. It consists of a MFC® process with modifications in the liquefaction cycle and a propane precooling instead of the mixed refrigerant circuit; no previous reference in the open literature was found for such arrangement.

Sammendrag

Ettersom verdens energiforbruk fortsetter å vokse, forventes det at naturgass og spesielt LNG bidrar betydelig til denne vekset. Mer enn 95% av de installerte LNG-anleggene bruker en precooling-syklus som den første fasen av prosessen. I dette arbeidet har en teknisk sammenligning mellom ulike precooling-sykluser for LNG prosesser blitt utført gjennom beregningsorientert simuleringer med Aspen HYSYS ®. Målet er å gi fremtidige prosjekter en klar idé om de tekniske fordelene/ulempene involvert i valg av prosessen for precooling syklus.

Precooling-kretsen er først behandlet som en frittstående syklus og deretter implementert i en hel væskeomgjøringsprosess, C_3MR- og MFC ®-prosessene ble brukt til dette formålet. Parametrene studert er i hovedsak ytelseskoeffisienten (β), varmevekslerens UA verdi, kompressorarbeid, volumetrisk inntaksstrømning og trykkforhold. To tilfeller, kalde (11 °C) og varme (30 °C) klimaforhold undersøkes for hver studie.

En tretrinns propan precooled-prosess var den mest energieffektive blant de studerte tilfellene, og viste seg å være bedre enn en totrinns blandet kjølemediumprosess (C_2/C_3) for begge klimaforhold. Men ytelsen i forhold til energiforbruk er ikke eneste parameter tatt hensyn til, derfor har et valg-diagram blitt laget. Blandet kjølemiddel precooling-syklusen er det foretrukne alternativ under kalde klimaforhold, på grunn av redusert andel som kan nås med en propan syklus temperatur begrensning. Andre tilfeller, for eksempel en ett stegs blandet kjølemedium-prosess og en blandet kjølemedium inkludert n-butan er også tatt hensyn til.

Basert på oppnådde resultater, har en ny og svært effektiv konfigurasjon for væskeomgjøring av naturgass blitt foreslått for relativt varme klimaforhold. Den består av en MFC ® prosess med modifikasjoner i LNG-syklusen, og propan-precooling istedenfor den blandete kjølekretsen, ingen tidligere referanse i den åpne litteraturen ble funnet for et slikt oppsett.

Preface

In the name of Allah, most Gracious, most Merciful

Five years of university studies are summarized in this final thesis, which has been written in only twenty weeks. This work has been carried out at the Department of Energy and Process Engineering in the Norwegian University of Science and Technology (NTNU); with the supervision of Prof. Carlos Dorao and Ph.D. student Luis Castillo, to whom I'm absolutely grateful for their time, advices and contributions to each step in the development of this thesis. I would like to thank also Prof. Jostein Pettersen from Statoil's Research Centre for useful advices and follow up when asked for.

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Nomenclature

A	Area of heat exchanger
APCI	Air Products and Chemicals, Inc.
AP-X	APCI's nitrogen expaned high capacity process
BTU	British Thermal Unit
C_2	Ethane
C_3	Propane
C_3MR	Propane precooled mixed refrigerant process
COP	Coefficient of performance
DMR	Dual mixed refrigerant process
FLNG	Floating liquefied natural gas
FPSO	Floating production, storage and offloading
H_P	Polytropic head
HX	Heat exchanger
k	Specific heat ratio
LMTD	Logarithmic mean temperature difference
LNG	Liquefied natural gas
LPG	Liquified petroleum gas
m	Mass flowrate
M	Molecular weight

MFC	Mixed Fluid Cascade
MMt/y	Million tonnes per year
MR	Mixed refrigerant
MTPA	Million tonnes per annum
n	Specific heat ratio
$n\text{-}C_4$	Normal butane
P	Pressure
PFHE	Plate fin heat exchanger
PPMRC	Propane precooled mixed refrigerant cascade
Qc	Heat transferred condenser
Qo	Heat transferred evaporator
R	Universal gas constant
RPM	Revolution per minute
S	Entropy
scf	Standard cubic feet
SMR	Single mixed refrigerant process
SWHE	Spiral wound heat exchanger
T	Temperature
Tc	Temperature of the cold side (absolute)
Th	Temperature of the warm side (absolute)
U	Overall heat transfer coefficient

W	Work
Wc	Work of the compressor
Wt	Work of the turbine
Z	Gas compressibility factor
β	Coefficient of performance
$\beta carnot$	Coefficient of performance for carnot cycle
ΔP	Pressure drop
$\Delta Tmin$	Minimum temperature difference
η_c	Isentropic efficiency compressor

Chapter 1. Introduction

From the early days of life, as each individual is born, the necessity to obtain energy in order to perform vital processes comes as a naturally given quality. However, as societies have developed, the energy role in quotidian life has increased significantly; nowadays energy is required to power homes, businesses, industries, transportation, and other daily life services. Driven by the earth's population growth, the worldwide demand for energy is increasing rapidly, and in the upcoming years it is expected to increase faster, especially due to the rapid developments of highly populated countries such as China and India [1].

In 2012 more than 85% of the worldwide primary energy consumption is being provided by fossil fuels, from which only the natural gas accounts for 24% [2]. Natural gas burns more cleanly than other fossil fuels, basically because it has less emissions of sulfur and carbon than, for instance, coal or oil; this is one of the reasons behind that the use of natural gas has grown so much and is expected to grow even more in the future [3].

Most of this natural gas is transported from the wellhead to a processing plant, and thereafter, to final consumers in gas transport pipelines. However, at remote locations or when the distance between the gas market and the source is long enough, liquefying the natural gas for transport has been widely implemented as a practical solution in the energy industry. Nowadays, more than 30% of the worldwide gas trading is done via liquefied natural gas (LNG) [4], and complex liquefaction processes are required in order to pass the gaseous natural gas to liquid.

The design of such liquefaction processes involves different selection of equipment (i.e. heat exchangers, compressors, etc.) and multiple process definitions (i.e. type of refrigerant, pressure levels, temperature differences, etc.). Only the liquefaction process represents between 40 and 57% of the total investment for the LNG value chain, and the major costs in this area are related to compressors/drivers and heat exchangers [5, 6]. Hence for any new plant development, selection of the appropriate liquefaction technology and associated equipment is very influential in reducing cost and increasing project feasibility.

Consequently, during the last decade an important amount of work has been focused on the design and optimization of LNG processes. Most of the developed liquefaction processes include a first stage that is well known as precooling stage where the natural gas is cooled down to a temperature that, depending on the precooling technology, varies from -30 to -50 ºC. One of the main differences between the precooling stages of the existing processes is the use of mixed or pure components as refrigeration fluid. More than 85% of the currently installed trains use pure component refrigerant, propane, in the precooling cycle [7]; however recently developed processes, such as the Dual Mixed Refrigerant and the Mixed Fluid Cascade process use mixed refrigerants for precooling.

The advantages of using a mixed or pure component refrigerant in the precooling stage are not well understood, basically because in most of the previous work about selection, thermal efficiency and energy consumption per mass unit of LNG (e.g. kWh/kg LNG) are the only benchmarks used to compare the different LNG technologies without mentioning the conditions of the judgment, such evaluations were made among others by Finn (2009), Shukri (2004) and Ransbarger (2007) [8-10]. That kind of comparison can be misrepresentative because the design premises are not consistent from project to project. The efficiency of the refrigeration compressors, the ambient temperature of the region, the feed gas composition, temperature and pressure are some of the factors that may influence the process energy consumption.

The technology choice for a new LNG project may depend on different criteria; for instance the selection may be influenced by economic, environmental, financial, license or technical issues. Since most of the economic evaluation data (i.e.: equipment price, license fees, etc.) is treated as confidential, the scope of this work will be based on the technical comparison of the different precooling arrangements of the known liquefied natural gas processes, in order to explore the advantages that each configuration may offer to the process and a LNG project in general. A reliable comparison between the possible configurations will provide future projects with a clear idea of the differences, and hence will ease selecting the appropriate technology, from the technical approach.

A theoretical background comes first in order to introduce the reader into the subject to be treated; an introduction to natural gas, liquefaction processes and thermodynamic definitions is given. Next, in Chapter 3, the simulation cases studied in order to perform the evaluation are presented; main parameters considered relevant for the reader are shown for each simulated case. Finally Chapter 4 presents the results of the simulated configurations, together with a comprehensive analysis of their meaning for the purpose of this work. The last chapters include conclusions reached and recommendations for further work.

Chapter 2. Theoretical Background

While oil is a liquid and coal is a solid, natural gas is originally found as a gaseous fossil fuel that occurs in the porous rock of the earth's crust either alone (non-associated natural gas) or with accumulations of petroleum (associated natural gas). For the latter case, the gas can exist as a cap above the petroleum layer and (when the reservoir pressure is sufficiently high) dissolved in the oil [11]. It is a colorless, odorless complex mixture of hydrocarbons with a heating value (i.e., the amount of heat produced by the combustion of a given quantity of fuel) that ranges from 900 to over 1200 BTU (British Thermal Unit) per scf (standard cubic feet) [12].

Based on the type of gas (associated or non-associated) and the geographical ubication of the field, raw natural gas composition can vary widely. The primary component is methane (CH_4), but it also contains ethane (C_2H_6), propane (C_3H_8), butane (C_4H_{10}) and heavier hydrocarbons (C_{5+}); non-hydrocarbons such as carbon dioxide (CO_2), hydrogen sulfide (H_2S) and nitrogen (N_2) may be present as well. Figure 2-1 shows a chart with typical raw natural gas composition.

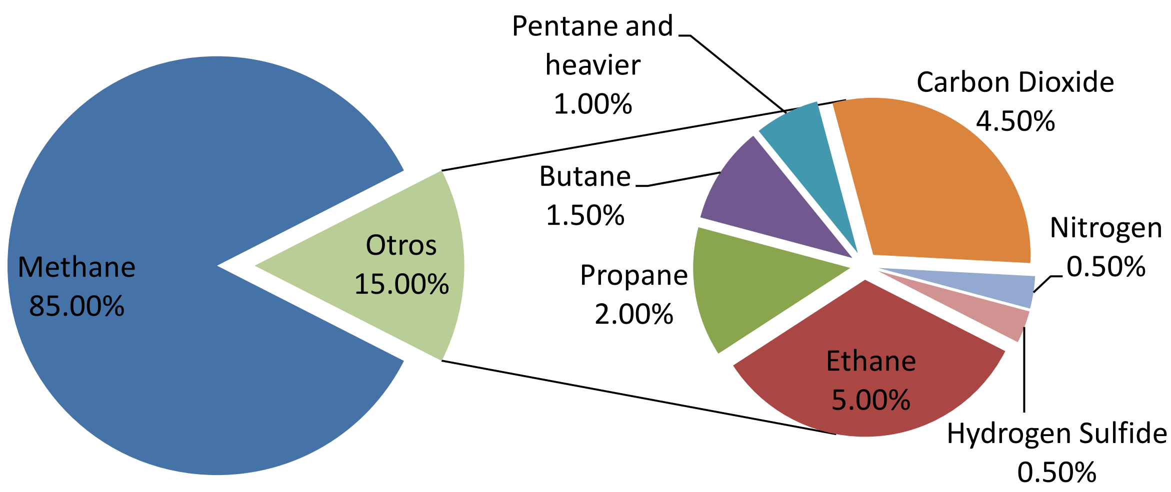

Figure 2-1. Typical natural gas composition [11]

In the worldwide energy industry natural gas plays a key role. As shown in Figure 2-2, global natural gas consumption is increasing as the total energy consumption year after year; by 2010 natural gas recorded its highest historical share in the energy consumption providing about 24% of the total, and it is expected to increase further more in the next years particularly due to electric power generation developments [13].

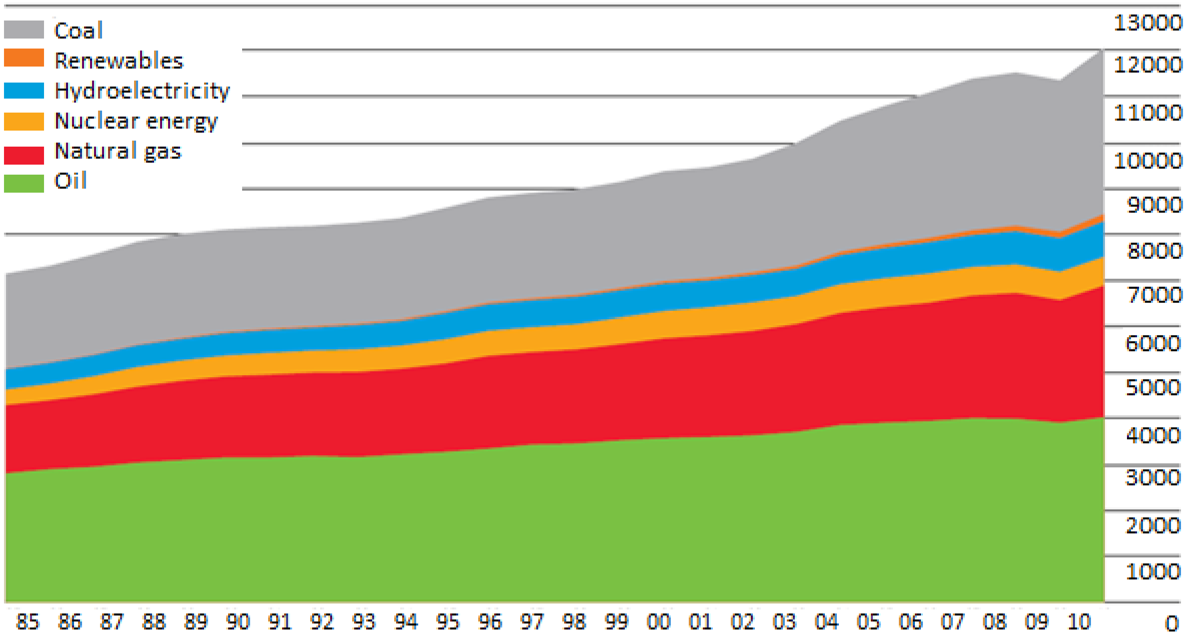

Figure 2-2. Historical world energy consumption (Million tonnes oil equivalent) [2]

Electric power generation is one of the recently growing applications of natural gas; it has become an attractive alternative fuel for new power generation plants because it offers low capital costs and favorable thermal efficiencies, with lower levels of potentially harmful byproducts that are released into the atmosphere (e.g.: Carbon Dioxide CO_2) [14]. Likewise natural gas is used extensively for heating in both residential and commercial sites, while for industrial purposes it is mainly used as process fuel and feedstock (petrochemical).

As a result of its increasing worldwide demand and undeniable environmental benefits compared to other fossil fuels, natural gas transport has become an important issue for the global energy supply. Most natural gas is transported from the wellhead to a processing plant, and thereafter, to final consumers in gas transport pipelines. However, at remote locations or when the distance between the gas market and the source is long enough, liquefying the natural gas for transport has been a major industrial operation [12].

2.1. Liquefied Natural Gas

Liquefied natural gas or usually referred to as LNG, is natural gas that has been processed and cooled down until condensation at atmospheric pressure (1 atm = 1,01325 bar). Since it is mainly composed by methane, natural gas bubble point temperature at atmospheric pressure lies around 104-110 K [14]; the bubble point temperature is defined as the state at a certain pressure in which the fluid is completely liquid and the first bubble of gas is formed. In comparison, one physical volume unit of LNG yields approximately 600 units of standard gas volume while it remains colorless, odorless, non-corrosive and non-toxic as in the gaseous phase.

This enormous reduction in physical volume of liquefied natural gas (LNG) relative to gaseous natural gas reduces transportation costs; it is indeed the cornerstone of the liquified natural gas business since the energy volumetric density increases (more energy per volumetric unit) allowing its long distance transport by ships across oceans to markets where pipelines are neither economic nor feasible [3].

Figure 2-3 shows natural gas transportation cost for different alternatives, it is evident that for long distances LNG becomes an economically feasible choice. Nevertheless, transport as LNG is a complex task which implies development of different components of what is so called the LNG value chain; this value chain from the gas field to the eventual consumer will be discussed in the next section.

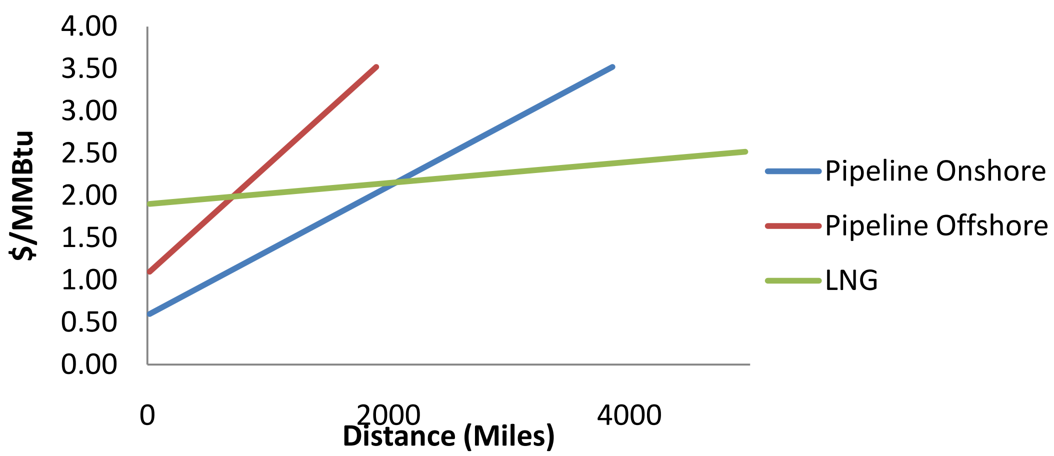

Figure 2-3. Natural gas transportation cost. [15]

An interesting concept under development is known as Floating LNG plants (FLNG) or also mentioned as "LNG FPSO" which stands for Floating Production, Storage and Offloading units for Liquified Natural Gas. It consists basically of ships with a LNG production plant on deck. The use of FLNG plants is expected to eliminate the transport of natural gas from well to plant (including transport of CO_2 back to the well for storage, when applicable). This technology has been developing in the late years and it is seen as one of the most promising solutions to monetize and exploit gas fields with long distance to shore or low amount of gas initially in place [16].

2.2. LNG Value Chain

When trying to bring gas reserves to market, its necessary to do it through a chain of separate but linked stages; for the liquefied natural gas sector these are basically: upstream gas production, liquefaction, shipping, and regasification [17]. Each of these components (shown in Figure 2-4) has its own set of technological challenges and investment criteria, but each is linked to the others in the sense that no one component is a viable business investment without the others. A brief explanation of each of these stages is given below.

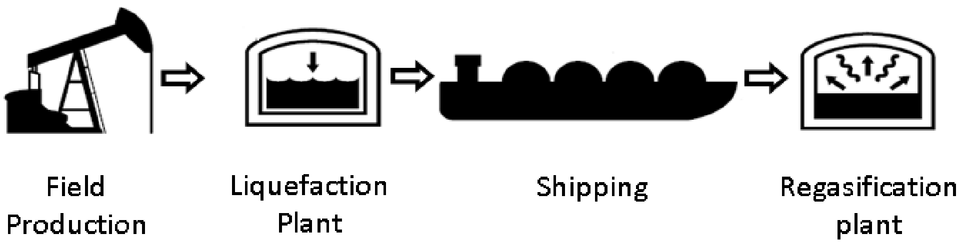

Figure 2-4. LNG value chain, main components. [14]

- Field production

In the early days, natural gas was often discovered as a less desirable byproduct of oil, but today's exploration is increasingly aimed at the discovery of exportable gas reserves. The field exploration and production for liquefied natural gas projects are identical to traditional gas fields, with identical gas wells, wellheads, and field processing facilities [17]. As mentioned before, natural gas may be found also in crude oil fields (associated natural gas).

Gas from a number of different fields may be combined prior to liquefaction. This stage may involve gas treatment to remove impurities or heavier hydrocarbons that can turn into liquid in the line between the field and the liquefaction plant.

- Liquefaction

After the gas leaves the upstream production facilities, it is metered and transported by pipeline to the liquefaction plant, this stage is the heart of any LNG project and it represents around 40-57 % of the LNG chain investment, depending on number of trains and location [18]. Before the gas can be liquefied, it must be treated to remove carbon dioxide, sulfur, mercury, heavy hydrocarbons and water, which can freeze or cause corrosion inside the heat exchangers [12]. Any heavier components removed in the plant (e.g., condensate and LPG) are shipped and sold separately, creating additional revenue for the project [14].

The liquefaction process is basically a complex refrigeration cycle (as will be explained later on) that consists of compressors (driven by steam or gas turbines, recently electrical motors), and heat exchangers, where heat from the incoming gas is transferred to the working fluid of the cycle, which in turn transfers heat to an outside coolant (air or water) [14]. There are a number of proprietary processes for natural gas liquefaction and even though each of the world's large baseload liquefaction plants is unique in design, they all perform a basic common task: first treating the gas to remove impurities and then liquefying it by cooling to around 104-110 K [19].

- Shipping

After liquefaction and storage, the LNG is loaded onto specially designed ships built around insulated cargo tanks. LNG ships historically were custom built for and dedicated to specific projects, sailing in regular service between the LNG supplier and one or more customers [20]. There are two basic types of cargo systems employed in the LNG fleet. Spherical and membrane tanks.

- Regasification

LNG cargoes are discharged at regasification terminals (also called receiving or import terminals) that are located in the overseas customer's country. A terminal consists of one or more docks, each with a set of unloading arms, LNG storage tanks, and vaporization equipment to move the regasified LNG into the pipeline system [21].

2.3. Refrigeration Thermodynamics

While heat in nature is transferred by itself from high to lower temperatures, the distinctive ability of a refrigeration cycle is that it can remove heat from an area with low temperature to one at higher temperature [22]. Conventional household refrigerators and air-conditioners do so, based on a vapour compression cycle. A vapour compression cycle, shown in Figure 2-5, consists mainly of four components in adition to the fluid pipes: compressor, condenser, expansion valve and evaporator; this type of cycles are the most common refrigeration systems in use nowadays [22].

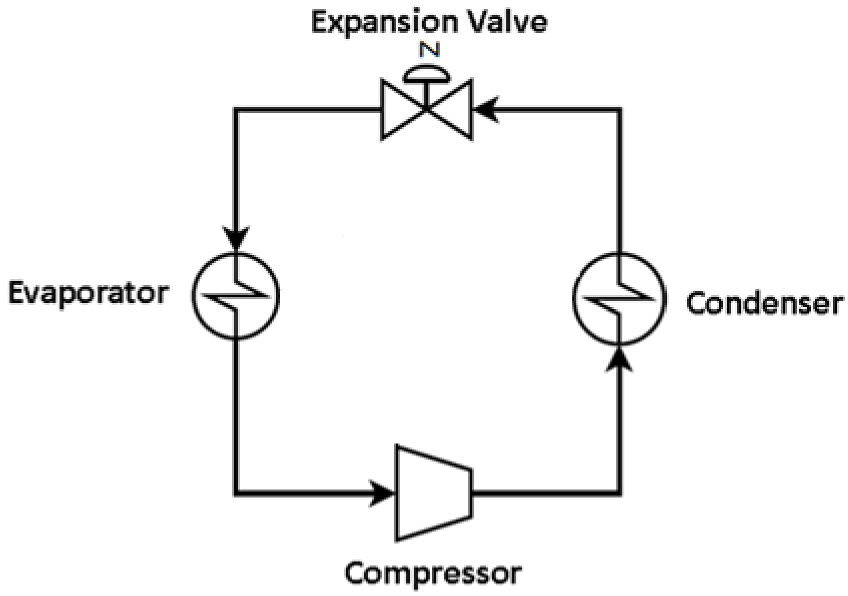

Figure 2-5. Basic refrigeration cycle, vapour compression cycle.

In order to introduce some important aspects related to refrigeration cycles, may be helpful to start with Carnot's cycle, since it is the ideal refrigeration process with the best possible efficiency [23]. A Carnot refrigeration cycle consists of 4 internally reversible processes, two adiabatic processes alternated with two isothermal processes. Quoting Moran and Shapiro [22] will ease the understanding of the term "reversibility":

"A process is called ***irreversible*** *if the system and all parts of its surroundings cannot be exactly restored to their respective initial states after the process has occurred. A process is* ***reversible*** *if both the system and surroundings can be returned to their initial state"*

Mathematically, the quoted statement is represented by Equation (2.1), where the equality applies when there are no internal irreversibilities as the system executes the cycle and the inequality when internal irreversibilities do exist. In this equation ∂Q represents the heat transfer at part of the system boundary (subscript "b") and T is given by the absolute temperature at that part of the boundary.

$$\oint \left(\frac{\partial Q}{T}\right)_b \leq 0 \tag{2.1}$$

By performing some analysis (may refer to [22]) it is possible to conclude that the value of this integral depends only of the end states, hence it represents the change in a system property, which is widely known as entropy and is represented by the symbol S. Equation (2.2) denotes the definition of entropy change in a differential basis.

$$dS = \left(\frac{\partial Q}{T}\right)_{\substack{int\\rev}} \tag{2.2}$$

Once given the definition of entropy it is possible to introduce the Carnot refrigeration cycle in detail, see Figure 2-6. W_T and W_C represent turbine and compressor work respectively, while Qo and Qc denote the heat transferred in the evaporator and condenser respectively. As mentioned before, the adiabatic processes are compression (1-2) and expansion (3-4), while the isothermal processes are condensation (2-3) and evaporation (4-1) of the working fluid.

Since the Carnot refrigeration cycle is made up of internally reversible processes, Equation (2.2) may be used to determine the amount of heat transferred in either the condenser or the evaporator; it also can be easily noticed that both compression and expansion represent what is so called an "isentropic" process (adiabatic and reversible), which means that the entropy of the system remains constant during the process execution.

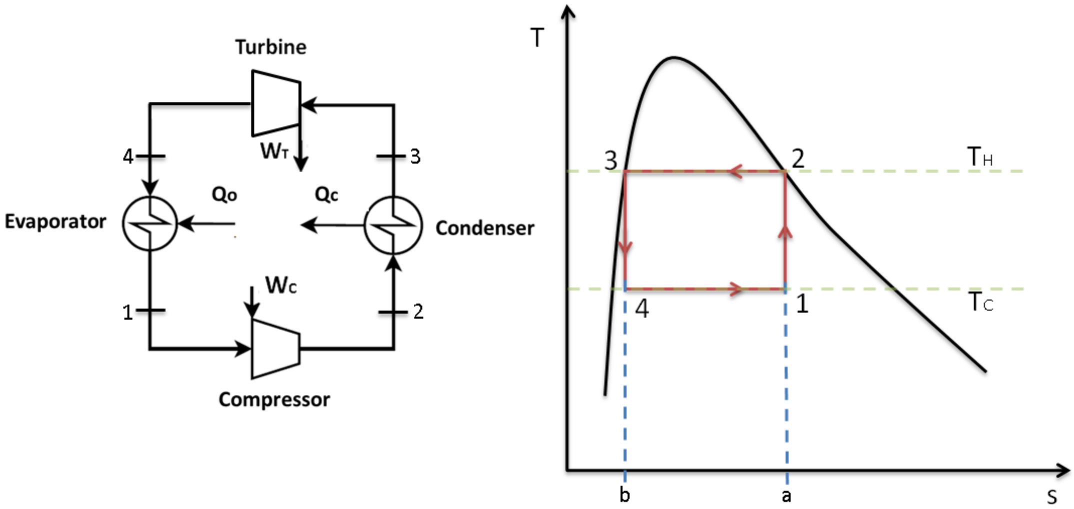

Figure 2-6. Carnot refrigeration cycle

Figure 2-7 shows the representation of the heat transfer calculated from Equation (2.2); while the total work of the cycle is obtained by introducing an energy balance derived from the first law of thermodynamics (Equation (2.3)), based on the fact the system is returned to its initial state (closed cycle) [22].

$$0 = \partial Q - \partial W \tag{2.3}$$

The net heat transfer that takes place during the cycle equals the net work done on the system. Where the net heat transfer is the difference between the heat rejected (Qc) and the heat added to the system (Qo); while the net work represents the difference between the compressor work (Wc) and the turbine work (W_T).

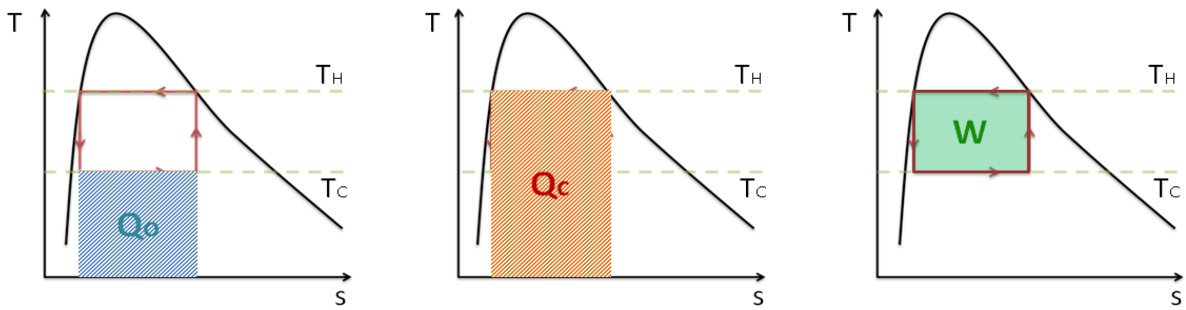

Figure 2-7. Heat tranfer represented in T-s diagram, Carnot cycle.

Thermodynamically, the performance of refrigeration cycles can be described as the ratio of the amount of energy added to the system (known as "refrigeration effect") to the net work input employed to achieve this effect, see Equation (2.4) [22]. This relation is well known as coefficient of performance (COP), and in this work will be represented by the Greek letter β.

$$\beta = \frac{Qo}{W} \tag{2.4}$$

For the Carnot cycle shown in Figure 2-6 the coefficient of performance can be computed by finding the areas represented in Figure 2-7 as Qo and W, which leads to Equation (2.5). This equation corresponds to the maximum theoretical coefficient of performance of any refrigeration cycle operating between regions at constant temperatures Tc and T_H [23].

$$\beta_{Carnot} = \frac{Qo}{W} = \frac{Tc.(s_a - s_b)}{(T_H - T_C).(s_a - s_b)} = \frac{Tc}{(T_H - T_C)} \tag{2.5}$$

Carnot's coefficient of performance (β_{Carnot}) represents the maximum theoretical β that could be obtained since reversible processes are not possible in reality [22]. One of the most remarkable differences between the Carnot refrigeration cycle and a practical applicable one is the heat transfer between the system fluid and both the cold (T_C) and hot region (T_H).

To understand this difference an extension of Newton's law of cooling must be introduced, and it's represented in Equations (2.6) and (2.7). These equations are extensively used to perform heat exchanger analysis, where Q denotes the heat transfer rate through the exchanger (evaporator or condenser), U represents the overall heat transfer coefficient, A is the surface area for heat transfer, and ΔT_{lm} represents the logarithmic mean temperature difference (also known as LMTD), in which ΔT_x and $\Delta T_{x+\Delta x}$ is the temperature difference between the interacting media at two arbitrary (but different) physical locations across the heat exchanger [24].

$$Q = U.A.\Delta T_{lm} \tag{2.6}$$

$$\Delta T_{lm} = \frac{\Delta T_x - \Delta T_{x+\Delta x}}{\ln \frac{\Delta T_x}{\Delta T_{x+\Delta x}}} \tag{2.7}$$

Based on equations (2.6) and (2.7) it's easily concluded that to achieve a rate of heat transfer in any real heat exchanger (HX), a temperature difference between the regions (T_H /T_C) and the system fluid (condenser/evaporator) is required. This limitation leads to a reduction in the coefficient of performance of the cycle; see Figure 2-8 where the pink shaded area represents the cycle (for the same refrigeration effect) in which the required temperature difference is taken into account. It is important to note that the mentioned cycle approaches the ideal (maximum COP) as the temperature difference approaches zero.

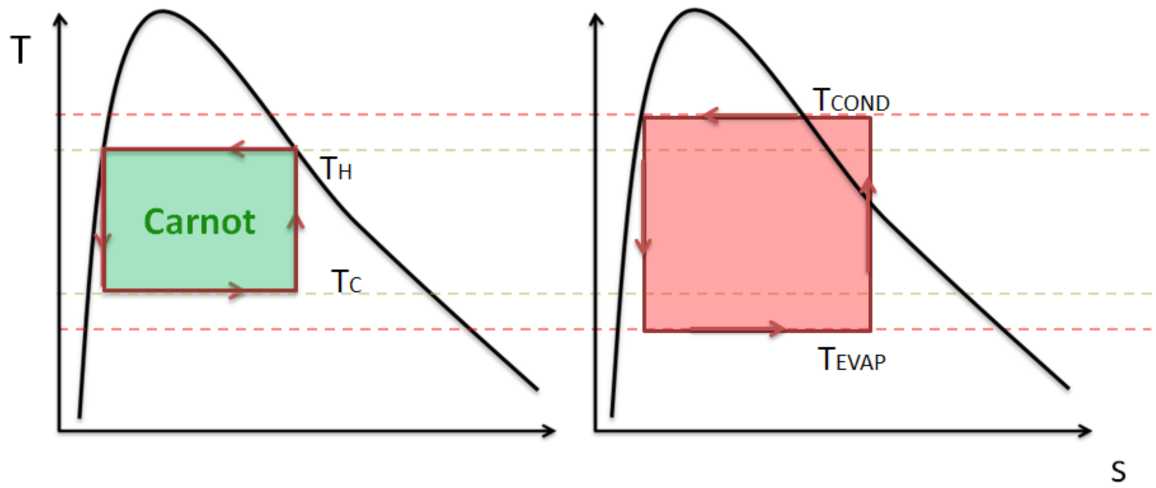

Figure 2-8. Cycle non-reversibilities, heat transfer.

Connected to the previous conclusion and mainly to equation (2.6) is also the relation between the area for heat transfer and the temperature difference in the HX, this relation defines a very important issue for the design stage since reducing the temperature difference (commonly known as temperature approach), doesn't lead only to an increase in the coefficient of performance, but also means an increase in the surface area of the heat exchanger. So as the area is increased, operational costs are reduced (the maximum achievable COP is approached), but capital costs increase (HX size), indicating that there is an economic optimum [25].

In addition to the heat transfer phenomena, there are two noteworthy features of Carnot's refrigeration cycle that make it impractical for real application [22]. First, the working fluid in the compression process (may refer to Figure 2-6, process 1-2) is a liquid-vapour mixture, which is generally avoided since the presence of liquid droplets can damage the compressor [26]. In actual systems the cycle is designed so the compressor has to handle only gaseous phase fluid.

On the other hand, the expansion process in the turbine (3-4) has to handle multiphase flow also, which has to be avoided as in the case of the compressor. In the expansion process the work produced by the turbine is relatively low compared to the required for the compressor, so the turbine is normally substituted by a simple throttling valve, which reduces the initial and maintenance expenses; the resulting cycle will be the vapour-compression cycle introduced at the beginning of this section (Figure 2-5).

Figure 2-9 illustrates the behavior exhibited by an actual vapour-compression refrigeration cycle. As shown in the figure, the heat tranfer irreversibilities are taken into account (temperature difference in HX). Irreversibilities are introduced also in the compression process (1-2), which is represented by the dashed line. One may compare cycle 1-2-3-4-1 and cycle 1-2s-3-4-1 in order to visualize the effect of irreversible compression, which is usually accounted for by using the isentropic compression efficiency (η_C) given by equation (2.8). It's also important to notice that the system fluid may be superheated at the outlet of the evaporator (state 1 in the figure below) and subcooled downstream the condenser (state 3).

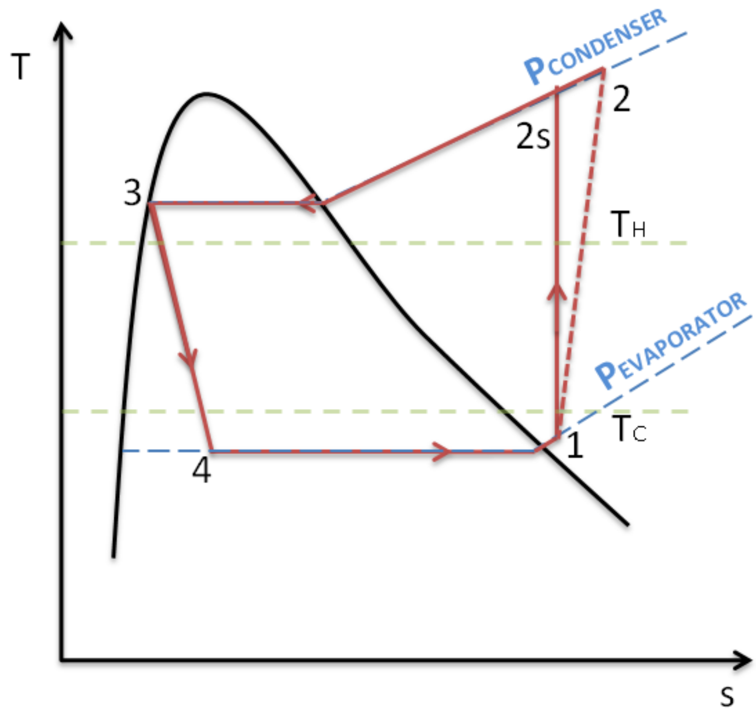

Figure 2-9. Temperature vs entropy diagram, actual vapour-compression cycle.

$$\eta_C = \frac{\left(\frac{\dot{W}_C}{\dot{m}}\right)_s}{\left(\frac{\dot{W}_C}{\dot{m}}\right)} \tag{2.8}$$

In order to reduce the effect of irreversible or non-isentropic compression a multiple stage compression system with intercooling might be used. This idea is shown in Figure 2-10 and is done basically by compressing the gas to a certain intermediate pressure, cooling the fluid by means of an intercooler and compressing again to meet the final required pressure. The outlined area in the figure represents the specific work reduction that can be achieved by means of a multistage compression with intercooling. Care should be taken since the cooling process has to be carried out in the superheated region of the phase envelope, crossing the two phase region damages the compressor because the inlet of the second stage will contain liquid.

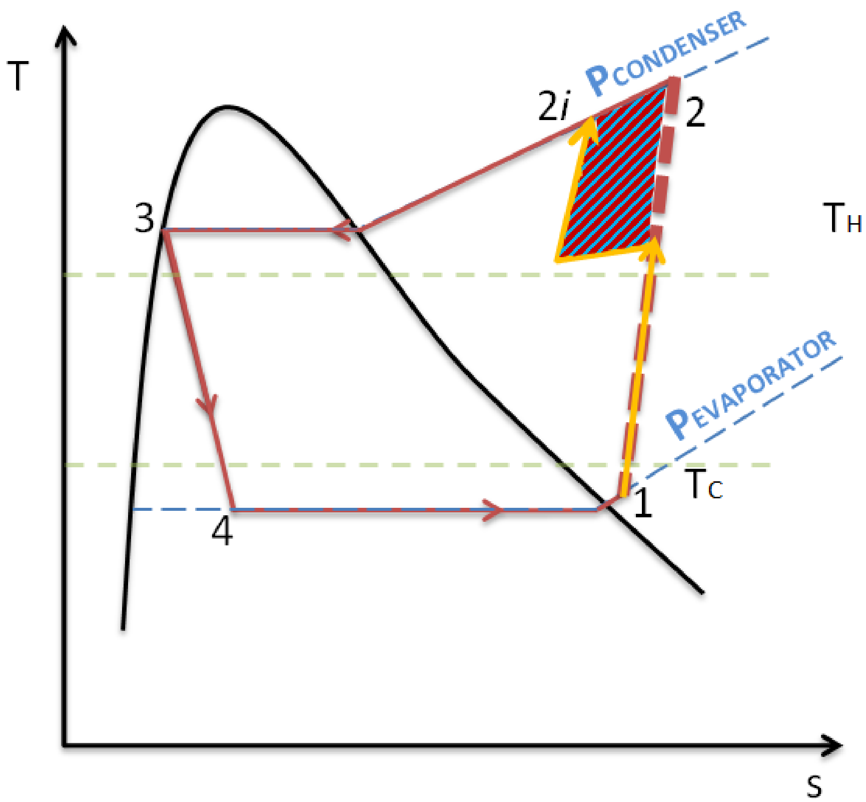

Figure 2-10. Temperature vs entropy diagram, multistage compression.

2.4. Refrigerants and Configurations

In the cycles presented above, a fluid was recurrently mentioned as the "working fluid" of the system, undergoing heat transfer, compression and expansion processes. The working fluids used in refrigeration cycles are commonly known as refrigerants. Each refrigerant has its particular properties and the selection of the appropriate refrigerant for each application is of great importance in the design of any cycle.

Natural gas liquefaction involves the use of mainly non-halogenated hydrocarbons such as methane (CH_4), ethane (C_2H_6), propane (C_3H_8), n-butane (n-C_4H_{10}) and ethylene (C_2H_4); whilst other refrigerants like carbon dioxide (CO_2) and nitrogen (N_2) are used less frequently. As has been shown in the previous section, the required temperatures of the refrigerant in the evaporator and the condenser are mainly determined by the temperatures of the cold and warm sides, respectively (may refer to Figure 2-8). In the case of refrigerants made out of a single component, these temperatures will set a specific value for the high pressure (condenser) and low pressure sides of the cycle (evaporator). Figure 2-11 shows the saturation pressure against temperature for the most commonly used LNG process refrigerants.

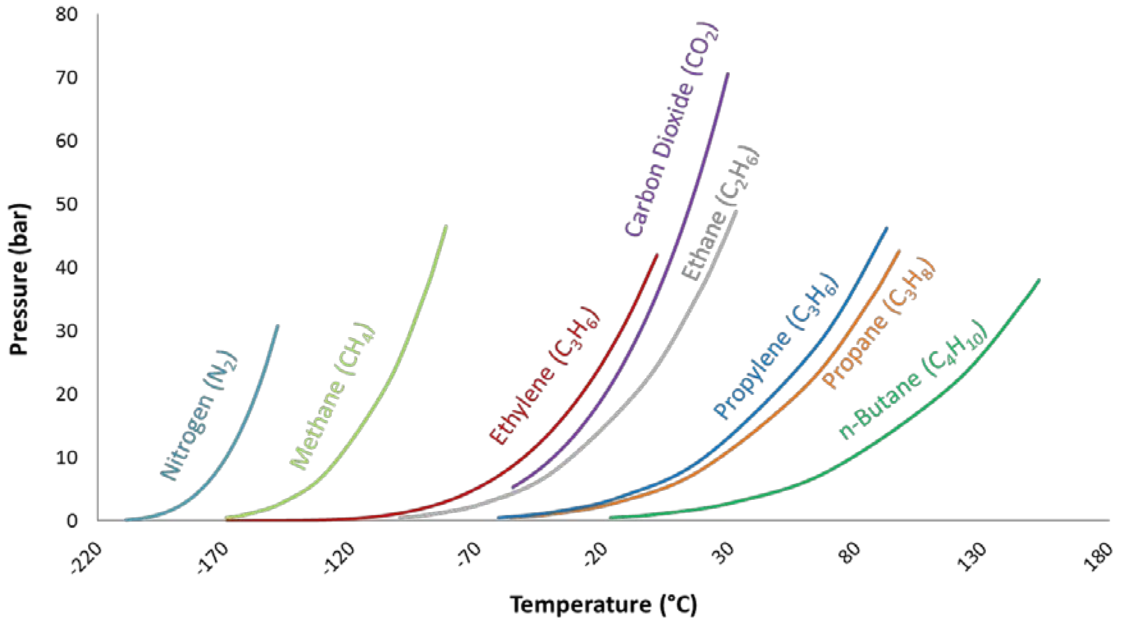

Figure 2-11. Saturation pressure related to temperature, LNG main refrigerants.

From the figure above, only methane (at pressures lower than 2 bar) and nitrogen seem to have the capacity to cover the entire LNG liquefaction process by means of a simple vapour compression cycle (Figure 2-5), which for LNG has to be carried out around the temperature range between -165 °C and 30 °C. Such alternative, for instance with methane (at 1,13 bar) is illustrated in a T-s diagram, may see Figure 2-12. Two important ideas shall be derived from the figure below. First, the condensation of the methane is carried out at supercritical conditions, which means that the process is actually a dense phase cooling. And the most important conclusion is that the power consumption of a natural gas liquefaction process with barely methane is far from the minimum possible (ideal), which implies a relatively small efficiency; this is shown with more detail in Figure 2-13 without accounting for compressor efficiency and expansion irreversibilities.

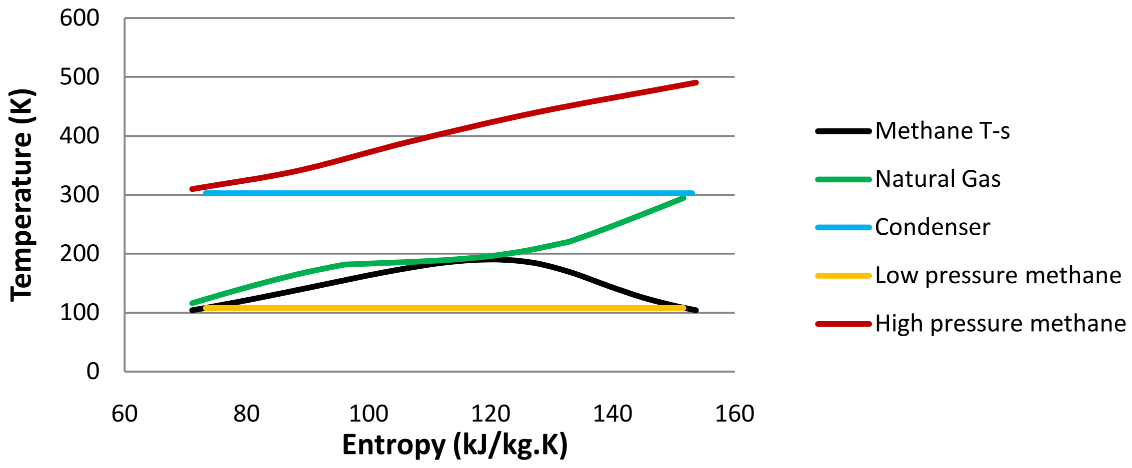

Figure 2-12. Methane T-s diagram, LNG liquefaction

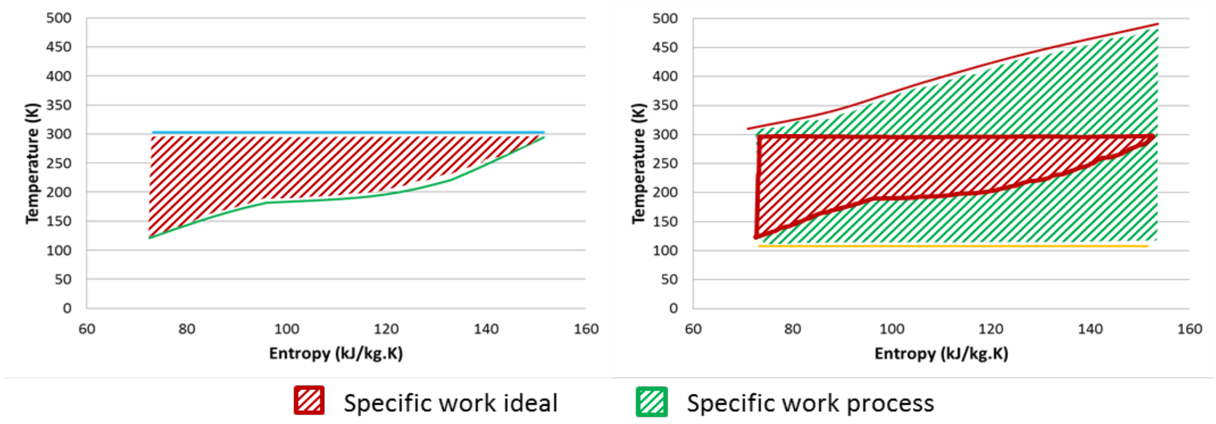

Figure 2-13. Specific work, simple vapour compression cycle for wide temperature cooling duty.

In this case, as in any other where the cooling task has to be performed over a wide temperature range, a simple refrigeration cycle does not complete the process efficiently, in other words the coefficient of performance (β) is not high enough; thus different solutions have to be considered. Two very well-known and widely applied solutions [27] are going to be discussed in this work; multilevel refrigeration and mixed refrigerant processes.

The principle of multilevel refrigeration is that the process is carried out at different pressure levels. A multilevel refrigeration might be performed using the same refrigerant for each pressure level, which is known as a multistage cycle; or using a different refrigerant for each cycle, such process is called a cascade cycle.

Whether the process is performed by one refrigerant or different ones for each stage, the idea is basically as depicted in Figure 2-14. Notice that the figures below (2.14-16) are illustrative and do not represent real values.

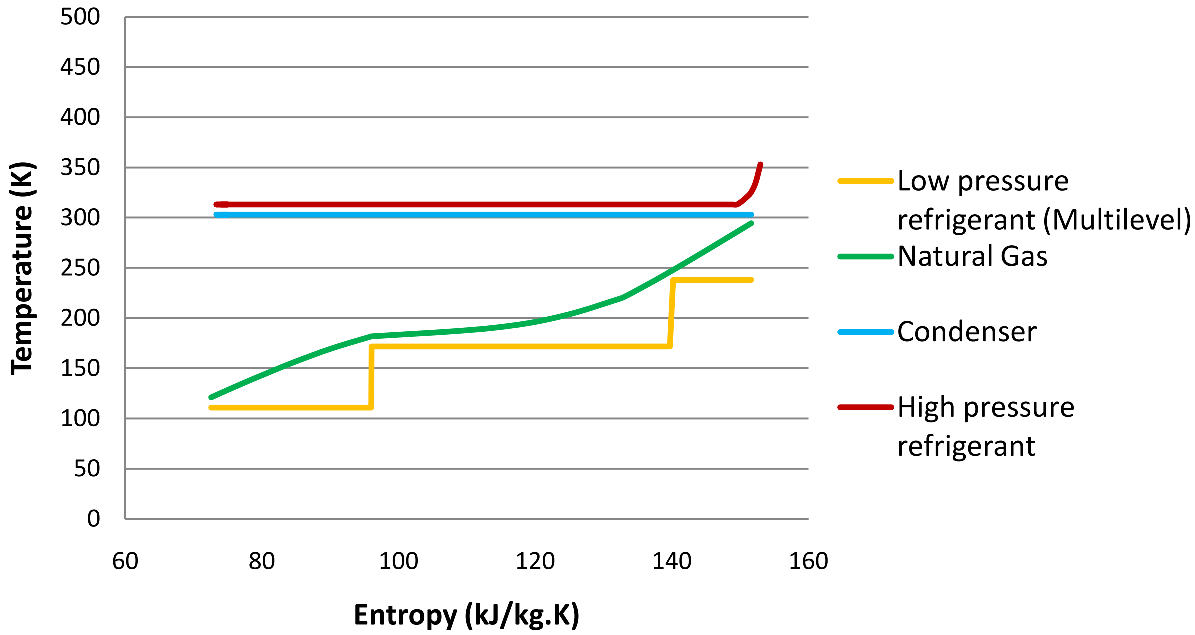

Figure 2-14. Multilevel refrigeration process, T-s diagram

When the process is a multistage cycle, a so-called multistage compressor is frequently used; these compressors are developed with one casing and several flow intakes in order to compress the same fluid from different intake pressures to a shared outlet pressure. On the other hand, if the process is a cascade cycle, the different refrigerants should not be mixed and therefore a compressor for each cycle will be required. In a cascade cycle each cycle can be designed as a multistage cycle, which means that for each cycle a different multistage compressor will be required.

The other alternative for achieving an efficient process is to use a mixed refrigerant as working fluid. Mixed refrigerants, unlike single component fluids, go through isobaric phase change processes at gliding temperature, delimited by the dew point and bubble point temperature of the mixture. A comparison of the phase change at constant pressure between a single component and a mixed component refrigerant is shown in Figure 2-15; the illustration shows also the dew and bubble point location for a mixed refrigerant at a certain pressure.

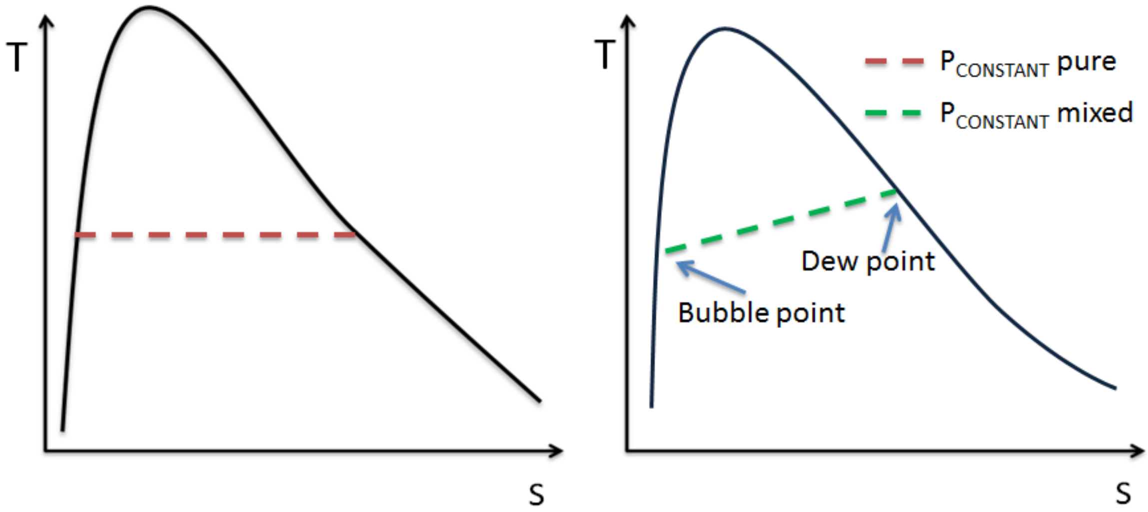

Figure 2-15. Phase change at constant pressure, propane and mixed refrigerant

Provided the right choice of compositions and pressures, a mixed refrigerant cycle can perform a natural gas liquefaction process as depicted in Figure 2-16. A very useful variation of the vapour compression cycle when using mixed refrigerants is the heat exchanger arrangement shown in Figure 2-17. It is a widely known configuration [27] where the compressed fluid exchanges heat with the expanded side in addition to the exchange with the external coolant in the condenser. Thus, the refrigerant is cooled down before being expanded; which means that a lower temperature is reached after the expansion device; the expense for such a benefit is an increase in the heat transfer area required for the process.

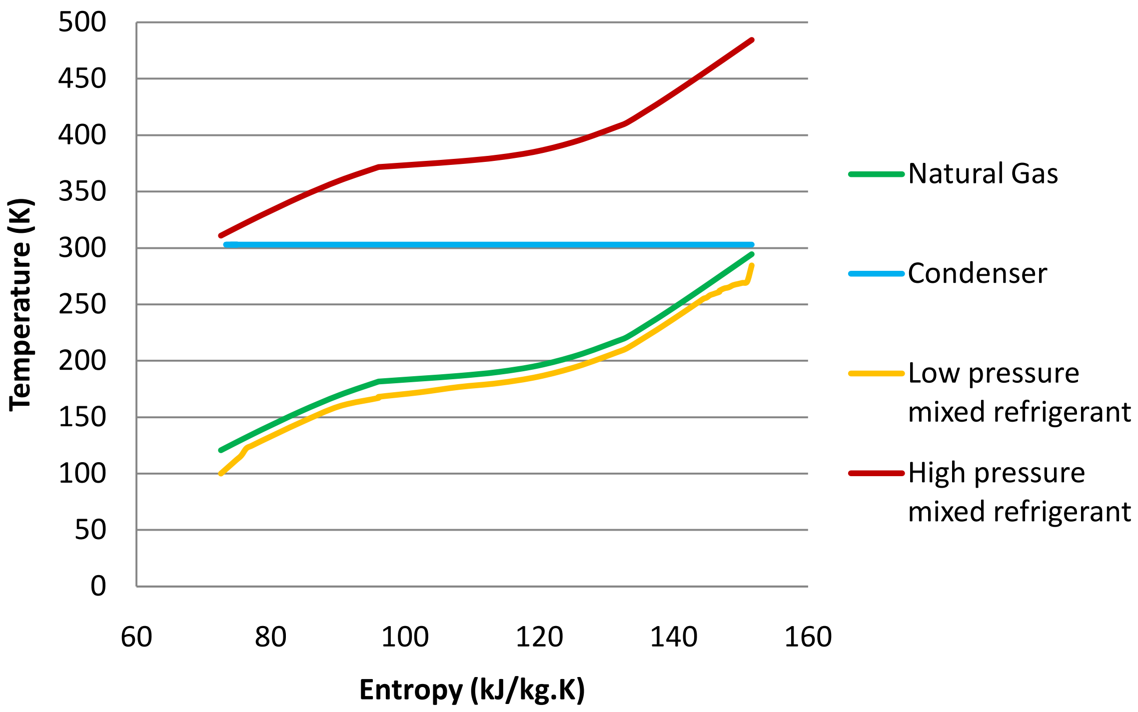

Figure 2-16. Mixed refrigerant refrigeration process, T-s diagram.

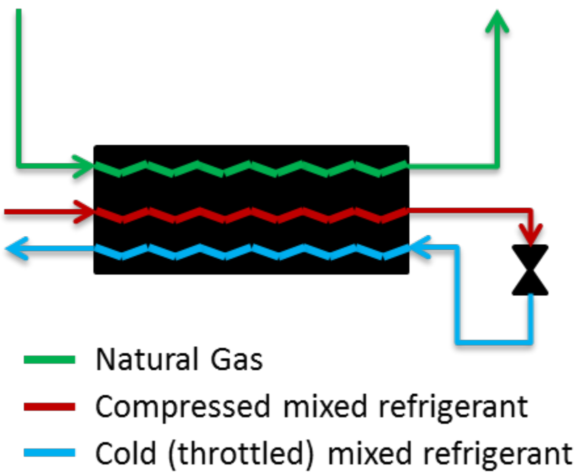

Figure 2-17. Heat exchanger arrangement for mixed refrigerant processes.

Based on the principles of refrigeration given above, the next section of this chapter will introduce briefly some of the most important natural gas liquefaction processes, with special focus on the processes to be studied in this work.

2.5. Natural Gas Liquefaction Processes.

In the early 1960's the first large-scale LNG plant started operation in Arzew, Algeria. The developed liquefaction process for that plant was a cascade cycle, using three refrigerants: methane, ethylene and propane proposed by Technip/Pritchard [7]. Since that first step, LNG production has grown significantly; and for each new project, recently engineered solutions have been offered by the leading licensors in order to achieve more energy efficient and economically rentable plants.

A natural gas liquefaction plant often consists of a number of parallel units, called trains, which can be considered as a standalone liquefaction cycle; this means that one process train can be shut down without affecting operations at adjacent trains. The use of multiple trains (2 or more) is due to the lack of capacity that a single train can offer. The capacity of a liquefaction train is primarily determined by the liquefaction process, the available size of the compressor and its driver, and the heat exchangers of the process. Figure 2-18 shows a historical train size development (in operation) with the detail of the installed technology proprietor; MMt/y indicates million tonnes per year or Megatonnes per annum (MTPA).

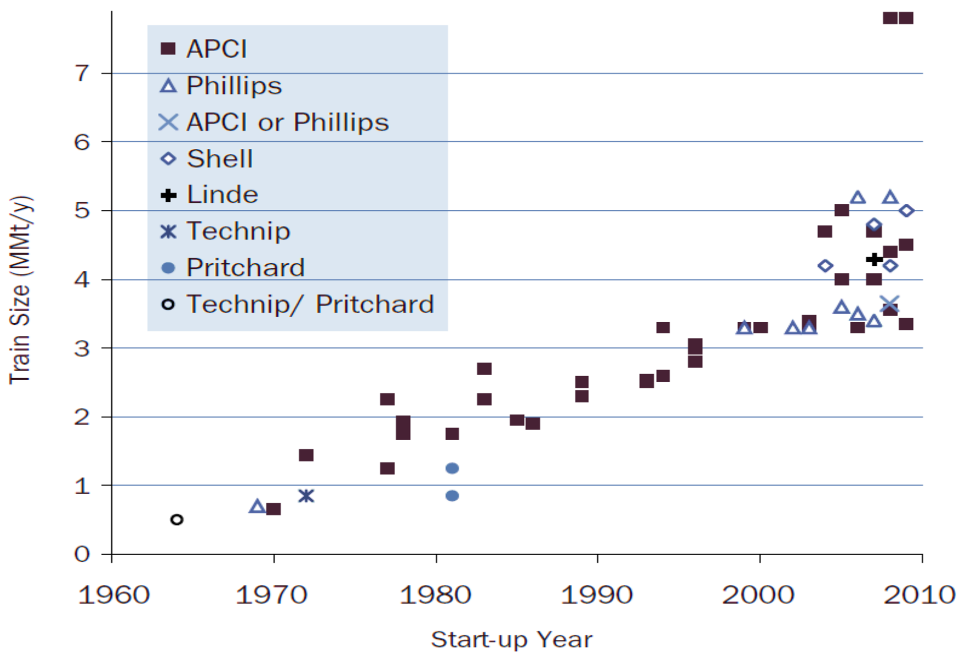

Figure 2-18. LNG train size growth and technology proprietary [28].

Whilst the single train capacity is an important factor due to the economies of scale, it is expected to reach a top value since it has to be compatible with the size of the gas field, upstream technology and LNG market needs [7]. Current technology innovation focuses therefore also on providing better efficiency, fuel economy and low emission processes.

It is important to note that recently small scale natural gas liquefaction processes are being developed in parallel to the large LNG trains; this is basically due to the desire to satisfy niche markets and exploit stranded gas reserves. The idea of a LNG FPSO or floating LNG is also providing the researchers with new challenges in this area, especially due to the reduced plot area and operation instabilities that may be present [29].

As seen in Figure 2-18, during the first decades, liquefaction process selection was homogenous, Air Products and Chemicals, Inc. (APCI) was the dominant choice and is still the leading licensor as shown in Figure 2-18; nevertheless in the last ten to fifteen years a considerable diversification has been the trend, licensors such as ConocoPhillips (previously Phillips), Shell and Linde/Statoil joined their technologies to the worldwide capacity.

The main natural gas liquefaction processes can be broadly classified into two groups based on the liquefaction process used, as described in Figure 2-19. Within the cascade processes, an example of those that use a single component refrigerant is the ConocoPhillips Optimized Cascade®. A cascade process using mixed refrigerants is the well-known Mixed Fluid Cascade (MFC®), developed by the Statoil and Linde LNG Technology alliance [30]. On the other hand, a mixed refrigerant process without precooling is one of the simplest processes available, patented by Black and Veatch, the Poly Refrigerant Integrated Cycle Operation (PRICO®) [31]. Finally, the mixed refrigerant processes with precooling can use a single component refrigerant in the precooling, such as the propane precooled, mixed refrigerant process (C_3MR) by APCI; whilst an example of a mixed refrigerant precooling is the Shell double mixed refrigerant (DMR) process.

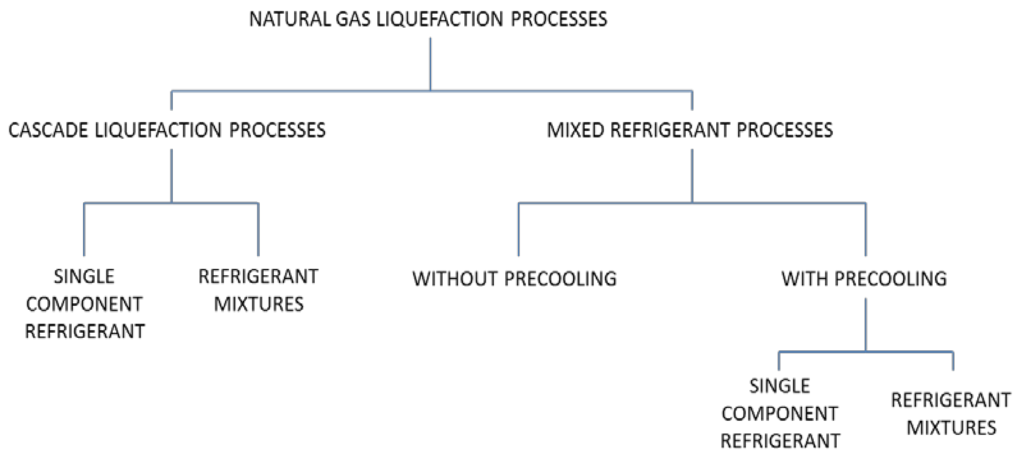

Figure 2-19. Classification of natural gas liquefaction processes

Table 2-1 gives on the other hand a perspective of the number of trains that each process license has in operation by 2010 (for further details see [32], [33], [34]). In the table, SMR stands for Single Mixed Refrigerant process; and AP-X® represents APCI's process that combines a C_3MR process with a closed nitrogen expander cycle [35]. Processes that use a precooling cycle account for 95,7 % of the worldwide installed trains (89 out of 93).

Table 2-1. LNG trains by liquefaction process, 2010.

Liquefaction Process	Licensor	Number of trains	% of Market
C_3MR Process	APCI	69	74,2 %
Optimized Cascade®	Phillips	9	9,7 %
SMR Process	APCI	4	4,3 %
Classic Cascade	Phillips	1	1,1 %
MFC® Process	Linde/Statoil	1	1,1 %
DMR Process	Shell	3	3,2 %
AP-X® Process	APCI	6	6,5 %

For the purpose of this work, two of the technologies mentioned are of particular importance, the propane precooled, mixed refrigerant process (C_3MR) and the mixed fluid cascade (MFC®). The first represents the highest proportion of the world's installed LNG production [28], and the latter is well known for its reduced energy consumption and high efficiency due to the use of refrigerant mixtures in cascade [36]. A brief explanation of these technologies is given below from the process approach, the equipment features will be considered in the next subchapter.

2.5.1. Propane Precooled, Mixed Refrigerant Process (C_3MR).

This process consists of two main refrigeration cycles, a precooling cycle and a liquefaction-subcooling cycle. The precooling cycle uses single component refrigerant, propane; whilst the liquefaction-subcooling cycle is operated with a mixed refrigerant. A typical flow diagram of this process is depicted in Figure 2-20.

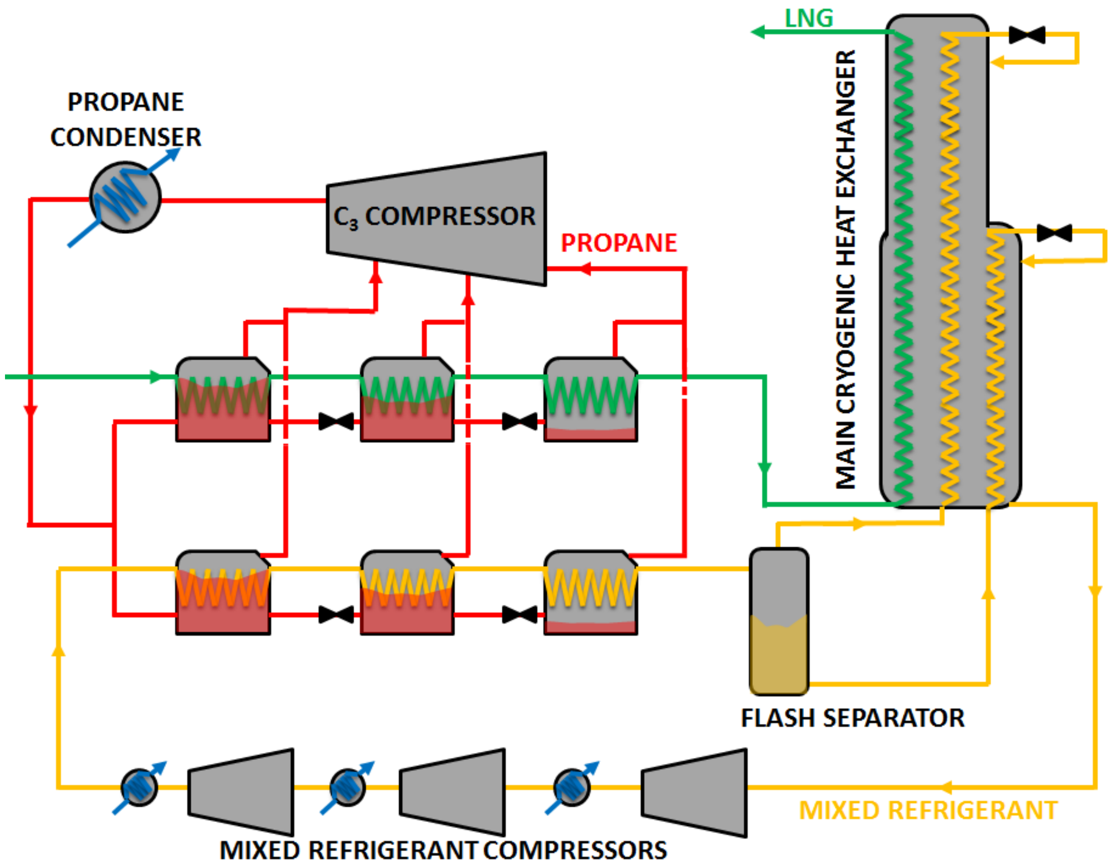

Figure 2-20. Propane precooled, mixed refrigerant process (C_3MR). Process Flow Diagram

In the precooling, a multistage refrigeration cycle is used at three or four pressure levels to exchange heat with the gas stream and the warm mixed refrigerant, cooling both streams down to around 238 K (-35 °C) [37]. The fluid circulation in this cycle is provided by a multistage compressor (with side streams) that compresses the vapour propane from each of the pressure levels to a common outlet pressure; at this pressure the propane stream undergoes heat exchange until it becomes liquid in the condenser. Once condensed, the propane stream is throttled multiple times throughout the precooling heat exchanger network where it is vaporized again.

In the liquefaction-subcooling cycle the rest of the process takes place; there the natural gas is further cooled down from 238 K to around 113 K (-160 °C) by the mixed refrigerant. The partially condensed mixed refrigerant from the precooling cycle is separated into vapour and liquid streams in a flash separator [38]. After that point both streams flow into the main cryogenic (multistream) heat exchanger, where the liquid stream is extracted in the first section/bundle and is expanded to be recirculated on the shell side. The gas stream goes all the way through both sections/bundles and at the top is throttled and recirculated on the shell side in the same way. Inside the heat exchanger the streams are mixed again and vaporized prior to the compression process, which can be carried out with more than one compressor, for instance a two or three compressor arrangement with intercooling [38].

2.5.2. Mixed Fluid Cascade Process (MFC®)

Three mixed refrigerants are used in this process in order to perform the precooling, liquefaction and subcooling duties. In this process the heat exchanger arrangement discussed before (may refer to Figure 2-17) is used for the three main cycles, and therefore multistream heat exchangers are required for each circuit.

The precooling cycle cools down the natural gas stream as well as both the liquefaction and subcooling refrigerant to around 223 K (-50 °C). The liquefaction cycle is responsible for cooling both the natural gas stream and the subcooling stage mixed refrigerant. Figure 2-21 is the process flow diagram for the MFC®, the process described may be easier to follow with help of the diagram.

The precooling cycle works as a multistage process since part of the refrigerant is throttled to an intermediate pressure, and used as the cold side in the first multi-stream heat exchanger. The rest is further subcooled in the second heat exchanger, to be expanded subsequently by means of a throttling valve. Once expanded it is used as the cold side in the second heat exchanger. Through both heat exchangers the precooling mixed refrigerant vaporizes while cooling the warm side streams;

after vaporization the streams are compressed to be liquefied in the precooling condenser.

After the precooling process, the liquefaction mixed refrigerant is further cooled down by its own cold (throttled) side. The throttled stream works as the cold side in the liquefaction heat exchanger, where it vaporizes while the warm side streams are cooled. Once vaporized, the refrigerant is compressed in the liquefaction compressor; to be cooled afterwards by the liquefaction cooler. Downstream the condenser, the liquefaction mixed refrigerant goes back to the precooling cycle.

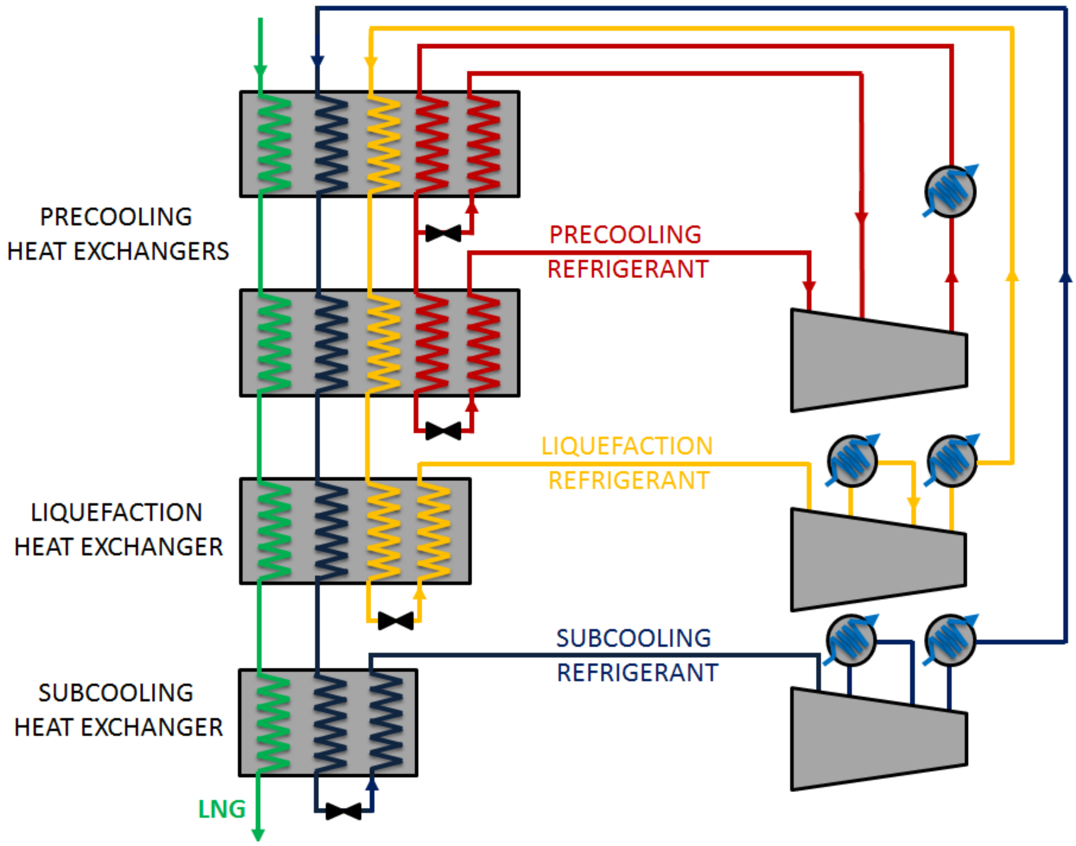

Figure 2-21. Mixed Fluid Cascade (MFC®). Process Flow Diagram.

Finally the subcooling mixed refrigerant, which has gone through the precooling, and liquefaction stage, enters the subcooling heat exchanger. In the same way as in the previous cycles, the throttled side vaporizes while the warm side streams (only natural gas stream and subcooling refrigerant) cool down. Downstream the subcooling heat exchanger, the refrigerant is compressed by the subcooling compressor and cooled down by the cooler of the subcooling stage. Then it goes back to the precooling and liquefaction stages. It is important to note that for all the cycles more than one compressor (and intercoolers) may be required [39].

2.6. LNG Process Equipment

Besides the multiple process definitions mentioned previously (i.e. type of refrigerant, pressure levels, temperature difference, number of cycles, etc.), natural gas liquefaction technologies involve selection of different equipment for the cycle operation. The liquefaction process equipment and installation can represent between 30 and 57% of the total investment in a LNG value chain, and the major costs in this area are related to compressors/drivers and heat exchangers [5, 10]. This section will give a brief explanation of the main equipment used in LNG processes, addressing the main differences between the existing technologies.

2.6.1. Heat Exchangers

The type of heat exchanger used depends on the selected type of refrigerant. A pure component refrigerant, for instance propane, can be vaporized efficiently in kettle-type heat exchangers. On the other hand, if a mixed refrigerant is used, a multistream heat exchanger is required. Two main types of multistream heat exchangers are widely used in the LNG industry; these are spiral (or coil) wound heat exchangers (SWHE) and plate fin (also known as brazed aluminum) heat exchangers (PFHE). The main differences between these two technologies are described in Table 2-2.

Table 2-2. Differences between plate fin and coil wound heat exchangers [40]

	Plate-fin	**Coil-wound**
Main features	Extremely compact Up to approx. 10 streams	Extremely robust Compact
Fluid requirements	Very clean Non-corrosive	No significant restrictions
Heating surface density	300-1000 m^2/m^3	50-100 m^2/m^3

	Plate-fin	Coil-wound
Equipment material	Aluminum	Aluminum, Stainless Steel, Carbon Steel
Design temperatures	-269 °C to +65 °C	All
Applications	Smooth operation Limited installation space	High temperature gradients High temperature differences
Prices	25-35 %	100 %

Figure 2-23 shows a picture of two units installed for the same performance, the unit in the right hand side is a spiral wound heat exchanger while the left hand side heat exchanger (smaller) is a plate-fin unit.

Since the precooling cycle of the C_3MR process uses propane as refrigerant, kettle-type heat exchangers are used in this section. A kettle heat exchanger refers basically to a unit in which the shell side stream is separated while the vaporization occurs; an example of the process is given in Figure 2-22. Two types of these heat exchangers are used in the LNG industry, one with a tube bundle into a shell (tube-shell kettle) and one with a plate fin heat exchanger submerged in the evaporation fluid (block-in-kettle). The block-in-kettle heat exchanger represents a greater investment, nevertheless it contains up to 10 times more heat transfer area per unit volume than a shell-tube unit [41].

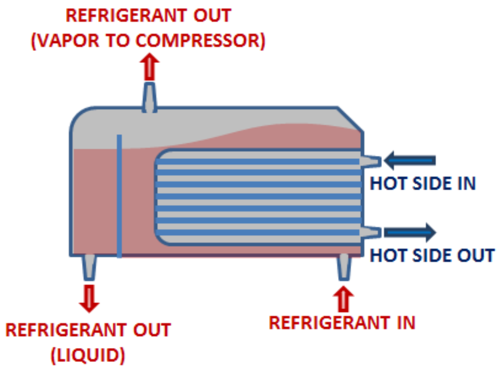

Figure 2-22. Kettle-type heat exchanger diagram

In contrast, the main cryogenic heat exchanger in the C_3MR process is a multistream heat exchanger, and the preferred choice is the coil-wound due to its robustness compared to the plate-fin heat exchangers [42]. In the MFC® process all the process heat exchangers are multistream-type. According to the process licensor, a detailed comparison may demonstrate that each multistream heat exchanger type has specific merits when it is placed at the proper place, hence plate-fin heat exchangers are chosen for the precooling circuit and coil-wound heat exchangers for the liquefaction and subcooling cycles [43].

Figure 2-23. Compactness of spiral wound heat exchanger versus plate fin heat exchanger [40]

2.6.2. Compressors

Each process requires specific treatment concerning cycle compressor selection; however, the main parameters studied for the selection are typically the same [10]. Power requirement, pressure ratio, volumetric flow of the suction stream and the efficiency of the compressor (Equation (2.8)) are the most important ones. For a specific compressor working with a given fluid the supplier provides a so-called compressor performance curve or compressor map in which the relation between the different operational parameters is given, an illustration of such map is shown in Figure 2-24. The highest efficiencies are usually achieved for pressure ratios between 2,5 and 5,0 [44].Pressure ratio and polytropic head are related through equation (2.9); hence both of them might be in the y-axis of the compressor map.

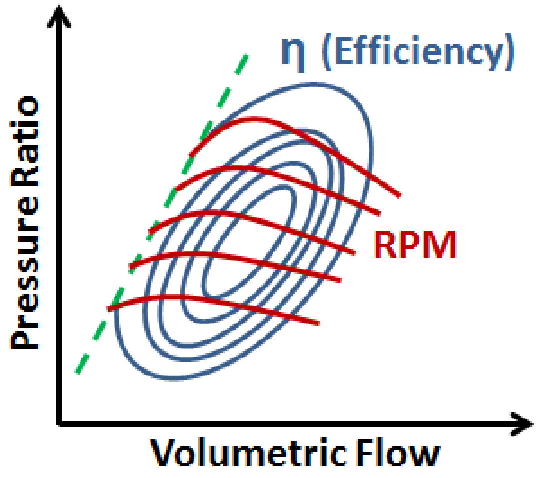

Figure 2-24. Compressor map illustration

$$H_p = \frac{n}{n-1}\frac{Z.R.T}{g.M}\left(\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right) \quad (2.9)$$

The larger used models of compressors and their respective technical specifications are shown in Table 2-3. This is important to take into account when developing a new project in order to remain within the design limits of proven solutions, and for the purpose of this work to know the limitations of the equipment capacity.

Table 2-3. Technical specifications of LNG compressors [45]

Model	RPM	Flow (m^3/h)	Side Streams	Maximum Power (MW)	LNG Duty
3MCL1800	2200 3600	200000 380000	1-3	120	Propane Precooling
MCL1800	2200 3600	200000 380000	-	120	Methane and mixed refrigerant subcooling. Ehtylene and mixed refrigerant liquefaction.

2.6.3. Compressor Drivers

The choice of drivers and their fit with the process and power generation system is considered the critical stage of the selection process in LNG projects [19]. For the first LNG plants the compressors were driven by steam turbines, providing a designer the ability to develop the process over wide ranges of power and speed. However, low efficiency, large equipment and required cooling system have led to their displacement by gas turbines.

Developments in gas turbines have been in pace with the growth in single train capacity, basically because the train size limitation has been set, for the most part, by the driver of the process (and its respective compressor) [10]. An overview of the most commonly used gas turbines specifications is given in Table 2-4.

Table 2-4. Gas turbine performance specifications [45]

Model	Power (MW)	Efficiency	Year startup
LM2500+	31,36	41,10 %	< 1989
LM6000	44,7	42,60 %	< 1989
Frame 5D	32,58	29,40 %	1989
Frame 6B	43,53	33,30 %	1995
Frame 7E	87,3	33,00 %	1996
Frame 9E	130,1	34,60 %	2007

Even though gas turbines are the most common drivers for existing LNG plants, recent developments are showing high interest in using electrical motors to drive the refrigerant compressors, as done by first time at the LNG plant in Hammerfest –Norway [39]. The reason behind this tendency is the increase in plant availability (larger production per year) that can be achieved since the performance of electrical motors is controlled almost stepless during operation whilst for gas turbines it depends on the fuel and air conditions. The power to supply the electrical motors may be obtained from the national power grid (if available) or generated by a power plant on site.

2.6.4. Cooling medium

The cooling medium of an LNG plant refers to the cold side fluid in the refrigerant condensers. More than 70% of the installed LNG trains use seawater as cooling medium, either by direct or indirect cooling; air cooled condensers account for the rest [46]. The main differences between seawater and air condenser are given in Table 2-5.

Table 2-5. Cooling medium differences, seawater versus air [46]

	Seawater	Air
Reliability	Depends on proper material selection (Fouling, corrosion). Good.	Depends on environmental conditions. Good.
Maintenance	Periodic cleaning required. Continuos self-cleaning systems also available.	No cleaning required.
Production	Stable	Varying daily. Affected by weather conditions
Environmental	Limitations in amount of processesed seawater to be returned. Discharge temperature restricted, marine life safety	Not important. High noise level

An air cooled plant usually gives a higher specific power than seawater cooled, due to the temperature difference that can be achieved between the streams [47]. Water cooled heat exchangers are designed for a temperature difference between the seawater and the refrigerant of around 5 °C; whilst in air cooled heat exchangers the difference is between 10 and 20 °C [48].

Chapter 3. Simulation Cases

This chapter will introduce the definitions and parameters used during the simulation work. Explanation of estimations, approximations and the path followed to build the processes into the simulation environment will be given. Aspen HYSYS®, a general purpose sequential modular process simulator was selected as the simulation software because it has features that accommodate many of the special requirements that are involved in natural gas liquefaction processes [27]. HYSYS® flowsheets of each of the processes described in this chapter may be found in Appendix A.

3.1. Stand-alone precooling cycles

In order to study the precooling stage of LNG processes in detail, a stand-alone simple refrigeration cycle is implemented first, see Figure 3-1 and Figure 3-2. The idea is to study the difference given by the use of a mixed refrigerant (made out of Ethane C_2 and Propane C_3) or a pure component refrigerant (only C_2 or C_3) in a simple cycle for precooling natural gas to 237 K (-36 °C), which is the reported minimum temperature that can be achieved with a propane precooling in order to avoid the risk of air entering the system [38].

The variables to be evaluated in this case are:

- Power of the compressor
- Heat exchanger UA value
- Suction volume of the compressor
- Pressure ratio in the compressor

The thermodynamic fluid package of Peng-Robinson ([49]) was used as the basis of the simulation. A Weighted model is chosen as the heat exchanger calculation method, since the software developer states that it represents an excellent model to deal with non-linear heat curve problems such as the phase change of fluids [50]. By selecting this model the heating curves are divided into intervals, and an energy balance is performed for each interval. A logarithmic mean temperature difference (LMTD) and UA value (Equation (2.6) and (2.7)) is calculated for each interval and the total UA is found by the sum of the values of the intervals.

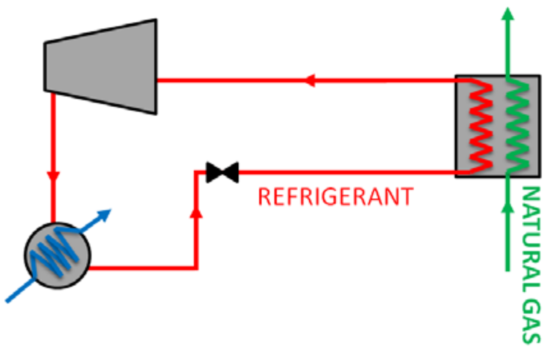

Figure 3-1. Simple refrigeration cycle configuration for pure component refrigerant

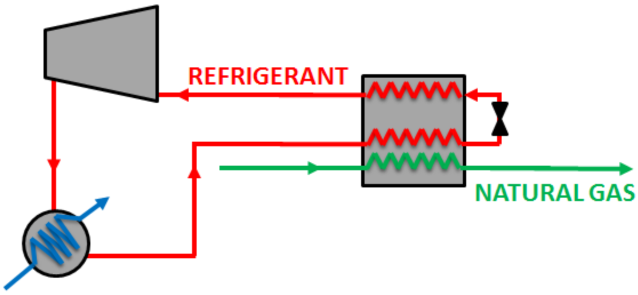

Figure 3-2. Simple refrigeration cycle configuration for mixed refrigerant

Table 3-1 shows the natural gas composition used for the simulation while other parameters set for this study case are shown in Table 3-2. The pressure after the compressor is calculated as the required for the refrigerant to be condensed at the condenser outlet temperature, which is set to 30 °C (assuming a relatively warm climate of 25 °C) [51]. On the other hand, the expansion pressure is determined by the minimum temperature difference that is set in the heat exchanger definition. Pressure drop through the equipment is not taken into account.

Table 3-1. Natural gas composition

Component	Fraction (%)
Methane (C_1)	89,7
Ethane (C_2)	5,5
Propane (C_3)	1,8
n-Butane (n-C_4)	0,1
Nitrogen (N_2)	2,9

Table 3-2. Parameters for simulation, simple refrigeration cycle

	Parameter	Value
Natural Gas Inlet	Temperature	30 °C
	Pressure	40 bar
	Flowrate	60000 kmol/h
Cycle parameters	Adiabatic efficiency compressors	75 %
	Temperature after condenser	30 °C
	ΔT_{MIN} condenser	5 °C
	ΔT_{MIN} heat exchanger	1 °C

Once the simulation for the simple cycle is done, the compressor is replaced by a two stage compression with intercooling, this is done in order to see mainly the improvements achieved in the energy consumption of the process and the pressure ratio, since the other factors (UA and suction volume) will remain unaltered. An illustrative sketch of the process flow diagram for this modified case is shown in Figure 3-3.

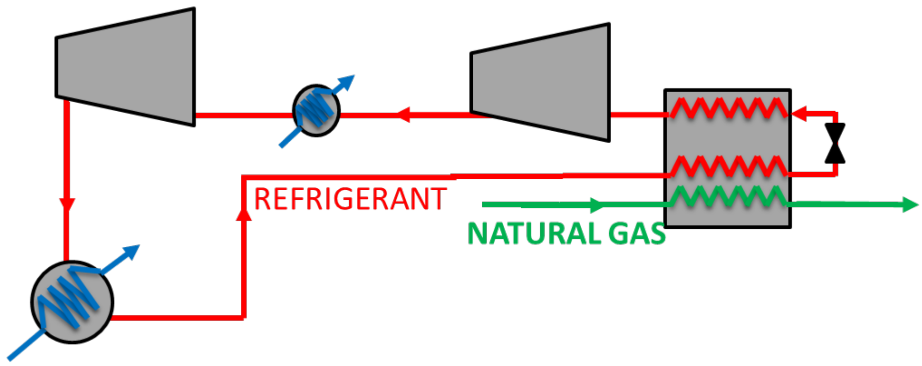

Figure 3-3. Simple refrigeration cycle using two stage compression with intercooling

Furthermore, the study is built with an important modification; the cooling duty is taken by a two stage cycle, this is exemplified in Figure 3-4. Since a two stage cycle is used, a new degree of freedom is introduced to the system and it is represented by the temperature of the natural gas between the first and the second stage of the cycle; which will determine the pressure level of the refrigerant in the first cycle, depending on the temperature difference set in the heat exchanger.

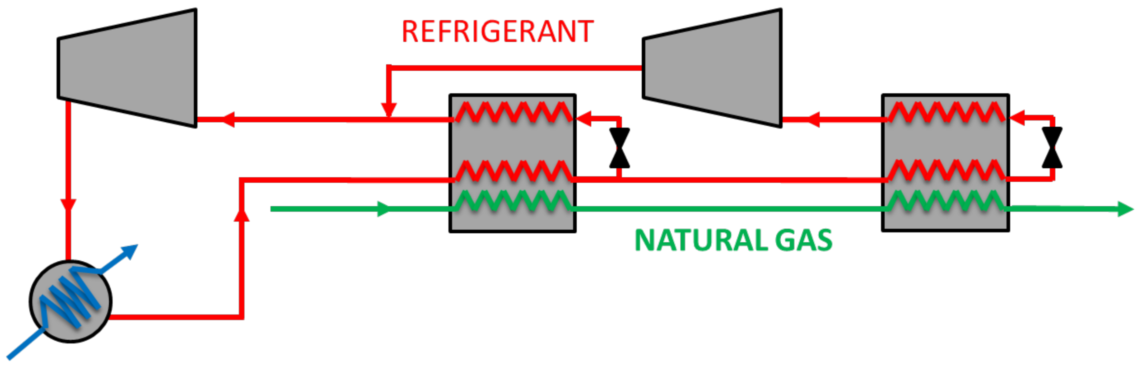

Figure 3-4. Two stage refrigeration cycle.

The last case for this section is a three stage cycle using propane as refrigerant, this is depicted in Figure 3-5. In this study the composition of the refrigerant is kept constant (pure C_3) since a three stage mixed refrigerant is unlikely to be implemented in LNG processes, basically due to complexity and equipment cost [52]. The study was made by evaluating the same parameters mentioned above but varying the first two pressure levels, which are determined by the natural gas outlet temperature from each stage; the third stage pressure is fixed by the natural gas outlet temperature and the temperature difference in the heat exchanger.

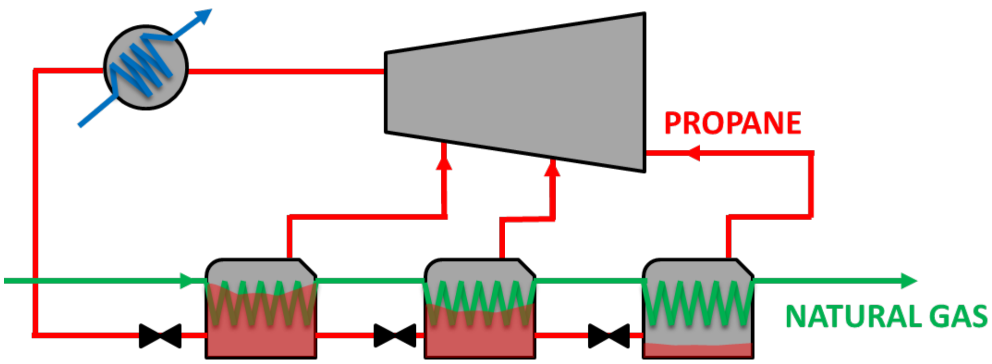

Figure 3-5. Three stage cycle using kettle type heat exchangers with propane

Since the heat exchangers available in the simulation software do not include kettle-type heat exchangers, these were simulated using tube-shell heat exchangers with a flash separator at the shell side outlet, an illustrative example of this setup is shown in Figure 3-6.

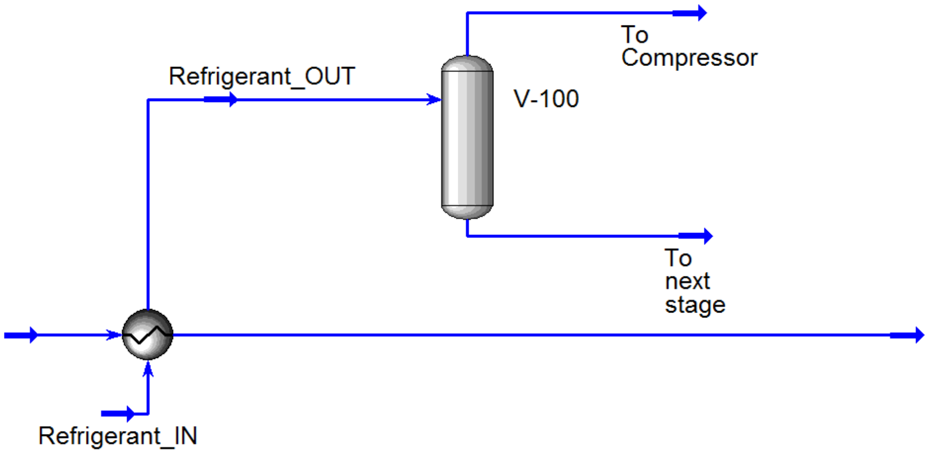

Figure 3-6. Kettle type heat exchanger implemented in HYSYS®

As a summary the cases studied in the stand-alone section and described above are:

- Simple refrigeration cycle. Variation of the refrigerant (C_2 and C_3) composition.
- Simple refrigeration cycle using two stage compression with intercooling. Variation of the refrigerant composition.
- Two stage refrigeration cycle. Variation of the refrigerant composition and the first stage pressure (temperature of natural gas between two stages).
- Three stage refrigeration cycle using kettle-type heat exchangers with propane. Variation of the first and second stage pressures (temperature of natural gas after each stage).

All these cases use the same heat load, since the natural gas enters at the same conditions (Table 3-2) and is cooled from 30 °C to -36 °C. A colder climate condition is also taken into account; for that condition the temperatures at the condenser outlet and the natural gas inlet are assumed to be 11 °C based on an ambient temperature of 6 °C. All the procedure described above is repeated for this cold climate condition.

3.2. Application of the different configurations to the C_3MR process

Based on the simulations done in the previous section, the effects on the whole liquefaction cycle performance are studied by implementing the most suitable conditions of each different configuration into a C_3MR process (Figure 2-20), which means that the three stage propane precooling circuit of the C_3MR process is replaced by the configurations given in Section 3.1.

Most suitable refers to the conditions that represent the clearest benefit in terms of the studied parameters for each case, these parameters ordered by relevance are: power consumption, UA value, suction volume and pressure ratio of the compressor. Notice that not necessarily the best conditions in the stand-alone cycle represent the most suitable ones for the whole liquefaction process, especially because the heating curves are altered by the flow of additional streams in the warm side. However, the values found from the stand-alone cycle give a fairly close approach to the most suitable condition in the entire liquefaction process, and function at least as starting point for finding the desired conditions.

In order to implement the C_3MR process, the main cryogenic heat exchanger bundles are modeled as separate LNG-type heat exchangers. Three recycle units are required at appropriate points with their respective estimated values because HYSYS® calculates unit operations subsequently, so the recycle streams are sequentially solved until the assumed values of the auxiliary stream match the calculated ones within a specified tolerance [50]. A three compressor arrangement with intercooling is implemented as the mixed refrigerant compression system.

For this section the natural gas inlet conditions are kept as given above (Table 3-1 and 3.2); pressure drop across equipment is taken into account and is shown in Table 3-3 along with other parameters of the simulation. Since the aim is to study the precooling cycle, the mixed refrigerant composition (Table 3-4) is taken from a C_3MR process optimized previously under the same conditions used in this work (may find [53]). It is important to notice that setting up parameters in the rest of the process is not the goal of this work and therefore it is avoided; otherwise the clarity of the results for the precooling circuit might be affected.

The pressures of the mixed refrigerant after expansion and in the compression train are chosen based on the reported values for the optimal process [53], as well as the mixed refrigerant flowrate; these may be found in Table 3-4 and Table 3-5.

Table 3-3. Main parameters for simulation, C_3MR case

	Parameter	Value
Precooling Cycle	ΔT_{MIN} Heat Exchanger	1 °C
	$\Delta P_{H.\ EX\ KETTLE}$ tube side	0,5 bar
	$\Delta P_{H.\ EX\ KETTLE}$ shell side	0,1 bar
	$\Delta P_{H.\ EX\ MULTISTREAM}$ hot side	5 bar
	$\Delta P_{H.\ EX\ MULTISTREAM}$ cold side	0,5 bar
Mixed Refrigerant Cycle	ΔT_{MIN} Heat Exchanger	1 °C
	$\Delta P_{HEAT\ EXCHANGER}$ hot side	5 bar
	$\Delta P_{HEAT\ EXCHANGER}$ cold side	0,5 bar
Cycle Parameters	$\Delta P_{CONDENSER}$	0,1 bar
	Temperature natural gas outlet	-154,5 °C

Table 3-4. Mixed refrigerant composition for C_3MR cycle

Component	Fraction (%)
Methane (C_1)	45,0
Ethane (C_2)	45,0
Propane (C_3)	2,0
n-Butane (n-C_4)	-
Nitrogen (N_2)	8,0

Table 3-5. Mixed refrigerant circuit parameters

Parameter	Value
Flowrate	117100,00 kmol/h
$P_{EXPANSION}$	5,4 bar
$P_{FIRST\ STAGE}$	22,9 bar
$P_{SECOND\ STAGE}$	33,8 bar
$P_{THIRD\ STAGE}$	48,0 bar

The cold climate condition is implemented using the same mixed refrigerant cycle parameters; a modification could be done, but it disturbs the equilibrium in the separator, so the refrigerant composition has to be changed. This will represent undesirable variables to enter the evaluation, and previous work has shown that the variation is not significant for the liquefaction/subcooling cycle as compared with the precooling [54].

Besides the cases based on the stand-alone cycles, the addition of n-Butane (n-C_4) to the refrigerant mixture is taken into account for this study. This is made with particular interest for the warm climate case, since, as can be seen in Figure 2-11, the temperature range indicates that n-C_4 might be a suitable component for the mixture. The optimization of the refrigeration process with ternary mixtures is a challenging task and many publications have been based on it, such as the referenced in [55-58]. Since the aim of this work is the evaluation, and dealing with complex optimization schemes is beyond the scope; the mixed refrigerant composition is selected from a tabular data series with optimized values for a dual mixed refrigerant process reported by Venkatarathnam [58]. The temperature range for the precooling in the optimized process is the same used for the warm climate condition, from 30 to -36 °C; nonetheless there is a difference in the natural gas composition as shown in Table 3-6. This difference in the composition might affect the optimality of the composition used, but not the validity of the evaluation and comparison.

Table 3-6. Natural gas composition, comparison between this work and Venkatarathnam's

	Fraction (%)	
Component	**This work**	**Venkatarathnam**
Methane (C_1)	89,7	87,5
Ethane (C_2)	5,5	5,5
Propane (C_3)	1,8	2,1
n-Butane (n-C_4)	0,1	0,5
i-Butane (i-C_4)	-	0,4
Nitrogen (N_2)	2,9	4

3.3. Mixed refrigerant precooling temperature relocation

One of the advantages of using a mixed refrigerant in the precooling cycle is that the restriction of the minimum temperature to be reached is relocated. If the component added to the mixture is lighter than the present one, then the minimum value moves to a lower temperature and vice versa. Figure 3-7 shows the minimum temperature that can be reached for different compositions of the refrigerant, based on a binary mixture of C_2 and C_3; notice that with pure propane this temperature is -36 °C, as mentioned before.

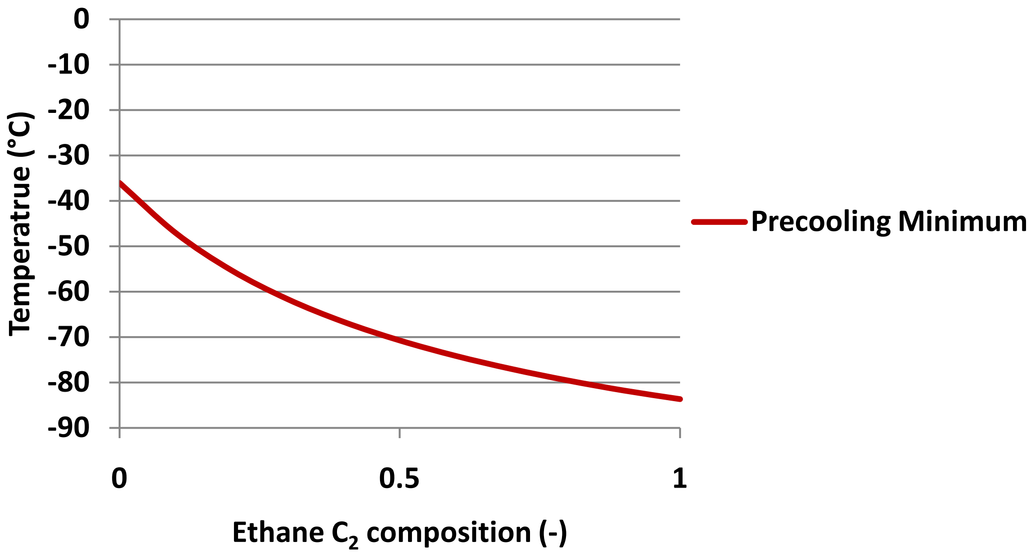

Figure 3-7. Minimum temperature to be reached in the precooling, composition variation

In the cases described on the previous sections of this chapter the precooling cycle is limited to -36 °C, which means that the mentioned benefit provided by the mixed refrigerant is not taken into account. In order to study this alternative, a stand-alone two stage cycle for precooling the natural gas stream (defined in Table 3-1 and Table 3-2) to -50 °C is simulated. The cold climate condition is also considered in this section.

3.4. Application of the different configurations to the MFC® process

Since in the study introduced on Section 3.3 the process reaches a different temperature than all the previously studied cases, this cycle is not equally comparable with the formerly described ones. In order to study the effects of such cycle a Mixed Fluid Cascade (MFC®) process is implemented, see Figure 2-21; this process fits the required purpose of the simulation because its precooling cycle cools the streams down to approximately -50 °C [59]. Data from an optimization work [60] on the MFC® process is used for the liquefaction and subcooling circuit in this simulation, the values are shown in Table 3-7 and Table 3-8. The natural gas inlet conditions for this case are different than the previously described in order to fit the optimized process parameters; nevertheless, the same natural gas flowrate is used, see Table 3-9 and Table 3-10.

Table 3-7. MFC® process liquefaction and subcooling refrigerant composition

	Liquefaction	Subcooling
Component	**Fraction (%)**	**Fraction (%)**
Methane (C_1)	4,02	52,99
Ethane (C_2)	82,96	42,45
Propane (C_3)	13,02	
Nitrogen (N_2)	-	4,55

Table 3-8. MFC® process liquefaction and subcooling cycle parameters

	Liquefaction	Subcooling
Parameter	**Value**	**Value**
Flowrate	30000 kmol/h	37620 kmol/h
$P_{EXPANSION}$	2,0 bar	2,0 bar
$P_{FIRST\ STAGE}$	-	28,38 bar
$P_{SECOND\ STAGE}$	20,58 bar	56,99 bar

Table 3-9. Natural gas composition, MFC® process simulation

Component	Fraction (%)
Methane (C_1)	88,80
Ethane (C_2)	5,70
Propane (C_3)	2,75
Nitrogen (N_2)	2,75

Table 3-10. Natural gas inlet conditions, MFC® process simulation

Parameter	Value
Temperature	11/30 °C
Pressure	61,5 bar
Flowrate	60000 kmol/h

The evaluation is made based on a modification in the precooling cycle of the MFC® process; in contrast with the C_3MR cases, the mixed refrigerant precooling of the MFC® is replaced by a three stage propane cycle. Care should be taken in this particular scenario because the liquefaction cycle parameters have to be redefined in order to cover the gap of temperatures left due to the restriction in temperature for pure propane, see Figure 3-8.

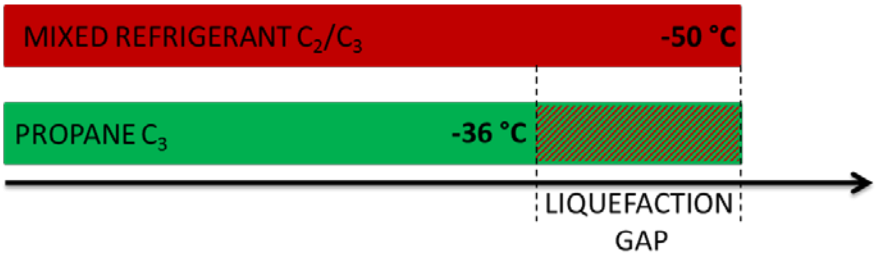

Figure 3-8. Temperature difference, propane and mixed refrigarant precooling

The redefined liquefaction cycle parameters for the propane precooled MFC® process are given in Table 3-11 and Table 3-12, while the subcooling cycle keeps the same parameters given in Table 3-8 and 3.7. As for the C_3MR processes, the evaluation is meant to be focused on the precooling cycle, therefore no modifications are introduced on the rest of the process when the climate condition is switched.

Table 3-11. Liquefaction refrigerant composition, MFC® with propane precooling

Component	Fraction (%)
Methane (C_1)	11,00
Ethane (C_2)	57,00
Propane (C_3)	32,00

Table 3-12. Liquefaction cycle pressures, MFC® with propane precooling

Parameter	Value
Flowrate	36600 kmol/h
$P_{EXPANSION}$	2,2 bar
$P_{COMPRESSION}$	21,3 bar

Chapter 4. Results and Discussion

In this chapter the main results obtained through the simulations are presented. It is divided in sections, following the structure introduced in the previous chapter. An analysis of the results and discussion is given as the results are presented, the theoretical background given in Chapter 2 and the simulation procedure from Chapter 3 are of great importance for the development of this chapter and consequently both will be frequently mentioned.

4.1. Stand-alone precooling cycles

4.1.1. Simple refrigeration cycle

The simple refrigeration cycle is the first of the cases to be studied in this section. The refrigerant composition is switched from pure propane (C_3) to pure ethane (C_2) passing through all the possible binary mixture combinations with C_2 and C_3. As mentioned in Chapter 3, two cases are considered, one under warm and one under cold climate conditions. For the natural gas stream being cooled until -36 °C, the difference in the heat load between the two cases is calculated and shown in Table 4-1.

Table 4-1. Heat load comparison, warm vs cold climate condition

	Heat load	Difference
Warm climate	48,35 MW	138,54 %
Cold climate	34,90 MW	100 %

Figure 4-1 shows the value of the compressor duty in Megawatts (MW) for different compositions of refrigerant; the minimum duty is found by using a mixture which is 20% ethane and 80% propane for the warm conditions, while for the cold climate it is found where the ethane composition is around 0,3. It is important to notice that for each composition there is more than one condition that can fit the constraints; the values shown are the best values found for each composition, the whole detailed study results are presented in Appendix B.

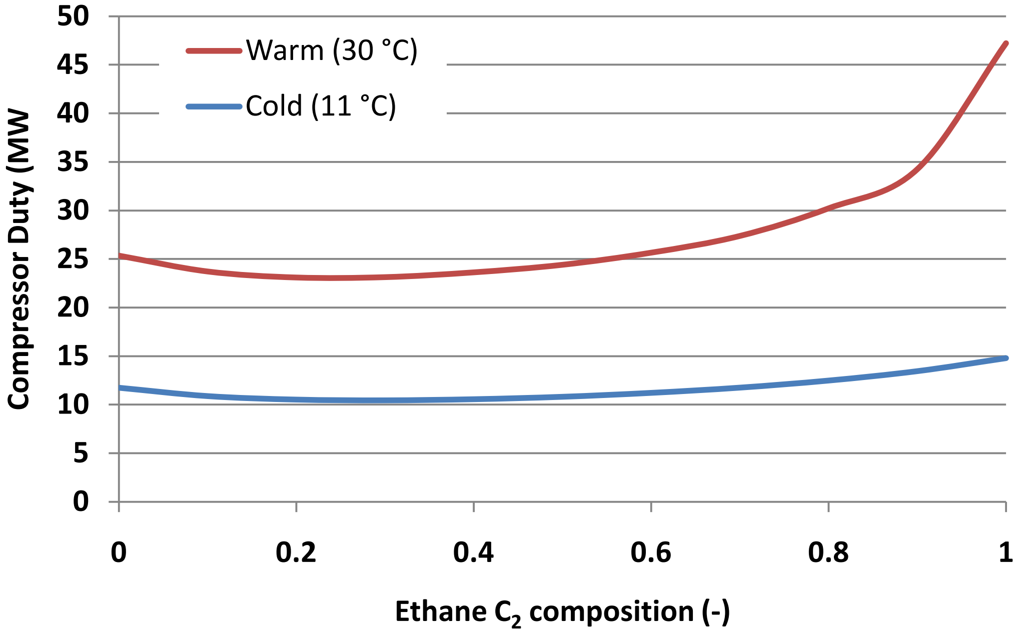

Figure 4-1. Compressor duty with variation of the refrigerant composition for simple cycle

A noteworthy aspect about the figure above is that using a pure propane refrigerant (C_2 composition equal to zero) gives a compressor duty 9,7 % higher than the minimum for the warm climate conditions, and 12,7 % higher than the minimum for the cold case. The large duty for warm climate when using pure ethane is because the temperature range is very distant from the saturation temperature of ethane (see Figure 2-11); which means that the temperature difference between the streams is far from ideal (large heat exchange losses).

The duty for the warm climate case is (in the most favourable comparison) 215 % of the cold climate duty. When comparing this with the values given in Table 4-1 for the heat load, the performance of the warm climate cycle seems to be in disadvantage. In order to compare the performance of the refrigeration cycles, the coefficient of performance (β) is calculated using Equation (2.4), the values are shown in Figure 4-2. The cold climate case indeed has a higher coefficient of performance for any composition of a C_2/C_3 mixture, and for the pure component cases.

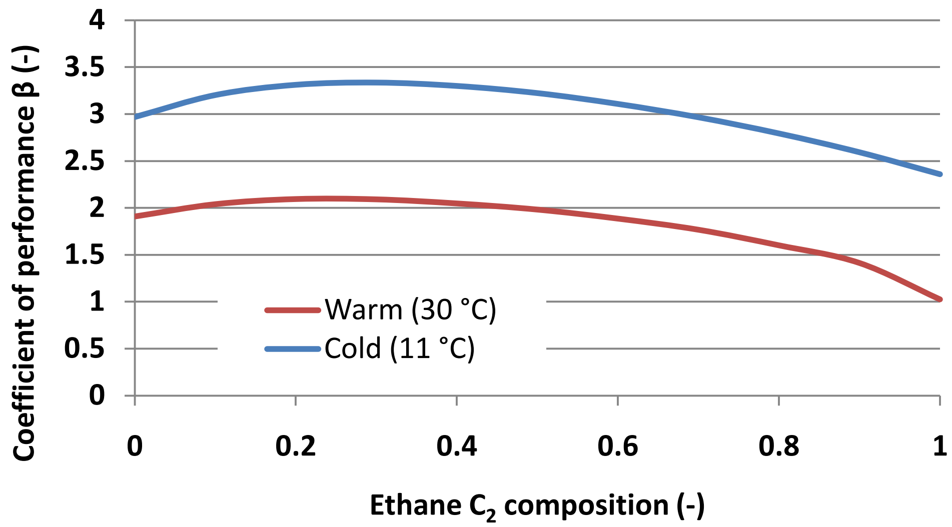

Figure 4-2. Coefficient of performance with variation of refrigerant composition for simple cycle

Another important parameter to take into account is the UA value in the heat exchanger. As mentioned in Chapter 2, the UA value is linked with the capital cost of this equipment and also gives a clue of the plot area required. Figure 4-3 shows the value for the UA parameter in kilojoule (kJ) per degree Celsius (°C) per hour (h) for different refrigerant compositions. As expected, the mixed refrigerant requires a higher value when compared to the pure refrigerant (C_2 composition equal zero and one); mainly because there is an additional stream flowing through the heat exchanger, compare Figure 3-1 and Figure 3-2.

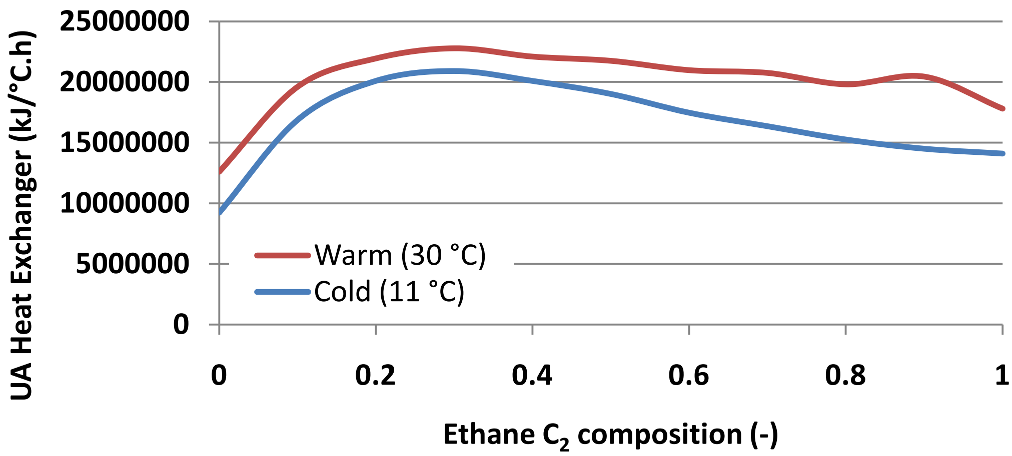

Figure 4-3. Heat exchanger UA with variation of the refrigerant composition for simple cycle

The extra stream flowing in the heat exchanger will increase the amount of heat transferred whilst the LMTD does not change significantly (Equations (2.6) and (2.7)), which represents a rise in the UA value. For both conditions (cold and warm climate) the exhibited behavior is mainly the same, with a slight reduction for the cold climate; this reduction is believed to be due to the reduction in heat load while the temperature difference through the heat exchanger does not increase significantly to compensate, see Figure 4-4 where the temperature profiles are shown for the case at 30% ethane/70% propane.

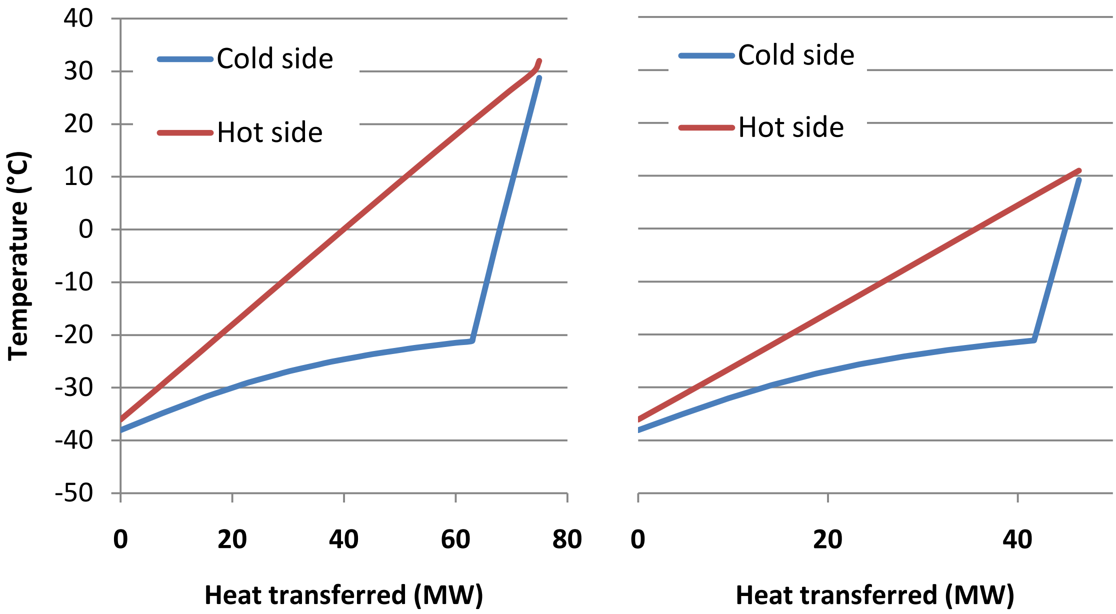

Figure 4-4. Temperature profile across the heat exchanger, mixture 0,3 C_2/0,7 C_3
Left figure: warm climate. Right figure: cold climate

It is essential to compare the behavior of the UA value with the compressor duty, Figure 4-3 and Figure 4-1 respectively. This is made by plotting both variables for the warm condition, in the same frame, as shown in Figure 4-5. The influence of using a mixed refrigerant is reproduced in the UA value much more than in the compressor power; for instance moving from pure propane to a mixture which is 20% ethane represents an increase of 74% in the UA value and only 9,7 % reduction in the compressor duty.

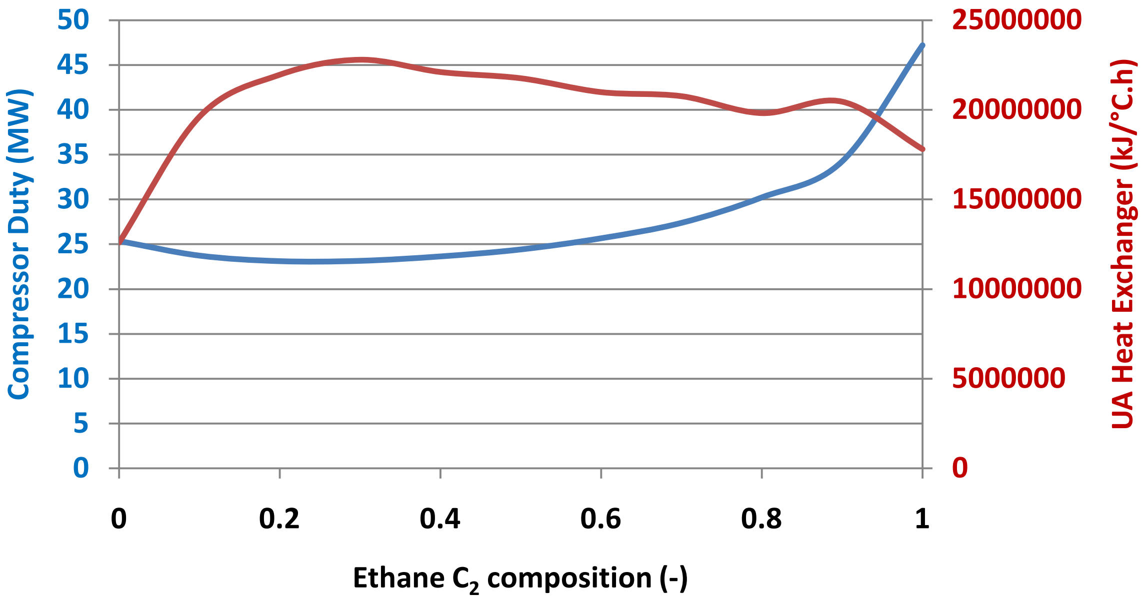

Figure 4-5. Compressor duty and UA value behavior comparison, simple cycle at warm condition

Besides its energy consumption, the compressor is studied by evaluating the volumetric flow in the suction side and the pressure ratio required; as mentioned in Chapter 2 these parameters are of great importance for the compressor performance and design limitations. The volumetric flow for different compositions is presented in Figure 4-6; using a mixed refrigerant benefits the suction volume when adding a lighter component, for instance a pure propane cycle requires a volumetric flow twice as large as a mixed refrigerant with 20% ethane. Both climate conditions give the same trend.

The pressure ratio (outlet/inlet) is plotted in Figure 4-7 for different refrigerant compositions. The inlet pressure is determined as the dew point at the natural gas outlet temperature plus temperature difference in the heat exchanger (approx. -37 °C), whilst the outlet pressure is given by the bubble point pressure at the temperature in the condenser (11 °C or 30 °C); for pure component fluids dew point and bubble point represent the same value and is known as saturation point, refer to section 2.6 for further details.

As ethane is added to pure propane (increasing C_2 composition) the pressure ratio is reduced because the saturation pressure for ethane is higher than for propane at any chosen temperature (Figure 2-11) and a higher value in the divisor will lead to a smaller quotient. The behavior of the curve is analogous for the warm and cold condition.

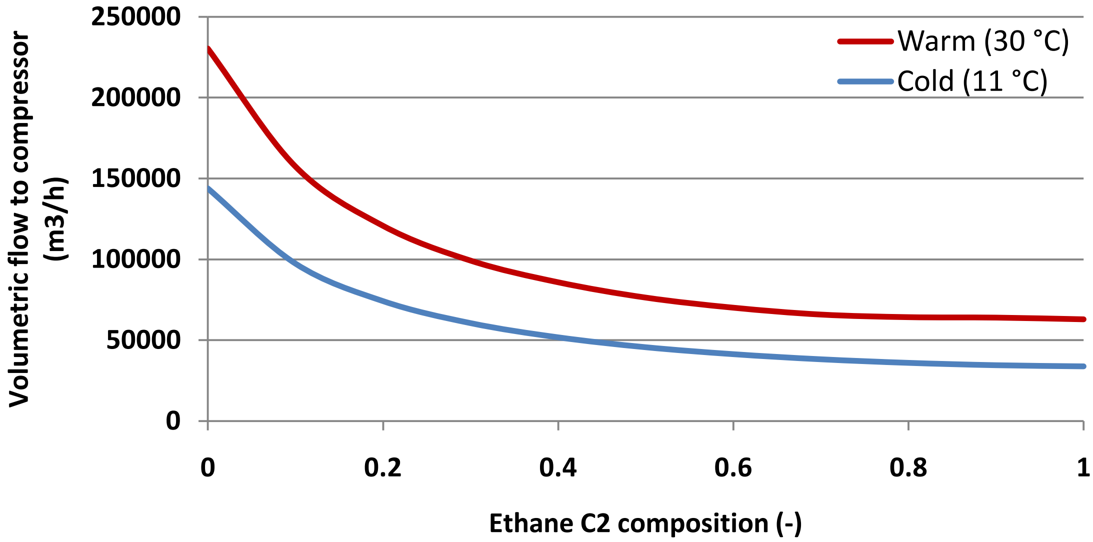

Figure 4-6. Compressor suction volume with variation of the refrigerant composition for simple cycle

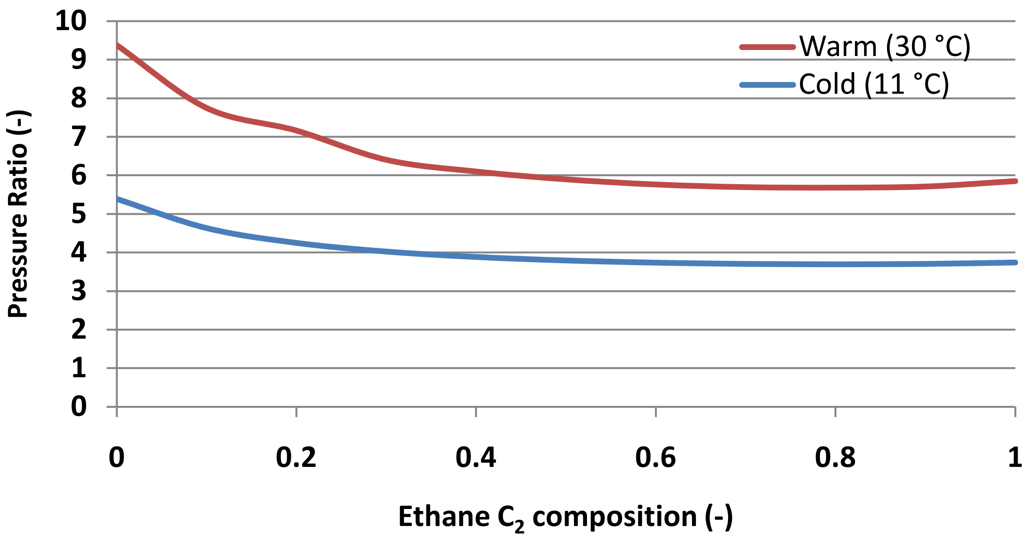

Figure 4-7. Compressor pressure ratio with variation of the refrigerant composition for simple cycle

A tabular summary of the results presented in this section is given below; Table 4-2 shows the results for the warm condition and Table 4-3 for the cold climate condition. The most suitable values are in red colored font.

Table 4-2. Summary for simple refrigeration cycle, warm climate condition

C_2 comp. (-)	Duty (MW)	$P_{EXPANSION}$ (bar)	$P_{COMPRESSION}$ (bar)	P_{RATIO} (-)	UA (kJ/°C.h)	Vol. flow (m^3/h)
1,00	47,22	8,31	48,57	5,84	17805699,70	62853,27
0,90	34,25	7,51	42,84	5,70	20449780,28	63902,97
0,80	30,22	6,74	38,19	5,67	19812439,84	64178,60
0,70	27,38	5,99	34,06	5,68	20755292,00	65791,33
0,60	25,66	5,26	30,26	5,75	20981343,57	70039,99
0,50	24,41	4,55	26,72	5,87	21754040,41	76288,58
0,40	23,63	3,85	23,36	6,07	22102054,50	85766,42
0,30	23,14	3,17	20,17	6,37	22787418,08	99181,87
0,20	23,10	2,50	17,12	6,85	21952005,45	120716,66
0,10	23,72	1,84	14,18	7,69	19609468,32	157422,00
0,00	25,35	1,21	11,35	9,35	12600265,76	230393,32

Table 4-3. Summary for simple refrigeration cycle, cold climate condition

C_2 comp. (-)	Duty (MW)	$P_{EXPANSION}$ (bar)	$P_{COMPRESSION}$ (bar)	P_{RATIO} (-)	UA (kJ/°C.h)	Vol. flow (m^3/h)
1,00	14,80	8,31	31,07	3,74	14097440,19	33764,51
0,90	13,46	7,51	27,81	3,70	14497764,77	34469,53
0,80	12,49	6,74	24,88	3,69	15256184,37	35930,51
0,70	11,77	5,99	22,18	3,70	16354447,76	38138,62
0,60	11,23	5,26	19,65	3,73	17467988,97	41300,64
0,50	10,82	4,55	17,25	3,79	19018203,54	45598,80
0,40	10,57	3,85	14,95	3,88	20095128,14	51723,11
0,30	10,46	3,17	12,74	4,02	20913763,92	60525,35
0,20	10,54	2,50	10,61	4,25	20101047,64	74154,49
0,10	10,89	1,84	8,54	4,63	16875748,31	97395,46
0,00	11,75	1,21	6,54	5,39	9219966,10	143898,21

4.1.2. Two stage compression with intercooling

The same cycle from the previous section is implemented using a two stage compression train with intercooling in order to achieve lower power consumption (see Figure 2-10). The study is run with the results from the previous section but with variation in the intermediate pressure of the compression train; again, the best cases for each composition are the ones given in the figures. The power consumption achieved is shown in Figure 4-8, where the duty for the single stage compression is also shown to ease the comparison. The dashed lines represent the compressor duty with single stage compression, and the regular lines (multistage compression) are from 7 to 11% below for both conditions.

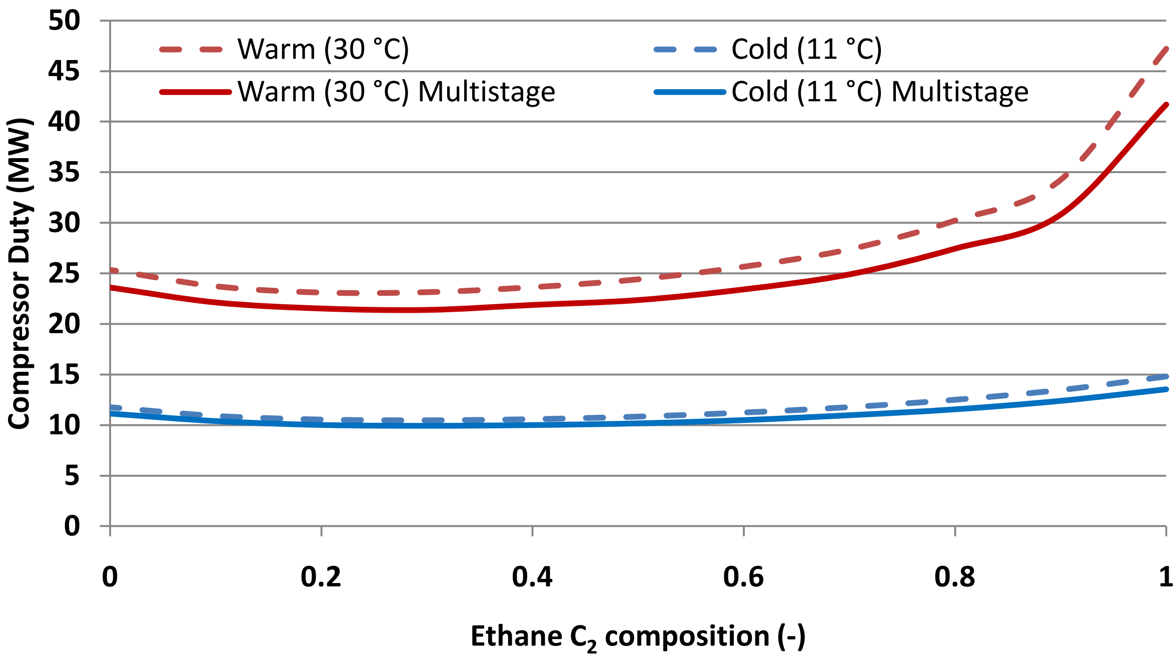

Figure 4-8. Compressor duty with variation of refrigerant composition, two stage compression

Another parameter improved by the use of a multistage compression train is the pressure ratio in the compressor. Figure 4-9 shows the value of the pressure ratio for the two compressors in the multistage train compared with the single stage value. The pressure ratio is approximately the same for both compressors in the compression train. This is basically because the values shown are the optimal, and performing an optimization analysis for the multistage compression process will lead to Equation (4.1), where P represents pressure and the subindexes INT, H and L represent intermediate, high and low pressure respectively; further details about the derivation of this relation may be found in Appendix C.

$$\frac{P_{INT}}{P_L} = \frac{P_H}{P_{INT}} \tag{4.1}$$

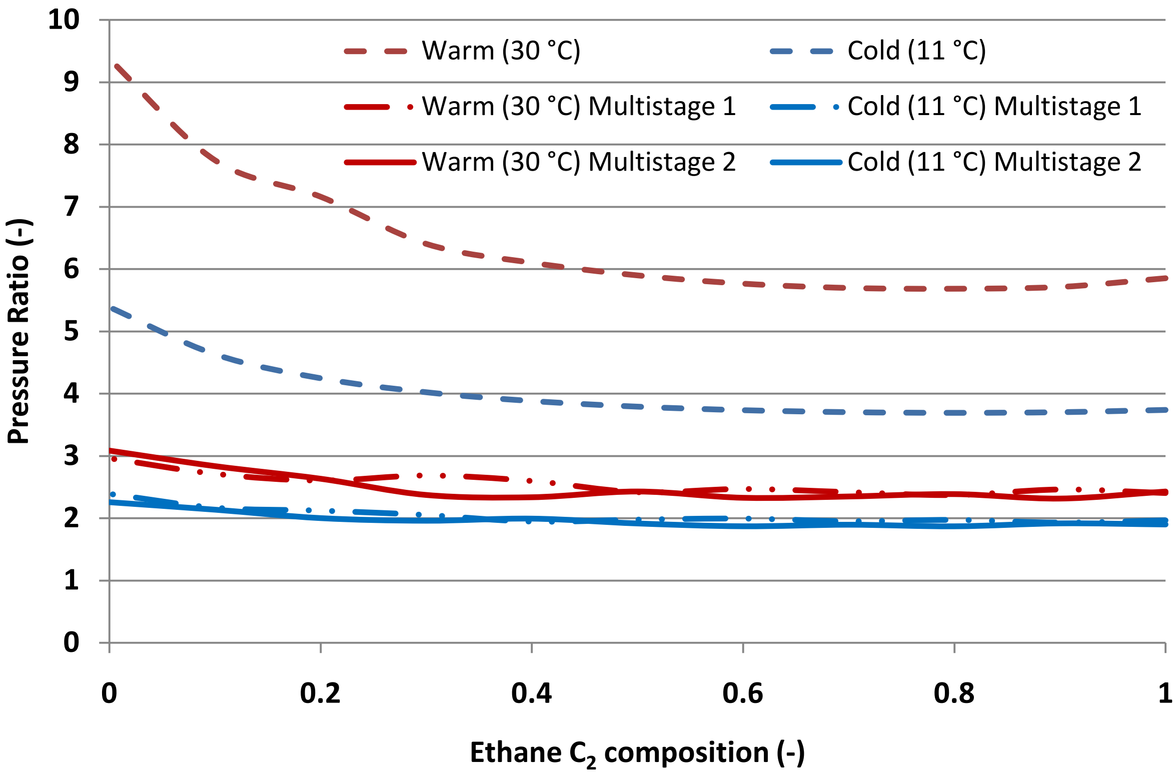

Figure 4-9. Pressure ratio with variation of refrigerant composition, two stage compression

The summary of the results is presented below, the UA value and the volumetric flow are excluded to avoid repetition (may be found in Table 4-2 and Table 4-3). The most suitable values are presented in red font.

Table 4-4. Summary for simple cycle with two stage compression, warm climate condition

C_2 comp. (-)	Duty (MW)	$P_{EXPANSION}$ (bar)	$P_{INTERMEDIATE}$ (bar)	P_{RATIO} (bar)	$P_{COMPRESSION}$ (bar)	P_{RATIO} (bar)
1,00	41,70	8,31	20,00	2,41	48,57	2,43
0,90	30,82	7,51	18,50	2,46	42,84	2,32
0,80	27,45	6,74	16,00	2,37	38,19	2,39

C_2 comp. (-)	Duty (MW)	$P_{EXPANSION}$ (bar)	$P_{INTERMEDIATE}$ (bar)	P_{RATIO} (bar)	$P_{COMPRESSION}$ (bar)	P_{RATIO} (bar)
0,70	24,91	5,99	14,50	2,42	34,06	2,35
0,60	23,42	5,26	13,00	2,47	30,26	2,33
0,50	22,37	4,55	11,00	2,42	26,72	2,43
0,40	21,89	3,85	10,00	2,60	23,36	2,34
0,30	21,40	3,17	8,50	2,68	20,17	2,37
0,20	21,54	2,50	6,50	2,60	17,12	2,63
0,10	22,14	1,84	5,00	2,71	14,18	2,84
0,00	23,61	1,21	4,50	3,71	11,35	2,52

Table 4-5. Summary for simple cycle with two stage compression, cold climate condition

C_2 comp. (-)	Duty (MW)	$P_{EXPANSION}$ (bar)	$P_{INTERMEDIATE}$ (bar)	P_{RATIO} (bar)	$P_{COMPRESSION}$ (bar)	P_{RATIO} (bar)
1,00	13,52	8,31	16,35	1,97	31,07	1,90
0,90	12,38	7,51	14,50	1,93	27,81	1,92
0,80	11,54	6,74	13,30	1,97	24,88	1,87
0,70	10,96	5,99	11,70	1,95	22,18	1,90
0,60	10,48	5,26	10,50	2,00	19,65	1,87
0,50	10,15	4,55	9,00	1,98	17,25	1,92
0,40	9,99	3,85	7,50	1,95	14,95	1,99
0,30	9,91	3,17	6,50	2,05	12,74	1,96
0,20	10,00	2,50	5,30	2,12	10,61	2,00
0,10	10,38	1,84	4,00	2,17	8,54	2,14
0,00	11,12	1,21	2,90	2,39	6,54	2,25

4.1.3. Two stage refrigeration cycle

As mentioned in Chapter 3, the simulation of the two stage refrigeration cycle brings a new degree of freedom to the process, given by the temperature of the natural gas between the first and second stage of the process. Therefore the study in this section is made for variation in composition and different intermediate temperatures of the natural gas. The analysis for all the results will be made first based on the warm climate condition. Figure 4-10 shows the compressor duty with variation of the refrigerant composition. The change in the intermediate temperature of the cycle does not affect significantly the compressor duty; the difference is almost undetectable in the figure.

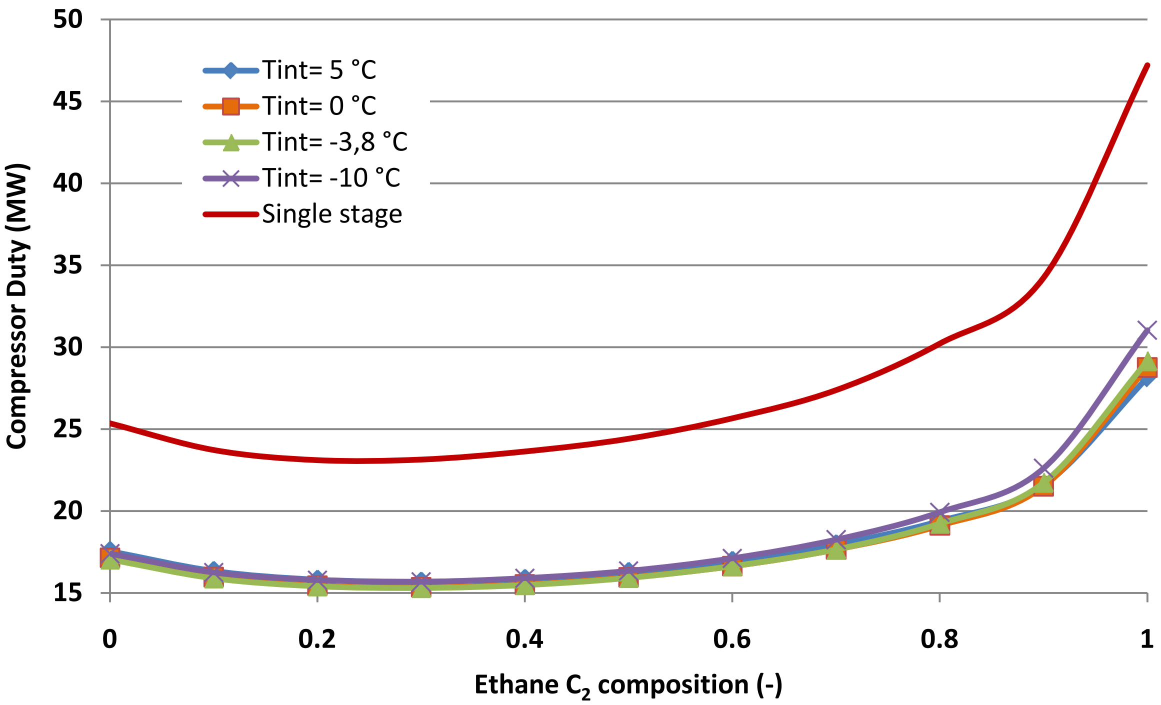

Figure 4-10. Compressor duty for two stage cycle, warm climate

What is remarkable from the figure above is the improvement achieved in terms of duty when comparing with a single stage cycle. The trend of the curve against variation of the refrigerant composition is similar for one and two stage cycle, however the offset between them provides the improvement. The duty for the single stage process is reduced between 32 and 38% by the use of a two stage cycle, depending on the refrigerant composition.

The UA value for this case is also an important variable to be studied. The sum of the UA values for the heat exchangers involved in a multistage cycle is supposed to be larger than the one for a single stage process. The reason behind this affirmation is that the temperature difference that is achieved in the multistage cycle is smaller than the single stage cycle (may see Figure 2-14), hence the heat transfer UA value is increased for the same heat load, according to Equations (2.6) and (2.7). The predicted behavior for the UA value is obtained through the simulations and shown in Figure 4-11; as for the compressor duty, the UA value is not significantly affected by the variation in the intermediate natural gas temperature.

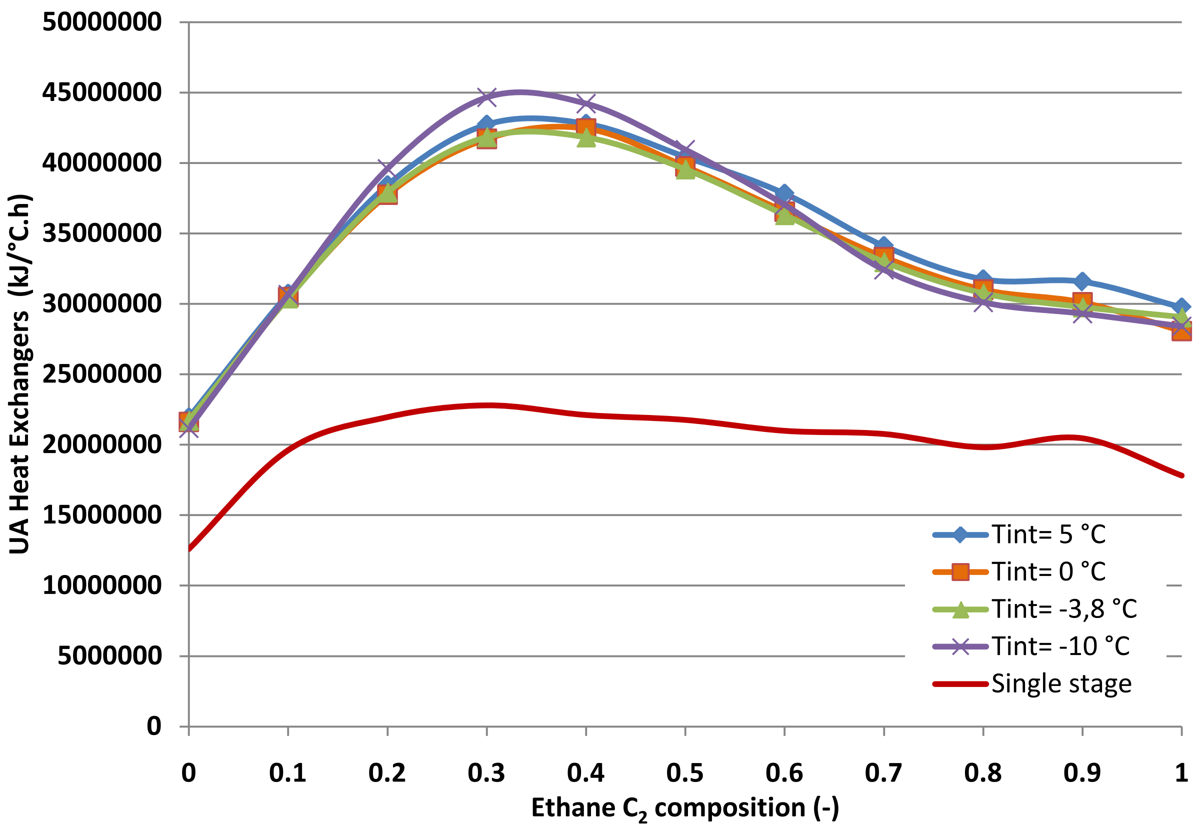

Figure 4-11. Heat exchangers total UA for two stage cycle, warm climate

The other two parameters to be taken into account for this case are the pressure ratio and the volumetric flow in the compressor suction. The pressure ratio behaves basically as the two stage compression case, with the exception that the intermediate pressure in this case is not subject to Equation (4.1) but to the temperature of the natural gas between the two stages. The curves for different intermediate temperatures and varying composition are shown in Figure 4-12; the dashed lines represent the second stage of the cycle. The main aspect about changing the intermediate temperature is the difference between the pressure ratio in the first stage and the second stage; for instance when Tint=-3,8 °C (green series) the dashed line is close the regular line, whilst for the case in which Tint= 5 °C (blue series) the values are far from each other.

Since varying the intermediate temperature of the natural gas stream has an effect on the pressure ratio of the compressor, it is important to evaluate which of the intermediate temperatures is more beneficial for the purpose of this study. As mentioned above, for Tint= -3,8 °C the values for the first and second cycle are almost the same, this is the best case because an equal split in the pressure ratio means that the process can remain within the design limitations for higher capacities.

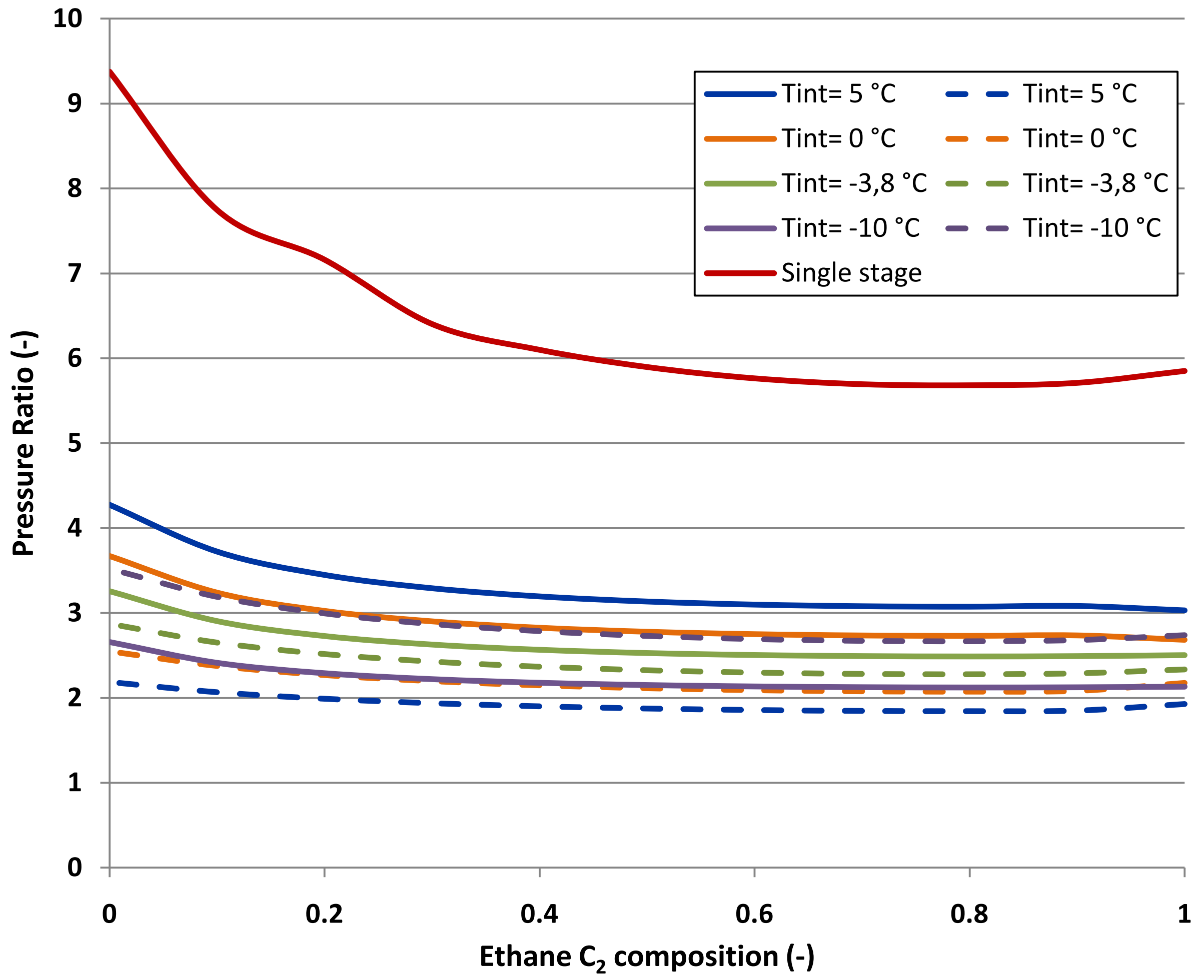

Figure 4-12. Pressure ratio for two stage cycle, warm climate

The same considerations set for the pressure ratio may be applied to the volumetric flow, which is shown in Figure 4-13. Since the curves are not distinguishable when plotted in the same frame, different frames are used to plot the first stage and second stage compressor flow, left and right frame respectively. The color sequence in the right side figure is exactly the opposite of the left side frame, for instance the value of Tint = 5 gives the lower volumetric flow for the first compressor, but the higher for the second.

As mentioned for the pressure ratio, a "sweet point" would be one in which the volumetric flows are equal for both compressors, providing higher capacity for the process. The best point from the simulated ones is at Tint = -3,8 °C as for the pressure ratio. A noteworthy aspect about the intermediate temperature is that the ideal point was found around halfway of the total heat to be removed from the stream; in this case the natural gas is cooled from 30 to -36 °C and the halfway temperature is -3 °C.

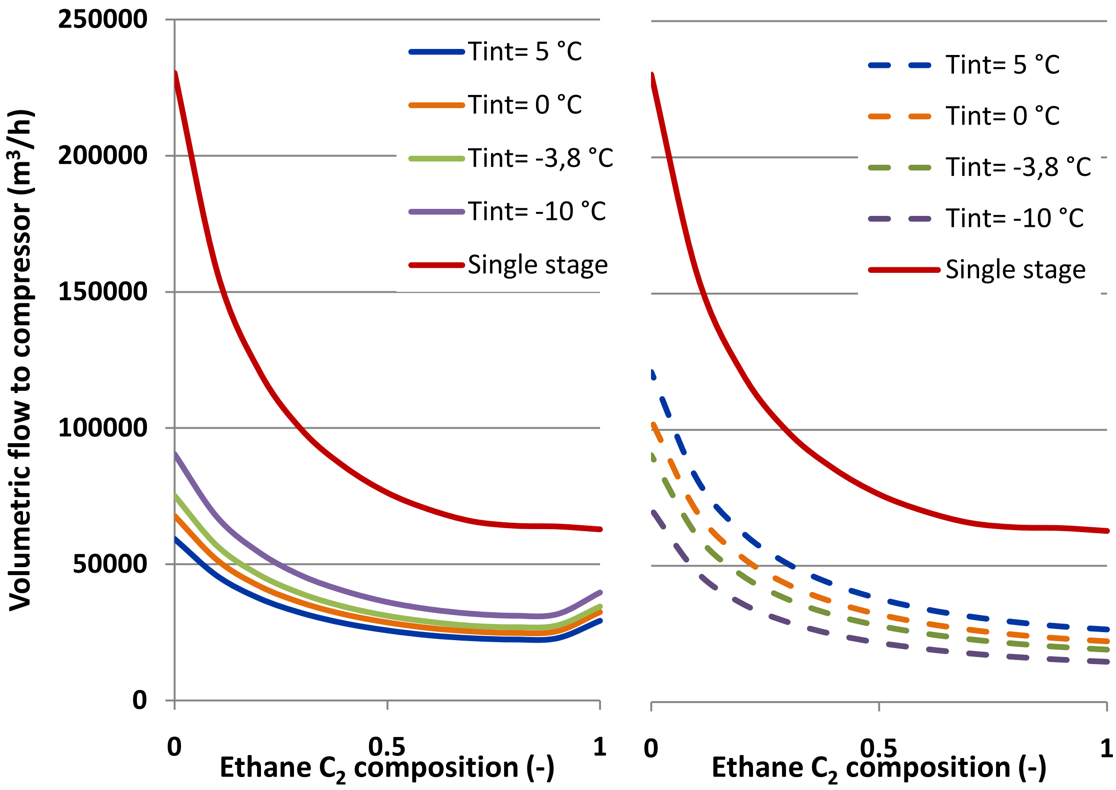

Figure 4-13. Compressors suction volume for two stage cycle, warm climate. Left figure: First stage compressor. Right figure: Second stage compressor

The cold climate condition results show the same behavior than the warm climate and hence the analysis from the latter may be applied to the first, nevertheless some differences are introduced. In order to see these differences and avoid repetition, the values under the cold climate condition for the best intermediate temperature; which was found to be -10 °C (around halfway from 11 to -36 °C) are compared with the warm climate condition at the best intermediate temperature, -3,8 °C as given above. The comparison is shown in Figure 4-14 and Figure 4-15 for the compressor duty and heat exchangers UA value respectively.

The compressor duty has the same behavior shown and explained for the simple stage cycle, the cold condition temperature range fits better with the refrigerant heating curve and therefore the curve has a smother variation for different refrigerant compositions. On the other hand, the UA value does not follow the trend shown for the single stage cycle. In this case the cold climate curve exceeds the warm climate one in the compositions where compressor duty is minimal (between 0,2 and 0,6 C_2). This can be explained by looking at Figure 4-16 and Figure 4-17; as said before the UA value is directly proportional to the heat load and inversely proportional to the temperature difference through the heat exchanger. Since the heat load for the cold climate is the smaller, the temperature difference in the heat exchanger for the cold climate is small enough to overpass the UA value obtained in the warm climate condition.

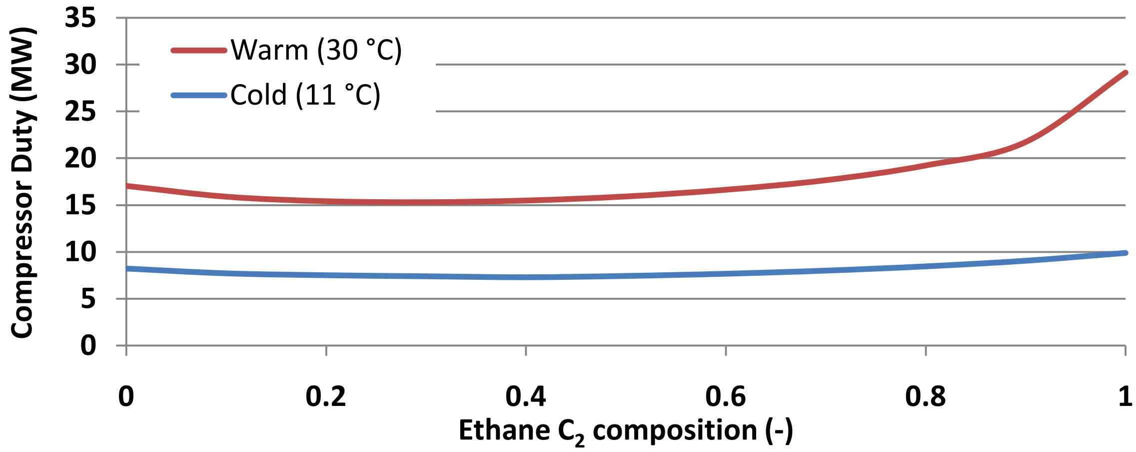

Figure 4-14. Compressor duty for two stage cycle, cold and warm climate comparison

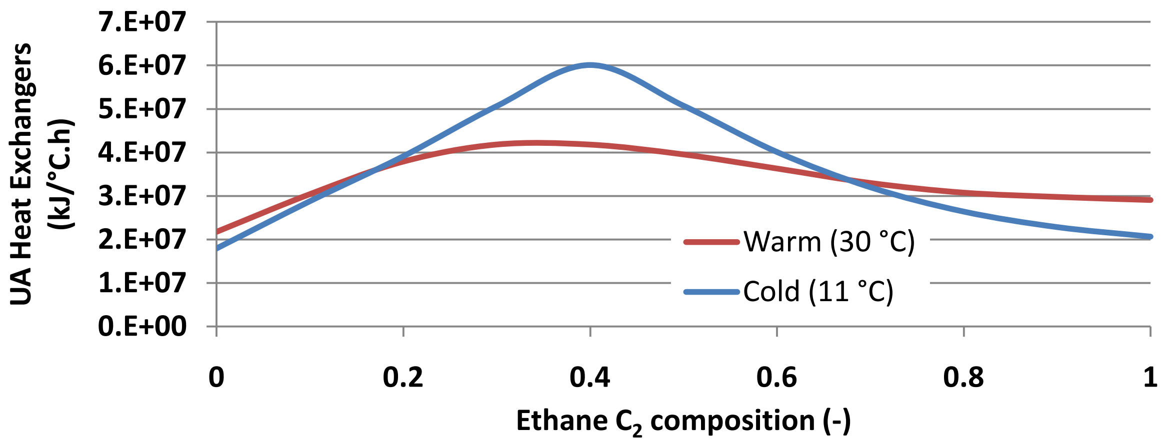

Figure 4-15. Heat exchangers UA for two stage cycle, cold and warm climate comparison

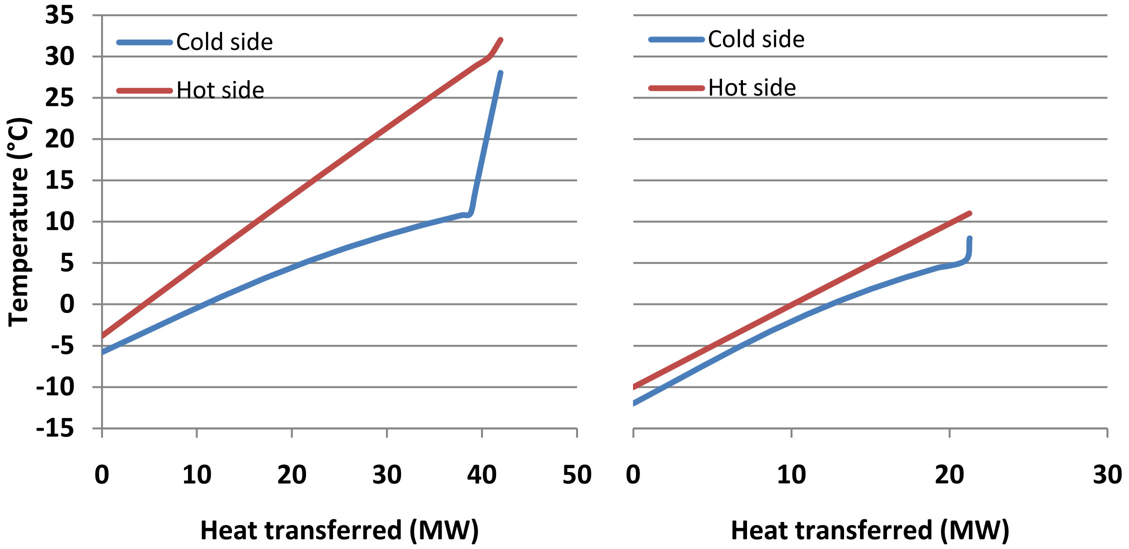

Figure 4-16. Temperature profile across first stage heat exchanger, mixture 0,4 C_2/0,6 C_3 Left figure: warm climate. Right figure: cold climate

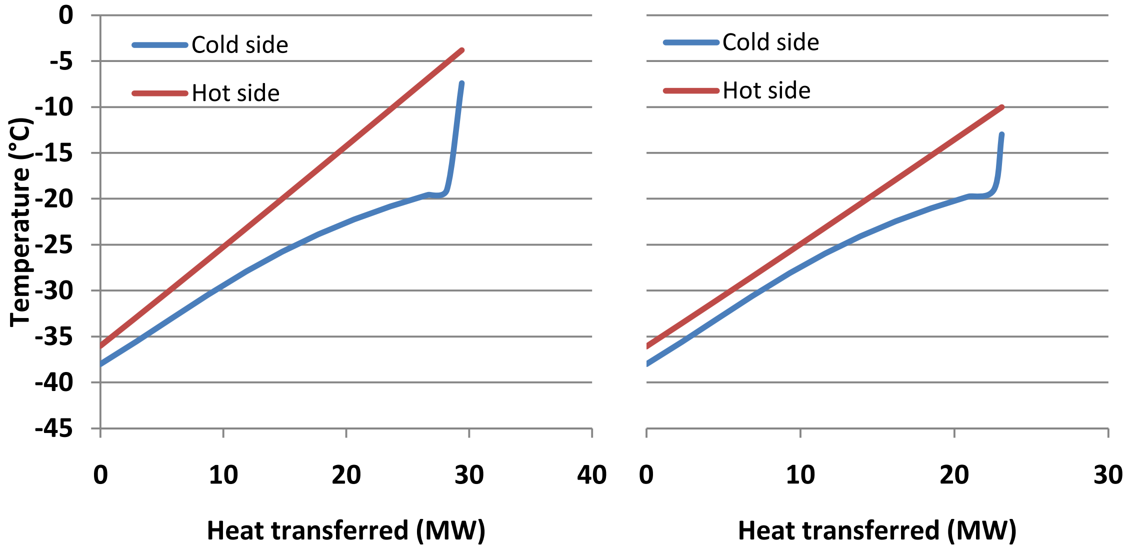

Figure 4-17. Temperature profile across second stage heat exchanger, mixture 0,4 C_2/0,6 C_3 Left figure: warm climate. Right figure: cold climate

The compressor suction volume and pressure ratio are also compared with the warm climate condition; the results are shown in Figure 4-18 and Figure 4-19 respectively, the dashed lines represent the first stage while the regular curves are for the second stage compressor. As for the previous cases the cold climate condition shows a similar trend to that of the warm case, with a slight reduction.

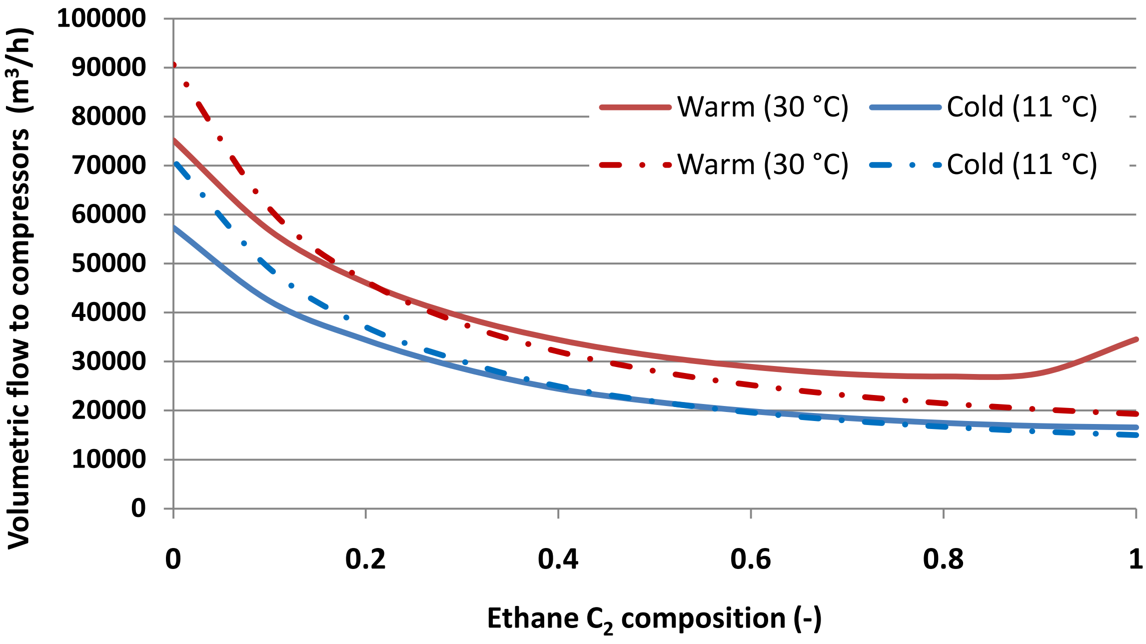

Figure 4-18. Compressor suction volume for two stage cycle, cold and warm climate comparison

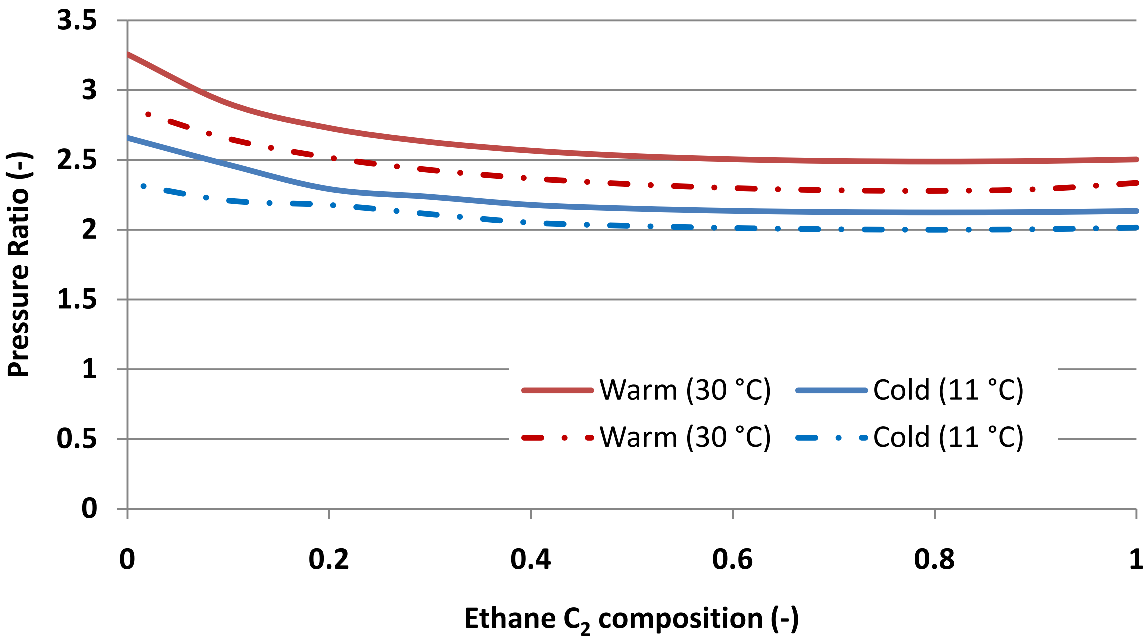

Figure 4-19. Compressors pressure ratio for two stage cycle, cold and warm climate comparison

The best intermediate temperatures mentioned above, -3,8 °C for the warm climate and -10 °C for the cold climate condition are used to provide a tabular summary for the different compositions. This is presented in Table 4-6 for the warm case and in Table 4-7 for the cold climate. The red colored rows represent the chosen most suitable compositions.

Table 4-6. Summary for two stage cycle, warm climate, Tintermediate -3,8 °C.

C2 comp. (-)	Duty (MW)	Vol. Flow C_1 (m3/h)	Vol. Flow C_2 (m3/h)	P_1 (bar)	P_2 (bar)	P_3 (bar)	UA_1 (kJ/°C.h)	UA_2 (kJ/°C.h)
1,00	29,16	34552,47	19314,71	8,31	20,80	48,57	20526828,8	8533230,2
0,90	21,73	27633,30	20252,15	7,51	18,71	42,84	19195911,0	10573528,9
0,80	19,23	26972,58	21478,61	6,74	16,76	38,19	19159316,3	11589032,4
0,70	17,66	27453,42	23087,81	5,99	14,92	34,06	19691979,6	13284596,0
0,60	16,64	28937,81	25211,96	5,26	13,17	30,26	21233478,4	15079758,8
0,50	15,92	31152,82	28073,34	4,55	11,49	26,72	22364593,9	17192969,3
0,40	15,48	34442,27	32021,69	3,85	9,87	23,36	22963366,4	18862745,6
0,30	15,30	39159,41	37711,64	3,16	8,31	20,17	22471705,3	19374193,5
0,20	15,41	46076,65	46427,95	2,49	6,80	17,12	20260247,8	17658292,2
0,10	15,89	56776,92	61192,16	1,84	5,35	14,18	16868559,7	13539050,6
0,00	17,04	75181,92	90619,97	1,21	3,95	11,35	12669919,6	9052987,8

Table 4-7. Summary for two stage cycle, cold climate, Tintermediate -10 °C.

C2 comp. (-)	Duty (MW)	Vol. Flow C_1 (m3/h)	Vol. Flow C_2 (m3/h)	P_1 (bar)	P_2 (bar)	P_3 (bar)	UA_1 (kJ/°C.h)	UA_2 (kJ/°C.h)
1,00	9,90	16551,84	14970,17	8,31	17,73	31,07	11206751,4	9423085,4
0,90	9,07	16828,72	15689,64	7,51	15,96	27,81	12306244,3	10516420,5
0,80	8,47	17468,99	16660,57	6,74	14,31	24,88	14021915,3	12359582,2
0,70	8,01	18458,58	17912,70	5,99	12,74	22,18	17288679,4	14690754,5
0,60	7,68	19855,63	19581,45	5,26	11,23	19,65	21861108,8	18226553,4
0,50	7,45	21777,23	21830,76	4,55	9,78	17,25	27736485,3	22975783,8
0,40	7,31	24426,68	24929,88	3,85	8,38	14,95	32214037,4	27855539,9
0,30	7,41	28582,99	30029,36	3,10	6,94	12,74	26111236,2	24516131,5
0,20	7,52	34396,02	37021,47	2,45	5,60	10,61	19054434,9	20044649,5
0,10	7,72	42339,36	48940,76	1,80	4,45	8,54	14911815,0	13859152,0
0,00	8,24	57324,21	71025,63	1,21	3,22	6,54	9003577,8	8906776,3

4.1.4. Three stage refrigeration cycle using propane

In this section a three stage cycle is implemented, as explained in Chapter 3 the refrigerant composition variation is not studied, but the pressure levels of the cycles in order to find the one that provides the best improvements for the process. The study will be shown first for the warm climate, Figure 4-20 and Figure 4-21 represent the compressors duty and the heat exchangers UA values respectively, notice that the values in the figures are the temperatures of the natural gas and not the pressure levels. The pressure levels are related to the temperature of the natural gas after each cycle by the temperature difference set in the heat exchanger, for example given 1°C temperature difference in the heat exchanger, if the second stage temperature is -15,6 °C, the propane saturation pressure at -16,6 °C is the pressure level for the second stage.

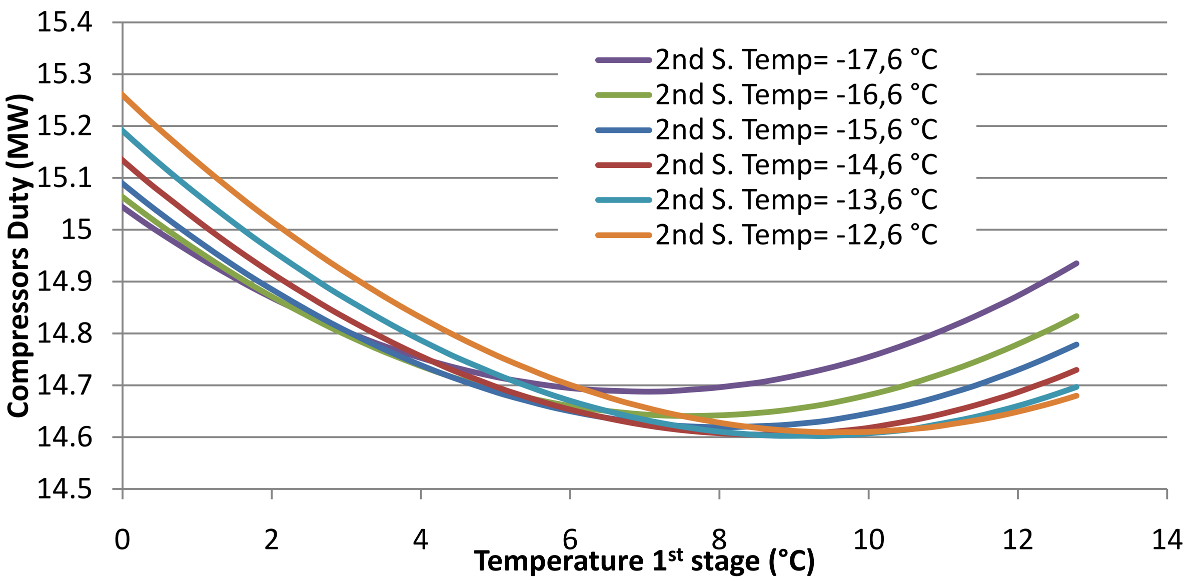

Figure 4-20. Compressors duty for three stage cycle with propane, warm climate

The variation of the temperature between the stages does not represent a significant change in those two values, for instance the minimum compressor duty is only 3% smaller than the maximum for a given composition. However, it is important to note that, as for the previously explained cases, the minimum work is linked to the higher UA for the heat exchanger and that this value is achieved by selecting 8 °C as the temperature for the first stage and -14,6 °C for the second stage. Notice that these values are fairly close to the ones obtained by splitting the heat load of the natural gas steam in three stages (30/**8 °C**, 8/**-14 °C**, -14/**-36 °C**). The volumetric flows into the compressor are shown in Figure 4-22 for the first stage and Figure 4-23 for the second and third stage.

It is important to notice from Figure 4-20 that for any value of the first and second stage temperature, the compressor duty is lower than the minimum value for a two stage cycle, which was found to be 15,3 MW (see Table 4-6).

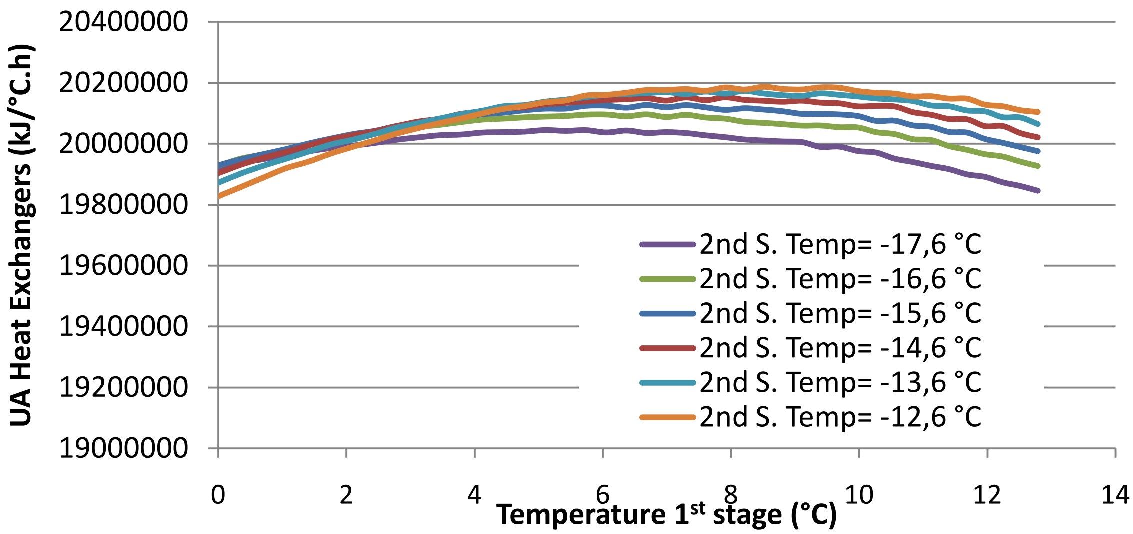

Figure 4-21. Heat exchangers UA value for three stage cycle with propane, warm climate

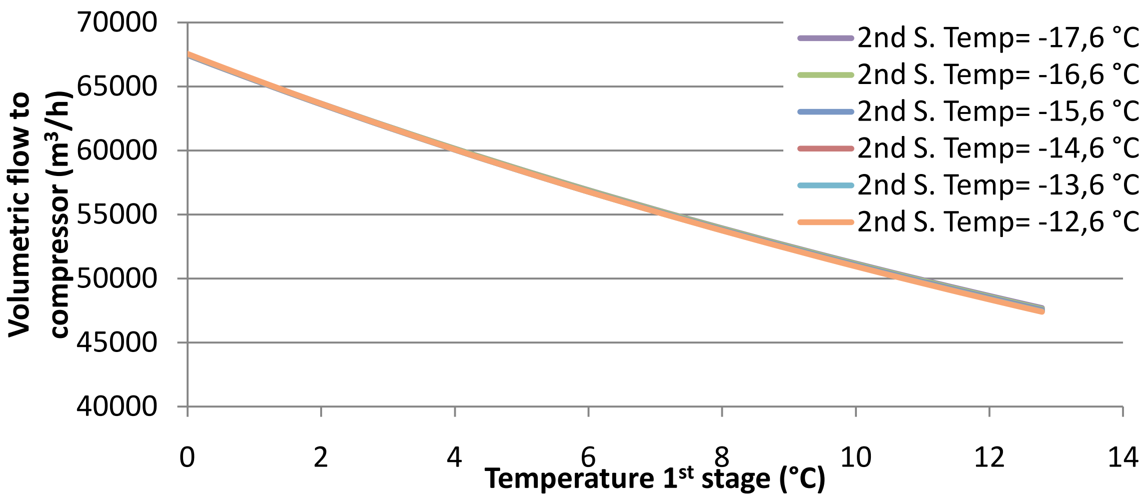

Figure 4-22. Compressor suction volume for three stage cycle with propane, first stage, warm climate

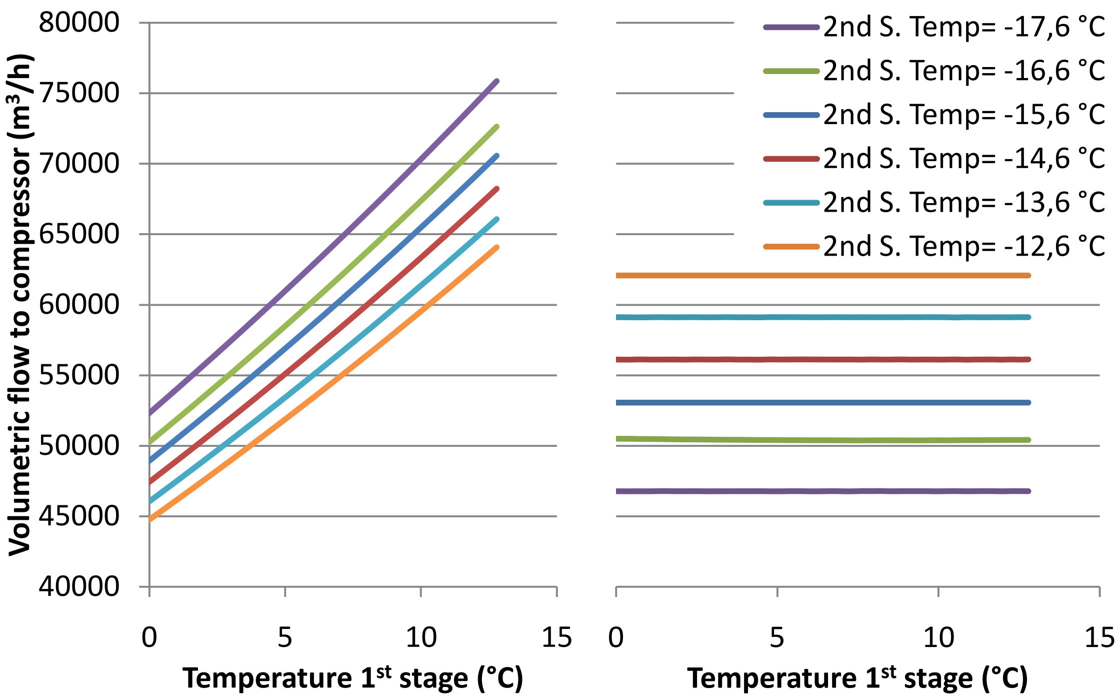

Figure 4-23. Compressors suction volume for three stage cycle with propane, warm climate Left figure: Second stage. Right figure: Third stage

The temperature of the natural gas after the second stage does not affect the first stage compressor; hence all the curves are overlapped in Figure 4-22 and are only affected by the variation of the first stage temperature. The same analysis may be made for Figure 4-23, where the third stage compressor (right frame) is not affected by the first stage temperature but only by the second stage and the curve shows a constant value for different values of the first stage temperature. The second stage compressor, on the other hand, is affected by both modifications and the curve trend clearly demonstrates it.

An optimal configuration of the cycle to achieve the largest capacity would be, as mentioned before, that the three compressors handle the same volumetric flow. For the temperatures that give the minimum work, 8 °C for the first and -14,6 °C for the second stage (stated above), the volumetric flows are almost equal, however the best split is obtained by setting the first cycle temperature equal to 6 °C (with -14,6 °C for the second stage) as might be seen in the figures above.

The results for the cold climate condition show the same behavior that has been shown and explained above, with a slight reduction in all the parameters (i.e.:

compressor duty, UA value and suction volume). However, in order to avoid a lack of information the figures with the cold climate results are presented below. For this case the most suitable values were found to be -4,2 °C for the first and -20 °C for the second stage.

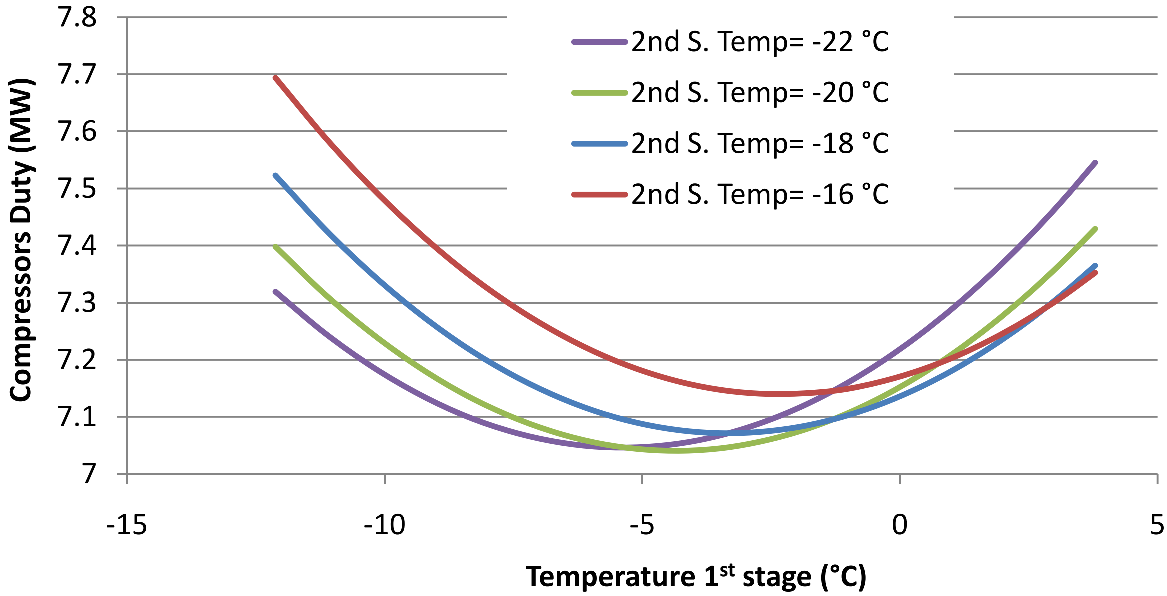

Figure 4-24. Compressors duty for three stage cycle with propane, cold climate

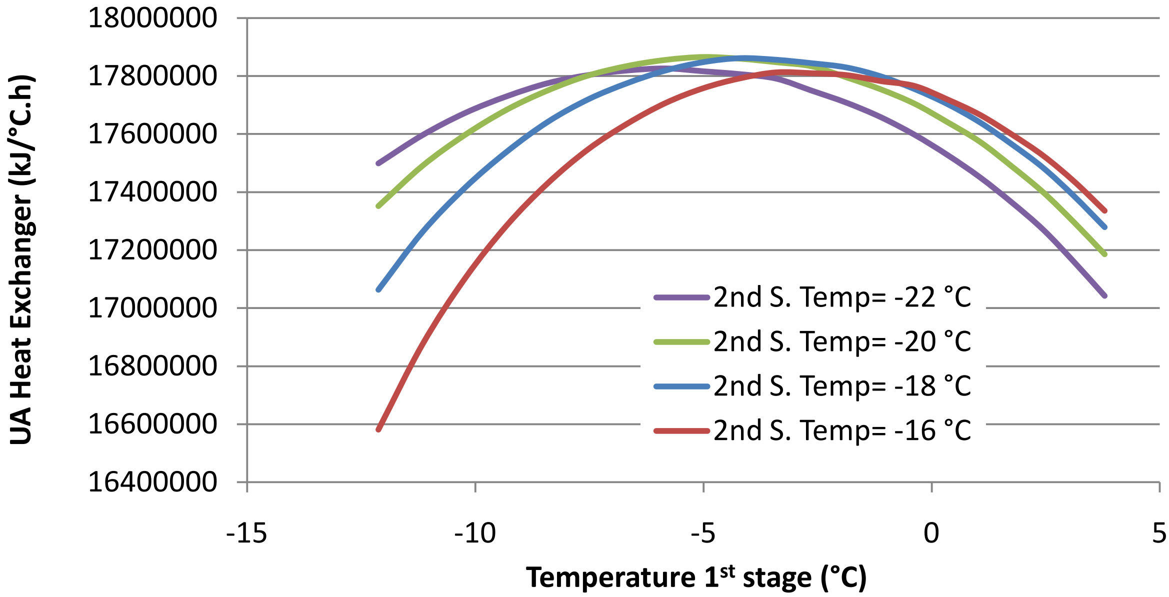

Figure 4-25. Heat exchangers UA value for three stage cycle with propane, cold climate

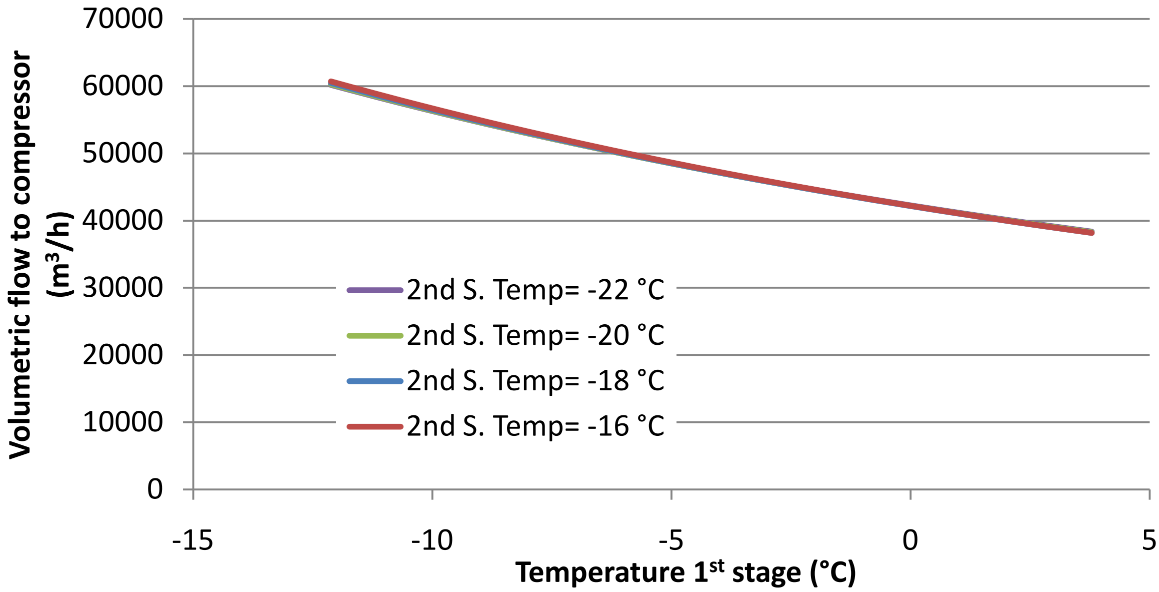

Figure 4-26. Compressor suction volume for three stage cycle with propane, first stage, cold climate

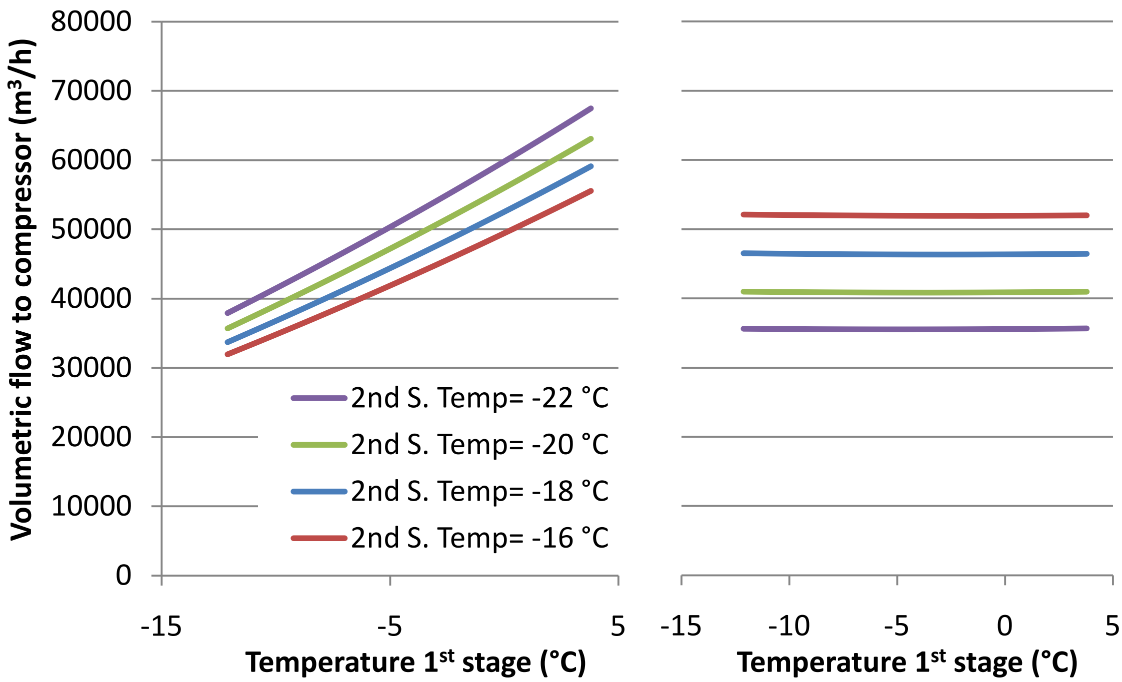

Figure 4-27. Compressors suction volume for three stage cycle with propane, cold climate Left figure: Second stage. Right figure: Third stage

4.2. Application of the different configurations to the C_3MR process

In this section the most suitable values for each configuration studied in Section 4.1 are implemented in an entire C_3MR process with a different precooling configuration in order to see the real limitations and actual values of the improvements. The flowrates of precooling refrigerant in each case are different from the stand-alone cycles due to the increase in heat load; hence these values are presented below in Table 4-8 for the warm climate.

Table 4-8. Conditions for different modifications in the precooling cycle for C_3MR process, warm climate

Three stage C_3

Temperature NG after 1st stage: 8,0 °C **Temperature NG after 2nd stage:** -14,6 °C
Molar flow to NG cycle: 13800 kgmole/h **Molar flow to MR cycle:** 70500 kgmole/h

Single stage MR

MR composition: 0,3 C_2/ 0,7 C_3 **Molar flow:** 78500 kgmole/h

Single s. MR. Multistage compression

MR composition: 0,3 C_2/ 0,7 C_3 **Molar flow:** 78500 kgmole/h
$P_{INTERMEDIATE}$: 8,5 bar

Two stage MR

MR composition: 0,3 C_2/ 0,7 C_3 **$T_{INTERMEDIATE}$ NG:** -3,8 °C
Molar flow 1st stage: 39000 kgmole/h **Molar flow 2nd stage:** 47000 kgmole/h

Two stage MR, n-butane included

MR composition: 0,2482 C_2/ 0,6416 C_3/ 0,6416 n-C_4 **$T_{INTERMEDIATE}$ NG:** 0 °C
Molar flow 1st stage: 32300 kgmole/h **Molar flow 2nd stage:** 47000 kgmole/h

As stated in Chapter 3 a supplementary consideration is studied in this particular case for the warm climate condition; the addition of n-butane to the mixed refrigerant. This is made taking the optimal composition from a previously optimized process [58], the intermediate temperature reported was used for this case.

The evaluation in this section is made by implementing the process with the previously given parameters and configurations in the precooling circuit and assess the differences between the configurations. The variables to be studied are the compressor duty, heat exchanger UA and volumetric flow at the suction of the compressor. The pressure ratio is not taken into account and may be found in the previous section because it is not affected by the heat load variation. Figure 4-28 shows the total compressor duty in Megawatts for the five configurations.

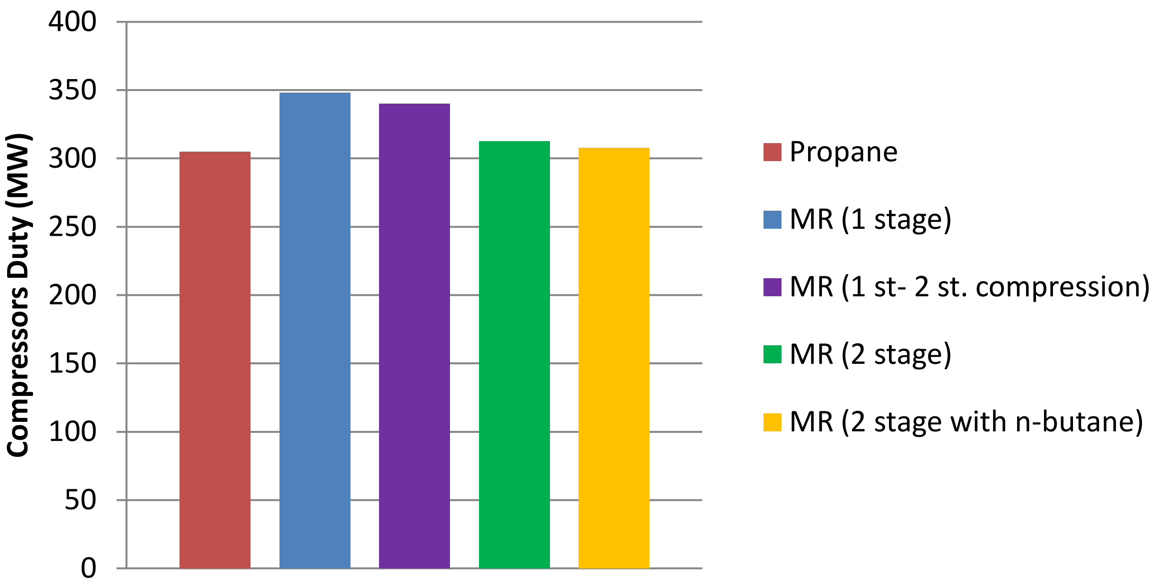

Figure 4-28. Compressor duty for C_3MR process with modifications applied, warm climate

As expected, the compressor duty, which is equivalent to the power requirement of the process, is lower for the propane precooled process. It is important to notice that, taking as a reference the propane case, the mixed refrigerant configuration with n-butane gives a power consumption only 0,88 % larger, while the two stage cycle without n-butane shows an increase of 2,53 %. The single stage cycle is 14,1 % and 11,5 % larger for the regular and multistage compression respectively. The share of the power consumption between the two cycles of the process is shown in Figure 4-29.

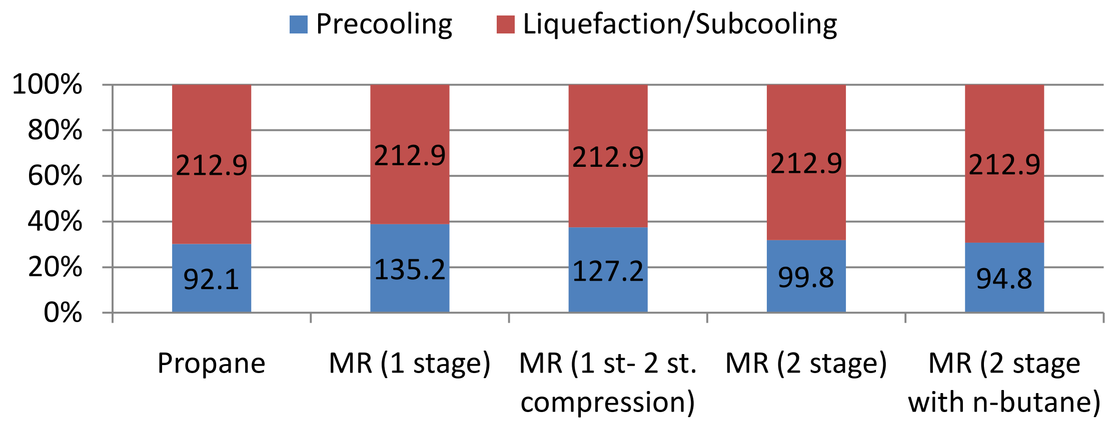

Figure 4-29. Power share between precooling and liquefaction cycle, warm climate

Besides the power consumption of the process, the UA value of the heat exchangers is an important criterion for the selection; the values for this parameter are shown in Figure 4-30. All the mixed refrigerant configurations give a higher UA value than the propane case. What is remarkable from the figure is the enormous value when n-butane is added to the two stage cycle; it is 3,37 times the propane case UA value, which is the lowest among the studied configurations. The two stage cycle using only C_2 and C_3 gives, on the other hand, a value 2,13 times that of propane.

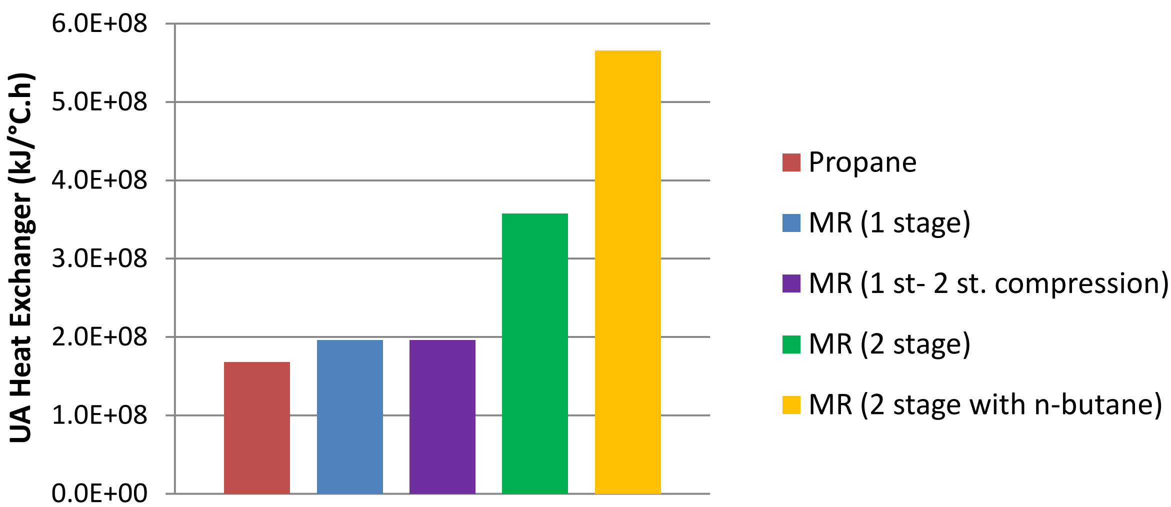

Figure 4-30. Heat exchangers UA for C_3MR process with modifications applied, warm climate

The volumetric flow at the suction of the compressor is also studied; this parameter is important because defines the maximum capacity of a given configuration. The capacity for the studied case is 60000 kgmole/h, which for a 315 days yearly operation is equivalent to 8,4 Megatonnes of LNG per annum (MTPA). This value is higher than the maximum single train capacity installed by 2012, and is used purposely to have a clearer scenario of the differences and limitations.

Since the entire C_3MR process is studied, the suction volume of the liquefaction/subcooling compressors might be the constraint, if it is larger than the value for the precooling circuit compressor. The suction volumes for the precooling compressor are shown in Figure 4-31 and for the liquefaction/subcooling cycle it is the same in all cases: 428437,8 m^3/h (the highest of the three compressors in the train). From the figure below using a single stage mixed refrigerant cycle for the precooling does not provide any improvement when referencing to a three stage precooling cycle. Nevertheless a two stage mixed refrigerant cycle gives a volumetric flow 24,5 % lower if n-butane is not part of the mixture, otherwise the reduction is only 8,1 %. Only the single stage mixed refrigerant precooling configurations (regular and with two stage compression) have a larger volumetric flow than the liquefaction/subcooling compressor, therefore only for those two cases the constraint in capacity is set by the precooling cycle.

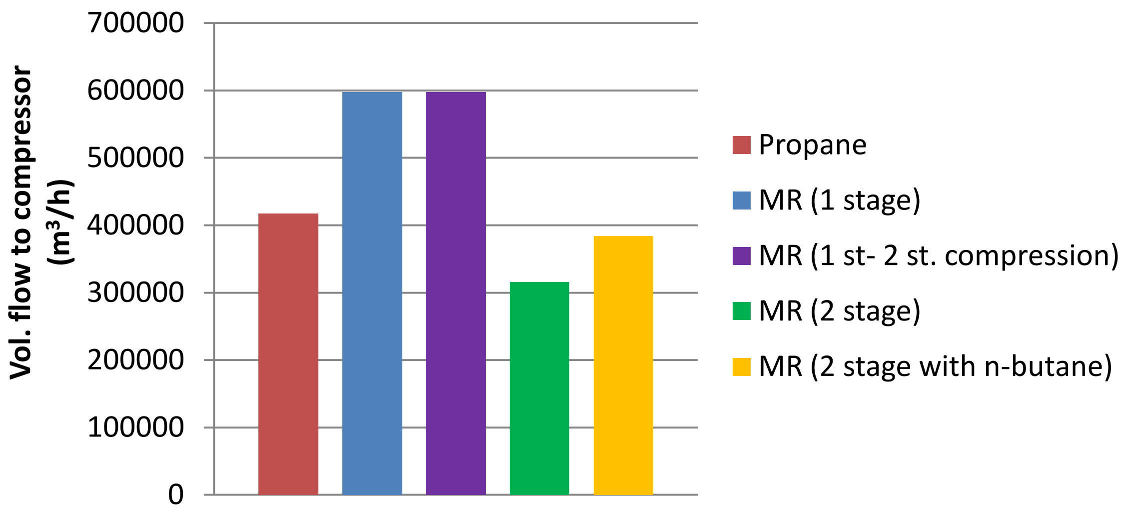

Figure 4-31. Precooling compressor suction volume for C_3MR process with modifications applied, warm climate

A tabular summary of the variables studied for the C_3MR process is given in Table 4-9. From the results shown above an important conclusion is reached; for the warm climate condition using a mixed refrigerant in the precooling cycle, instead of a three stage propane circuit for the C_3MR process, does not benefit the process in terms of energy efficiency, heat exchanger UA value or capacity of the process; the latter because the capacity limitation when using a propane precooled process is not provided by the precooling compressor but by the liquefaction/subcooling units. Nevertheless, the complexity of the process is indeed reduced by the use of a mixed refrigerant in the precooling, as well as the number of main unit operations required, see Table 4-10.

Table 4-9. Summary of the results for the C_3MR cycle with modifications, warm climate

Precooling configuration	Duty (MW)	UA (kJ/°C.h)	Vol. Flow (m^3/h)	Massflow (kg/h)	COP (-)
Three stage propane	305,0	167700003,8	417746,3	3743791,2	0,78
MR (1 stage)	348,1	196300245,5	597749,6	3131276,3	0,68
MR (1 st- 2 st. compression)	340,2	196310156,8	597744,1	3131276,3	0,70
MR (2 stage)	312,8	357782520,2	315740,5	3430442,8	0,76
MR (2 stage with n-butane)	307,7	565821996,2	384204,3	3339282,0	0,77

Table 4-10. Main process equipment count, C3MR cycle with modifications

Precooling configuration	Compressor	Process heat exchanger	Condenser Intercooler	Sum
Propane	4	7	4	15
MR (1 stage)	4	2	4	10
MR (1 st- 2 st. compression)	5	2	5	12
MR (2 stage)	4	3	4	11
MR (2 stage with n-butane)	4	3	4	11

Furthermore, as for the warm climate case, the implementation of the different configurations into the C_3MR cycle for the cold climate condition is done by using the parameters chosen in the previous section as most suitable and the values shown in Table 4-11.

Table 4-11. Conditions for different modifications in the precooling cycle for C3MR process, cold climate

Three stage C_3	**Temperature NG after 1st stage:** -4,2 °C; **Temperature NG after 2nd stage:** -20,0 °C; **Molar flow to NG cycle:** 8550 kgmole/h; **Molar flow to MR cycle:** 51700 kgmole/h
Single stage MR	**MR composition:** 0,3 C_2/ 0,7 C_3; **Molar flow:** 59500 kgmole/h
Single s. MR, multistage compression	**MR composition:** 0,3 C_2/ 0,7 C_3; **Molar flow:** 59500 kgmole/h; **$P_{INTERMEDIATE}$:** 6,5 bar
Two stage MR	**MR composition:** 0,4 C_2/ 0,6 C_3; **$T_{INTERMEDIATE}$ NG:** -10 °C; **Molar flow 1st stage:** 24300 kgmole/h; **Molar flow 2nd stage:** 42000 kgmole/h

The compressor duty for the different configurations is shown in Figure 4-32; taking the propane precooled process as a reference, the two stage arrangement is only 1% larger, while the single stage mixed refrigerant configuration gives a compressor duty which is 5,5 % and 6,7 % larger for the regular and multistage compression train respectively. The difference between the configurations in terms of power consumption is smaller than in the warm climate case, this is shown in Figure 4-33, where the raise compared to the reference is plotted for both climate conditions and the different configurations.

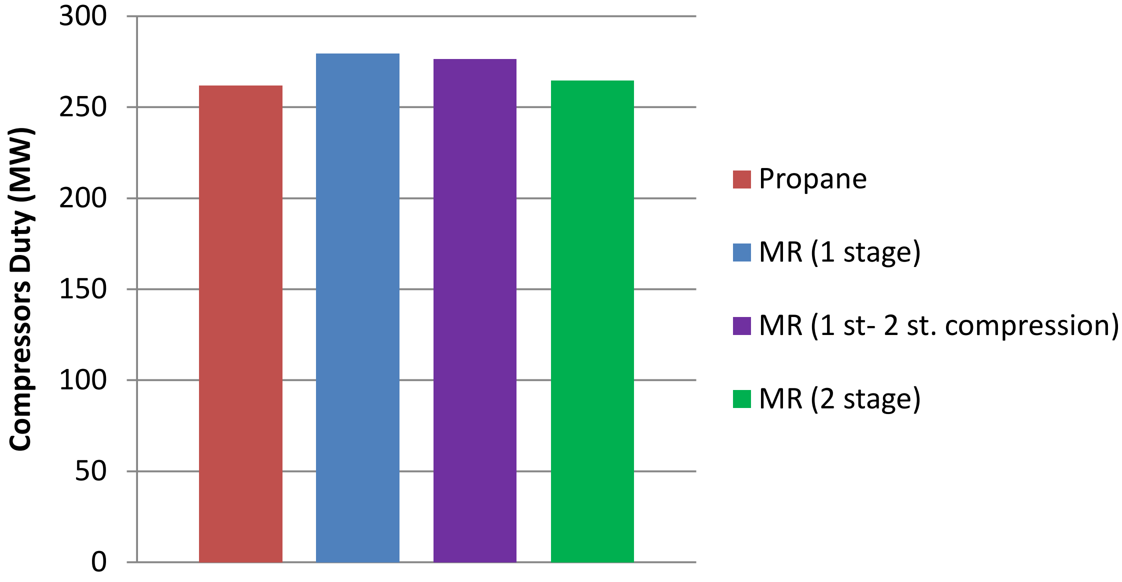

Figure 4-32. Compressor duty for C_3MR process with modifications applied, cold climate

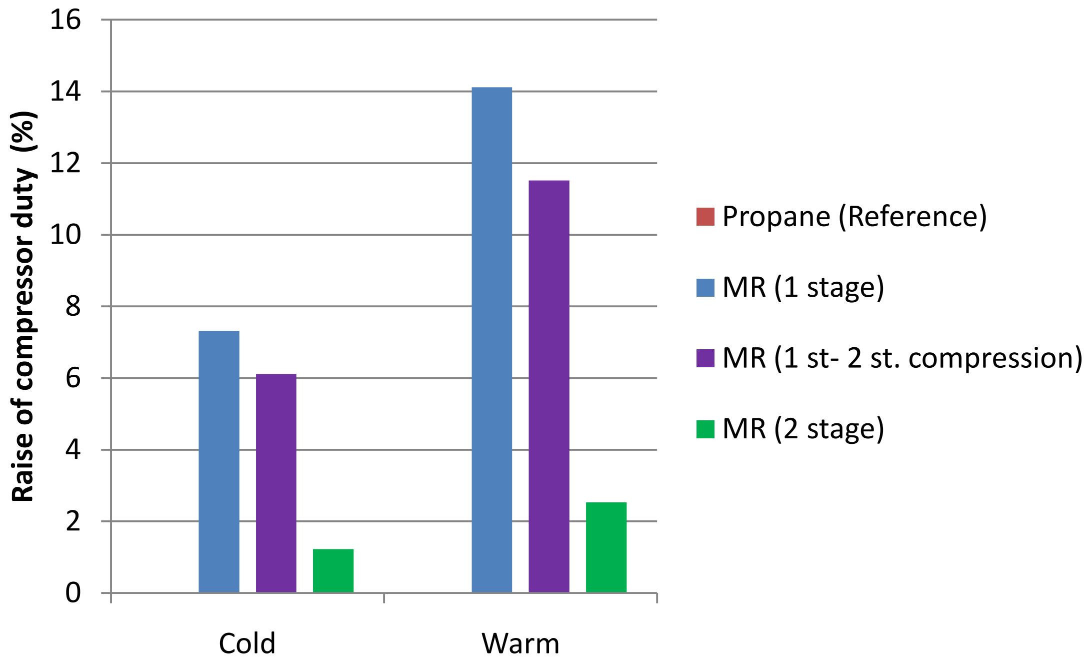

Figure 4-33. Compressor duty variation, comparison of cold and warm climate

Under the cold climate condition the difference is reduced because the mixed refrigerant case has a smaller temperature difference through the heat exchanger, this has been explained in the previous section, see Figure 4-16 and Figure 4-17. The share of the precooling cycle in the process energy consumption is presented below in Figure 4-34. It is important to mention that the share in this case ranges between 20 and 26 percent while for the warm condition it was between 30 and 39 percent. This is basically because the temperature of the precooling stage is fixed to -36 °C, which means that the precooling cycle will represent a higher portion of the process if the ambient temperature is higher and vice versa.

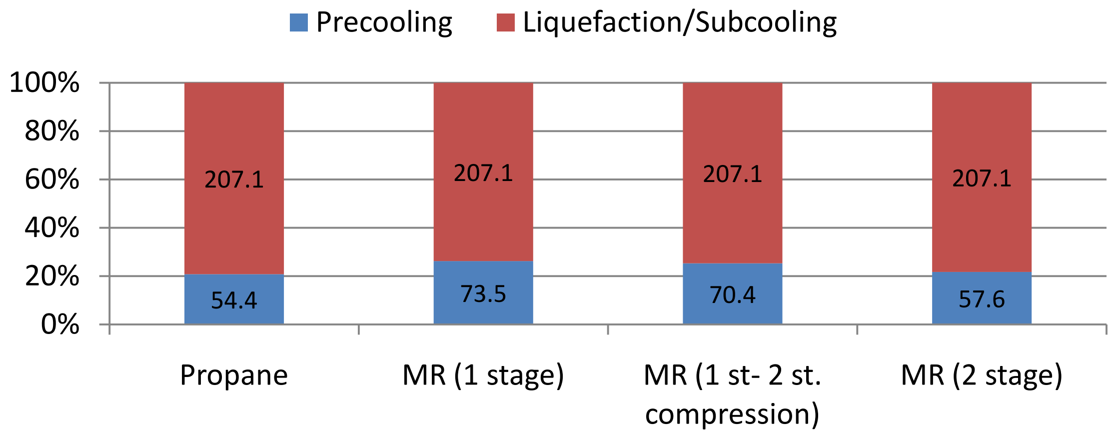

Figure 4-34. Power share between precooling and liquefaction cycle, cold climate

The heat exchanger UA value for the cold climate condition is shown in Figure 4-35, as previously explained, the mixed refrigerant cases have larger UA values than the propane precooled. However, the increase in the UA value for the cold climate is, as opposed to the compressor duty, higher for the cold climate.

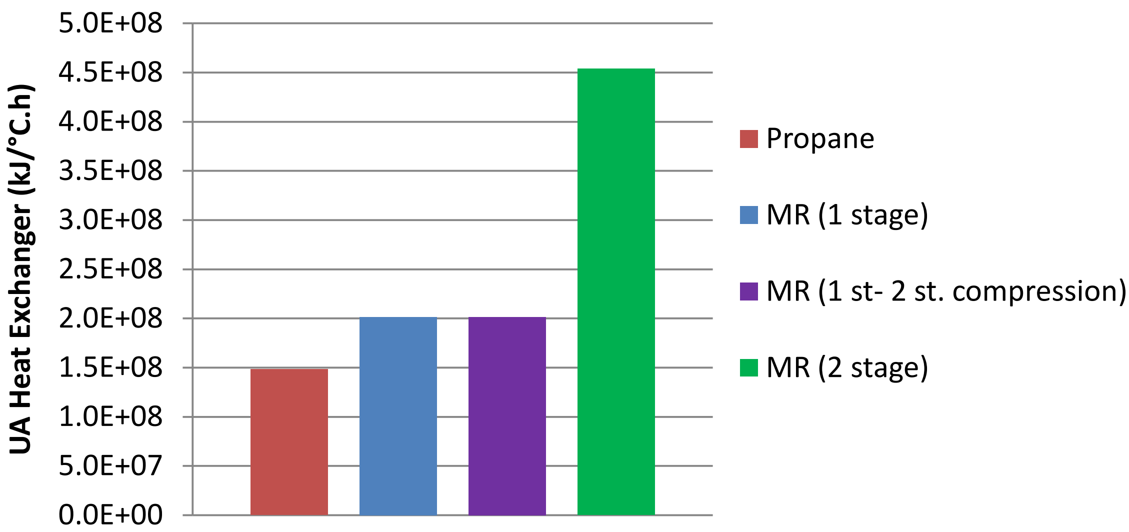

Figure 4-35. Heat exchangers UA for C_3MR process with modifications applied, cold climate

Moreover, the volumetric flow at the compressor suction is shown in Figure 4-36. Using a mixed refrigerant in a two stage cycle gives a significant improvement in terms of volumetric flow; the value is reduced by 44%, if the propane case is taken as the reference. The mixed refrigerant in one stage cycle does not benefit the suction volumetric flow; this value is increased 6%. Since the liquefaction/subcooling circuit is not altered, the volumetric flow of this cycle is the same for all the cases, 428756 m³/h. For the four cases studied the volumetric flow of the precooling compressor does not exceed the value of the liquefaction/subcooling circuit. Therefore the constraint in capacity is not set by the precooling cycle. A tabular summary for the different configurations studied under the cold climate condition is given in Table 4-12.

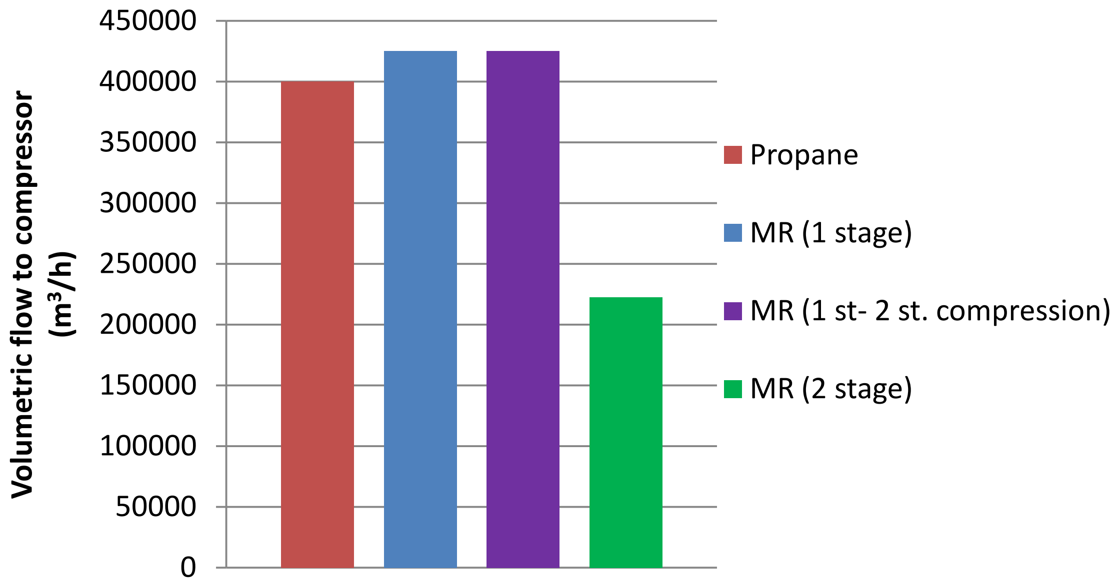

Figure 4-36. Precooling compressor suction volume for C3MR process with modifications applied, cold climate

Table 4-12. Summary of the results for the C3MR cycle with modifications, cold climate

Precooling configuration	Duty (MW)	UA (kJ/°C.h)	Vol. Flow (m^3/h)	Massflow (kg/h)	COP (-)
Propane	261,43	148314276,8	400244,5	2656844,3	0,86
MR (1 stage)	280,54	201570432,6	425114,2	2373387,8	0,80
MR (1 st- 2 st. compression)	277,41	201567905,1	425114,2	2373387,8	0,81
MR (2 stage)	264,64	453994745,8	222483,4	3430442,8	0,85

Based on the variables studied, the selection of a precooling circuit with a mixed refrigerant, whether it is a one or two stage cycle, is not the favored alternative for the cold climate condition. When compared to the propane precooling arrangement (three stages) for a C_3MR process, the only variable that represents a benefit is the volumetric flow to the compressor, which may reduce the size of this equipment; however it is attached to an increase in the heat exchanger UA value, which is correspondingly linked with the heat exchanger size. The capacity of the process in this case is also restricted by the mixed refrigerant cycle and therefore the reduction of the volumetric flow for the precooling does not represent a benefit in terms of capacity increase.

It is noteworthy to remember that the precooling cycle for the cold climate case represents around 20 % of the entire process power requirement. This might be seen as a weak point of this configuration since the precooling equipment, for instance the heat exchangers, cost around 70% less than the ones required for the rest of the liquefaction process; see Section 2.6. Based on this, a precooling cycle that covers a wider range of temperatures may represent an interesting case of study, especially for the cold climate condition; this possibility is addressed in the next sections.

4.3. Mixed refrigerant precooling temperature relocation

In this section a stand-alone cycle to cool the natural gas until -50 °C is studied. As might be seen in Figure 3-7, the ethane composition in a binary mixture (C_2/C_3) has to be higher than 0,13 in order to achieve this goal. Based on the previously done studies the cycle to be implemented is a two stage cycle with an intermediate temperature which is approximately half of the temperature span of the process, the chosen values are -10 °C for the warm and -20 °C for the cold climate case. The single stage cycle configurations were excluded due to less importance for the effect of the evaluation, as has been learnt in the previous sections.

The results for this case are, as expected, similar to the ones obtained for the two stage cycle in the stand-alone studies (Section 4.1). Therefore the analysis of each variable is not presented again, but the tabular summary for the cold and warm climate condition studies is shown below, Table 4-13 contains the results for the warm case and Table 4-14 for the cold condition. The values considered most suitable are in red colored font. Notice that for the warm climate case the volumetric flows for the compressors are far from equal, which means that the chosen intermediate temperature is not the appropriate one. A case with an intermediate temperature of -15 °C is taken into account. The results are shown in Table 4-15, the red colored row represents the most suitable values.

Table 4-13. Summary for two stage cycle (to -50 °C), warm climate, Tintermediate -10 °C

C2 comp. (-)	Duty (MW)	Vol. Flow C_1 (m3/h)	Vol. Flow C_2 (m3/h)	P_1 (bar)	P_2 (bar)	P_3 (bar)	UA_1 (kJ/°C.h)	UA_2 (kJ/°C.h)
1,00	39,68	44699,16	38981,59	5,14	17,73	46,65	23435024	12101936
0,90	33,12	40856,66	40819,35	4,65	15,97	41,23	20686814	12585945
0,80	29,58	40145,59	43378,96	4,17	14,31	36,77	20539249	13714453
0,70	27,26	40971,98	46779,09	3,70	12,74	32,79	21348299	15192629
0,60	25,70	43141,44	51327,41	3,24	11,23	29,12	22320716	17082631
0,50	24,57	46379,74	57512,70	2,79	9,78	25,69	24081289	19435634
0,40	23,87	51253,31	66170,18	2,34	8,38	22,45	25110368	22319304
0,30	23,58	58521,48	78771,21	1,91	7,03	19,36	24687903	23813266
0,20	23,76	69503,11	98692,48	1,47	5,71	16,40	22371813	22325506
0,15	24,04	76936,50	113687,93	1,26	5,07	14,96	20712303	20810496

Table 4-14. Summary for two stage cycle (to -50 °C), cold climate, Tintermediate -20 °C

C2 comp. (-)	Duty (MW)	Vol. Flow C_1 (m3/h)	Vol. Flow C_2 (m3/h)	P_1 (bar)	P_2 (bar)	P_3 (bar)	UA_1 (kJ/°C.h)	UA_2 (kJ/°C.h)
1,00	18,36	30067,72	27071,45	5,14	13,49	31,07	13671148	10031390
0,90	16,79	30469,57	28571,02	4,65	12,17	27,81	14301778	11080979
0,80	15,67	31555,32	30571,59	4,17	10,91	24,88	15498928	12680387
0,70	14,82	33326,12	33065,97	3,70	9,71	22,18	17307330	14906822
0,60	14,19	35879,15	36395,55	3,24	8,55	19,65	19346094	18143014
0,50	13,72	39423,43	40727,58	2,79	7,43	17,25	21631011	21835270
0,40	13,43	44368,62	46815,41	2,35	6,34	14,95	23317154	25948562
0,30	13,34	53739,65	55799,30	1,91	5,27	12,74	23524252	28562886
0,20	13,47	62057,44	69831,37	1,48	4,24	10,61	21301760	25218944
0,15	13,68	69632,93	80510,82	1,26	3,74	9,57	18722960	21642997

Table 4-15. Summary for two stage cycle (to -50 °C), warm climate, Tintermediate -15 °C

C2 comp. (-)	Duty (MW)	Vol. Flow C_1 (m3/h)	Vol. Flow C_2 (m3/h)	P_1 (bar)	P_2 (bar)	P_3 (bar)	UA_1 (kJ/°C.h)	UA_2 (kJ/°C.h)
1,00	40,69	49834,57	32861,81	5,14	15,50	46,65	25583819	10984677
0,90	33,26	44694,65	34703,07	4,65	13,97	41,23	21184610	12557341
0,80	29,77	44365,33	36918,64	4,17	12,53	36,77	21145811	13534965
0,70	27,30	45670,87	39888,14	3,70	11,15	32,79	21538027	15539514
0,60	25,90	48270,41	43765,45	3,24	9,83	29,12	22534693	17782321
0,50	24,77	52205,87	49027,66	2,79	8,55	25,69	23908195	20660141
0,40	23,88	58387,35	56405,01	2,34	7,31	22,45	23330837	23933691
0,30	23,59	66728,62	67031,49	1,91	6,10	19,36	23866085	24738047
0,20	23,84	79545,93	84060,24	1,47	4,93	16,40	22241741	23055998
0,15	24,25	89392,29	96993,38	1,26	4,36	14,96	20125514	20892482

Due to the difference in the temperature range, the only parameter that may be compared between the cycle implemented in this section and the ones from Section 3.1 is the coefficient of performance (β); this is shown in Figure 4-37. The cycles reaching a lower temperature (regular curves) give a smaller coefficient of performance than the processes until -36 °C (dashed curves) for both climate conditions. The temperature interval for the cycles reaching a lower temperature is obviously higher, which gives less possibility to adapt the process heating curves in order to reduce the irreversibilities.

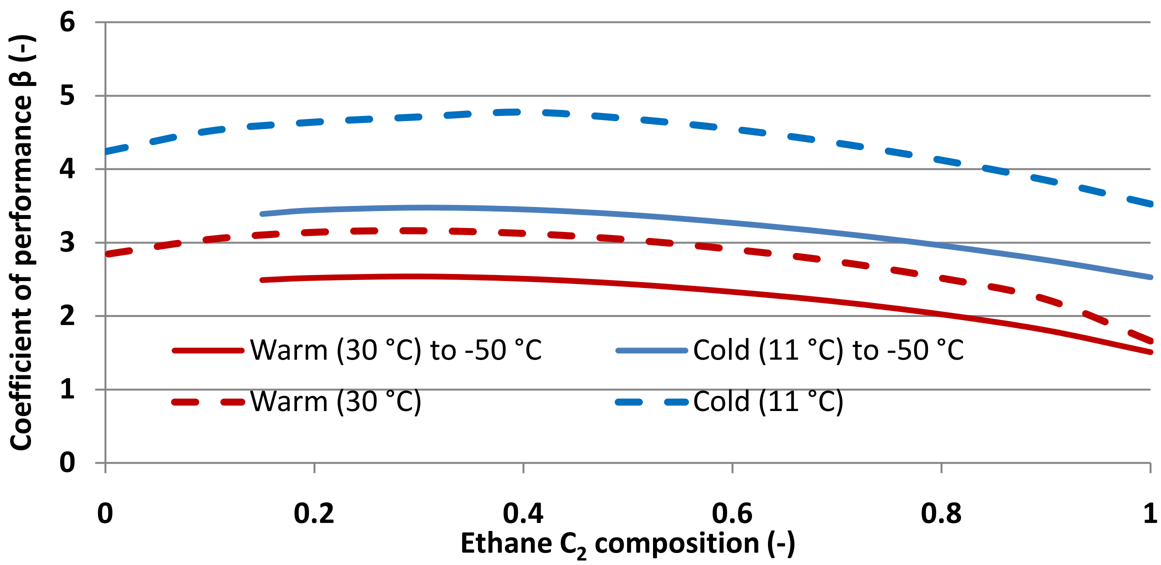

Figure 4-37. Coefficient of performance comparison for two stage cycle, different temperature range

4.4. Application of the different configurations to the MFC® process

Even though the coefficient of performance for the stand-alone cycle reaching -50 °C is clearly lower than the one reaching -36 °C, the study of a Mixed Fluid Cascade (MFC®) process is considered important for the purpose of this work, basically, due to the benefits that could be provided to the entire cycle if the precooling cycle is a mixed refrigerant circuit that cools the natural gas down to -50 °C.

The study is carried out for both climate conditions considered for the previous cases. The conventional mixed refrigerant cycle used in the MFC® process is compared with a three stage propane cycle. The parameters found as most suitable and used for the simulation of the precooling cycle of each case are given in Table 4-16. Once more the temperature of the natural gas after the cycle is presented, yet it is linked to the pressure level of the stage through the temperature difference in the heat exchanger.

Table 4-16. Precooling parameters for the MFC process studies

		Condition	
Precooling	**Parameter**	**Warm**	**Cold**
Propane three stage, to -36 °C	Temp. NG 1^{st} stage (°C)	6,5	-4,5
	Temp. NG 2^{nd} stage (°C)	-16,5	-20
	Molar flow to NG (kgmole/h)	13850	10170
	Molar flow to $MR_{LIQUEFACTION}$ (kgmole/h)	47750	25740
	Molar flow to $MR_{SUBCOOLING}$ (kgmole/h)	29900	15550
	Precooling heat load (MW)	52,43	41,18
Mixed refrigerant two stage, to -50 °C	C_2/C_3 composition (-)	0,3/0,7	0,377/0,623
	$T_{INTERMEDIATE}$ (°C)	-15	-18

		Condition	
Precooling	**Parameter**	**Warm**	**Cold**
	Molar flow 1st stage (kgmole/h)	52800	47330
	Molar flow 2nd stage (kgmole/h)	25700	22300
	Precooling heat load (MW)	64,27	51,62

The values shown in Table 4-16 do not match in all the cases with the values found in the stand-alone cycle studies as the most suitable. For instance, in the propane case under the warm climate condition the optimal temperatures of the first and second cycle were found to be 8 °C and -14,6 °C respectively, for the stand-alone cycle. This difference in the optimal parameters for the cycle is due to the heat load variation provided by the addition of two streams to the precooling heat exchanger (the mixed refrigerant stream from the liquefaction and subcooling cycle). The values from the table above were found to be the most suitable for the entire process simulation and the results from these simulations are shown below.

Figure 4-38 shows the compressor duty for the processes under the warm climate condition. The MFC® process has a power consumption 3,7% above the one for the propane precooled. The duty share among the cycles for the propane precooled is 0,35/0,24/0,41; whilst for the MFC® case it is 0,42/0,18/0,4. When replacing the mixed refrigerant precooling cycle by a propane circuit (until -36 °C) the heat load in the precooling is reduced, but increased in the liquefaction cycle; which for this case has been favorable since the liquefaction cycle performs the cooling more efficiently, leading to an overall lower process power consumption.

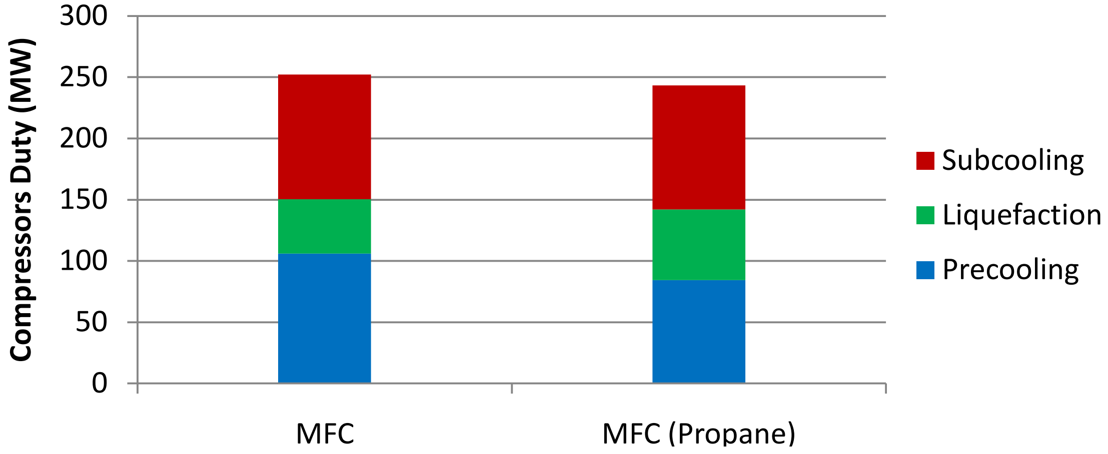

Figure 4-38. Compressor power comparison, MFC® with modifications, warm climate

Furthermore, the UA value of the heat exchangers in both processes is taken into account, this is shown in Figure 4-39. The value in the precooling cycle is higher for the mixed refrigerant case. This is due to the increase in heat load linked to the temperature range, which goes from 30 to -50° C; in addition to this, the mixed refrigerant heat exchanger has an extra stream flowing through it, which as explained before increases the UA value for this configuration.

An important result to highlight is the UA value for the liquefaction cycle; even though the heat load of this cycle in the MFC® process is lower than in the propane precooled, the latter has a smaller UA value; which is thought to be due to the shorter temperature difference in the heat exchanger.

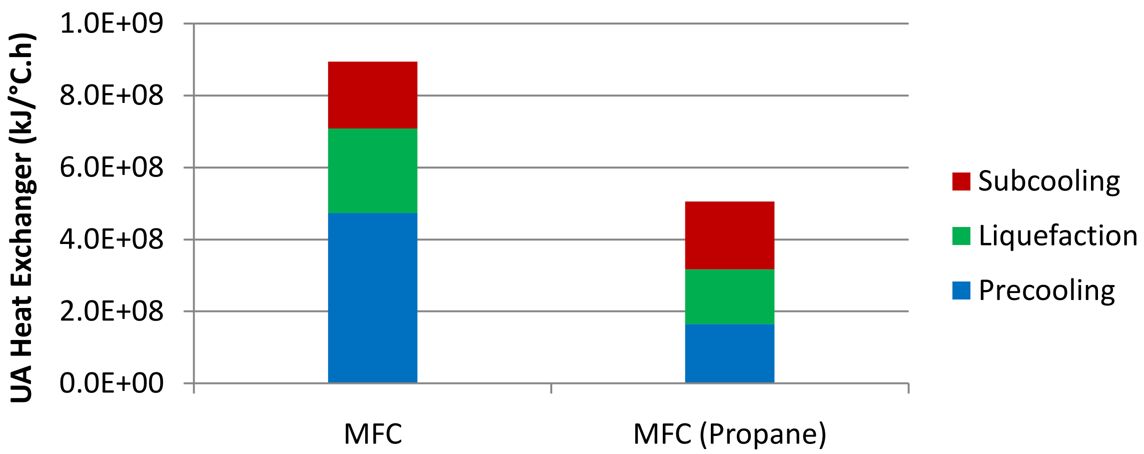

Figure 4-39. Heat exchanger UA values comparison, MFC® with modifications, warm climate

The last parameter considered is the suction volume of the compressor; from the previously studied cases the use of a mixed refrigerant has been beneficial in terms of this parameter (the volume has been reduced), however in this case the suction volume for the mixed refrigerant case is larger than for the propane case, as it is shown in Figure 4-40. Using a mixed refrigerant provides a smaller volumetric flow than a propane circuit indeed, but when the same heat load is set for both cases. In this case the heat load for the precooling cycle of the MFC® process is higher than for the MFC® with propane precooling; therefore it counteracts the reduction, which for the given case results in a higher volumetric flow.

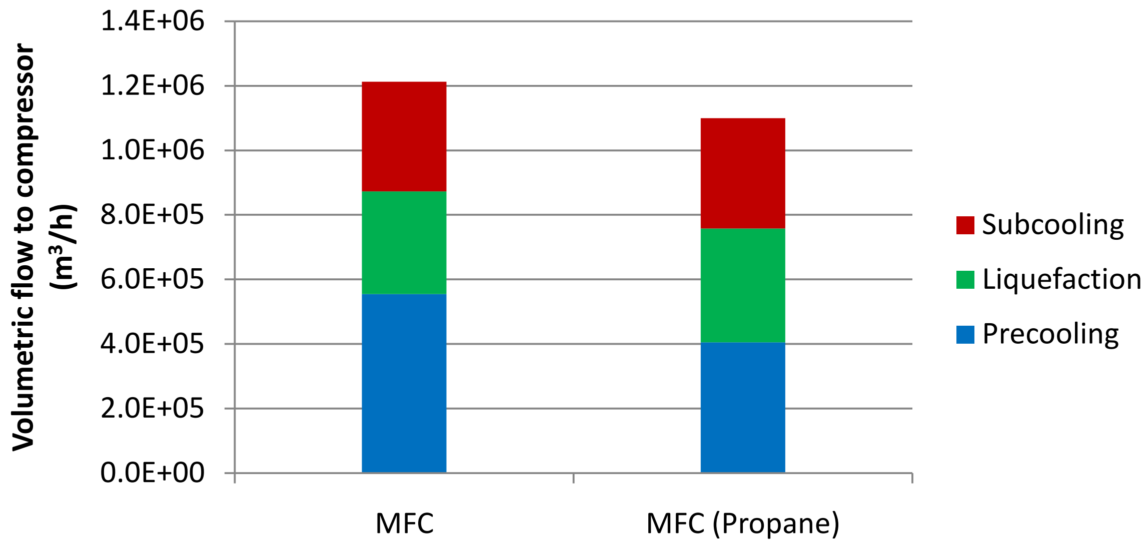

Figure 4-40. Compressor suction volume comparison, MFC® with modifications, warm climate

Under the warm climate condition, the use of a three stage propane precooling circuit instead of the conventional two stage mixed refrigerant cycle for the MFC® process represents a benefit in terms of power consumption, UA of the heat exchangers and suction volume of the compressors (process capacity). One of the main disadvantages of the mixed refrigerant precooling is the wide temperature span to be covered (from 30 °C to -50 °C), which makes the reduction of heat transfer irreversibilities a challenging task and likewise increases the volume suction of the compressor significantly.

On the other hand, the results for the cold climate condition simulations are shown below. The total compressor duty when using a mixed fluid cascade is only

1,6% above the value for a process with a propane precooling arrangement. The share of duties between the cycles (precooling/liquefaction/subcooling) is 0,3/0,25/0,45 for the MFC® and 0,2/0,34/0,46 for the propane precooled process.

The UA value for the process heat exchangers is given in Figure 4-42, as expected the mixed refrigerant precooling cycle has a larger UA value than the propane precooling circuit. Whereas due to the heat load increase, the liquefaction UA value is higher for the propane precooled process. The subcooling heat exchangers have the same value since, as described in Chapter 3, no variations in the subcooling cycle are made when the precooling configuration is modified.

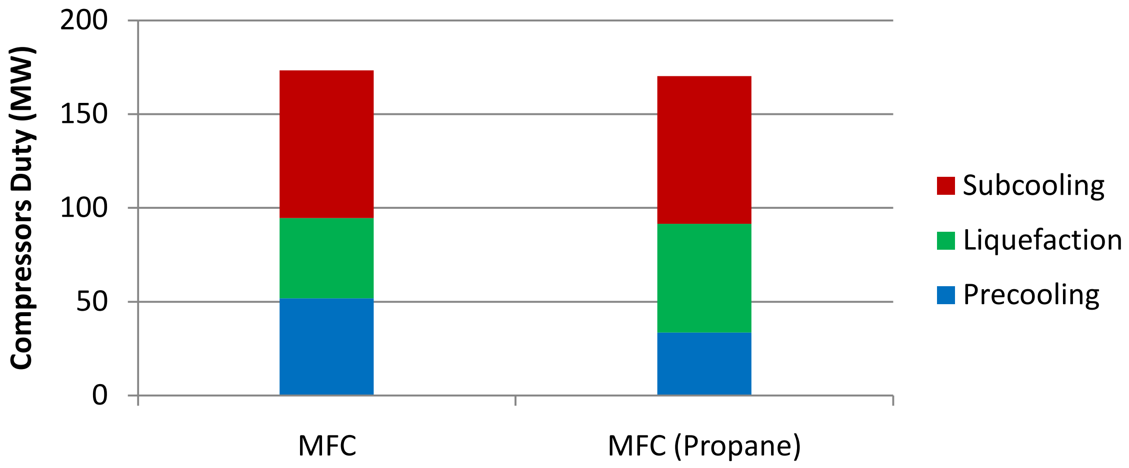

Figure 4-41. Compressor power comparison, MFC® with modifications, cold climate

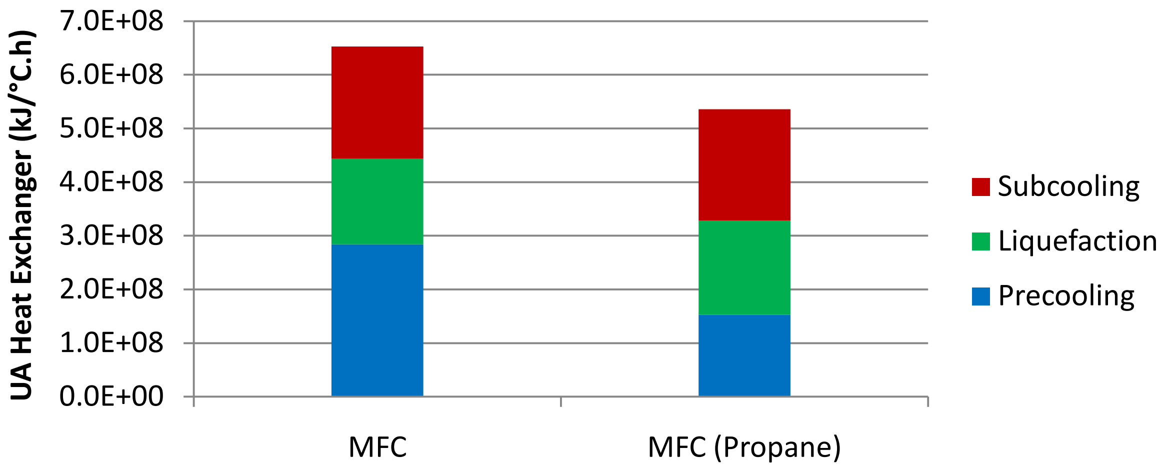

Figure 4-42. Heat exchanger UA values comparison, MFC® with modifications, cold climate

The volumetric flow comparison is represented in Figure 4-43 for the cold climate condition; in this case the heat load in the precooling counteracts the volumetric reduction effect of using a mixed refrigerant, but does not overpass it as in the warm climate case. Therefore the overall result is a smaller volumetric flowrate. When a propane precooling configuration is used, the process capacity is limited by the liquefaction compressor, which has a value of 338855 cubic meters per hour. On the other hand, if the conventional mixed refrigerant precooling is used, the limitation is set by the subcooling compressor, with a value of 258048 cubic meters per hour. Nevertheless, as might be seen in the figure below, the share between the three cycles is almost equal for the MFC® process.

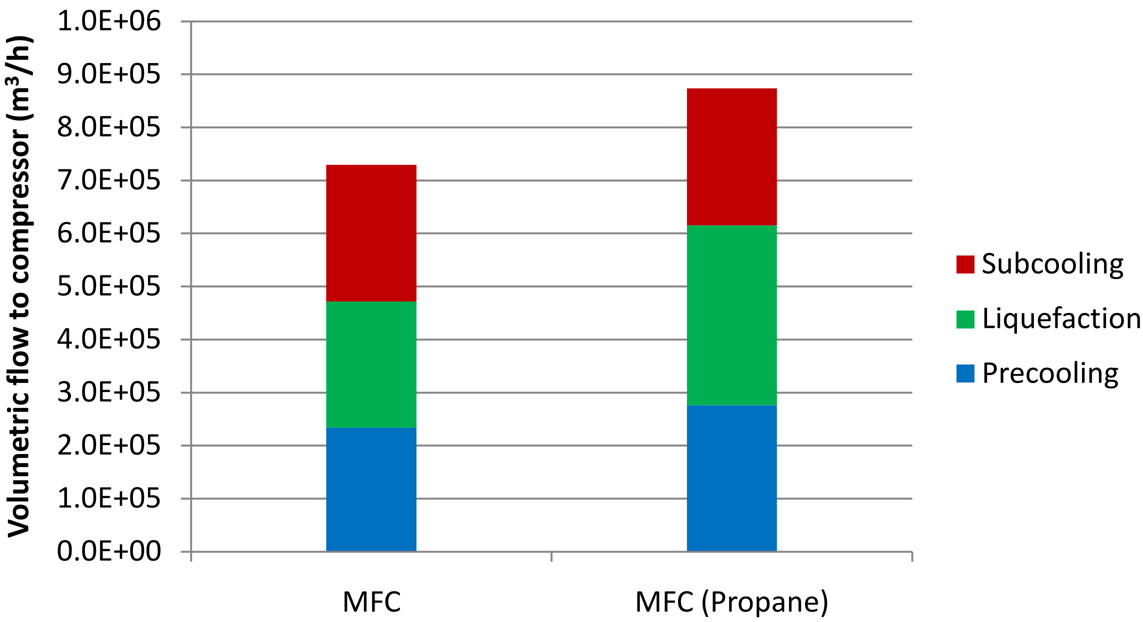

Figure 4-43. Compressor suction volume comparison, MFC® with modifications, cold climate

In spite of its higher power consumption, which is only 1,6% above the propane precooled case, the mixed fluid cascade with the conventional mixed refrigerant precooling cycle is the favored choice under the cold climate conditions. First, due to the capacity limitation that is introduced by the volumetric flow in the compressor, which also represents a larger compressor. In second place, due to the small share that the precooling cycle has in the propane precooled MFC, which, as mentioned before, represents a disadvantage because the precooling equipment, since it does not deal with cryogenic temperatures has a lower capital cost.

4.5. Overall specific work comparison

In this section the natural gas liquefaction processes studied are compared based on a benchmark criterion which is used frequently to compare different technologies; the work required to liquefy a unit mass of natural gas. These values are presented on kilowatt hour per kilogram LNG in Table 4-17. Based on the obtained results, the processes considered important for the purpose of the comparison are as follows:

- C_3MR process (A)
- C_3MR process, two stage mixed refrigerant cycle in the precooling (B)
- MFC® process (C)
- MFC® process with three stage propane cycle in the precooling (D)

Table 4-17. Specific work comparison for the main studied processes, kWh/kg LNG

Process	Cold	Warm
A	0,246	0,287
B	0,249	0,294
C	0,163	0,237
D	0,160	0,229

As it is shown in Table 4-17, the MFC® process with a three stage propane precooling cycle (Process D) has the lower specific work among the processes presented for both climate conditions. A process flow diagram for this configuration is proposed in Figure 4-44. The precooling circuit uses propane (red colored streams) in a three stage cycle, whereas the liquefaction (yellow) and subcooling (blue) circuits are mixed refrigerant processes. The illustration given represents a three stage cycle in the precooling; a four stage cycle could be beneficial for cases were the ambient temperature is high enough, or when the efficiency needs to be improved (e.g.: to fit existing equipment capacity).

Based on the studies done this process is to be implemented in projects where the aim is low process efficiency with ambient conditions above 20 °C. Besides the energy efficiency, when this process is compared to the C_3MR process, it does not require a flash separator, which means that the composition of the refrigerant does not depend on specific conditions, reducing, therefore the complexity of the process. The equipment count for this process is increased when compared to the MFC® and the C_3MR processes; see Section 2.5.

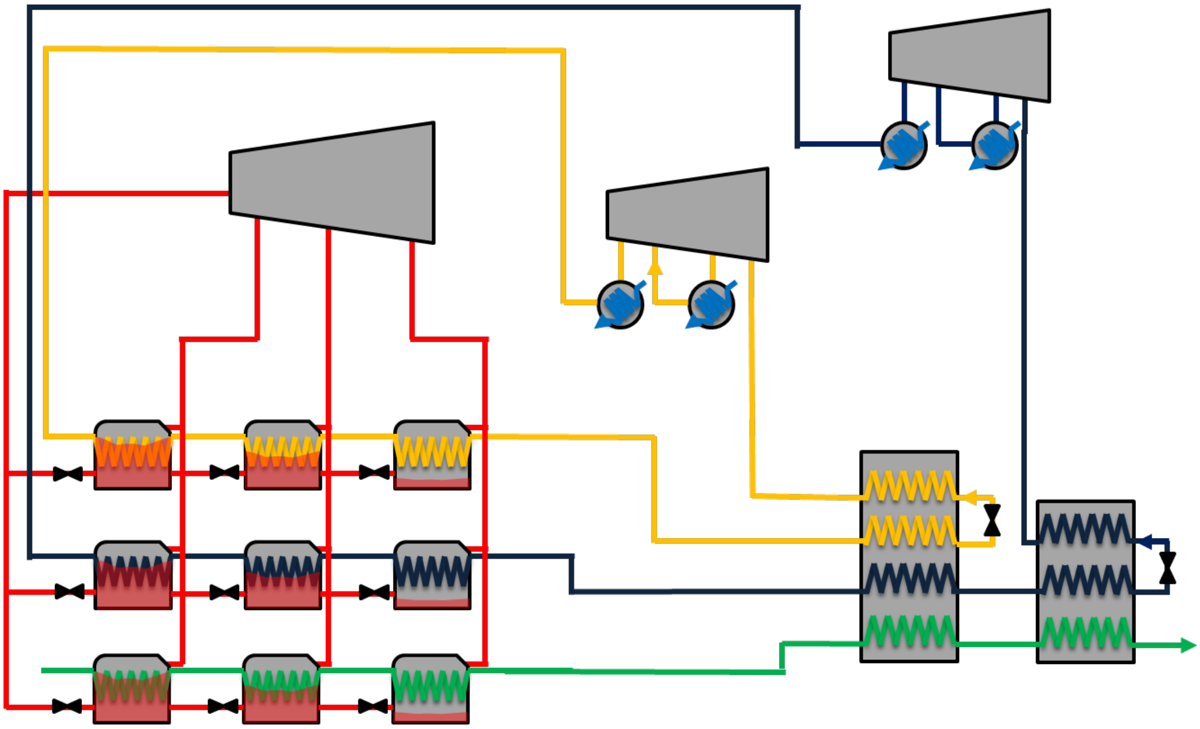

Figure 4-44. Process flow diagram of proposed high efficiency process, MFC modified

Chapter 5. Conclusions

The main differences related to technical performance between choosing a pure component or a mixed refrigerant configuration for the precooling cycle of LNG processes were addressed through this report. Many configurations were studied for the precooling cycle; a single stage mixed refrigerant process was found to be the less beneficial in terms of compressor duty, heat exchanger UA, volumetric flow at compressor suction (compressor size/process capacity) and pressure ratio (related to the compressor efficiency) for both climate conditions studied (warm -25 °C-, cold -6 °C-).

In terms of energy efficiency, a three stage propane configuration was found to be a better alternative than a two stage mixed refrigerant (C_2/C_3) arrangement under both climate conditions. The selection of such a three stage cycle with propane represents as well the best choice concerning the heat exchangers UA value; however, the equipment count is larger than the one for a mixed refrigerant cycle. Using n-butane as part of the mixed refrigerant cycles was considered, it provided some improvements in the compressor duty for the warm climate case but the volumetric flow and the UA value were significantly increased, this alternative was therefore discarded.

Although the heat exchanger UA and the compressor duty do not benefit from a two stage mixed refrigerant configuration (compared to the three stage propane), this arrangement has shown to be a possible alternative to improve (reduce) the volumetric flow of the compressors for any of the climate conditions. Under the cold climate, however, the short temperature range gives the propane precooled process a disadvantage in terms of capacity share in the entire process. The mixed refrigerant configuration with cooling duties until -50 °C is a suggested solution for increasing the share in this case.

A technical selection chart is given below, the purpose of this is to provide, based on the obtained results, a basis for future selection of the precooling stage configuration; only the most promising configurations are shown in the chart. The criteria used for the selection are power consumption (power), UA value of the heat exchangers (UA HX), volumetric flow to the compressors (comp.) and the share that the precooling cycle has in the entire process. The choice recommended for each case is also given in this chart, notice that these are technical recommendations and that the ultimate choice will remain on the hands of project developer, dependent on project specific variables and new developed technologies.

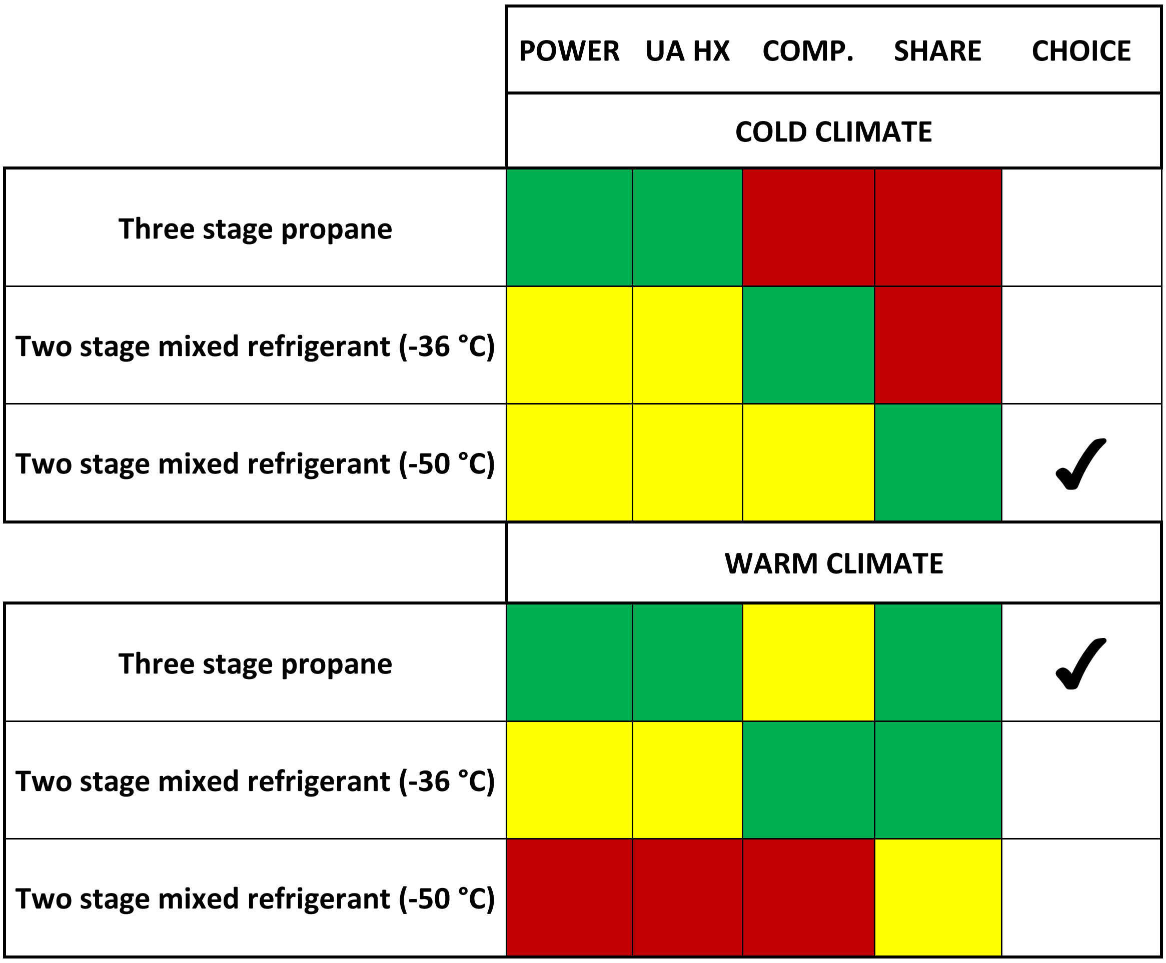

Figure 5-1. Selection chart for precooling stage of LNG processes

The chart is made using green for the most recommended process under each parameter; consequently, yellow color means that the process may be considered as an alternative based on this parameter since, even though it is not the best, it is not far from it. Finally the red colored parameters represent those in which the process is far from the most recommended and therefore based on those parameters such process is neither suggested nor convenient.

In addition to the main goal achieved, which is the technical comparison of different precooling configurations, a new, highly efficient configuration for natural gas liquefaction has been suggested for warm climate temperature conditions (25 °C) based on the results obtained; this configuration consists of 3 different cycles (as the MFC® process) in which the precooling circuit is a three stage propane instead of a mixed refrigerant; no previous reference in the open literature was found for such arrangement. A suggestion for the name of this process is propane precooled mixed refrigerant cascade (PPMRC).

Chapter 6. Recommendations for Further Work

Since the most recent developments are showing high interest on floating LNG facilities, it is highly recommended to study the differences between the available liquefaction technologies and configurations in order to provide the project developers with technical differences and if possible a selection criteria. For floating LNG the selection criteria is influenced by the small available plot area and the required robustness for the process, since it has to withstand different challenges regarding motion and flow stability.

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Appendix A: HYSYS® flowsheets simulated

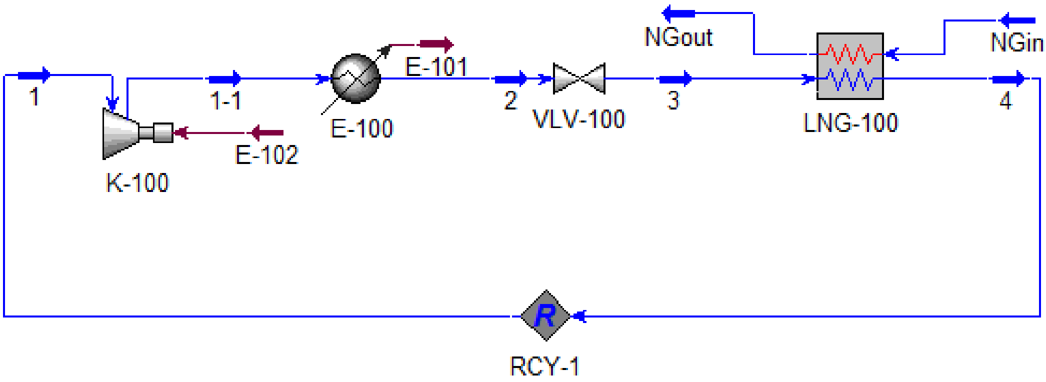

Figure A-1. Simple refrigeration cycle for pure component refrigerant, standalone

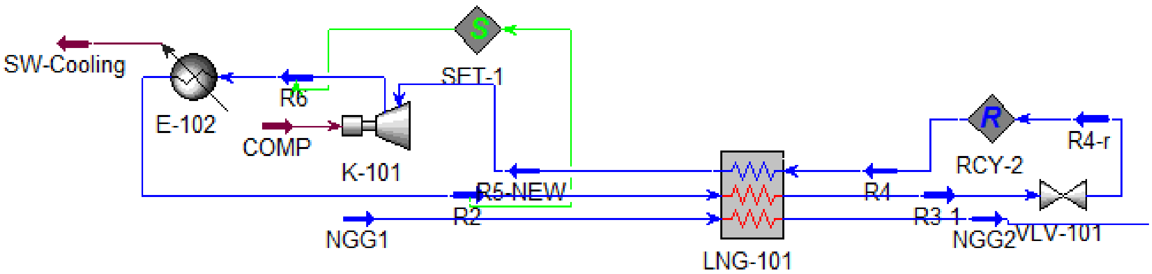

Figure A-2. Simple refrigeration cycle for mixed refrigerant, standalone

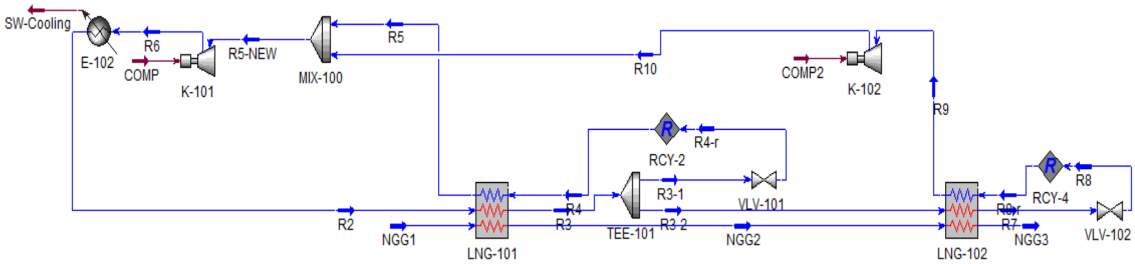

Figure A-3. Two stage refrigeration cycle for mixed refrigerant, standalone

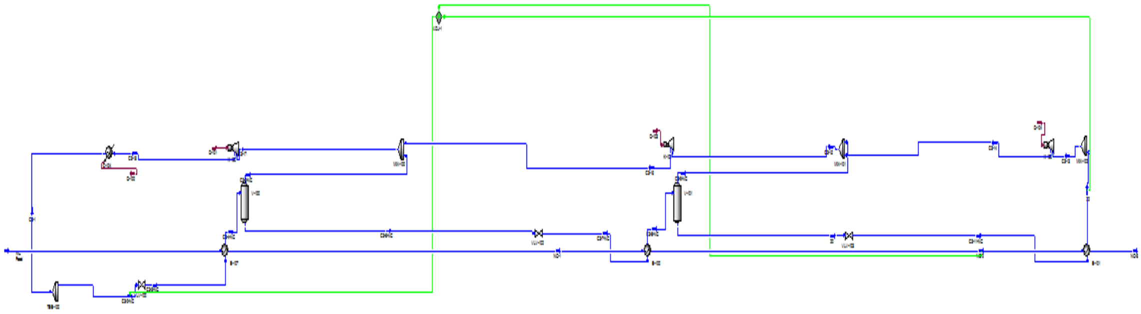

Figure A-4. Three stage propane refrigeration cycle, standalone

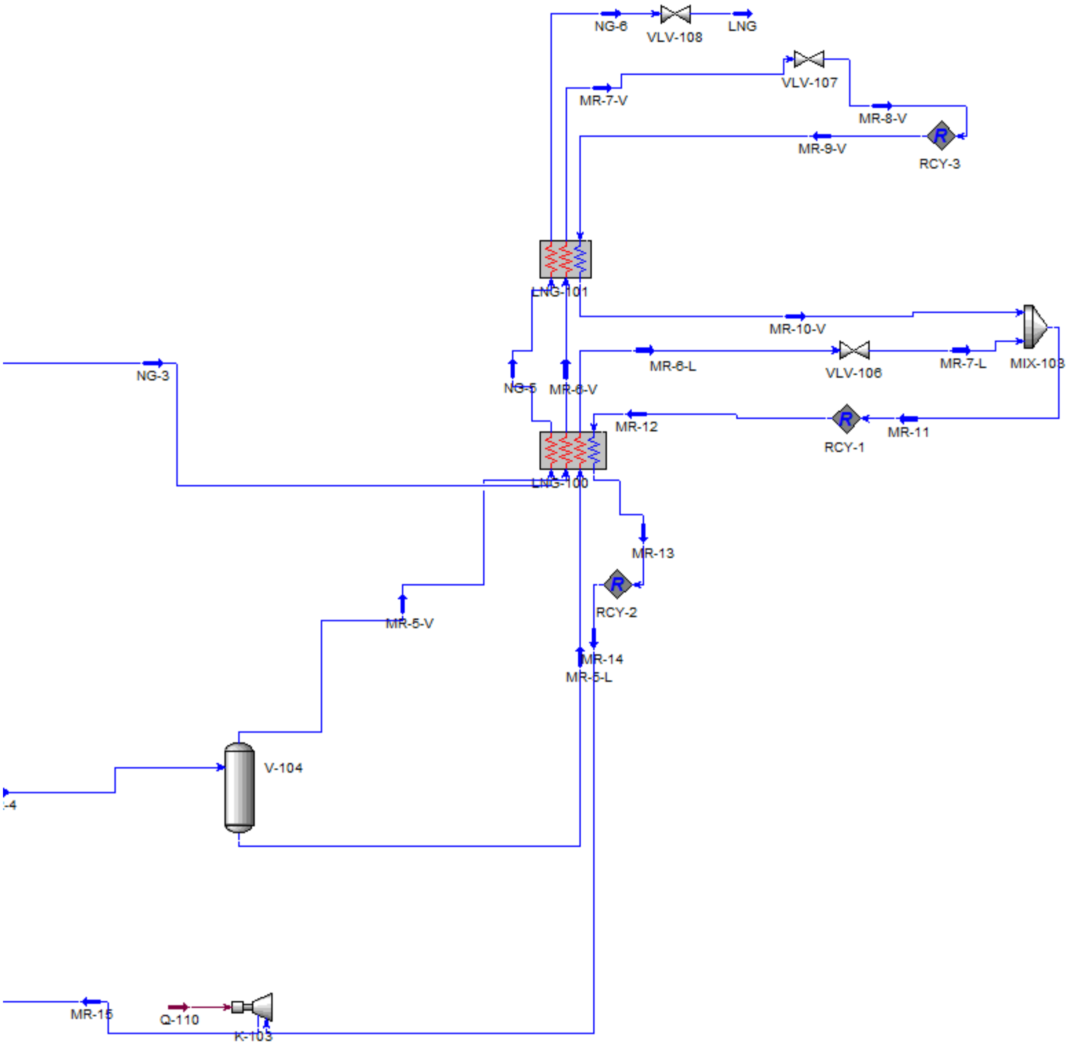

Figure A-5. Liquefaction cycle for C$_3$MR process studies

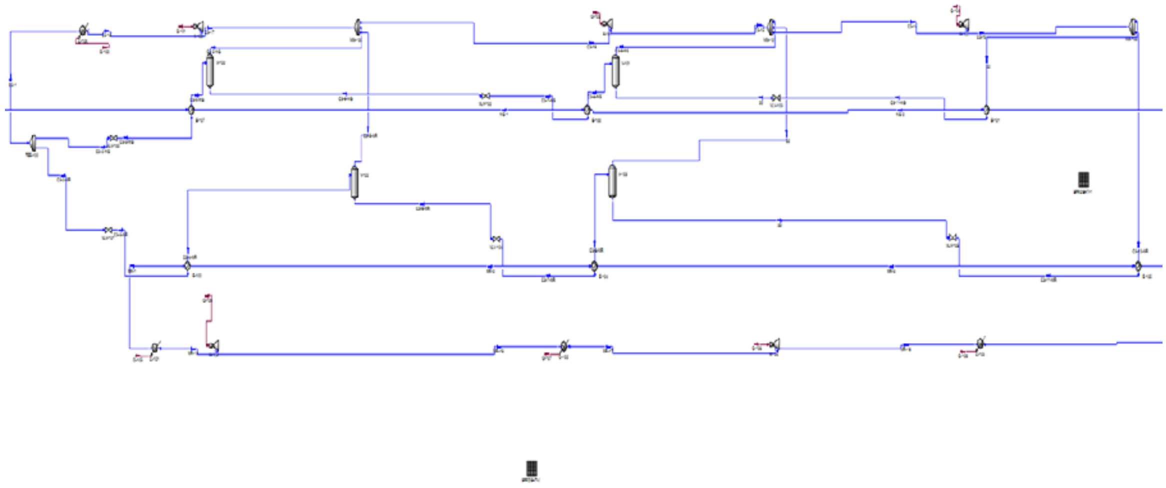

Figure A-6. Three stage propane precooling cycle for C_3MR process studies

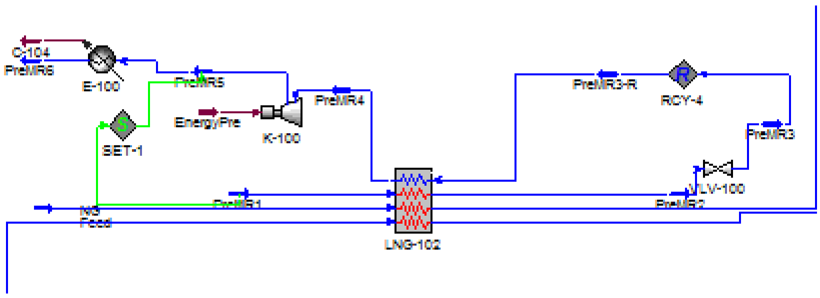

Figure A-7. Single stage mixed refrigerant precooling cycle for C_3MR process studies

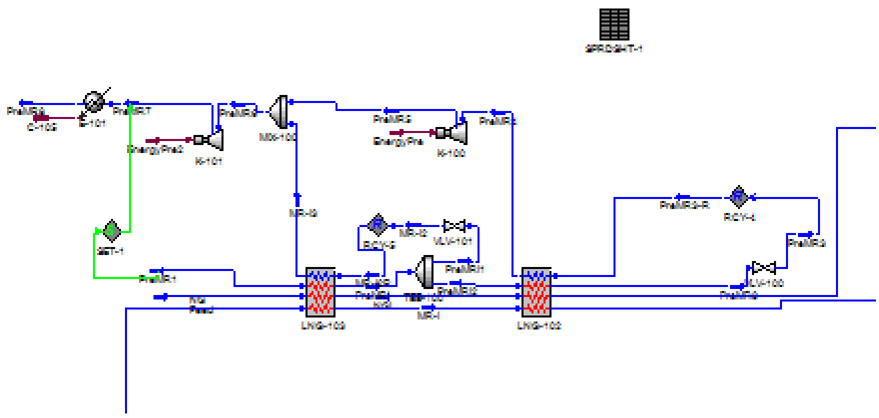

Figure A-8. Two stage mixed refrigerant precooling cycle for C_3MR process studies

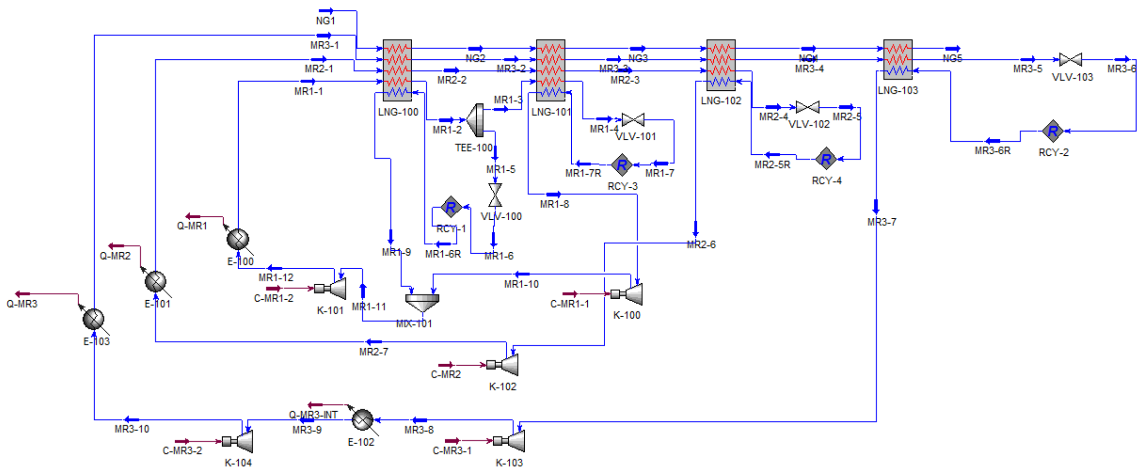

Figure A-9. MFC® process

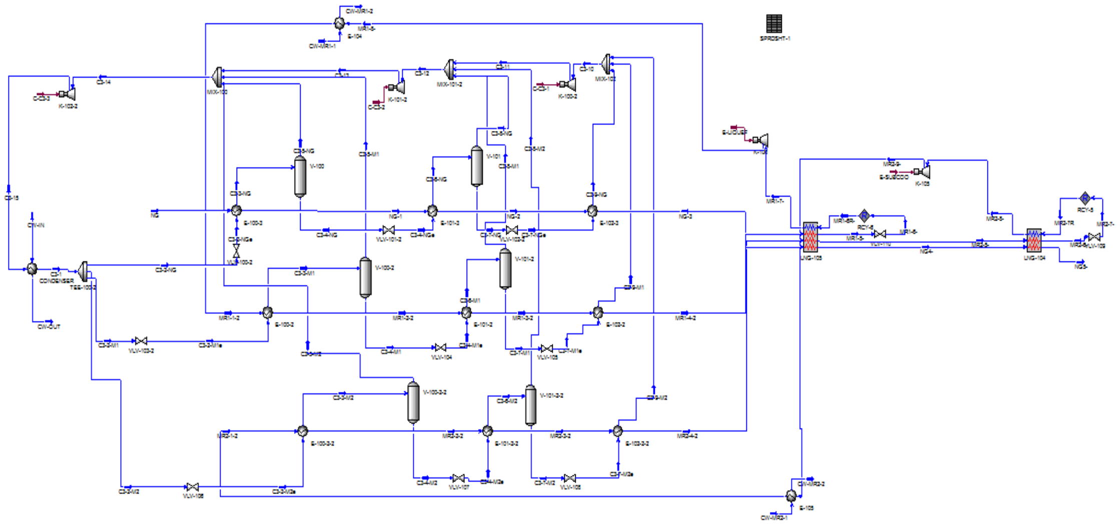

Figure A-10. MFC® process with propane precooling cycle

Appendix B: Stand-alone study details

Table B-1. Parameters for the stand-alone single stage cycle, warm climate

C2= 1	Pressure exp= 830 kPa			C2= 0,9	Pressure exp= 750 kPa		
C3= 0				C3= 0,1			
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)	Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
27500,00	47,22	169988478,32	27,97	20500,00	122703306,67	34,08	29,80
28500,00	47,68	171632171,30	26,02	20750,00	123306873,45	34,25	28,04
29000,00	48,13	173266652,05	24,12	21000,00	123905022,96	34,42	26,32
29500,00	48,58	174892225,90	22,29	21250,00	124497853,91	34,58	24,63
30000,00	49,03	176509185,16	20,51	21500,00	125085464,32	34,75	22,98
30500,00	49,48	178117806,32	18,79	21750,00	125667947,96	34,91	21,36
31000,00	49,92	179718356,88	17,12	22000,00	126245396,07	35,07	19,77
31500,00	50,36	181311090,29	15,50	22250,00	126817897,36	35,23	18,22
32000,00	50,80	182896249,03	13,92	22500,00	127385538,03	35,38	16,69
32500,00	51,24	184474065,14	12,39	22750,00	127948401,92	35,54	15,20
33000,00	51,68	186044763,07	10,90	23000,00	128506570,47	35,70	13,74
33500,00	52,11	187608548,29	9,45	23250,00	129060122,89	35,85	12,30
34000,00	52,55	189165636,35	8,05	23500,00	129609136,23	36,00	10,89
34500,00	52,98	190716227,82	6,68	23750,00	130153685,45	36,15	9,51
35000,00	53,41	192260501,10	5,35	24000,00	130693843,50	36,30	8,15
35500,00	53,83	193798636,03	4,05	24250,00	131229681,40	36,45	6,82
36000,00	54,26	195330805,23	2,79	24500,00	131761268,27	36,60	5,51
36500,00	54,68	196857174,51	1,56	24750,00	132288671,45	36,75	4,23
37000,00	55,10	198377903,18	0,36	25000,00	132811956,54	36,89	2,97
37500,00	55,53	199893144,38	-0,81	25250,00	133732710,80	37,15	4,40
38000,00	55,95	201403045,41	-1,95	25500,00	133846426,38	37,18	0,52
38500,00	56,36	202907748,02	-3,06	25750,00	134776385,33	37,44	1,92
39000,00	56,78	204407388,63	-4,15	26000,00	134865169,84	37,46	-1,84
39500,00	57,20	205902098,66	-5,21	26250,00	135804233,78	37,72	-0,46
40000,00	57,61	207392004,68	-6,24	26500,00	135868654,59	37,74	-4,13
40500,00	58,02	208877228,76	-7,25	26750,00	136816726,99	38,00	-2,77
41000,00	58,43	210357888,56	-8,23	27000,00	136857325,58	38,02	-6,33
41500,00	58,84	211834097,60	-9,20	27250,00	137814312,90	38,28	-4,99
42000,00	59,25	213305965,41	-10,14	27500,00	137831606,46	38,29	-8,46
42500,00	59,66	214773597,77	-11,06	27750,00	138797418,05	38,55	-7,14
43000,00	60,07	216237096,83	-11,96	28000,00	138791900,91	38,55	-10,52
43500,00	60,47	217696561,26	-12,83	28250,00	139766448,95	38,82	-9,23
44000,00	60,88	219152086,46	-13,69	28500,00	139738593,94	38,82	-12,52

44500,00	61,28	220603764,62	-14,54	28750,00	140721793,26	39,09	-11,24
45000,00	61,68	222051684,95	-15,36	29000,00	140672052,97	39,08	-14,44
45500,00	62,08	223495933,76	-16,16	29250,00	141663820,98	39,35	-13,19
46000,00	62,48	224936594,55	-16,95	29500,00	141592628,94	39,33	-16,31
46500,00	62,88	226373748,19	-17,73	29750,00	142592885,54	39,61	-15,07
47000,00	63,28	227807472,99	-18,48	30000,00	142500657,27	39,58	-18,12
47500,00	63,68	229237844,81	-19,22	30250,00	143509324,73	39,86	-16,90
48000,00	64,07	230664937,20	-19,95	30500,00	143396458,79	39,83	-19,87
48500,00	64,47	232088821,39	-20,66	30750,00	144413461,67	40,11	-18,67
49000,00	64,86	233509566,54	-21,36	31000,00	144280340,57	40,08	-21,57
49500,00	65,26	234927239,63	-22,04	31250,00	145305605,63	40,36	-20,39
50000,00	65,65	236341905,73	-22,71	31500,00	145152596,71	40,32	-23,22
50500,00	66,36	238903878,87	-20,92	31750,00	146186052,84	40,61	-22,05
51000,00	66,43	239162467,56	-24,02	32000,00	146013509,09	40,56	-24,82
51500,00	67,15	241740229,13	-22,25	32250,00	147055087,23	40,85	-23,67
52000,00	67,21	241971735,30	-25,27	32500,00	146863348,07	40,80	-26,37
52500,00	67,93	244564944,05	-23,54	32750,00	147912981,08	41,09	-25,23
53000,00	67,99	244770164,83	-26,48	33000,00	147702373,07	41,03	-27,88
53500,00	68,72	247378491,47	-24,78				
54000,00	68,77	247558186,70	-27,64				
54500,00	69,49	250181313,28	-25,97				
55000,00	69,54	250336207,95	-28,76				
55500,00	70,27	252973827,27	-27,12				
56000,00	70,31	253104613,75	-29,84				
56500,00	71,04	255756428,82	-28,23				
57000,00	71,07	255863768,84	-30,89				
57500,00	71,81	258529492,44	-29,30				
58000,00	71,84	258614019,12	-31,90				
58500,00	72,58	261293373,22	-30,34				
59000,00	72,60	261355692,70	-32,87				
59500,00	73,35	264048408,10	-31,34				
60000,00	73,36	264089101,22	-33,81				
60500,00	74,11	266794917,11	-32,31				
61000,00	74,12	266814540,84	-34,73				
61500,00	74,87	269533204,42	-33,24				
62000,00	74,87	269532293,27	-35,61				
62500,00	75,63	272263559,37	-34,15				
63000,00	75,62	272242626,66	-36,46				
63500,00	76,39	274986257,42	-35,03				
64000,00	76,37	274945796,44	-37,29				
64500,00	77,14	277701560,99	-35,88				
65000,00	77,13	277656842,76	-38,04				

C2= 0,8 / C3= 0,2	Pressure exp= 672 kPa			C2= 0,7 / C3= 0,3	Pressure exp= 598 kPa		
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)	Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
18000,00	107388134,73	29,83	30,90	16500,00	98106656,57	27,25	30,47
18125,00	107766253,38	29,94	31,77	16625,00	98461888,16	27,35	31,37
18250,00	107909695,19	29,97	28,71	16750,00	98574062,79	27,38	27,95
18375,00	108290755,18	30,08	29,57	16875,00	98932508,46	27,48	28,84
18500,00	108423859,53	30,12	26,56	17000,00	99032437,01	27,51	25,49
18625,00	108807846,23	30,22	27,42	17125,00	99394086,29	27,61	26,37
18750,00	108930763,29	30,26	24,47	17250,00	99481940,35	27,63	23,09
18875,00	109317662,67	30,37	25,31	17375,00	99846783,27	27,74	23,96
19000,00	109430537,65	30,40	22,42	17500,00	99922986,37	27,76	20,75
19125,00	109820335,54	30,51	23,26	17625,00	100291203,85	27,86	21,61
19250,00	109923309,59	30,53	20,42	17750,00	100355174,49	27,88	18,46
19375,00	110315992,60	30,64	21,25	17875,00	100726387,99	27,98	19,31
19500,00	110409201,87	30,67	18,46	18000,00	100778953,44	27,99	16,23
19625,00	110804756,85	30,78	19,28	18125,00	101153329,70	28,10	17,07
19750,00	110888333,32	30,80	16,55	18250,00	101194532,13	28,11	14,05
19875,00	111286747,43	30,91	17,36	18375,00	101572063,69	28,21	14,88
20000,00	111361086,40	30,93	14,68	18500,00	101601784,56	28,22	11,92
20125,00	111762364,43	31,05	15,49	18625,00	101982458,61	28,33	12,75
20250,00	111827011,81	31,06	12,85	18750,00	102001138,27	28,33	9,84
20375,00	112231118,36	31,18	13,65	18875,00	102384956,50	28,44	10,66
20500,00	112286477,38	31,19	11,06	19000,00	102392594,35	28,44	7,81
20625,00	112693390,82	31,30	11,85	19125,00	102779549,52	28,55	8,62
20750,00	112739689,04	31,32	9,30	19250,00	102776273,64	28,55	5,82
20875,00	113149415,39	31,43	10,09	19375,00	103166359,13	28,66	6,63
21000,00	113186480,51	31,44	7,59	19500,00	103152292,52	28,65	3,88
21125,00	113599009,12	31,56	8,36	19625,00	103545502,24	28,76	4,68
21250,00	113627340,75	31,56	5,91	19750,00	103519479,58	28,76	1,98
21375,00	114042658,23	31,68	6,68	19875,00	103915754,33	28,87	2,77
21500,00	114062174,10	31,68	4,26	20000,00	103880611,64	28,86	0,13
21625,00	114480269,81	31,80	5,02	20125,00	104280005,59	28,97	0,91
21750,00	114491073,00	31,80	2,65	20250,00	104234396,67	28,95	-1,69
21875,00	114911936,61	31,92	3,40	20375,00	104636903,89	29,07	-0,92
22000,00	114914127,16	31,92	1,07	20500,00	104582094,88	29,05	-3,47
22125,00	115337748,62	32,04	1,82	20625,00	104986553,29	29,16	-2,70
22250,00	115331423,67	32,04	-0,48	20750,00	104921378,00	29,14	-5,21
22375,00	115757793,21	32,15	0,26	20875,00	105330149,19	29,26	-4,45
22500,00	115743047,02	32,15	-2,00	21000,00	105253629,95	29,24	-6,91

22625,00	116172155,17	32,27	-1,26	21125,00	105665494,75	29,35	-6,16
22750,00	116149079,30	32,26	-3,48	21250,00	105578934,54	29,33	-8,58
22875,00	116580916,85	32,38	-2,75	21375,00	105993891,64	29,44	-7,84
23000,00	116549600,21	32,37	-4,94	21500,00	105897377,68	29,42	-10,21
23125,00	116984158,24	32,50	-4,22	21625,00	106315425,12	29,53	-9,48
23250,00	116944687,18	32,48	-6,37	21750,00	106209043,27	29,50	-11,81
23375,00	117381957,03	32,61	-5,66	21875,00	106630179,25	29,62	-11,08
23500,00	117334415,45	32,59	-7,77	22000,00	106514012,76	29,59	-13,38
23625,00	117774388,73	32,72	-7,06	22125,00	106938235,73	29,71	-12,66
23750,00	117718858,13	32,70	-9,15	22250,00	106812364,96	29,67	-14,92
23875,00	118161526,70	32,82	-8,45	22375,00	107240481,04	29,79	-14,20
24000,00	118098086,30	32,81	-10,50	22500,00	107104176,18	29,75	-16,42
24125,00	118543442,29	32,93	-9,80	22625,00	107534569,98	29,87	-15,71
24250,00	118472169,07	32,91	-11,82	22750,00	107389520,19	29,83	-17,90
24375,00	118920204,85	33,03	-11,13	22875,00	107822998,57	29,95	-17,19
24500,00	118841173,62	33,01	-13,12	23000,00	107668468,30	29,91	-19,35
24625,00	119291881,82	33,14	-12,44				
24750,00	119205165,30	33,11	-14,40				
24875,00	119658538,81	33,24	-13,72				
25000,00	119564207,72	33,21	-15,65				
25125,00	120020239,64	33,34	-14,98				
25250,00	120474836,94	33,47	-14,30				
25375,00	120093626,33	33,36	-17,49				
25500,00	120553633,50	33,49	-16,82				
25625,00	121012189,88	33,61	-16,15				
25750,00	120612249,74	33,50	-19,27				
25875,00	121076218,00	33,63	-18,61				
26000,00	121538719,18	33,76	-17,96				

0,6	Pressure exp= 525 kPa			0,5	Pressure exp= 453 kPa		
0,4				0,5			
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)	Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
15500,00	92033395,08	25,56	28,51	14500,00	87554231,81	24,32	28,88
15600,00	92306849,47	25,64	29,24	14600,00	87818486,94	24,39	29,64
15700,00	92377316,84	25,66	26,28	14700,00	87886057,39	24,41	26,42
15800,00	92653121,60	25,74	27,01	14800,00	88152597,72	24,49	27,17
15900,00	92714346,69	25,75	24,10	14900,00	88211302,10	24,50	24,01
16000,00	92992180,83	25,83	24,82	15000,00	88480207,06	24,58	24,75
16100,00	93044745,86	25,85	21,96	15100,00	88529584,73	24,59	21,65
16200,00	93324819,97	25,92	22,68	15200,00	88800874,37	24,67	22,39
16300,00	93368628,22	25,94	19,87	15300,00	88840545,65	24,68	19,34

16400,00	93650902,22	26,01	20,58	15400,00	89114236,35	24,75	20,07
16500,00	93685015,81	26,02	17,82	15500,00	89143226,16	24,76	17,08
16600,00	93969435,25	26,10	18,53	15600,00	89419250,61	24,84	17,81
16700,00	93997105,57	26,11	15,82	15700,00	89438695,16	24,84	14,87
16800,00	94283769,18	26,19	16,51	15800,00	89717057,90	24,92	15,59
16900,00	94301953,82	26,19	13,85	15900,00	89727033,02	24,92	12,71
17000,00	94590812,71	26,28	14,54	16000,00	90007737,50	25,00	13,42
17100,00	94600611,99	26,28	11,92	16100,00	90008320,02	25,00	10,59
17200,00	94891664,65	26,36	12,61	16200,00	90291369,25	25,08	11,30
17300,00	94893156,12	26,36	10,03	16300,00	90282635,18	25,08	8,51
17400,00	95186401,35	26,44	10,71	16400,00	90568032,00	25,16	9,22
17500,00	95178547,92	26,44	8,17	16500,00	90550055,66	25,15	6,48
17600,00	95473945,50	26,52	8,85	16600,00	90837802,95	25,23	7,18
17700,00	95461118,84	26,52	6,35	16700,00	90810656,40	25,23	4,49
17800,00	95756760,50	26,60	7,02	16800,00	91100757,18	25,31	5,18
17900,00	95735657,66	26,59	4,57	16900,00	91064266,60	25,30	2,53
18000,00	96035513,37	26,68	5,24	17000,00	91356967,51	25,38	3,22
18100,00	96004375,65	26,67	2,82	17100,00	91311459,46	25,36	0,62
18200,00	96306417,16	26,75	3,48	17200,00	91606270,35	25,45	1,30
18300,00	96267334,14	26,74	1,10	17300,00	91552042,51	25,43	-1,26
18400,00	96571562,58	26,83	1,76	17400,00	91849217,89	25,51	-0,59
18500,00	96524593,07	26,81	-0,58	17500,00	91786080,11	25,50	-3,10
18600,00	96831009,61	26,90	0,07	17600,00	92085624,19	25,58	-2,44
18700,00	96775728,79	26,88	-2,24	17700,00	92013635,15	25,56	-4,91
18800,00	97084816,77	26,97	-1,59	17800,00	92315552,37	25,64	-4,25
18900,00	97021784,85	26,95	-3,86	17900,00	92234768,27	25,62	-6,68
19000,00	97332569,66	27,04	-3,22	18000,00	92539063,39	25,71	-6,03
19100,00	97262314,23	27,02	-5,46	18100,00	92449537,81	25,68	-8,42
19200,00	97575291,61	27,10	-4,82	18200,00	92756215,96	25,77	-7,77
19300,00	97497366,39	27,08	-7,02	18300,00	92657999,82	25,74	-10,13
19400,00	97812538,73	27,17	-6,39	18400,00	92967066,52	25,82	-9,48
19500,00	97726992,80	27,15	-8,56	18500,00	92860208,08	25,79	-11,81
19600,00	98044362,21	27,23	-7,94	18600,00	93171669,24	25,88	-11,16
19700,00	97951243,74	27,21	-10,07	18700,00	93056214,19	25,85	-13,45
19800,00	98270812,47	27,30	-9,45	18800,00	93370076,10	25,94	-12,81
19900,00	98170168,00	27,27	-11,56	18900,00	93246067,55	25,90	-15,07
20000,00	98491938,50	27,36	-10,94	19000,00	93562336,95	25,99	-14,43
20100,00	98812793,47	27,45	-10,33				
20200,00	98488669,74	27,36	-13,74				
20300,00	98813748,04	27,45	-13,13				
20400,00	99137895,22	27,54	-12,52				
20500,00	98795445,18	27,44	-15,86				

0,4	Pressure exp= 383 kPa			0,3	Pressure exp= 315 kPa		
0,6				0,7			
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)	Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
14000,00	85079235,09	23,63	26,77	13000,00	83047230,46	23,07	28,54
14100,00	85140402,04	23,65	23,38	13100,00	83302459,36	23,14	29,35
14200,00	85403478,40	23,72	24,14	13200,00	83388839,07	23,16	25,65
14300,00	85453840,36	23,74	20,82	13300,00	83646650,15	23,24	26,45
14400,00	85719382,62	23,81	21,58	13400,00	83721154,22	23,26	22,83
14500,00	85759038,05	23,82	18,32	13500,00	83981550,50	23,33	23,62
14600,00	86027049,96	23,90	19,07	13600,00	84044262,36	23,35	20,07
14700,00	86056096,90	23,90	15,87	13700,00	84307245,04	23,42	20,85
14800,00	86326582,05	23,98	16,61	13800,00	84358372,25	23,43	17,38
14900,00	86345115,88	23,98	13,48	13900,00	84623947,71	23,51	18,15
15000,00	86618077,93	24,06	14,21	14000,00	84663554,33	23,52	14,74
15100,00	86626077,55	24,06	11,13	14100,00	84931726,54	23,59	15,51
15200,00	86901633,58	24,14	11,87	14200,00	84959921,06	23,60	12,17
15300,00	86899311,62	24,14	8,84	14300,00	85230694,54	23,68	12,93
15400,00	87177235,49	24,22	9,57	14400,00	85247580,42	23,68	9,65
15500,00	87164781,62	24,21	6,60	14500,00	85520960,20	23,76	10,41
15600,00	87445195,43	24,29	7,32	14600,00	85526635,63	23,76	7,19
15700,00	87422572,67	24,28	4,40	14700,00	85802627,24	23,83	7,94
15800,00	87705481,33	24,36	5,12	14800,00	85797185,38	23,83	4,79
15900,00	87672766,86	24,35	2,25	14900,00	86075794,90	23,91	5,53
16000,00	87958175,67	24,43	2,96	15000,00	86059324,01	23,91	2,43
16100,00	87915442,91	24,42	0,14	15100,00	86340558,04	23,98	3,17
16200,00	88203357,63	24,50	0,85	15200,00	86313141,59	23,98	0,13
16300,00	88150676,22	24,49	-1,92	15300,00	86597007,28	24,05	0,86
16400,00	88441103,06	24,57	-1,22	15400,00	86558724,07	24,04	-2,13
16500,00	88378538,94	24,55	-3,94	15500,00	86845229,14	24,12	-1,41
16600,00	88671484,60	24,63	-3,25	15600,00	86796153,43	24,11	-4,34
16700,00	88599100,03	24,61	-5,92	15700,00	87085306,15	24,19	-3,62
16800,00	88894571,70	24,69	-5,24	15800,00	87025507,74	24,17	-6,51
16900,00	88812425,39	24,67	-7,87	15900,00	87317316,99	24,25	-5,79
17000,00	89110430,76	24,75	-7,19	16000,00	87246861,32	24,24	-8,63
17100,00	89018577,87	24,73	-9,77	16100,00	87541336,58	24,32	-7,92
17200,00	89319125,15	24,81	-9,10	16200,00	87460284,83	24,29	-10,71
17300,00	89217617,40	24,78	-11,64	16300,00	87757436,16	24,38	-10,00
17400,00	89520715,34	24,87	-10,97	16400,00	87665845,30	24,35	-12,74
17500,00	89409601,01	24,84	-13,48	16500,00	87965683,44	24,43	-12,05
17600,00	89715258,90	24,92	-12,81	16600,00	87863606,29	24,41	-14,74
17700,00	89594582,85	24,89	-15,28	16700,00	88166142,60	24,49	-14,05
17800,00	89902810,56	24,97	-14,61	16800,00	88053627,93	24,46	-16,70
17900,00	89772614,37	24,94	-17,04	16900,00	88358874,44	24,54	-16,01

18000,00	90083422,33	25,02	-16,38	17000,00	88235966,97	24,51	-18,62
18100,00	89943744,65	24,98	-18,77				

0,2	Pressure exp= 239 kPa			0,1	Pressure exp= 183 kPa		
0,8				0,9			
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)	Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
12500,00	83169679,64	23,10	26,60	11800,00	85089793,57	23,64	28,81
12575,00	83362243,82	23,16	27,22	11875,00	85284679,99	23,69	29,45
12650,00	83438153,97	23,18	24,30	11950,00	85400528,99	23,72	26,31
12725,00	83632240,66	23,23	24,91	12025,00	85597045,61	23,78	26,95
12800,00	83703438,15	23,25	22,04	12100,00	85703856,57	23,81	23,86
12875,00	83899047,54	23,31	22,65	12175,00	85901964,43	23,86	24,49
12950,00	83961621,40	23,32	19,82	12250,00	86001054,89	23,89	21,46
13025,00	84158739,28	23,38	20,43	12325,00	86200755,50	23,94	22,08
13100,00	84215489,60	23,39	17,65	12400,00	86292193,02	23,97	19,10
13175,00	84414130,85	23,45	18,25	12475,00	86493487,93	24,03	19,72
13250,00	84463881,90	23,46	15,51	12550,00	86575420,58	24,05	16,77
13325,00	84664048,09	23,52	16,11	12625,00	86778315,47	24,11	17,39
13400,00	84706854,20	23,53	13,41	12700,00	86856556,62	24,13	14,50
13475,00	84908547,13	23,59	14,00	12775,00	87061045,08	24,18	15,12
13550,00	84944730,44	23,60	11,35	12850,00	87129910,71	24,20	12,27
13625,00	85147951,53	23,65	11,94	12925,00	87335998,65	24,26	12,88
13700,00	85177880,63	23,66	9,32	13000,00	87397462,22	24,28	10,07
13775,00	85382637,76	23,72	9,90	13075,00	87605151,63	24,33	10,68
13850,00	85404955,85	23,72	7,32	13150,00	87659270,34	24,35	7,92
13925,00	85611245,28	23,78	7,91	13225,00	87868563,37	24,41	8,52
14000,00	85627616,41	23,79	5,36	13300,00	87915393,50	24,42	5,80
14075,00	85834637,33	23,84	5,94	13375,00	88126292,44	24,48	6,40
14150,00	85844293,27	23,85	3,44	13450,00	88165886,28	24,49	3,72
14225,00	86053659,77	23,90	4,01	13525,00	88378393,62	24,55	4,31
14300,00	86054262,89	23,90	1,54	13600,00	88410803,26	24,56	1,67
14375,00	86265164,87	23,96	2,11	13675,00	88624921,61	24,62	2,27
14450,00	86260782,96	23,96	-0,32	13750,00	88650196,31	24,63	-0,34
14525,00	86473229,83	24,02	0,25	13825,00	88865928,43	24,68	0,25
14600,00	86461590,08	24,02	-2,16	13900,00	88883449,19	24,69	-2,32
14675,00	86675586,76	24,08	-1,59	13975,00	89100800,39	24,75	-1,73
14750,00	86658676,91	24,07	-3,96	14050,00	89112609,31	24,75	-4,26
14825,00	86874224,12	24,13	-3,40	14125,00	89331578,39	24,81	-3,68
14900,00	86849429,47	24,12	-5,74	14200,00	89335738,65	24,82	-6,18
14975,00	87066534,13	24,19	-5,18	14275,00	89556331,11	24,88	-5,60

15050,00	87283147,28	24,25	-4,62	14350,00	89553495,63	24,88	-8,06
15125,00	87128179,77	24,20	-8,35	14425,00	89775715,07	24,94	-7,48
15200,00	87347627,16	24,26	-7,80	14500,00	89765989,69	24,93	-9,91
15275,00	87565762,70	24,32	-7,24	14575,00	89989839,82	25,00	-9,34
15350,00	87398205,11	24,28	-10,89	14650,00	89973237,24	24,99	-11,73
15425,00	87619996,32	24,34	-10,34	14725,00	90198722,01	25,06	-11,16
15500,00	87841277,13	24,40	-9,80	14800,00	90175277,93	25,05	-13,53
15575,00	87654437,04	24,35	-13,38	14875,00	90402401,49	25,11	-12,96
15650,00	87878589,30	24,41	-12,84	14950,00	90372153,95	25,10	-15,29
15725,00	88102221,30	24,47	-12,30	15025,00	90600920,63	25,17	-14,73
15800,00	87899937,81	24,42	-15,82	15100,00	90829161,14	25,23	-14,17
15875,00	88126462,28	24,48	-15,28	15175,00	90657094,05	25,18	-17,89
15950,00	88352456,42	24,54	-14,74	15250,00	90888335,83	25,25	-17,34

0	Pressure exp= 121 kPa		
1			
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
11200,00	90891604,67	25,25	30,54
11275,00	91096205,75	25,30	31,21
11350,00	91264883,25	25,35	27,85
11425,00	91471195,46	25,41	28,51
11500,00	91631221,59	25,45	25,21
11575,00	91839244,64	25,51	25,87
11650,00	91990710,92	25,55	22,63
11725,00	92200444,72	25,61	23,27
11800,00	92343439,47	25,65	20,09
11875,00	92554884,06	25,71	20,73
11950,00	92689492,50	25,75	17,60
12025,00	92902648,12	25,81	18,24
12100,00	93028952,47	25,84	15,16
12175,00	93243819,54	25,90	15,79
12250,00	93361899,17	25,93	12,76
12325,00	93578478,26	25,99	13,39
12400,00	93688409,75	26,02	10,40
12475,00	93906701,61	26,09	11,03
12550,00	94008558,88	26,11	8,09
12625,00	94228564,43	26,17	8,72
12700,00	94322418,81	26,20	5,82
12775,00	94544139,15	26,26	6,45
12850,00	94630137,29	26,29	3,60
12925,00	94853545,06	26,35	4,21

13000,00	94931753,74	26,37	1,40
13075,00	95156868,23	26,43	2,02
13150,00	95227326,40	26,45	-0,75
13225,00	95454147,36	26,52	-0,14
13300,00	95516924,83	26,53	-2,86
13375,00	95745452,12	26,60	-2,26
13450,00	95800616,76	26,61	-4,94
13525,00	96030850,32	26,68	-4,34
13600,00	96078468,09	26,69	-6,99
13675,00	96310407,93	26,75	-6,39
13750,00	96350542,92	26,76	-9,00
13825,00	96584189,17	26,83	-8,40
13900,00	96613971,32	26,84	-10,98
13975,00	96849497,59	26,90	-10,39
14050,00	96874013,60	26,91	-12,92
14125,00	97111275,07	26,98	-12,33
14200,00	97128363,62	26,98	-14,84
14275,00	97367363,26	27,05	-14,25
14350,00	97377070,76	27,05	-16,72
14425,00	97617812,55	27,12	-16,14
14500,00	97620184,02	27,12	-18,58
14575,00	97862669,98	27,18	-18,00
14650,00	97857749,85	27,18	-20,41
14725,00	98101984,20	27,25	-19,83
14800,00	98089813,50	27,25	-22,20
14875,00	98335799,98	27,32	-21,63
14950,00	98316418,60	27,31	-23,97
15025,00	98564161,16	27,38	-23,40
15100,00	98811334,87	27,45	-22,83
15175,00	98646183,37	27,40	-26,58
15250,00	98896567,89	27,47	-26,01
15325,00	99146372,89	27,54	-25,45
15400,00	98963897,41	27,49	-29,13
15475,00	99216933,86	27,56	-28,57
15550,00	99469379,94	27,63	-28,00
15625,00	99269690,12	27,57	-31,62
15700,00	99525389,16	27,65	-31,06
15775,00	99780486,74	27,72	-30,50
15850,00	99563684,12	27,66	-34,06
15925,00	99822057,11	27,73	-33,50
16000,00	100079817,32	27,80	-32,95

Table B-2. Parameters for the stand-alone single stage cycle, cold climate

C2= 1	Pressure exp= 830 kPa			0,9	Pressure exp= 750 kPa		
C3 = 0				0,1			
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)	Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
13000,00	53221256,63	14,78	9,48	12000,00	48428528,40	13,45	9,00
13200,00	53370259,94	14,83	6,79	12200,00	48542679,29	13,48	5,95
13400,00	53512850,22	14,86	4,16	12400,00	48649130,10	13,51	2,98
13600,00	53649137,28	14,90	1,60	12600,00	48748005,04	13,54	0,09
13800,00	53779230,76	14,94	-0,89	12800,00	48839423,27	13,57	-2,72
14000,00	53903234,20	14,97	-3,32	13000,00	48923496,25	13,59	-5,45
14200,00	54021248,10	15,01	-5,69	13200,00	49000331,70	13,61	-8,11
14400,00	54133368,30	15,04	-8,00	13400,00	49070031,42	13,63	-10,70
14600,00	54239686,59	15,07	-10,25	13600,00	49132692,04	13,65	-13,22
14800,00	54340289,48	15,09	-12,44	13800,00	49188405,20	13,66	-15,67
15000,00	54435263,50	15,12	-14,58	14000,00	49237257,72	13,68	-18,07
15200,00	54524688,09	15,15	-16,67	14200,00	49279331,73	13,69	-20,39
15400,00	54608640,31	15,17	-18,71	14400,00	49314704,82	13,70	-22,66
15600,00	54687193,94	15,19	-20,70	14600,00	49343450,19	13,71	-24,88
15800,00	54760419,56	15,21	-22,64	14800,00	49365636,72	13,71	-27,03
16000,00	54828384,74	15,23	-24,54	15000,00	49382517,55	13,72	-28,90
16200,00	54891154,15	15,25	-26,39	15200,00	49404929,73	13,72	-29,08
16400,00	54948789,66	15,26	-28,20	15400,00	49441547,00	13,73	-29,25
16600,00	55001350,52	15,28	-29,97	15600,00	49493162,62	13,75	-29,42
16800,00	55048893,42	15,29	-31,69	15800,00	49561009,31	13,77	-29,58
17000,00	55091472,67	15,30	-33,38	16000,00	49640295,77	13,79	-29,73
17200,00	55129140,28	15,31	-35,03	16200,00	49732857,03	13,81	-29,88
17400,00	55161946,09	15,32	-36,64	16400,00	49836468,18	13,84	-30,03
17600,00	55192268,31	15,33	-38,06	16600,00	49950006,66	13,88	-30,17
17800,00	55224337,82	15,34	-38,06	16800,00	50073777,39	13,91	-30,30
18000,00	55267811,51	15,35	-38,06	17000,00	50207224,57	13,95	-30,43
18200,00	55326261,46	15,37	-38,06	17200,00	50365733,71	13,99	-30,55
18400,00	55390742,05	15,39	-38,06	17400,00	50518201,87	14,03	-30,68
				17600,00	50679406,05	14,08	-30,79
				17800,00	50850325,39	14,13	-30,90
				18000,00	51029823,74	14,17	-31,01
				18200,00	51218097,15	14,23	-31,12
				18400,00	51414538,12	14,28	-31,22

0,8	Pressure exp= 672 kPa		
0,2			
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
11200,00	44966732,97	12,49	8,88
11400,00	45057505,24	12,52	5,49
11600,00	45139455,03	12,54	2,20
11800,00	45212616,85	12,56	-0,99
12000,00	45277113,52	12,58	-4,09
12200,00	45333059,68	12,59	-7,11
12400,00	45380563,73	12,61	-10,04
12600,00	45419726,76	12,62	-12,89
12800,00	45450643,04	12,63	-15,66
13000,00	45473400,24	12,63	-18,35
13200,00	45488148,28	12,64	-20,97
13400,00	45494176,63	12,64	-23,50
13600,00	45505818,96	12,64	-23,71
13800,00	45543022,19	12,65	-23,92
14000,00	45594834,55	12,67	-24,13
14200,00	45665527,61	12,68	-24,33
14400,00	45754369,89	12,71	-24,52
14600,00	45855444,47	12,74	-24,71
14800,00	45979442,27	12,77	-24,89
15000,00	46112120,20	12,81	-25,07
15200,00	46258638,11	12,85	-25,24
15400,00	46418314,16	12,89	-25,41
15600,00	46592139,14	12,94	-25,57
15800,00	46774651,29	12,99	-25,73
16000,00	46966584,48	13,05	-25,89
16200,00	47156004,69	13,10	-26,04
16400,00	47345540,52	13,15	-26,19
16600,00	47531258,00	13,20	-26,33
16800,00	47723944,37	13,26	-26,47
17000,00	47908754,67	13,31	-26,61

0,7	Pressure exp= 598 kPa		
0,3			
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
10500,00	42353913,47	11,76	9,49
10700,00	42431234,83	11,79	5,76
10900,00	42497786,76	11,80	2,15
11100,00	42555136,92	11,82	-1,35
11300,00	42603172,21	11,83	-4,75
11500,00	42640829,16	11,84	-8,06
11700,00	42668873,99	11,85	-11,26
11900,00	42687400,49	11,86	-14,37
12100,00	42696495,72	11,86	-17,39
12300,00	42697838,08	11,86	-20,28
12500,00	42704805,22	11,86	-20,49
12700,00	42739801,58	11,87	-20,70
12900,00	42798554,04	11,89	-20,91
13100,00	42877505,14	11,91	-21,11
13300,00	42980837,21	11,94	-21,30
13500,00	43103184,57	11,97	-21,49
13700,00	43243074,03	12,01	-21,68
13900,00	43403750,94	12,06	-21,86
14100,00	43577847,07	12,10	-22,04
14300,00	43756940,87	12,15	-22,21
14500,00	43935787,05	12,20	-22,38
14700,00	44116206,39	12,25	-22,55
14900,00	44295314,67	12,30	-22,71
15100,00	44470664,63	12,35	-22,87
15300,00	44652601,21	12,40	-23,02
15500,00	44832822,65	12,45	-23,17
15700,00	45011938,19	12,50	-23,32
15900,00	45190904,24	12,55	-23,46
16100,00	45372527,44	12,60	-23,60
16300,00	45548215,39	12,65	-23,74
16500,00	45726994,60	12,70	-23,88
16700,00	45906193,66	12,75	-24,01
16900,00	46084515,95	12,80	-24,14

0,6	Pressure exp= 525 kPa			0,5	Pressure exp= 453 kPa		
0,4				0,5			
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)	Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
10000,00	40422828,80	11,23	8,09	9400,00	38965167,68	10,82	10,22
10200,00	40488671,97	11,25	4,09	9600,00	39033224,04	10,84	5,85
10400,00	40542260,31	11,26	0,22	9800,00	39088924,58	10,86	1,62
10600,00	40584613,59	11,27	-3,53	10000,00	39132680,23	10,87	-2,47
10800,00	40616361,54	11,28	-7,16	10200,00	39164466,75	10,88	-6,43
11000,00	40637331,96	11,29	-10,69	10400,00	39184396,98	10,88	-10,27
11200,00	40647618,14	11,29	-14,10	10600,00	39192571,52	10,89	-13,99
11400,00	40647306,61	11,29	-17,42	10800,00	39189076,65	10,89	-17,60
11600,00	40645103,53	11,29	-18,73	11000,00	39190924,92	10,89	-18,36
11800,00	40671301,28	11,30	-18,92	11200,00	39229195,39	10,90	-18,52
12000,00	40727117,65	11,31	-19,11	11400,00	39301480,57	10,92	-18,69
12200,00	40812311,95	11,34	-19,30	11600,00	39404637,41	10,95	-18,85
12400,00	40921149,45	11,37	-19,48	11800,00	39541022,29	10,98	-19,00
12600,00	41056779,38	11,40	-19,66	12000,00	39696066,86	11,03	-19,16
12800,00	41214483,72	11,45	-19,83	12200,00	39851070,66	11,07	-19,31
13000,00	41382042,12	11,50	-20,00	12400,00	40008479,38	11,11	-19,46
13200,00	41545860,44	11,54	-20,17	12600,00	40164866,16	11,16	-19,60
13400,00	41718002,75	11,59	-20,33	12800,00	40323712,96	11,20	-19,74
13600,00	41884757,34	11,63	-20,49	13000,00	40476789,91	11,24	-19,88
13800,00	42052171,05	11,68	-20,65	13200,00	40635435,95	11,29	-20,02
14000,00	42222194,97	11,73	-20,80	13400,00	40792161,58	11,33	-20,15
14200,00	42387722,53	11,77	-20,95	13600,00	40949862,68	11,37	-20,28
14400,00	42557954,18	11,82	-21,10	13800,00	41107864,66	11,42	-20,41
14600,00	42724992,18	11,87	-21,24	14000,00	41266343,76	11,46	-20,54
14800,00	42892908,85	11,91	-21,38	14200,00	41422816,07	11,51	-20,66
15000,00	43063499,66	11,96	-21,52	14400,00	41580780,04	11,55	-20,79
15200,00	43227524,41	12,01	-21,66	14600,00	41738808,84	11,59	-20,91
15400,00	43397969,05	12,05	-21,79	14800,00	41900709,33	11,64	-21,02
15600,00	43568077,86	12,10	-21,92	15000,00	42056794,79	11,68	-21,14
15800,00	43735978,00	12,15	-22,05	15200,00	42212705,61	11,73	-21,25
16000,00	43903172,48	12,20	-22,17	15400,00	42371242,75	11,77	-21,36
16200,00	44072267,66	12,24	-22,30	15600,00	42530330,50	11,81	-21,47
16400,00	44242303,05	12,29	-22,42	15800,00	42689015,73	11,86	-21,58
16600,00	44410003,91	12,34	-22,54	16000,00	42849684,32	11,90	-21,68
16800,00	44577811,19	12,38	-22,65				
17000,00	44742452,92	12,43	-22,76				

0,4 / 0,6	Pressure exp= 383 kPa		
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
9000,00	38060524,30	10,57	9,24
9200,00	38129305,76	10,59	4,60
9400,00	38184918,06	10,61	0,11
9600,00	38227509,51	10,62	-4,23
9800,00	38257212,03	10,63	-8,44
10000,00	38274141,23	10,63	-12,51
10200,00	38278396,08	10,63	-16,45
10400,00	38272720,06	10,63	-19,08
10600,00	38297377,43	10,64	-19,22
10800,00	38365209,25	10,66	-19,35
11000,00	38470178,98	10,69	-19,48
11200,00	38606557,59	10,72	-19,61
11400,00	38753246,69	10,76	-19,73
11600,00	38899614,15	10,81	-19,86
11800,00	39046417,91	10,85	-19,98
12000,00	39192913,89	10,89	-20,10
12200,00	39339760,40	10,93	-20,21
12400,00	39482390,19	10,97	-20,33
12600,00	39633929,39	11,01	-20,44
12800,00	39781009,86	11,05	-20,55
13000,00	39928628,83	11,09	-20,66
13200,00	40077143,71	11,13	-20,77
13400,00	40220791,24	11,17	-20,87
13600,00	40372121,25	11,21	-20,97
13800,00	40521932,88	11,26	-21,08
14000,00	40671220,95	11,30	-21,18
14200,00	40820631,60	11,34	-21,27
14400,00	40968854,71	11,38	-21,37
14600,00	41121166,80	11,42	-21,46
14800,00	41268906,41	11,46	-21,55
15000,00	41416177,64	11,50	-21,65
15200,00	41566395,93	11,55	-21,74
15400,00	41716190,24	11,59	-21,82
15600,00	41866547,91	11,63	-21,91
15800,00	42010723,61	11,67	-22,00
16000,00	42164787,55	11,71	-22,08

0,3 / 0,7	Pressure exp= 315 kPa		
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
8600,00	37675017,81	10,47	9,24
8800,00	37753591,19	10,49	4,29
9000,00	37818078,49	10,51	-0,49
9200,00	37868644,91	10,52	-5,12
9400,00	37905436,67	10,53	-9,59
9600,00	37928581,46	10,54	-13,92
9800,00	37938188,75	10,54	-18,11
10000,00	37935930,13	10,54	-21,13
10200,00	37967579,56	10,55	-21,23
10400,00	38043597,70	10,57	-21,33
10600,00	38165788,60	10,60	-21,43
10800,00	38303696,83	10,64	-21,52
11000,00	38442610,39	10,68	-21,62
11200,00	38581441,96	10,72	-21,71
11400,00	38720035,37	10,76	-21,80
11600,00	38861009,31	10,79	-21,89
11800,00	38999903,36	10,83	-21,97
12000,00	39141371,83	10,87	-22,06
12200,00	39284357,66	10,91	-22,14
12400,00	39422594,02	10,95	-22,23
12600,00	39563191,10	10,99	-22,31
12800,00	39709532,49	11,03	-22,39
13000,00	39849727,18	11,07	-22,46
13200,00	39987684,62	11,11	-22,54
13400,00	40130194,78	11,15	-22,62
13600,00	40269987,93	11,19	-22,69
13800,00	40412946,90	11,23	-22,76
14000,00	40554823,29	11,27	-22,84
14200,00	40698850,99	11,31	-22,91
14400,00	40841610,65	11,34	-22,98
14600,00	40988241,33	11,39	-23,05
14800,00	41130083,66	11,43	-23,11
15000,00	41270522,58	11,46	-23,18

0,2	Pressure exp= 239 kPa			0,1	Pressure exp= 183 kPa		
0,8				0,9			
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)	Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
8200,00	37935263,85	10,54	10,22	7900,00	39218279,18	10,89	9,35
8400,00	38034322,55	10,57	4,92	8100,00	39341737,77	10,93	3,77
8600,00	38116971,36	10,59	-0,19	8300,00	39449083,86	10,96	-1,61
8800,00	38185657,55	10,61	-5,12	8500,00	39540547,06	10,98	-6,80
9000,00	38240100,89	10,62	-9,89	8700,00	39616334,69	11,00	-11,82
9200,00	38279564,58	10,63	-14,51	8900,00	39676632,48	11,02	-16,68
9400,00	38305089,46	10,64	-18,98	9100,00	39721605,23	11,03	-21,37
9600,00	38314937,58	10,64	-23,30	9300,00	39751397,29	11,04	-25,92
9800,00	38329170,38	10,65	-24,66	9500,00	39767423,08	11,05	-29,98
10000,00	38396489,35	10,67	-24,72	9700,00	39808950,88	11,06	-30,00
10200,00	38507668,32	10,70	-24,79	9900,00	39910111,81	11,09	-30,03
10400,00	38645518,66	10,73	-24,85	10100,00	40055639,67	11,13	-30,06
10600,00	38781919,78	10,77	-24,91	10300,00	40197495,70	11,17	-30,09
10800,00	38918324,43	10,81	-24,97	10500,00	40342917,65	11,21	-30,12
11000,00	39056904,27	10,85	-25,02	10700,00	40490410,06	11,25	-30,15
11200,00	39197407,69	10,89	-25,08	10900,00	40635357,46	11,29	-30,17
11400,00	39333920,44	10,93	-25,14	11100,00	40782499,34	11,33	-30,20
11600,00	39468109,46	10,96	-25,19	11300,00	40925631,82	11,37	-30,23
11800,00	39610190,23	11,00	-25,25	11500,00	41070991,38	11,41	-30,25
12000,00	39749271,48	11,04	-25,30	11700,00	41219666,19	11,45	-30,28
12200,00	39887814,70	11,08	-25,35	11900,00	41365905,67	11,49	-30,30
12400,00	40026530,69	11,12	-25,41	12100,00	41512263,28	11,53	-30,32
12600,00	40166161,59	11,16	-25,46	12300,00	41657058,43	11,57	-30,35
12800,00	40307052,44	11,20	-25,51	12500,00	41805267,27	11,61	-30,37
13000,00	40447630,37	11,24	-25,56	12700,00	41951921,79	11,65	-30,40
13200,00	40586221,31	11,27	-25,61	12900,00	42098687,05	11,69	-30,42
13400,00	40726510,33	11,31	-25,65	13100,00	42245320,86	11,73	-30,44
13600,00	40870846,37	11,35	-25,70	13300,00	42392561,76	11,78	-30,46
13800,00	41006921,57	11,39	-25,75	13500,00	42537413,15	11,82	-30,49
14000,00	41147197,09	11,43	-25,79	13700,00	42686698,80	11,86	-30,51
14200,00	41288530,24	11,47	-25,84	13900,00	42833288,71	11,90	-30,53
14400,00	41430015,76	11,51	-25,88	14100,00	42982501,59	11,94	-30,55
14600,00	41569510,67	11,55	-25,93	14300,00	43129027,11	11,98	-30,57
14800,00	41715874,95	11,59	-25,97	14500,00	43276519,95	12,02	-30,59
15000,00	41853557,46	11,63	-26,01	14700,00	43425128,36	12,06	-30,61
				14900,00	43571660,62	12,10	-30,63

0	Pressure exp= 121 kPa		
1			
Flow (kmol/h)	Power (MW and kJ/h)		Tout (°C)
7600,00	42301100,60	11,75	9,17
7800,00	42468387,49	11,80	3,29
8000,00	42618179,70	11,84	-2,38
8200,00	42750784,83	11,88	-7,86
8400,00	42864488,97	11,91	-13,15
8600,00	42962433,67	11,93	-18,26
8800,00	43043673,01	11,96	-23,21
9000,00	43108401,34	11,97	-28,00
9200,00	43156791,76	11,99	-32,65
9400,00	43188996,86	12,00	-37,15
9600,00	43231061,46	12,01	-38,06
9800,00	43334986,34	12,04	-38,06
10000,00	43494399,71	12,08	-38,06
10200,00	43665208,60	12,13	-38,06
10400,00	43836017,43	12,18	-38,06
10600,00	44006826,29	12,22	-38,06
10800,00	44177635,15	12,27	-38,06
11000,00	44348444,01	12,32	-38,06
11200,00	44519252,88	12,37	-38,06
11400,00	44690061,74	12,41	-38,06
11600,00	44860870,60	12,46	-38,06
11800,00	45031679,48	12,51	-38,06
12000,00	45202488,30	12,56	-38,06
12200,00	45373297,17	12,60	-38,06
12400,00	45544106,03	12,65	-38,06
12600,00	45714914,90	12,70	-38,06
12800,00	45885723,76	12,75	-38,06
13000,00	46056532,62	12,79	-38,06
13200,00	46227341,47	12,84	-38,06
13400,00	46398150,32	12,89	-38,06
13600,00	46568959,19	12,94	-38,06
13800,00	46739768,04	12,98	-38,06
14000,00	46910576,92	13,03	-38,06
14200,00	47081385,75	13,08	-38,06
14400,00	47252194,63	13,13	-38,06
14600,00	47423003,48	13,17	-38,06
14800,00	47593812,37	13,22	-38,06
15000,00	47764621,23	13,27	-38,06

Appendix C: Multistage compression

Consider a process in which a gas at certain conditions of pressure (P_1) and temperature (T_1) has to increase its pressure to a given P_2. If the process will be carried out through two consecutive stages with intercooling, the amount of work required to do the task is given by Equation C.1, under adiabatic conditions.

$$W = R\frac{k}{k-1}\left[T_1\left(\left(\frac{P_i}{P_1}\right)^{\frac{k-1}{k}} - 1\right) + T_i\left(\left(\frac{P_2}{P_i}\right)^{\frac{k-1}{k}} - 1\right)\right] \tag{C.1}$$

W represents the sum of the work of the first and second stage compressors, R is the universal gas constant, k is the ratio of specific heat at constant pressure to the specific heat at constant volume (Cp/Cv), P_1 and T_1 are known inlet pressure and temperature, P_i and T_i are the pressure and temperature of the intermediate stage, whilst P_2 is the pressure to be reached by the process.

In order to optimize such a two stage compression process, based on the minimum work to be achieved, a differentiation of W is performed and the resulting expression is set to equal zero. The result of the optimization is shown in Equation C.2.

$$P_{i(optimal)} = \left(\frac{T_i}{T_1}\right)^{\frac{k}{2k-2}}\sqrt{P_3.P_1} \tag{C.2}$$

For most of the practical applications the given equation can be further simplified if it is assumed that the temperature achieved by the intercooler between stages (T_i) is equal to the inlet temperature of the compressor (T_1), this is shown in Equation C.3.

$$P_{i(optimal)} = \sqrt{P_3.P_1} \tag{C.3}$$