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Future CO_2 refrigeration systems for hot climates

Master Thesis

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Future CO_2 refrigeration systems for hot climates

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by
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To Mum and Dad

Preface

The present thesis was prepared at the Section of Thermal Energy, Department of Mechanical Engineering, Technical University of Denmark (DTU). It is submitted as final requirement to complete the MSc "Sustainable Energies, Thermal Energy" and it is written as a monograph.

The work was carried out for approximatively six months, from January 2015 to July 2015, under the supervision of Assistant Professor Wiebke Brix Markussen (DTU) and co-supervision of Researchers Martin Ryhl Kaern (DTU).

An external partner was part of this project, the Danish company *The Danfoss Group*. They kindly offered their expertise in the sector, sharing information and confidential findings.

Un ringraziamento è poi dovuto alla professoressa Luisa Rossetto (Università degli studi di Padova) che ha eseguito un'ulteriore attenta revisione del lavoro.

København, 1st July 2015 Stefano Bissoli.

Abstract

During the last decades, the refrigeration industry was one of the sector mainly involved in the increasing concern about sustainability and environmental issues.
From the 1950s until recently, the refrigerants used were only CFCs and similar. Unfortunately, these well-performing refrigerants were discovered to be main responsible of ozone depletion (CFCs and HCFCs) and increased green house effect (HFCs). For this reason, the cold industry is asked to move on and find sustainable alternatives. Between the so called "natural refrigerants", carbon dioxide seems to be the most promising solution.
The main disadvantage of CO_2 used in traditional vapour-compression cycle is the transcritical operation, prominent especially when the ambient temperatures are high. This project deals with possible solutions aimed to improve the performances of CO_2 vapour-compression cycle applied in hot climates. The tools considered are parallel compression, mechanical subcooling and the use of ejectors for expansion power recovery. Firstly, these solutions are investigated under fixed design conditions. Subsequently, they are simulated during a real-operation year using part-load modelling of the cycles.
As the reader will discover from this report, the ejector cycle seems to be the most promising solution, whether when comparing the cycles under design conditions or during a real year of operation (Rome's weather conditions). The PCE cycle offers also interesting improvements, followed by the mechanical subcooling cycles.

Keywords. Carbon dioxide, refrigeration systems, vapour-compression cycles, ejectors, parallel compression, mechanical subcooling.

Contents

Nomenclature

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Abbreviations	
CFC	chlorofluorocarbon
CO_2	carbon dioxide
EERC	ejector expansion refrigeration cycle
EES	engineering equation solver
EPR	expansion power recovery
GWP	global warming potential
HCFC	hydrochlorofluorocarbon
HFC	hydrofluorocarbon
IHE	interna heat exchange
MS	mechanical subcooling
ODP	ozone depletion potential
PCE	parallel compression economization
PI	piping and instrumentation
R744	carbon dioxide
TERC	turbine expander refrigeration cycle
TEWI	total equivalent warming impact
VCRC	vapour compression refrigeration cycle

Symbols		
COP	coefficient of performances	[-]
c_p	specific heat at constant pressure	[J/kgK]
ΔT	temperature difference	[°C]
e	evaporation enthalpy	[J/kg]
h	enthalpy	[J/kg]
m	mass flow rate	[kg]
η	efficiency	[-]
p	pressure	[bar]
$\dot{Q}$	heat flux	[W]
r	ratio of the motive flow and the total ejector mass flow rate	[-]
s	entropy	[J/kgK]
T	temperature	[°C]
u	velocity	[m/s]
UA	UA-value of the heat exchanger	[W/m^2K]
$\dot{W}$	mechanical power	[W]
ω	entrainment ratio	[-]
x	quality	[kg/kg]

Subscripts	
0	design conditions
compr	compressor
cond	condensation
d	diffuser
eco	economizer
evap	evaporation
gc	gas cooler
i	inlet
id	ideal or reversible process
is	isentropic
lift	lift
m	motive
main	main cycle
max	maximum
n	normalized
o	outlet
opt	optimal
s	suction
sub	subcooler or subcooling cycle
sc	subcooling

1 Introduction

During the last decades, the refrigeration industry was forced to face some radical changes. The use of CFCs (chlorofluorocarbons), starting from the 1950s, was a turning point for this sector. These refrigerants were particularly suitable for the refrigeration applications and ensured high performances of the traditional vapour-compression systems. Anyway, in the middle of the 1980s, CFCs were discovered to be incredibly harmful for the planet we are living on. They are considered one of the main responsible of the stratospheric ozone's depletion phenomena. Unsuccessful attempts to replaced the CFCs with similar compounds were made, they are the HCFCs (hydrochlorofluorocarbons) and more recently the HFCs (hydrofluorocarbons). Anyway, the former still cause ozone depletion if released in the atmosphere, while the second emphasises the green house effect.
All this awareness about the negative environmental impact of the traditional refrigerants it is quite recent. Even more recent is the growing attention for the "health" of our planet. The turning point was probably the Montreal Protocol (1987), a huge international agreement against the compounds mainly responsible of the ozone depletion. From that moment on, the concentration of the media and public opinion on the sustainable topics started to grow, day after day.
The Montreal Protocol already included actions against CFCs and HCFCs. These refrigerant are almost off the market nowadays, only their "brothers", the HFCs are still trying to survive the arising sustainable wave. Recently, also the HFCs started to be target of political actions in order to decrease the human impact on the Earth. Thus, it is clear why the cold chain industry is now (more than ever) looking for a new sustainable solution able to replace the traditional refrigerants, once and for all.
Actually, carbon dioxide (CO_2) is not a new solution. It was widely studied and used for refrigeration application before the CFC were even born. It was abandoned only due to the lower performances respect these artificial refrigerants. Now that CFCs and similar will progressively disappear from the market, CO_2 is probably the most reliable solution for the refrigeration industry. Its impact on the green house effect and/or the ozone depletion is almost negligible. It is more than available on our planet, it is cheap and produced as secondary product in many industrial processes. Moreover, on average, the CO_2 can be considered as a refrigerant with

very good properties for refrigeration applications.
The main problem arising from the use of carbon dioxide is its low critical point. When this refrigerant is used in traditional vapour-compression refrigeration cycles and the ambient temperatures are, on average, quite high, the cycle runs for most of the hours during the year as transcritical. This implies considerable inefficiencies in the cycle, if compared to the same application using the "traditional" refrigerants. This project is focused exactly on this. Here the aim is to study possible solutions able to improve the performances of vapour-compression CO_2 cycles, especially when used in hot climates.
The main tools considered are three: mechanical subcooling, parallel compression and the use of ejectors. These solution are here studied and compared each other to determine their potential to improve the simple stage vapour-compression CO_2 cycle.
In the first part of this report, the three devices used to improve CO_2 cycles are studied under specific design conditions. These conditions are specifically determined in order to assess the behaviour of these systems when applied in hot climates. Some assumptions are made in order to simplify the analyses. For instance, the isentropic efficiency of the compressors is considered constant, like the cooling capacity required to the systems. At the end of this first part, the cycles are compared each other and with the 1-stage reference cycle in order to assess eventual performances improvements.
The second part of the project is again about behaviour of CO_2 vapour-compression systems under fixed design conditions. Here, more complex cycles are considered in order to assess the possibility to reach even greater performances improvements. The tools studied in the first part of the project are now combined together. The complexity of the cycles increases as, hopefully, the COP of the systems. Not only the already investigated solutions are improved. This part deals also with other possible devices usable to improve CO_2 vapour-compression cycle operating in hot climates. This represents probably the most creative part of the project. A huge number of different solution are implemented, simulated and compared each other. Hopefully, many hints for future studies can be taken from this part of the report.
Finally, the third part of the project is about the part-load operation of the studied cycles (mechanical subcooling, parallel compression and ejector cycle). From the point of view of possible future application of these systems, this is probably the most interesting part. Studies about the design conditions of the considered cycles give important hints about the systems' behaviour under specific conditions. Anyway, a refrigeration system usually faces continuously changing working conditions along one typical year of operation. The part-load investigation allows for an accurate computational simulation of the system's behaviour when the working conditions are changing. In this way, it is possible to assess which one of the considered solutions is the most promising considering one entire year of operation. The considered cycles were investigated under the weather conditions of Rome, Italy. The location was chosen on the base of the temperate/hot climate that makes the Italian city particularly suitable for the investigations here performed.
The idea is to perform a step by step analysis of possible improvements of CO_2 vapour-compression cycles. Firstly, they are investigated under fixed conditions, compared each other and with other possible solution. After that, the focus goes back to the three initial solutions

investigated and they are tested for real working conditions during an year in Rome, Italy. In this way, the project is able to quantify the improvements produced by the considered solutions for fixed conditions (initially) and (after) for real-conditions of operation.

The need for an accurate investigation of CO_2 transcritical systems came from the recent interest of the Danish company DANFOSS for supermarkets' CO_2 refrigeration cycles. As reported by Sharma et al., based on the U.S. supermarkets average, leakages of refrigerants are estimated between 3% and 25% of the charge [1]. The wide range is due to the fact that new and old equipments have different performances. Leakages of "traditional" refrigerants have a great impact on the ambient and contribute directly to ozone depletion (CFCs and HCFCs) and global warming (HFCs). *"Supermarkets are intensively energy consuming with constantly increasing number of installations"* [2]. For this reason, the direct impact of refrigerants leakages from supermarkets applications is an aspect that is becoming more and more important day by day. The attention is gradually shifting to CO_2 that seems the most promising solution among the "natural refrigerants".
"Furthermore, the operation of refrigeration systems contributes to global warming indirectly" [1]. Greenhouse gasses are typically released during generation and transmission of the electricity required by the supermarkets. This is an even more important issue in CO_2 transcritical systems because they are characterized by lower COPs respect cycles using "traditional" refrigerants. *"Transcritical CO_2 systems tend to be more popular in moderate climates such as Northern Europe where the refrigeration system operates a majority of the time in the more efficient subcritical mode"* [1]. The so called "booster" CO_2 transcritical cycle is a common solution for cold climates. As reported by Minetto et al., *"the booster cycle is actually the state-of-art solution for North Europe, where near 3000 installations, with variable lay-outs, are currently running only on CO_2"* [3]. Recently, more components for CO_2 systems are appearing on the market and this is paving the way for a massive usage of $CO_2 - only$ cycles [4]. It must be remembered that the first supermarkets carbon dioxide applications were only in cascade systems or in secondary loops (so CO_2 combined with "traditional" refrigerants), due to the lower knowledge and components availability. Nowadays these problems are almost overtaken and interest is arising about the possibility to use efficiently transcritical $CO_2 - only$ systems even in hot climates. Differently from the cited cascade and secondary loop systems, transcritical systems offers the possibility to use CO_2 as unique refrigerant in the system: this allows for the shift to "natural" refrigerants that the cold industry is looking for. The solutions already introduced, mechanical subcooling, PCE and ejectors, ensure performances' improvements of the low-efficient CO_2 transcritical cycles and are quite interesting expedients even for supermarket applications.
Anyway, as already said, the environmental impact of "traditional refrigerants" and the necessity to reach the highest possible COPs are common problems of all the refrigeration sectors. The refrigeration industry as an all needs to shift to environmentally friendly refrigerants. Moreover, with the increasing global consumptions, highly efficient systems are required in order to produce with lowered input requirements. For these reasons, in this project the

investigation is kept quite general. The CO_2 transcritical refrigeration cycles' improvements are tested for no particular application. In this way, the results obtained show trends that can be applied to whichever refrigeration sector.

All the results presented in this report were obtained with computational simulations based on the software EES (Engineering Equation Solver) [5]. The data processing was performed using common spread sheets.

2 Carbon dioxide as a refrigerant

During the last decades, the refrigeration industry was forced to face great changes due to the increasing concern about environmental issues.
From the 1950s, the CFCs were hugely employed as refrigerants in the cold industry. They ensured very good performances but they were found to be one of the main responsible of the stratosphere's ozone depletion. As an attempt to reduce the environmental impact, the refrigerants HCFCs were introduced due to the lower ozone depletion impact. Anyway, common idea is to move forward also from these freons due to the still high environmental impact.
The HFCs were expected to be the final solution due to the almost insignificant effect on the stratosphere's ozone but they highly increase the green house effect with consequent serious implications on the Hearth's climate change.
Lorentzen summarized very well the situation that is now facing the refrigeration industry: *"In the present situation, when the CFCs and in a little longer prospective the HCFCs are being banned by international agreement, it does not seem very logical to try to replace them by another family of related halocarbons, the HFCs, equally foreign to nature"* [6]. Right now, the cold industry is looking for a permanent solution that will not be banned or regulated in the long run. Great interest is pointed to "natural" refrigerants, like water, air, hydrocarbons, ammonia and carbon dioxide.

2.1 Old refrigerants

The halocarbons refrigerants first appeared on the market in the 1930s, when the R12 was introduced. What characterizes the history of these refrigerants is the velocity they spread through almost all the refrigeration applications. Anyway, this is not a surprising aspect of their stories. The CFCs were strongly promoted by their own producers. Moreover, the CFC's companies put incredible efforts to allow also the refrigeration technology to follow the refrigerants' changes. Finally, CFCs systems were characterized by *"simple and relatively cheap operation, not without interest to the service firms and refrigerants supplier"*, as reported by

Characteristics of some refrigerants

	R-12	R-22	R-134a	R-407C[a]	R-410A[b]	R-717	R-290	R-744
ODP/GWP[c]	1/8500	0.05/1700	0/1300	0/1600	0/1900	0/0	0/3	0/1
Flammability/toxicity	N/N	N/N	N/N	N/N	N/N	Y/Y	Y/N	N/N
Molecular mass (kg/kmol)	120.9	86.5	102.0	86.2	72.6	17.0	44.1	44.0
Normal boiling point[d] (°C)	−29.8	−40.8	−26.2	−43.8	−52.6	−33.3	−42.1	−78.4
Critical pressure (MPa)	4.11	4.97	4.07	4.64	4.79	11.42	4.25	7.38
Critical temperature (°C)	112.0	96.0	101.1	86.1	70.2	133.0	96.7	31.1
Reduced pressure[e]	0.07	0.10	0.07	0.11	0.16	0.04	0.11	0.47
Reduced temperature[f]	0.71	0.74	0.73	0.76	0.79	0.67	0.74	0.90
Refrigeration capacity[g] (kJ/m^3)	2734	4356	2868	4029	6763	4382	3907	22545
First commercial use as a refrigerant [14]	1931	1936	1990	1998	1998	1859	?	1869

[a] Ternary mixture of R-32/125/134a (23/25/52, %).
[b] Binary mixture of R-32/125 (50/50, %).
[c] Global warming potential in relation to 100 years integration time, from the Intergovernmental Panel on Climate Change (IPCC).
[d] ASRAE handbook 2001 fundamentals.
[e] Ratio of saturation pressure at 0 °C to critical pressure.
[f] Ratio of 273.15 K (0 °C) to critical temperature in Kelvin.
[g] Volumetric refrigeration capacity at 0 °C.

Figure 2.1 – Characteristics of some traditional and natural refrigerants including CO_2 (R-744) [7]

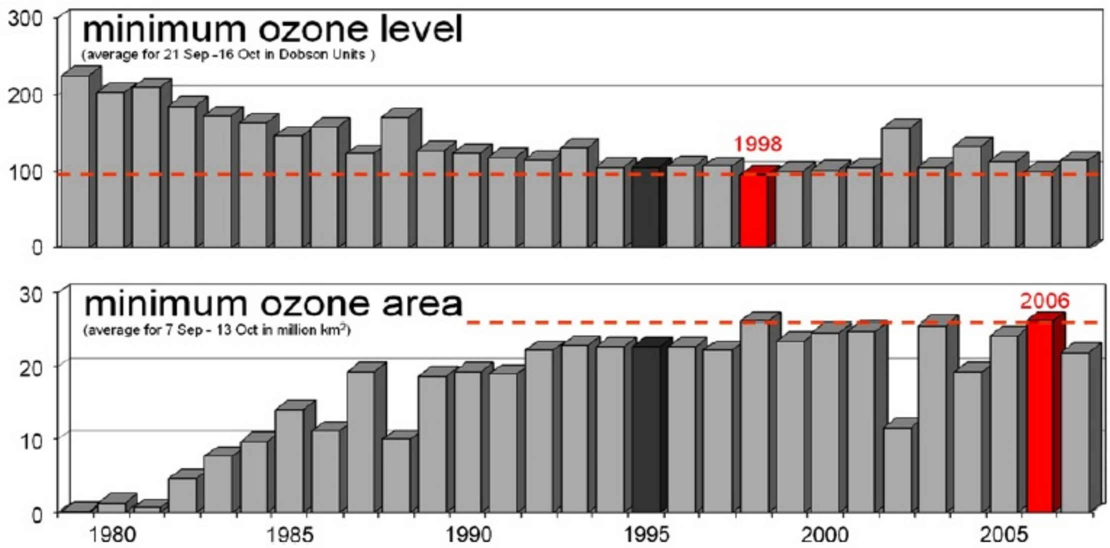

Figure 2.2 – Minimum ozone level and area for the last 30 years [8]

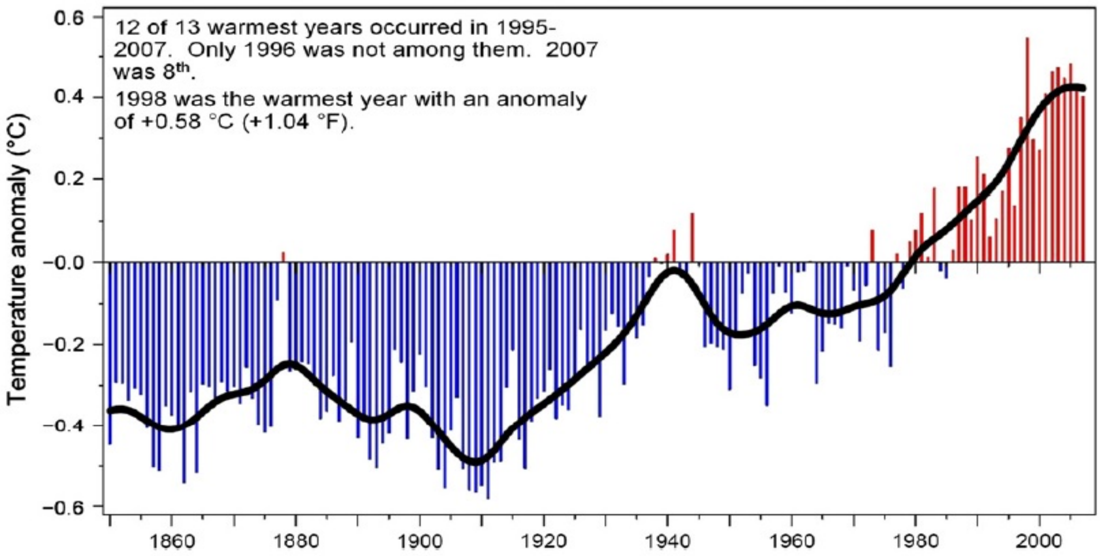

Figure 2.3 – Global temperatures rise [8]

Lorentzen [6]. All these factors, together with the unquestionable suitability of this refrigerants for cold production applications, determined the overwhelming success of the refrigerants. During "the young" years of CFCs, only ammonia was still used for large industrial application, all the other sectors of the cold production were dominated by halocarbons refrigerants.
Figure 2.1 presents a comparison between some "traditional" refrigerants (CFC, HCFC and HFC) with ammonia, propane and carbon dioxide (probably the most promising "natural refrigerants"). CFCs and HCFCs are characterized by an alarming impact on the stratospheric ozone. Since the Vienna Convention (1985) and the following Montreal Protocol (1987), all the industrialized world started a unite action against them. The Montreal Protocol already disposed for the abandonment of CFCs and set increasing limits in the HCFCs use. As known, these international agreements represents mainly guidelines for the participant countries. Moreover, they usually apply only to developed countries. Anyway, the road is already defined: CFCs and HCFCs are destined to leave the market.
The efforts made in the last decades to save the precious ozone layer seem to be giving some results. Figure 2.2 presents the minimum ozone level for the last thirty years. It is called "ozone hole" an area with less than 220 Dobson units. As visible, the worst moment was in 1998 but recently the ozone level seems to stabilize or even improve slowly. As reported by Calm, *"the progress in ozone recovery is even more evident when measured by global mean ozone rather than ozone in the isolated Antarctic vortex"* [8].
The HFCs, expected to be the long term solution from the environmental point of view, still have a really high value of GWP. The global warming potential (GWP) is used to compare the effect of a chemical on the green house effect using as reference the CO_2 (that for this reason has $GWP_{CO_2} = 1$). The increasing green house effect is as worrying as the ozone layer depletion. As reported by Calm, *"the 1990s are the warmest decade in the last millennium and*

the 20th century is its warmest century" [8]. Figure 2.3 is here reported to prove the increasing global temperatures. The human responsibility for the global warming is as clear as it is for the ozone depletion. The HFCs were one of the numerous compounds included in the Kyoto protocol (1997) and even if it is not in force right now, its echo has been felt all over the world. If the Kyoto Protocol can be interpreted as a symbolic act, the sunset of HFCs as refrigerants was definitively established with the European Parliament vote of 2006. With this first international act against HFCs, it was established the abandon of HFC-134a as refrigerant for automotive air-conditioning systems.
For this reason, the refrigeration industry is looking forward for new environmentally friendly long term solution: the "natural refrigerants".

2.2 New refrigerants

As reported by Pearson [9], nowadays the fluids considered as "natural refrigerants" are five. Air can be used for some applications without phase change reaching also quite low temperatures. Anyway, the Brayton cycle has really low efficiency and this limits its application. Water vapour is strongly limited by its minimum evaporation temperature of 0°C. This restricts considerably the range of application excluding air conditioning, refrigeration industry and freezing processes. Hydrocarbons, like also ammonia and CO_2 have a larger range of applications possible. The limit on hydrocarbons use is set by the allowed maximum refrigerant charge. For this reason, they cannot be used in large refrigeration systems, due to its safety issues. On the other hand, ammonia is quite suitable for large systems but not for domestic use or for cars' air conditioning systems, due to its safety issues. Concluding, due to its favourable characteristics, carbon dioxide, is the only "natural refrigerant" that gives some real hope regarding the future employment in all-kind-of-size applications. The peculiarities of CO_2 as a refrigerant will be discussed in the following chapters. For the sake of knowledge also a brief illustration of the CO_2 history is reported here.

2.3 History of the refrigerant carbon dioxide

Briefly, the history of CO_2 refrigeration will be here addressed using as base the studies of Pearson [9]. From the paper figure 2.4 is reported. It gives an interesting overview on the refrigerants used along the cold production history. Figure 2.5 serves for the same purpose. Effective and extremely clear (from the reader point of view) is the choice to divide the refrigerants' history into four main steps.
The discovery of CO_2 is dated back to the 18th century. The credit has to be given to the Scottish physician James Black. During his experiments on heating magnesium carbonate he came to the carbon dioxide's discovery. Anyway, Black was not interested in refrigeration applications. It seems that the first to propose the use of CO_2 as refrigerant was Alexander

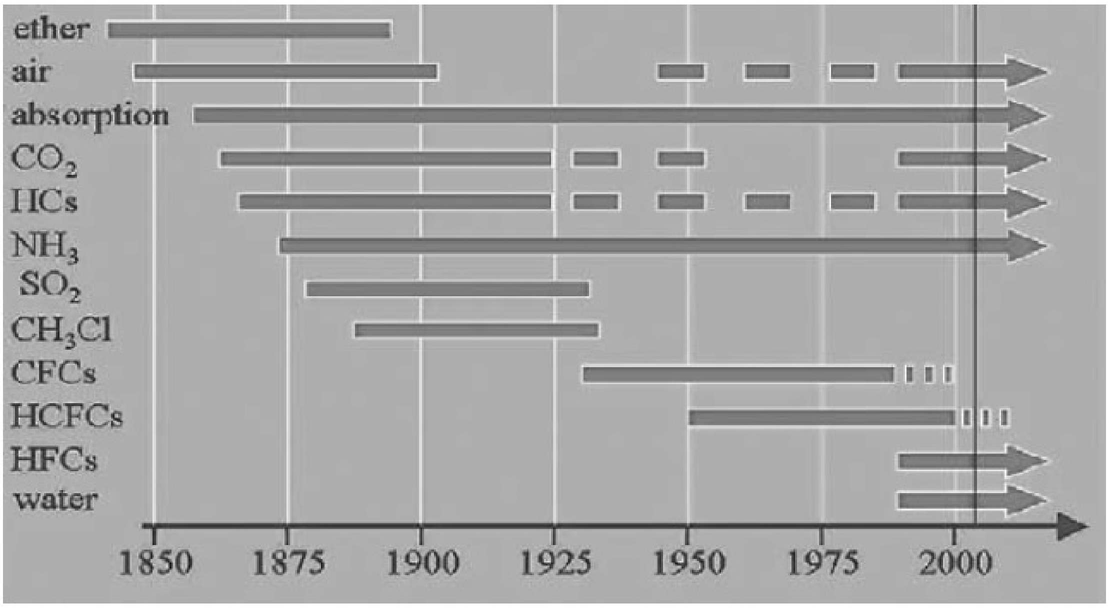

Figure 2.4 – Refrigerants' time line [9]

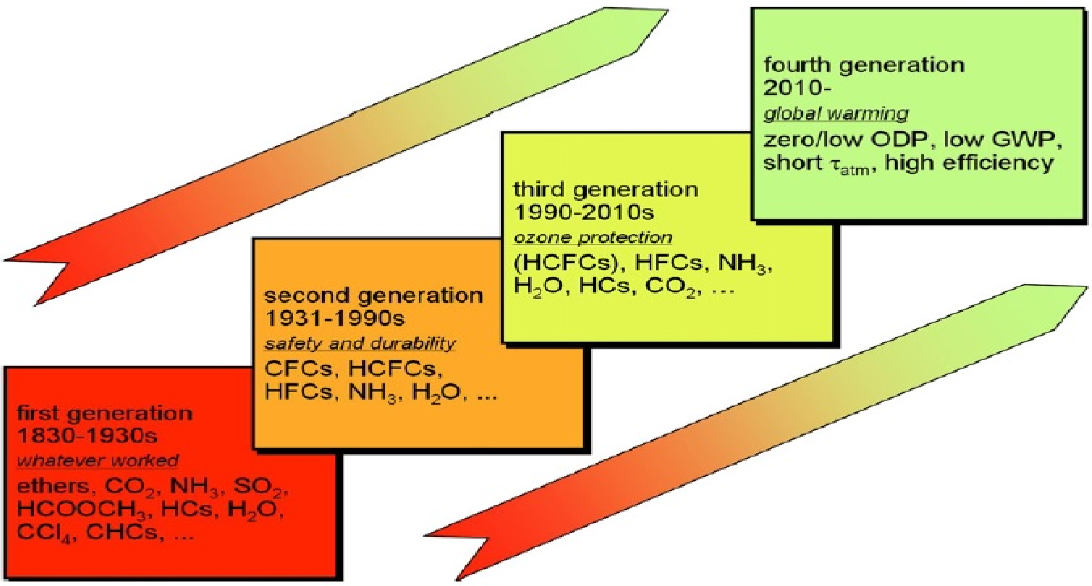

Figure 2.5 – Refrigerants' time line divided in four main steps, as reported by Calm [8]

Twining with his 1850 British patent.
The American Thaddeus Lowe was the first, in 1866, to build a CO_2 compressor. It was originally meant to fill with hydrogen military balloon but was after adapted for carbon dioxide applications. Apparently, Lowe was able to make ice using a close loop carbon dioxide cycle but he never patented his idea. The first carbon dioxide system ever built seems to be the one of Lowe. At the same time, other refrigerants were more appreciated and widely experimented, the use of CO_2 in close loop refrigeration systems were delayed respect due to the really high pressures required into the system.
Starting from 1887, CO_2 became a valuable option in the refrigeration industry thanks to the work of Raydt, Linde and Windhausen. They filed some patents stating some of the advantages of this refrigerant, like its safety, the possibility to produce cold at any temperature and the fact that was cheaper than the other used refrigerants. One of the first application of carbon dioxide was on marine refrigeration. Starting from the 1880s until the middle of 1900 CO_2 was the main refrigerant in this application due to the safety that it ensures. Ammonia was still the leading refrigerant for stationary applications.
From the 20th century, also ammonia started to generate some safety concerns and so started to lose its leadership in the refrigerant market. For instance, in 1932, Frick Company started the production of a system with double refrigerant: CO_2 for low temperature application and ammonia for higher temperatures. Other improvements were attempted in order to improve the carbon dioxide cycles, especially when operating in transcritical conditions. An example is the so called multiple-effect compression introduced by Voorhees in 1905. Anyway, carbon dioxide was not able to take an important place in the refrigeration scene of the time.
From the middle of the 20th century, CO_2 was really abandoned due to the appearance of the CFCs. As already said, the halocarbons rapidly grabbed the spotlight from all the other refrigerants for some decades.
Carbon dioxide is having only now a real chance to reach a leadership position in the refrigerant market. Starting from the late 20th century, the environmental concerns forced the cold industry to look for environmentally friendly solutions. This increasing consideration of the environmental impacts brought back to the scene some "old" refrigerants like ammonia and carbon dioxide. In particular, during the last decades, CO_2 was object of an year by year increasing number of studies. Figure 2.6 shows the increasing number of papers on the topic presented at the IIR conferences and congresses. Gustav Lorentzen was the one that started these IIR conferences about new environmentally friendly refrigerants. He is probably the main pioneer of this carbon dioxide's reborn, also for his 1990 patent about a transcritical CO_2 air conditioning system for cars. In 1992, Lorentzen himself and Pettersen used a CO_2 air conditioning systems prototype to get the first experimental results. Their paper and some other results obtained in the same years pulled the carbon dioxide reborn as refrigerant.

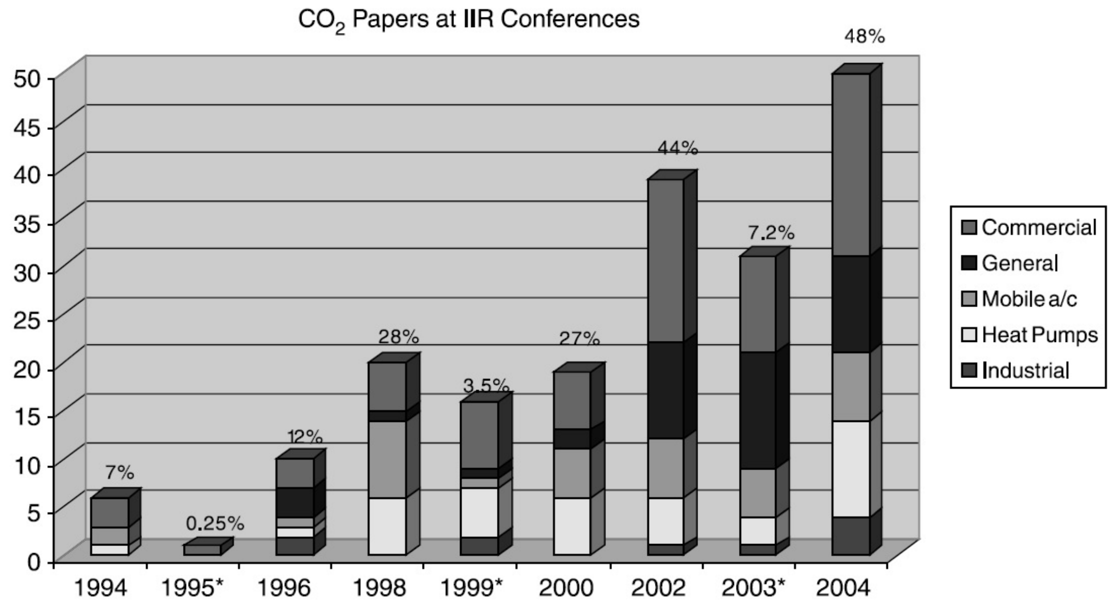

Figure 2.6 – Carbon dioxide's papers presented at the IIR conferences and congresses [9]

2.4 Carbon dioxide peculiarities

Looking at table 2.1, it is worthy to cite Lorentzen, *"the absolutely ideal refrigerant in every aspect does not exist. All the available compounds have their weak points, which must be taken into account in the design and operation of the systems"* [10]. Moreover, most of the time a proper refrigerant can be defined only relatively to the specific application considered. Anyway, CO_2 is the most promising solution among the environmentally-friendly refrigerants. As reported in figure 2.1, it is non-flammable and non-toxic. From the personal safety point of view, *"it is at least as good as the best of the halocarbons"* [6]. From the ambient safety point of view, CO_2 is probably the best refrigerant available so far, if excluding air and water. Its global warming potential (GWP) is very small and it does not have ozone depletion potential (ODP). In case of leakages at liquid state, around 50% evaporates while the rest deposits at solid state and can be easily handled (the solid state is the common "dry ice"). Of course, a big release of CO_2 can be harmful if inhaled by humans. Anyway, a good-working ventilation system is enough to ensure safety for workers, costumers or who else around the system. In its paper, Lorentzen unmasked also the legendary danger that CO_2 system would provide due to their high operation pressures [6]. Moreover, carbon dioxide presents another fundamental peculiarity of a good refrigerant, it is available everywhere and almost costless.
From the global warming point of view the reader cannot be completely convinced. CO_2 is surely another green house gas but its impact is much smaller if compared with the halocarbons. Moreover, CO_2 is produced by a great number of other industrial activities. The refrigeration industry just uses this secondary product postponing its release [10].
Let's enter now more in the details of the carbon dioxide's properties. CO_2 presents a volu-

metric refrigeration capacity much higher than the one of the "traditional" refrigerants. This is due to the high vapour density, as the volumetric refrigeration capacity is defined as the product between latent heat of evaporation and vapour density. Additionally, this refrigerant can be exploited in vapour-compression refrigeration cycles below 0°C.

Figure 2.8 presents the pressure-enthalpy and the temperature-entropy diagrams for carbon dioxide.

The carbon dioxide's critical temperature is quite low, 31.1°C. Over this temperature is not possible to release heat through a condensation. For this reason, vapour-compression cycles operating in warm climates are forced to work in transcritical conditions. "Transcritical" means that the low-pressure side of the cycle is subcritical while the high-pressure side is supercritical. The transcritical operation is characterized by high throttling and superheat losses. This reduces considerably the performances of $CO_2 - only$ systems operating in hot climates. Anyway, *"in some cases -the author is referring to small size applications- this can be compensated by much improved compressor performances as a result of very low pressure ratio and small volume requirement"* [6]. For bigger size application, solutions to deal with throttling and superheat losses are required. This is actually the main point of this report.

The phase diagram of CO_2 is presented in figure 2.7. The reduced pressure at 0°C is considerably higher than the one of the other refrigerants (figure 2.1). This peculiarity together with the low critical pressure implies that *"the low-side conditions will be much closer to the critical point than with conventional refrigerants"* [7]. This means lower p ratio in CO_2 cycles.

One of the main aspects to underline regarding the CO_2 behaviour concerns the abrupt property change close to the critical point. In the supercritical region, the temperature does not go together with the pressure any more. This independence of the two variables allow for settings that bring to a maximum of the system performances (this aspect will be developed later in this report). Figure 2.9 reports enthalpy and entropy decreasing with temperature in a constant pressure gas cooling process. It is visible the great change in the properties passing through the critical point.

Figure 2.10 shows that the vapour pressure of carbon dioxide is higher than the one of other refrigerants and the curve is more steep. Consequently, smaller temperature change are faced when the pressure changes. For this reason, *"the temperature change associated with a pressure drop in the evaporator will become smaller"* [7]. Figure 2.11 shows the already cited volumetric refrigeration capacity for carbon dioxide and other refrigerants.

The rapid property change during isobaric supercritical processes close to critical point is surely one of the main aspects when dealing with CO_2 systems. Figure 2.12 presents the isobaric specific heat for carbon dioxide. The behaviour of this parameter close to the critical point has a great impact on heat exchange processes and must be kept in mind when analysing them.

All the results provided in this report are based on EES (engineering equation solver) calculations. This software bases its CO_2 calculations on the fundamental equation of state developed by Span and Wagner [7].

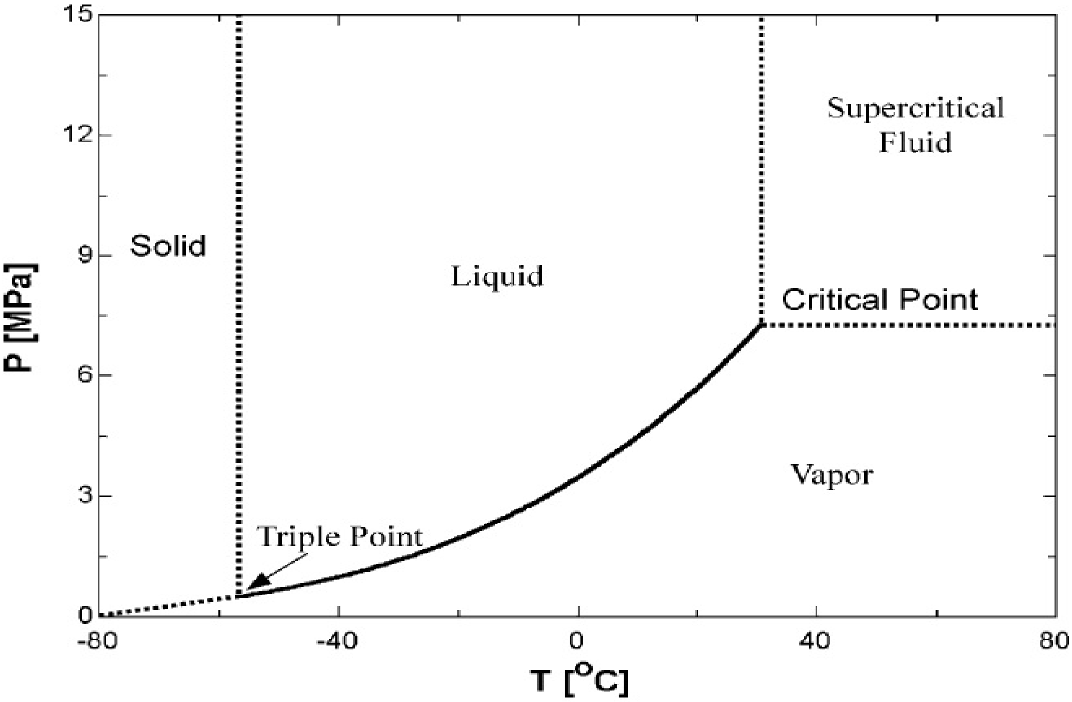

Figure 2.7 – Phase diagram of CO_2 [7]

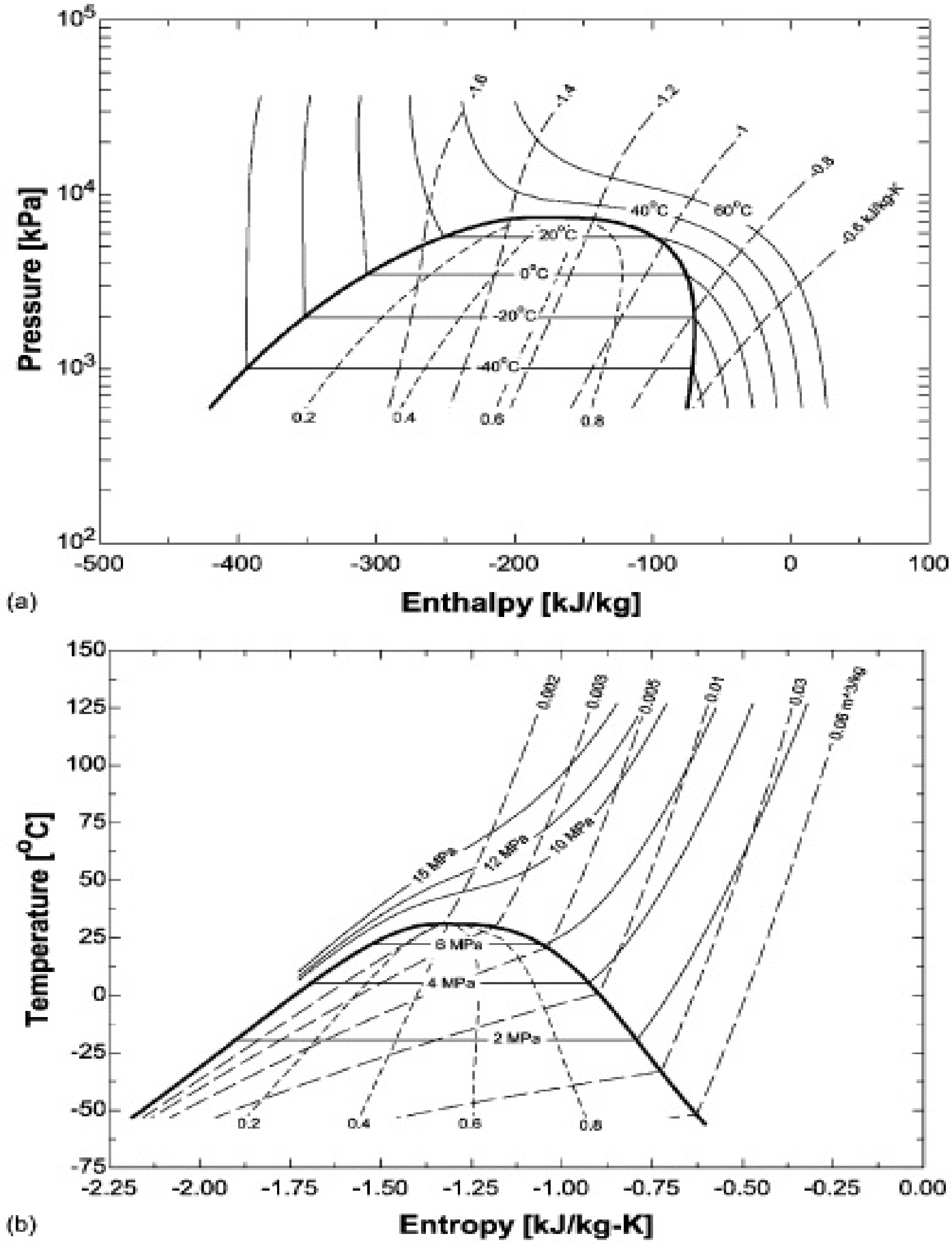

Figure 2.8 – (log)p-h diagram and T-s diagram for CO_2 [7]

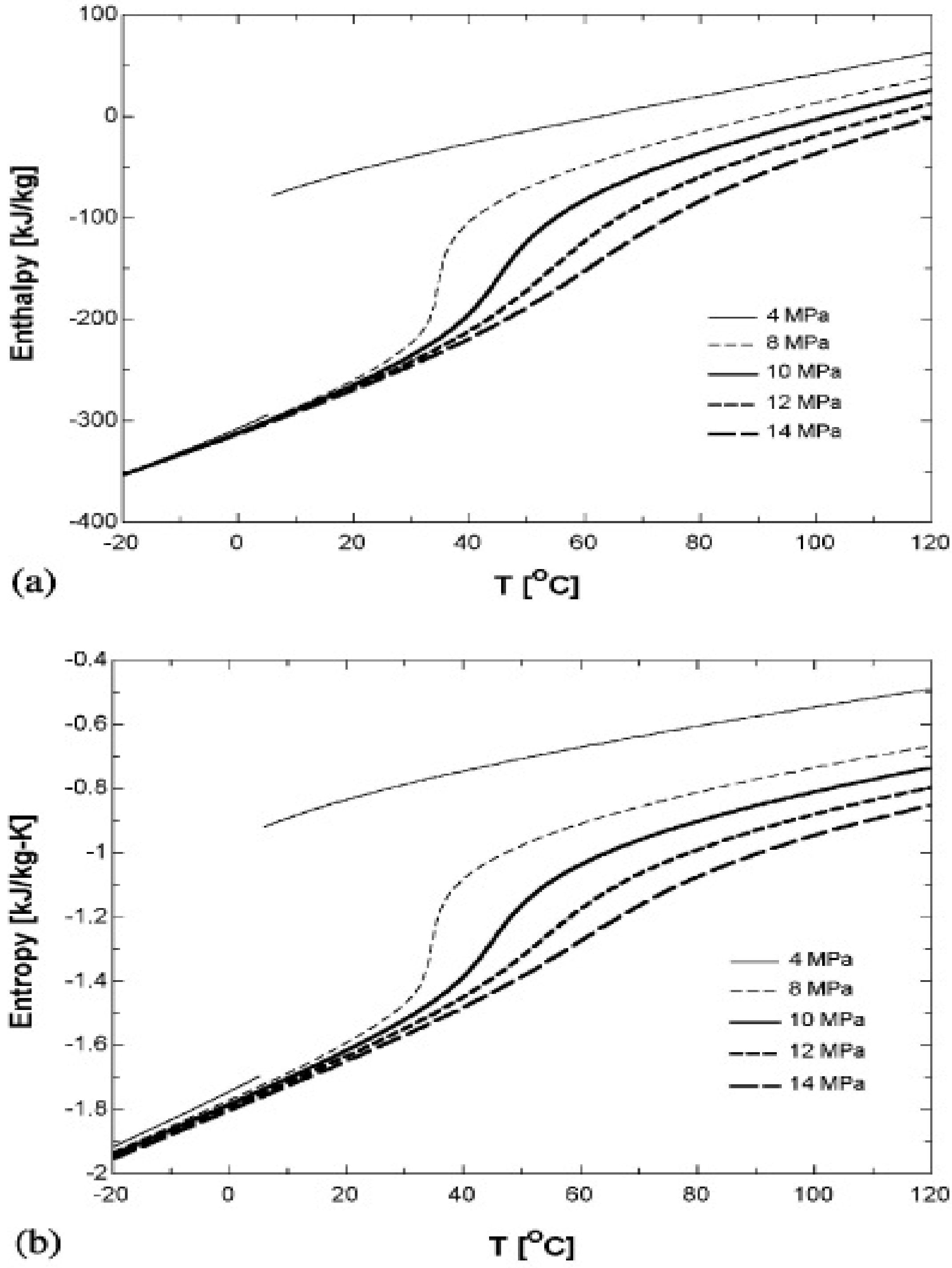

Figure 2.9 – Enthalpy and entropy change across the critical point going through a constant pressure gas cooling process [7]

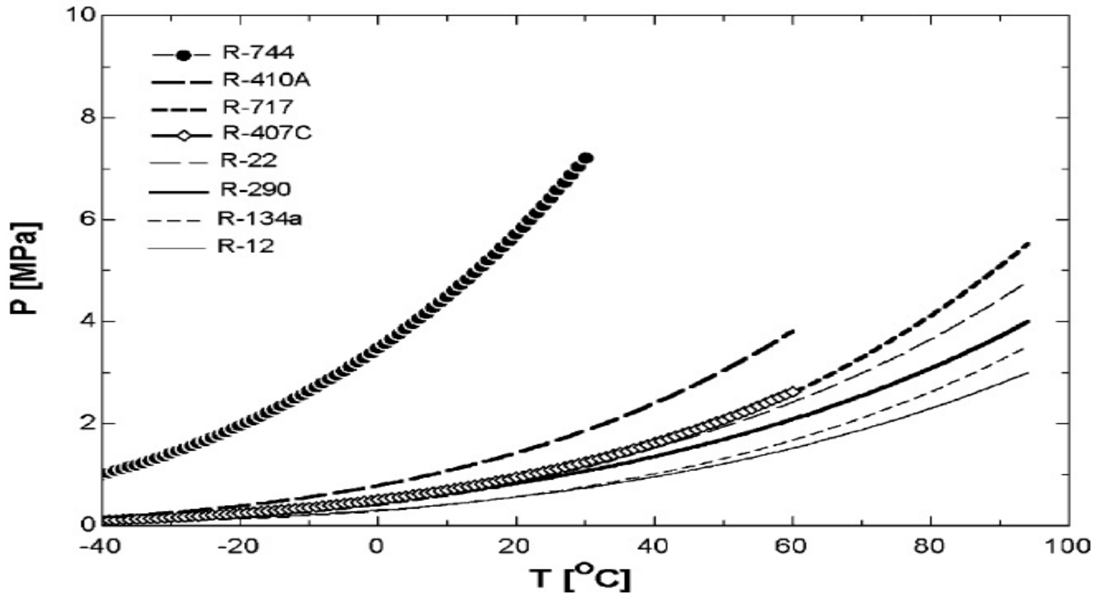

Figure 2.10 – Vapour pressure for some refrigerants [7]

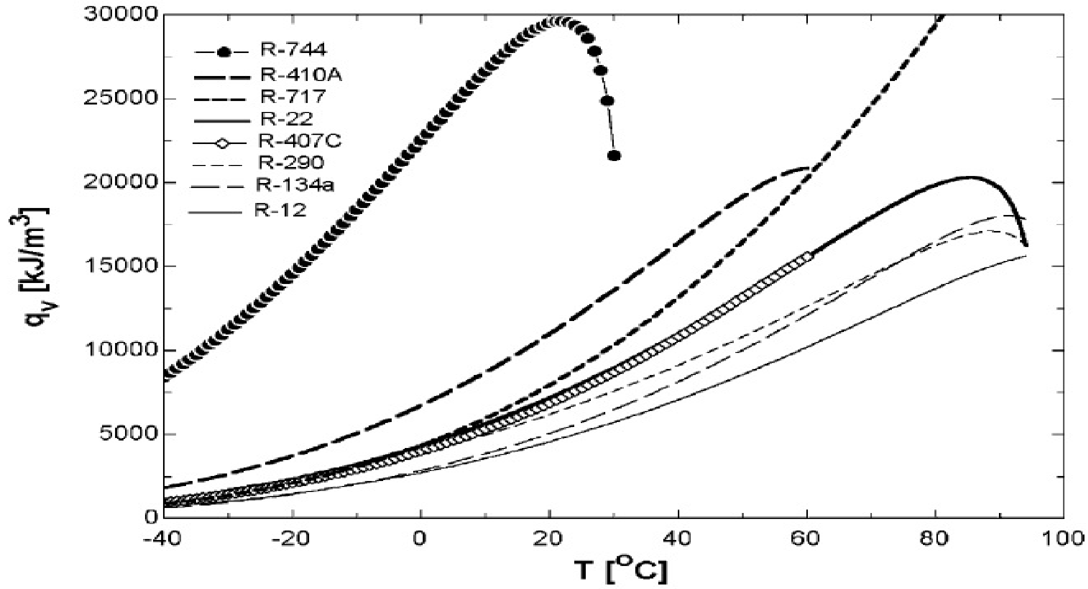

Figure 2.11 – Volumetric refrigeration capacity for some refrigerants [7]

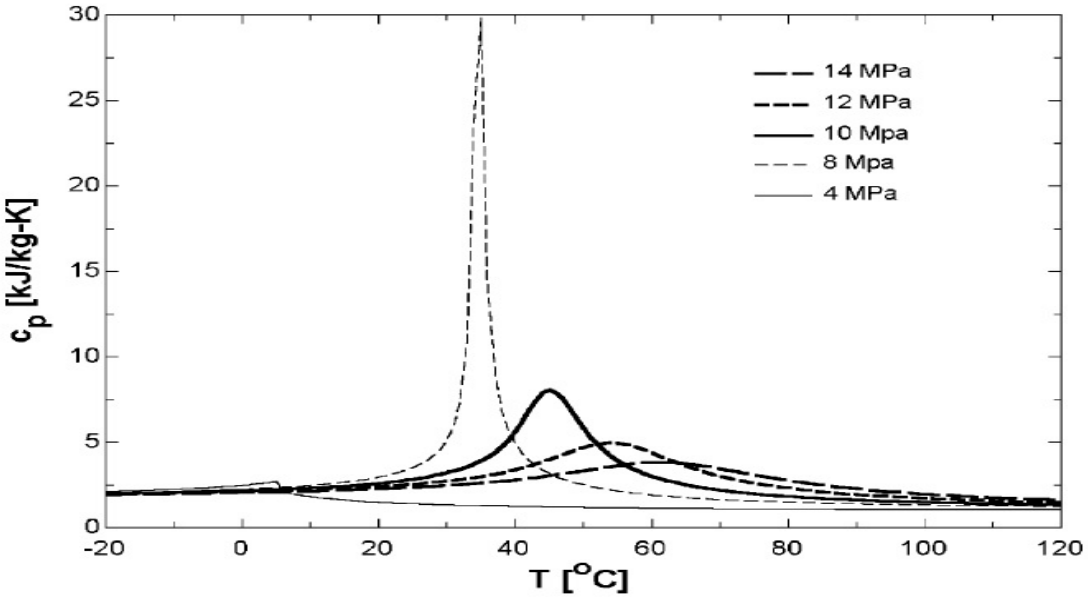

Figure 2.12 – Isobaric specific heat for CO_2 [7]

3 State-of-art of the CO_2 vapour-compression technology

Starting with the early 1990s, carbon dioxide came back on the refrigeration scene. Due to the increasing concern regarding the environmental impact of traditional freons, the CFCs' applications started to decrease until they were completely abandoned. Lorentezen and Pettersen in 1992 published what seems to be the first paper about carbon dioxide refrigeration in this new era of natural refrigerants rebirth [11]. From that moment on, an exponential increasing interest has been shown for this environmentally friendly refrigerants.
Lorentzen and Pettersen pointed their attention on the cars' air-conditioning systems. This sector still offers considerable space for CO_2 applications due to the banned use of HFC-134A in 2006 (European parliament's vote). This political act not only stimulate the air-conditioning industry to work on new refrigerants, it represent the first international action against the use of HFC. The HFCs were discovered to be one of the major responsible of the ongoing hearth's climate change. For this reason, they will be dismissed completely in some years and the refrigeration industry needs to find a valuable alternative. For all its suitable properties and also for the lack of valid alternatives, CO_2 seems to be the most promising replacement.
In this chapter a brief presentation about the CO_2 vapour-compression technology is addressed. Particular attention will be focused on the importance of the choice of the gas cooler pressure in transcritical systems. Finally, some issue concerning carbon dioxide system will be introduced. The solution or the attenuation of these challenges is left for the next chapters (being this the main topic of the project).

3.1 Vapour-compression refrigeration technology

The PI diagram of a single-stage vapour-compression refrigeration cycle is proposed in figure 3.1. The same figure includes also the (log)p-h diagram for the same cycle. The refrigerant evaporates going through the evaporator (from state 4 to 1). During this isobaric process, the heat $\dot{Q}_e$ is extracted from the cooled space and in this way the refrigeration effect is performed. After that, the refrigerant is compressed until state 2, here its conditions allow for an heat exchange with the ambient surrounding the cooled space. The heat exchange is performed in a

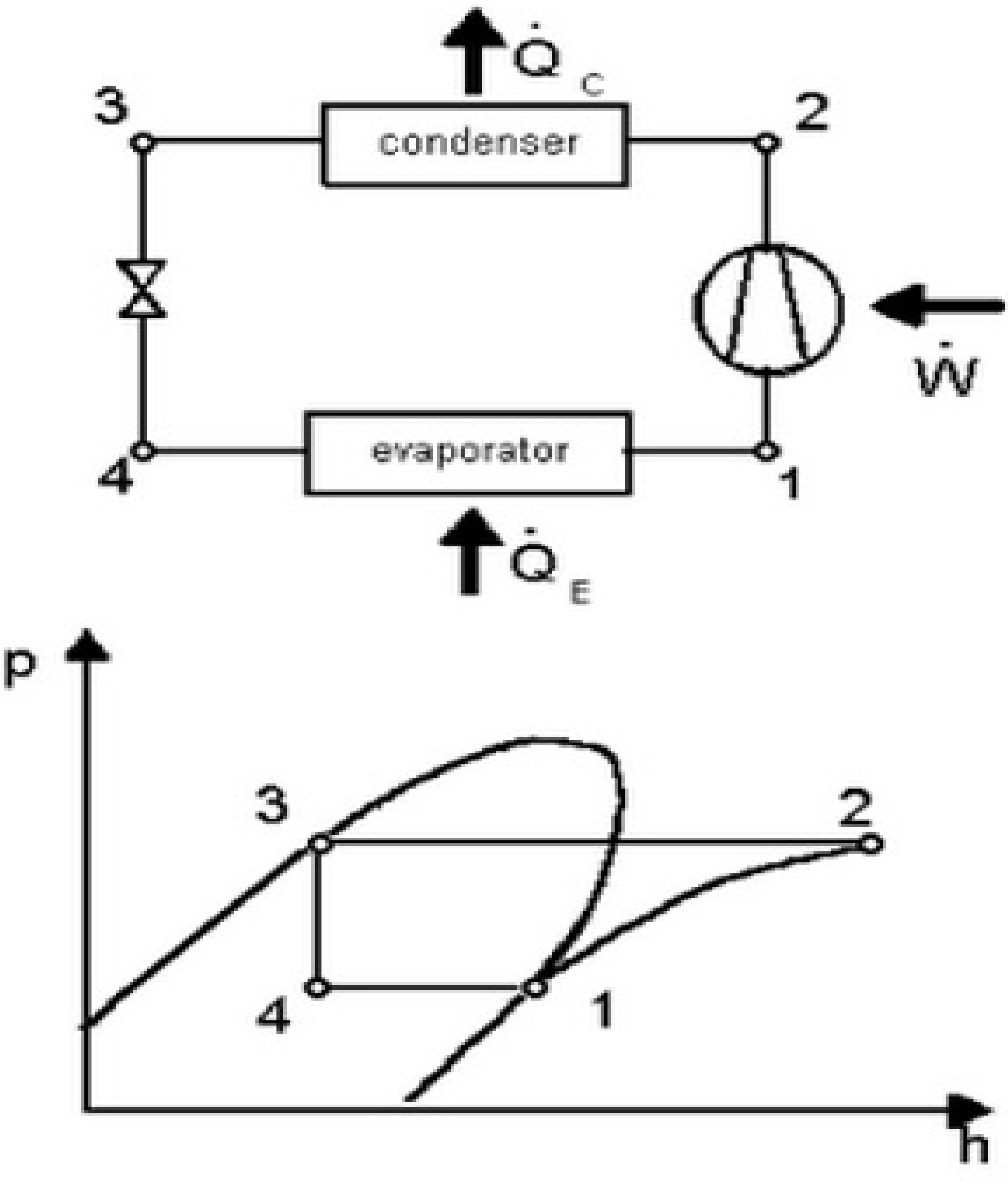

Figure 3.1 – PI diagram and (log)p-h diagram, 1 stage vapour-compression refrigeration cycle

condenser (subcritical operation) or gas cooler (transcritical/supercritical operation). Finally, the refrigerant is throttled in an expansion valve (state from 3 to 4) until the evaporation pressure is reached.
Before going in deep with the specific refrigerant here considered, it is worthy to remember some basic concepts. Citing Lorentzen, *"the efficiency, or COP, of any reversible (loss-free) process, working between given temperature limits, is exactly the same and completely independent of the properties of the working medium used."* [10]. When considering real cycles, even the refrigerant used has its impact and the main losses faced can be summarized as follow:

- compressor loss. It depends mainly on the pressure level and pressure ratio, together with some fluid thermal properties.
- condenser/gas cooler losses. It is related to the temperatures involved in the heat exchange and heat transfer properties of the fluid.
- throttling loss. It depends on the ratio $\frac{c_{p,liq}}{e}$
- evaporator loss. Like for the condenser/gas cooler, It is related to the temperatures involved in the heat exchange and heat transfer properties of the fluid.
- flow resistance loss. This is specifically connected to the refrigerant's properties.

What is important to remember is that the impact of all these losses is all connected to the system design. Theoretically, the system's losses can be reduced to whatever level, if the budget for the technological expenditures is infinite. Thus, it is clear that the optimization of the system is an economical matter. When comparing the performances of different refrigerants, must be kept in mind that the comparison is relative to the specific system used. One refrigerant can give higher system's performances than another because the cycle, as it is designed, is more suitable for that specific refrigerant. Maybe changing the application or the the cycle's design, other refrigerants result in better performances.

3.2 CO_2 transcritical cycles technology

Due to the very low critical point of CO_2, vapour-compression refrigeration cycles working with this fluid are forced to work for many hours along the year in transcritical conditions. Especially when considering warm climates, the high ambient temperatures force the system to a supercritical heat rejection. The dominant transcritical operation is what mainly distinguish carbon dioxide from the other refrigerants. For this reason some words are used here to describe it.
Transcritical operation means that the evaporation temperature is below the critical point, while the heat rejection temperatures are over it. The evaporation is still performed in a two-phase device, like in the common subcritical cycles. Instead, the condenser is replaced

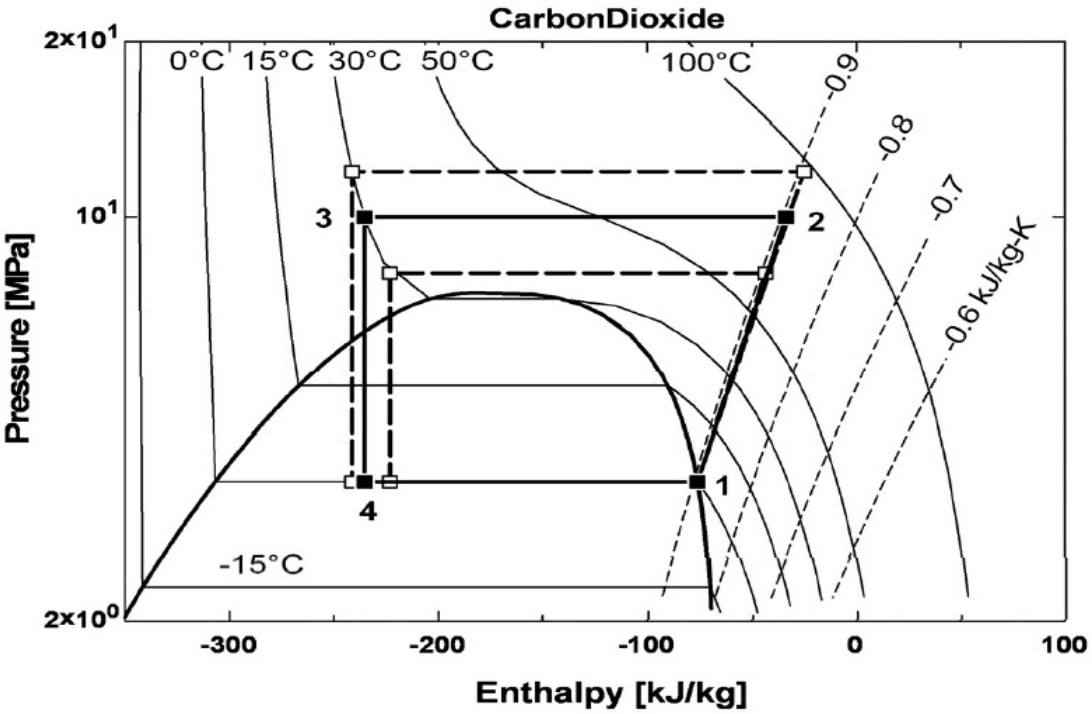

Figure 3.2 – 1-stage transcritical carbon dioxide vapour-compression basic cycle [12]

with a gas cooler: the fluid does not change phase during the heat rejection (differently from what happens in subcritical condensers).
Figure 3.2 presents the (log)p-h diagram for a basic 1-stage vapour-compression transcritical CO_2 cycle. The evaporation (state from 4 to 1) is assumed to happen at 0°C, after that the refrigerant is compressed from state 1 to 2. The gas cooling process (state from 2 to 3) is isobaric but not iso-temperature. The gas cooler outlet temperature is here assumed to be 30°C. The cycle is closed by an isenthalpic expansion from state 3 to 4. In the evaporator the refrigeration effect is performed by the carbon dioxide: an heat flux is removed from the cooled space's air.

3.3 Optimal high pressure for transcritical cycles

In case of subcritical operation, the condensing pressure is determined by the condensing temperature (in turn determined by the ambient temperature). In fact, during the two-phase operation, pressure and temperature are coupled. This is not true any more when considering supercritical operation: the pressure is independent from the temperature.
Coming back to figure 3.2, keeping constant the gas cooler outlet temperature, the system can operate at different gas cooler pressures (by mean of the expansion valve that controls the back pressure). Increasing p_{gc}, two opposite effects are visible on the system. The enthalpy

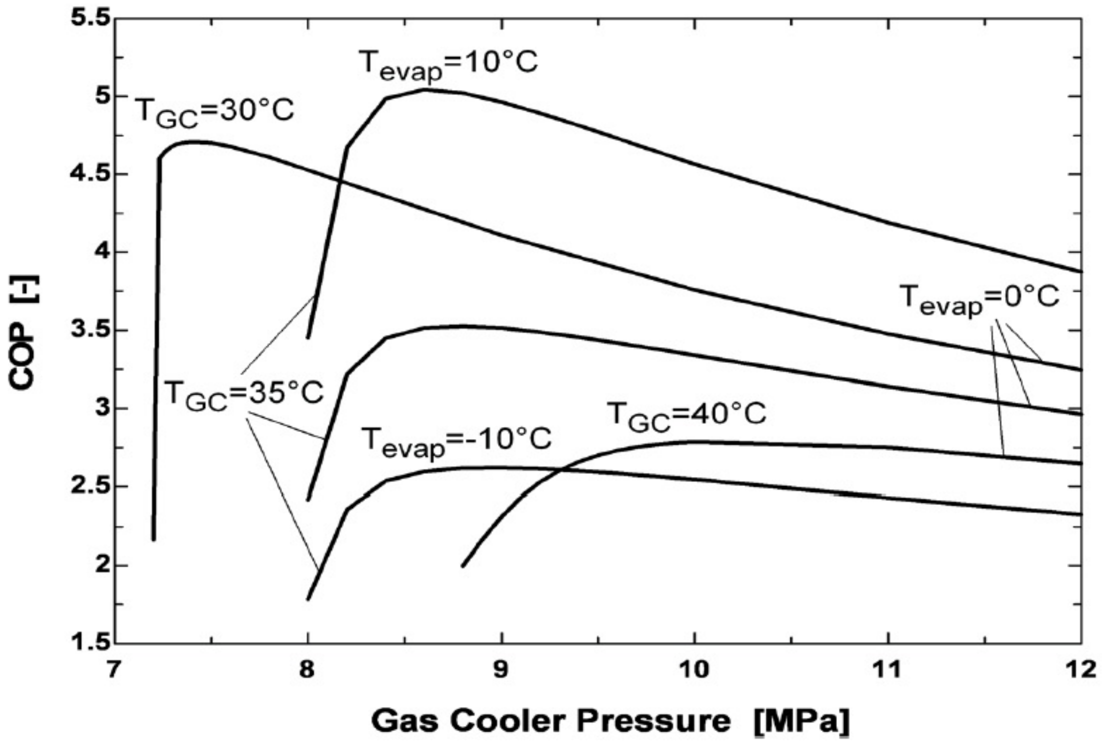

Figure 3.3 – Carbon dioxide transcritical cycle's COP vs. gas cooler pressure, different gas cooler outlet temperatures and evaporation temperatures are considered [12]

difference across the evaporator increases, producing a positive effect on the COP. Also the enthalpy difference across the compressor increases but this produces a negative effect on the system's COP. As a result of a trade-off between this two contributions, an optimal gas cooler pressure can be found for each application. Figure 3.3 reports the results of an investigation conducted by Groll and Kim [12] about the optimal gas cooler pressure. The cycle they used is the one already introduced with figure 3.2. As visible in figure 3.3, for each transcritical cycle's configuration an optimum COP can be found. Evaporation temperature and gas cooler outlet temperature strongly affect the position of this maximum.
This behaviour of the system can be quite surprising if compared to conventional subcritical systems. In fact, for subcritical operation the COP drops with the increasing condensation temperature, due to the increasing compressor work and decreasing enthalpy difference through the evaporator.
The importance to maximize the system's COP is well underlined by Kauf [13]. In a society facing increasing energy consumptions, the energy security is becoming a very current topic. *"To keep the energy necessary for propulsion of the refrigeration system small, the COP of the system has to be high"* [13]. With these words, the author of the paper stresses the importance of the choice of the cycle's high pressure to obtain high system's COP. Kauf made a more accurate investigation on this optimum high pressure than Groll and Kim. In his paper, he demonstrated that the evaporation pressure has a good influence on the COP but the optimal gas cooler pressure does not change significantly. He concluded saying that mainly ambient temperature and gas cooler outlet temperature influence the position of the optimum high pressure.
In all the CO_2 vapour-compression refrigeration cycles, the high side pressure is subject of regulation in order to reach the optimum COP for each ambient temperature and/or to control the cooling capacity [7].

3.4 Components design for CO_2 systems

The peculiar behaviour of CO_2 and relative refrigeration systems is reflected in the expedients that must be adopted in the cycle's components design. CO_2 systems are characterized by much lower compression ratio respect what happens when using other "traditional" refrigerants. This potentially results in better compressor efficiencies. On the other hand, CO_2 systems can reach pressure up to 140 bar, a level much higher than the one of traditional vapour-compression systems. For this reason, the compressors used for CO_2 applications require thicker shells.
Due to the fact that CO_2 has much higher volumetric refrigeration capacity than "traditional" refrigerants, *"the volumetric flow rate is much smaller for the same cooling capacity"* [12]. Consequently, *"the compressor chamber size for CO_2 applications is smaller than a current refrigeration compressor for the same cooling capacity"* [12].
Another issue regarding the compressor concerns the leakages. Due to the high pressures

Ambient Air Temp.		28 °C (82 °F)	35 °C (95 °F)	46 °C (115 °F)	52 °C (125 °F)	57 °C (135 °F)
Capacity (kW)	R-22	12.84	11.98	10.63	9.95	9.32
	R-410A	13.01	11.92	10.20	9.32	8.50
	Rel* (%)	1.3%	-0.5%	-4.0%	-6.3%	-8.8%
COP	R-22	3.79	3.11	2.26	1.92	1.64
	R-410A	3.99	3.19	2.19	1.79	1.47
	Rel* (%)	5.3%	2.4%	-3.4%	-6.9%	-10.7%

*Rel = 100% ([R-410A value] – [R-22 value])/[R-22 value]

Figure 3.4 – R-22 and R410A capacity and COP for different ambient temperatures [14]

reached in this component, the design must be carefully determined in order to minimize the fluid losses.

The high volumetric refrigeration capacity allows also for a reduction of the piping size. Consequently, even the space occupied by the heat exchangers can be reduced up to 30% [12] (comparison made with components from halocarbons cycles). *"Because of the small area over which the pressure acts in a small-diameter tube, this geometry is ideal for high pressure systems"* [12]. As a result, high pressures are not problematic for the heat exchangers.

Investigations between the possible compressor's oil resulted in two major candidates for CO_2 systems: polyolester (POE) and polyalkylene (PAG) [12].

3.5 Impact of elevated ambient temperatures on vapour-compression refrigeration cycles

The aim of this project is to study the performances of CO_2 refrigeration systems exploited in hot/warm climates. High ambient temperatures entail a reduction of the system's performances for all the vapour-compression refrigeration cycle. In particular, carbon dioxide systems are penalized by high ambient, because of the CO_2 low critical point.

The impact of elevated ambient temperature on vapour-compression refrigeration cycles can be understood starting from the ideal Carnot inverse cycle. From its COP's formula (3.1) it is visible that the COP decreases along with the increase of T_{cond}, assuming a constant T_{evap}.

$$COP_{Carnot} = \frac{T_{evap}}{T_{cond} - T_{evap}} \tag{3.1}$$

In other words, high ambient temperatures reduce the COP of the Carnot inverse cycle.
This conclusion, derived from the ideal cycle, can be extend to the irreversible vapour-compression refrigeration cycle, no matter which refrigerant it is used in the cycle. The COP reduction will be even higher for not-ideal cycles than for the Carnot cycles. The magnitude of this reduction depends on the refrigerant used [15].
Going through some literature, Chin and Spatz [14] studied the different performances of R-22 and R-410A at variable ambient temperatures. Their results are quite interesting because they were able to show how the performance of R-410A systems are higher than the one of R-22 cycles for low ambient temperature, while they are overtaken from the second when the ambient temperatures are higher. Anyway, both the two systems face a decrease of the COP with the increasing ambient T, see figure 3.4. More studies can be found in the literature, like the one of Yana Motta and Domanski [16]. They compared much more refrigerants than the previous studies founding that R-410A seems to be the refrigerant with higher performance degradation with the increasing ambient temperature, while R-134A is the one with the lowest.
As already discussed, considering subcritical operation, the increasing ambient temperature affects the condensation temperature and consequently the condensing pressure (in two-phase conditions pressure and temperature are coupled): the warmer is the ambient, the higher is the condensing pressure. Consequently, the enthalpy difference across the evaporator is decreasing and the enthalpy difference across the compressor increases. As a result, the COP of the system decreases with increasing ambient temperature.

3.6 Lower performances of transcritical vapour-compression refrigeration cycles

Carbon dioxide is characterized by a very low critical point. This implies that CO_2 vapour-compression systems work for many hours along the year in transcritical conditions, especially when running in hot climates.
Figure 3.5 from Kim et al. [7] presents a comparison between the same vapour-compression cycle using R-134A (subcritical functioning) and CO_2. The temperature-entropy diagram is chosen to underline the increased thermodynamic losses faced with transcritical operation. Equal T_{evap} and minimum heat rejection temperature are assumed. As depicted by the figure, transcritical operation brings higher average heat rejection temperature and higher throttling losses. Consequently the work required in the CO_2 transcritical cycle is higher.
The gliding temperature during the heat rejection can be a great advantage when considering heat pumps. The profile of the heated water or air can be perfectly matched by the cooled refrigerant profile decreasing considerably the thermodynamic losses. On the other hand, when the heat rejection is not a topic of interest the gliding temperature just increases (more than necessary) the average heat rejection temperature. In this way, the gliding temperature is not an advantage when considering refrigeration systems. Anyway, in this situations, the thermodynamic losses can be reduced with a well-performed design of the heat exchanger.

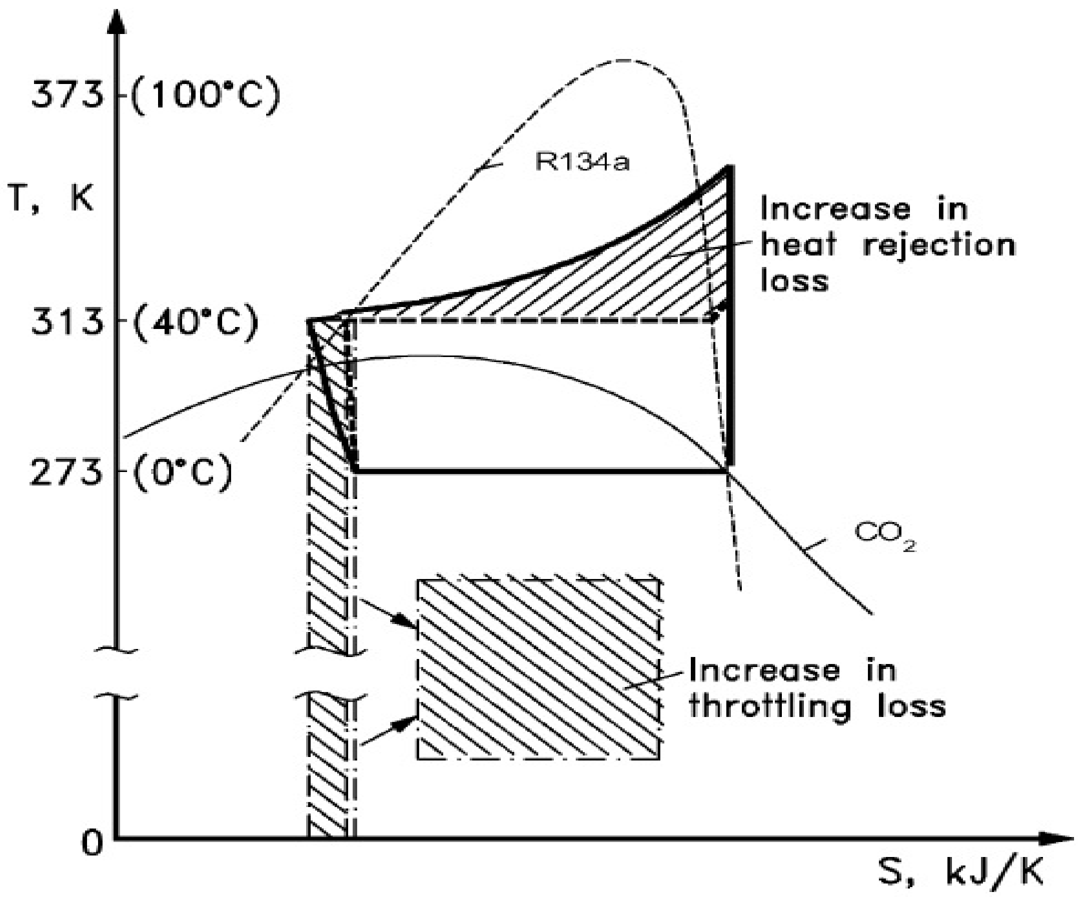

Figure 3.5 – Thermodynamic cycle comparison between R-134A and CO_2 [7]

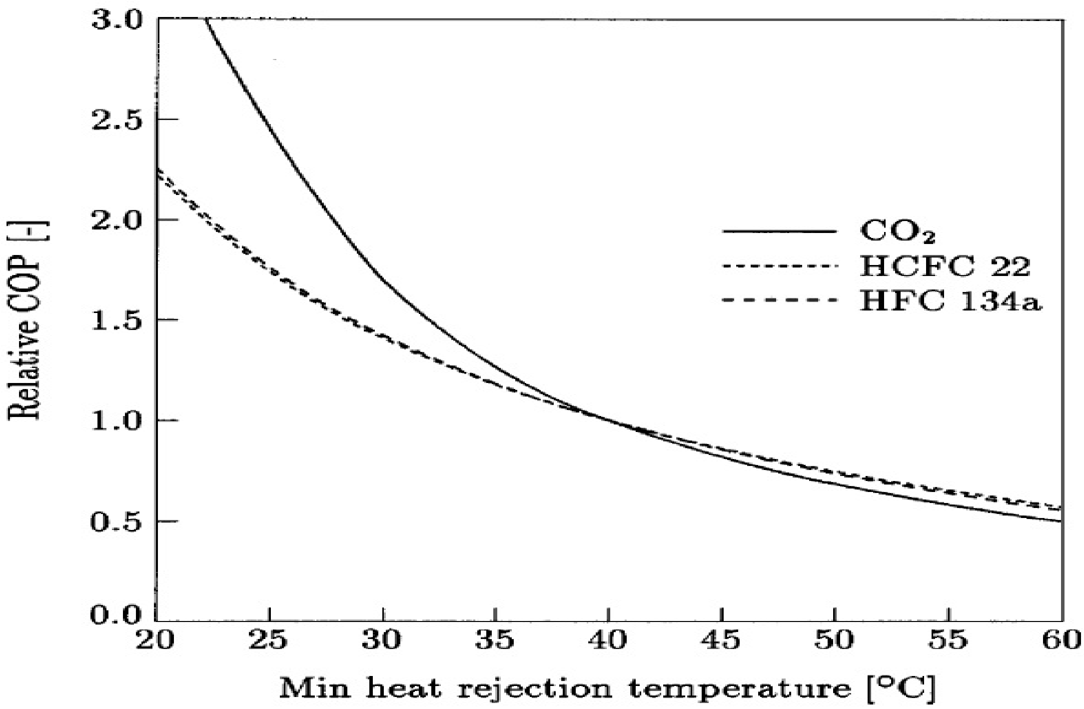

Figure 3.6 – Relative COP vs. minimum heat rejection temperature CO_2 [7]

The refrigerant temperature can be allowed to approach really closely the heat sink temperature (usually air or water) at the outlet of the gas-cooler. In this way the average heat rejection temperature is reduced.
Summing up, transcritical CO_2 systems are strongly affected by the glide heat rejection temperature and by the throttling losses. For this reason the outlet temperature from the gas cooler is one of the most important parameters in this kind of systems. Again from the same paper [7], figure 3.6 enlightens the impact of the gas cooler outlet temperature on the system's COP. The COP is here normalized with its value at 40°C. The figure offers also a comparison with other "traditional" refrigerants. As visible, the higher is the minimum heat rejection temperature, the lower is the system's COP. Moreover, the impact of this minimum temperature on the COP is much higher for CO_2 systems than for the others presented in the figure. Concluding, it is clear how important is that the outlet gas cooler temperature is as close as possible to the inlet heat sink temperature to achieve good performances in CO_2 systems.

Independently from the operation of the CO_2 systems, transcritical or subcritical, the COP decreases with increasing ambient temperature. Moreover, carbon dioxide systems are further penalized from the COP point of view when operating in transcritical conditions (for the reasons just explained). For these reasons, it becomes even more important the choice of the correct gas cooler pressure. A wise setting of the p_{gc} allows the system to reach higher values of COP when the outdoor temperature is high and the system is running transcritical.

Of the same importance is a proper gas cooler design that determines the gas cooler outlet temperature.

3.7 Thermodynamic cycle's improvements

As already expresses, the aim of this work is the study of the thermal performances of CO_2 refrigeration cycles, especially when operated in hot climates. The purpose of the project is to overcame the already cited limitations of CO_2 systems when dealing with high ambient temperatures and, consequently, a high number of transcritical operation hours.
The possible modifications to the basic cycle are many and different. Three main expedients are here discussed: mechanical subcooling, parallel compression and the use of an ejector. All these improvements are aimed to overcome the discussed issues faced by CO_2 systems, improving the system's performances in hot climates operation.
The improved CO_2 cycles are firstly studied for the same design conditions. Using equal boundaries conditions, it allows for a comparison of their performances. Anyway, real systems face every hour and every day along the year different ambient conditions. For this reason, the studied cycles are analysed even in part-load operation, with particular attention for the different (hour by hour) compressor work profiles.

Part I

Design operation of the studied cycles

4 Design conditions

The first part of this project concerns the comparison of different improved CO_2 vapour-compression refrigeration systems, working at the same design conditions. As a reference for this comparison, the basic one-stage vapour-compression CO_2 refrigeration cycle is used. As an attempt to improve the performances of this system, especially when dealing with high ambient temperatures, three main modification of the basic cycle are considered: mechanical subcooling, parallel compression and use of an ejector are the cycle's improvements here considered. The three devices just cited are widely described in the following chapters, so no words are used here to depicts their operation. The reader should continue his/her trip to understand how they work.
The investigations here performed continue with more design operation analyses in the second part of this report. Further improvements of the three solutions taken into account in the first part are proposed. Additionally, even new ideas for further improvements of CO_2 vapour-compression cycles are suggested. Interesting comparison of a wide range of CO_2 vapour-compression cycles, under design conditions, are offered in these two parts of the report. The two parts are here introduced together because they complete each other and give a comprehensive overview of possible boosts for CO_2 cycles.
All the results obtained and here presented come from simulations made with the support of the software EES (Engineering Equation Solver). Once that the thermodynamic calculations were made, spread sheets were used to collect different data and work on them.

4.1 Selected boundaries conditions

Same boundaries conditions and assumptions are required when investigating the different CO_2 cycles. Only in this way a fair comparison of them is performed. The modelling approach adopted in this project is well summarized by Sarkar, *"the entire system has been modelled based on the energy balance of individual components of the system. Steady flow energy equations based on first law of thermodynamics have been employed in each case and specify energy quantities are used"* [17]. Here the main assumptions that associate all the studied cycle are presented. More detailed assumptions regarding only specific cycles will be introduced when required.

- Steady-state and steady-flow processes are assumed for all the components.
- The compression process is adiabatic but not isentropic. The compressor is characterized by a given isentropic efficiency η_{is}. This efficiency is assumed to be 0.9 and it is constant. No particular assumptions are made about displacement or other geometrical characteristics of the compressor. It is assumed as always able to compress the required amount of refrigerant.
- The process in the throttling valve is isenthalpic.
- When a separator is used, two flows come out from it: one is saturated vapour and the other is saturated liquid.
- The processes of mixing and separation are isobaric.
- Evaporation and gas cooling/condensing processes are isobaric.
- Pressure drop and heat losses are neglected. The only heat rejection happens at the gas cooler.
- The required cooling capacity is fixed to 60kW for all the cycles. The choice of this value is quite random. It came to the author mind after a trip in Trento during January 2015. At that time DANFOSS was starting a new ejector system in a supermarket characterized by a required cooling capacity of 60kW. Anyway, this is a quite random value and it is as good as another. For instance, Minetto et al. performed a similar analysis assuming a cooling load of 100kW [3].
- The evaporation pressure is set to 30bar. The fluid coming out of the evaporator is considered saturated vapour. It is worthy to specify that this is not a realistic assumption: *"evaporator superheat is always required to ensure that no liquid enters the compressor"* [1]. Anyway, for sake of simplicity, the superheat is here considered nil.
- The outlet temperature of the refrigerant from the gas cooler is given and equal to 40°C. This temperature depends on the ambient conditions and the approach temperature achieved in the gas cooler. The chosen value has the purpose to simulate very hot ambient conditions (the gas cooler is, in fact, assumed as air-cooled). During the design investigation of the cycles this parameter was not changed. It is obvious that a reduction of it brings COP improvements, but the purpose of this first part of the project is to analyse the systems behaviour under extremely warm conditions.

No requirements are presented for the cycle's high pressure. The gas cooler pressure is optimized, for the given operating conditions, in each one of the considered cycle. Consequently, each cycle is characterized by its own value of p_{gc} (see chapter 3 for more informations about

the existence of an optimal gas cooler pressure).
The parameter primarily used to evaluate the performances of the considered cycles is the coefficient of performances (COP). It is defined as the ratio between the output of the system and its input:

$$COP = \frac{output}{input} = \frac{cooling\ capacity}{compressor(s)\ work} \tag{4.1}$$

The higher is this parameter, the lower are the consumptions to produce the same amount of output. The other way around, the higher is this parameter, the highest is the system's output using the same amount of energy input. What the COP represents is very important in a society like ours. Day by day, the population increases, along with the share of it that seeks better life conditions. The energy consumptions of our planet are barely sustainable in the long run. For this reason, energy savings are a very current topic nowadays. A high value of the system's COP means energy savings with respect to another system that produce the same amount of output but with lower COP.
To be more precise during the COP calculation, the input required by the system should include, additionally to the compressors work, the pumps and fans work. Anyway, the choice here made was to simplify the system following what suggested by Sharma et al. *"Condenser/gas cooler and evaporator fan power consumption was assumed to be roughly equal for all the systems, so fan power was not included in the COP calculations"* [1]. Due to the fact that the analysis here performed is structurally similar to the one of the cited authors, the same assumption is here made.

5 One-stage vapour-compression CO_2 transcritical cycle, design conditions

The first part of this project concerns the comparison of different CO_2 cycles operating at the same design conditions. As a reference for this comparison, the one-stage CO_2 vapour-compression refrigeration cycle was chosen. Due to the high gas cooler outlet temperature assumed for the design operation (40°C), all the cycles are operating in transcritical conditions. Such a high temperature was chosen with the purpose to compare the different cycles for a hot climate situation, during transcritical operation.

5.1 System description

The PI diagram of the 1-stage vapour-compression refrigeration cycle was already presented in chapter 3. The refrigerant evaporates in the evaporator until saturated conditions, here the refrigeration effect is performed. The air of the cooled space is cooled down due to the heat absorbed by the evaporating CO_2. Assuming that the superheating after the evaporator is nil, the saturated vapour enters in the compressor. At the outlet of the compressor the refrigerant reaches the gas cooler pressure, the process inside the compressor is determined by its isentropic efficiency. In the gas cooler the fluid is cooled down in a isobaric process. The temperature along the heat rejection is not constant and reaches a value of 40°C at the outlet of the gas cooler. The cycle is closed by the throttling valve that reduces the refrigerant's pressure to the evaporation one.

5.2 System analysis

As initially explained all the cycles were compared using the same boundaries conditions defined in chapter 4. The gas cooler outlet temperature was fixed to 40°C for all the cycles, while, nothing was said about the gas cooler pressure. The reason for this is that each one of the cycle have its own optimal gas cooler pressure, for the given working conditions. In

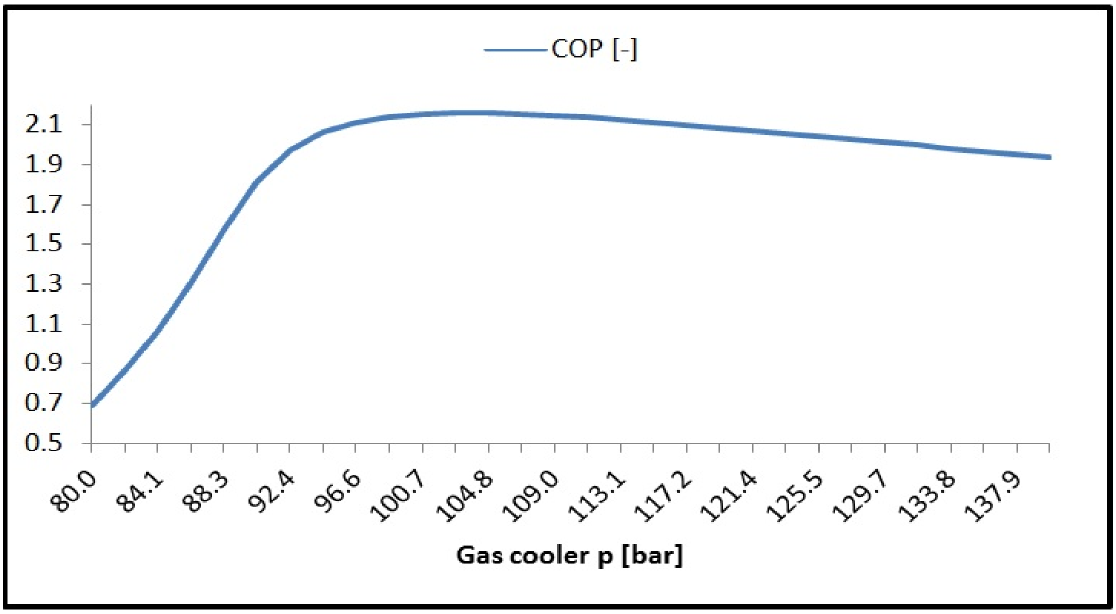

Figure 5.1 – COP vs. gas cooler pressure, 1-stage reference cycle

all the cycles considered the gas cooler pressure was optimized using the min/max function provided by the software EES. This function was able to automatically determine which p_{gc} gives the maximum COP, keeping constant all the remaining system's parameters.
Figure 5.1 represents the COP of the reference 1-stage cycle for different gas cooler pressures. As visible, an inverted U-shape is depicted, demonstrating that an optimum can be found for the gas cooler pressure.

CO_2 transcritical cycles allow for the use of "natural" refrigerants. These fluids reduce considerably the direct environmental impact respect the traditional refrigerants, as CFCs, HCFCs and HFCs. The main issue connected to CO_2 transcritical cycles regards the lower performances respect cycles using synthetic refrigerants for a given application, especially if compared for high ambient temperatures applications. The transcritical operation brings a high degree of irreversibility in the cycle, in particular localized in the expansion valve. Low efficiency means high consumptions and, consequently, high indirect environmental impact. To make the CO_2 transcritical cycle a competitive alternative, some solutions must be adopted to improve the performances of the basic system. This is exactly the aim of this project.

5.3 System performances

The results coming from the computational simulation are presented in table 5.1. The optimal gas cooler pressure and the resulting maximum COP are shown. The compressor work is

COP	$p_{gc,opt}$	T_{max}	W_{compr}
[-]	[bar]	[°C]	[kW]
2.162	103.6	93.97	27.76

Table 5.1 – System performances for 1-stage vapour-compression CO_2 transcritical cycle at design conditions

also presented along with the maximum temperature in the cycle. Very high temperature are usually avoided in vapour-compression cycles due to the compressor's oil issues that can arise.

6 CO_2 transcritical cycle with parallel compression economization, design conditions

"The major drawback of the use of carbon dioxide as a refrigerant is its low critical temperature which makes it impossible to avoid transcritical cycles when rejecting heat at ambient temperature typical of the summer season of most of the countries in the temperate belt" [19]. In other words, due to the low critical point, carbon dioxide systems offer lower performances for a given application, if compared for warm climates applications. The reasons of this are mainly the considerably exergy losses during the gas cooling process and the the throttling one, for transcritical operation. The non-constant temperature of heat rejection and the high pressure difference of these cycles are the major causes of the low COPs in hot climates conditions of operation.
Parallel compression economization (PCE) is one of the solutions here studied in order to improve the COP of CO_2 vapour-compression transcritical refrigeration cycles. Figure 6.1, provided by the work of Sarkar and Agrawal [18], presents the PI diagram and the (log)p-h diagram of the system.
The main purpose of this improved cycle is to reduce the losses of the throttling process. This is achieved introducing what is the major difference respect the reference cycle: the throttling process is two-staged, in the middle a separator is positioned and the two streams arising from it are compressed, independently, up to the gas cooler pressure. It is important to stress that the two flows going out from the separator take different paths and they are not mixed until the end of the compression process. The beneficial effect of the separator is well described by Chesi et al, *"once the separation took place it is possible to remove the flash gas from the fluid by means of an auxiliary compressor: this prevents the flash vapour from entering the evaporator, where it would not provide any useful effect, thus being sequentially compressed by means of wasted work"* [19]. Moreover, due to the separator, the refrigerant quality at the evaporator inlet is lower than the one for the 1-stage reference cycle. This is another positive effect in fact it increases the cooling capacity.
During the synthetic refrigerants era, the focus of the studies was mainly the fluid characteristics. With the new "natural refrigerants" the kind of research changes. A lot of effort must be put into the design of the components and the improvements of the thermodynamic cycles [19]. This is particularly true for carbon dioxide. Its peculiar properties require considerable

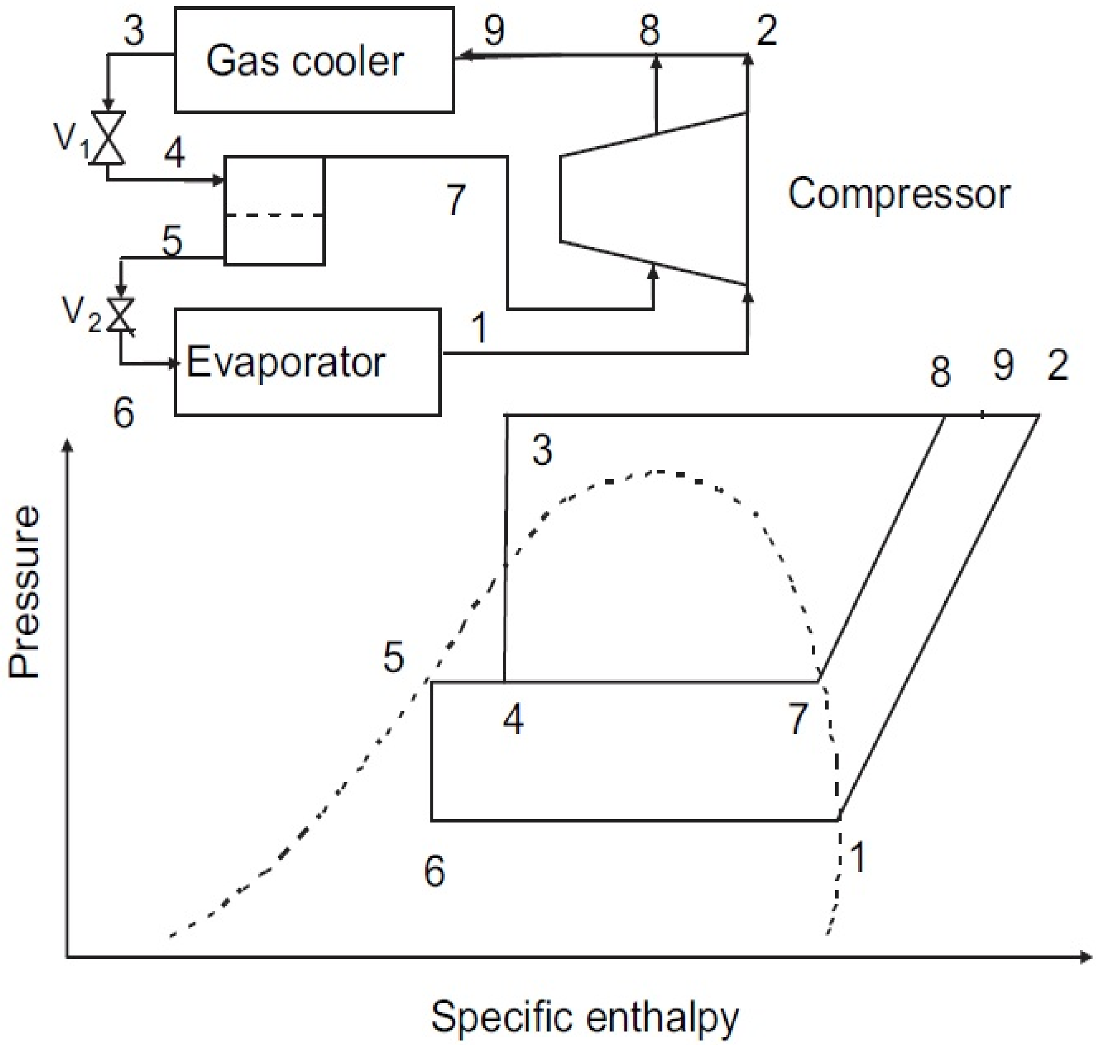

Figure 6.1 – PI diagram and (log)p-h diagram, transcritical CO_2 cycle with parallel compression economization [18]

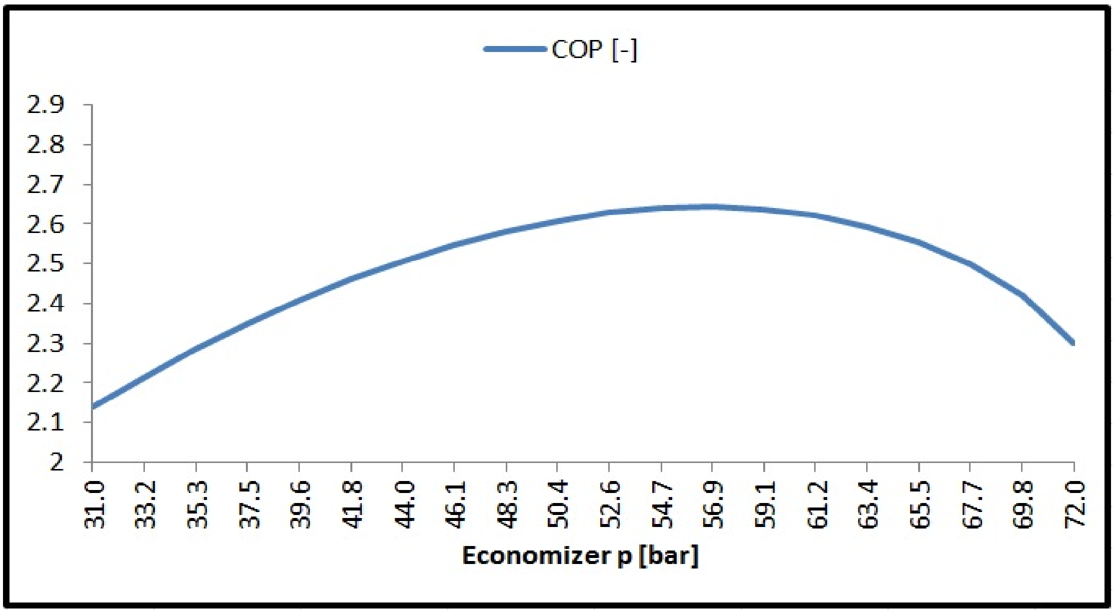

Figure 6.2 – COP vs. economizer pressure, transcritical CO_2 cycle with parallel compression economization

attention to the system planning to achieve efficient enough performances. As reported in the work of Bell [20], PCE is a technique that appears to be very beneficial in CO_2 system while its benefits are much lower for other systems, like hydrocarbons ones.

6.1 System description

The system requires an additional throttling valve, a secondary compressor and a liquid-vapour separator, usually called economizer. The pressure of the refrigerant coming from the gas cooler is firstly reduced until the economizer pressure by means of the primary expansion valve V_1 (states from 3 to 4). The vapour line coming out the separator (state 7) is compressed to the gas cooler pressure (states from 7 to 8). On the other hand, the liquid line (state 5) goes through the evaporator to provide the cooling effect required to the system. After the evaporator, also this stream is compressed to the gas cooler pressure (states from 1 to 2). At the end of the two simultaneous compressions, the streams are mixed (process assumed as isobaric). The mixed flow resulting (state 9) goes through the gas cooler.
In the literature it is common to run into PCE cycles using internal heat exchanger (IHE). *"The internal heat exchanger is not used in this study due to its negligible effect on performance"* [20].

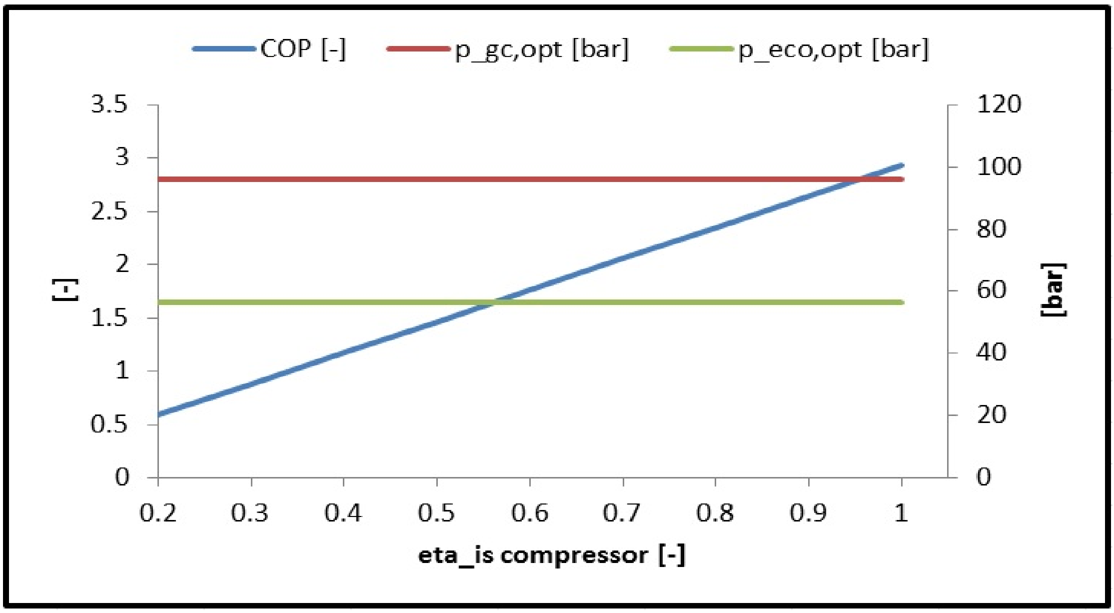

Figure 6.3 – COP, optimal gas cooler pressure and optimal economizer pressure vs isentropic efficiency of the two compressors, transcritical CO_2 cycle with parallel compression economization

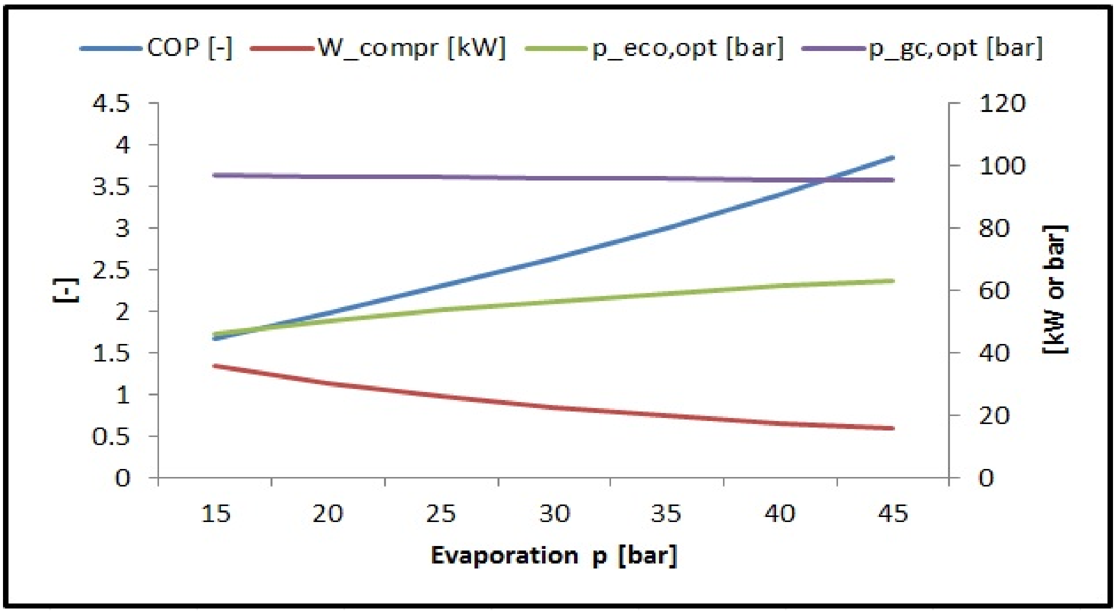

Figure 6.4 – COP, optimal gas cooler pressure, optimal economizer pressure and compressor work vs. evaporation pressure, transcritical CO_2 cycle with parallel compression economization

6.2 System analysis

The operating conditions of the cycle are the same common conditions used for all the studied cycle (see chapter 4 for the assumptions at the base of this investigation). The assumed ambient state forces the system to transcritical operation. For this reason, the gas cooler pressure must be optimized for the given conditions.
It was found that even an optimal economizer pressure can be found for the given operating conditions. Figure 6.2 presents the COP as a function of the economizer pressure. The graph was created keeping constant all the reaming parameters of the cycle, even the gas cooler pressure. The inverted U-shape presented in the figure shows the possibility to find and optimum also for the separator pressure. For this reason, the optimization of the cycle was performed with a simultaneous maximization of the COP leaving gas cooler and separator pressure free to vary.
The same system behaviour is documented in the literature, for instance in the paper proposed by Sarkar and Agrawal. Citing their work, *"the existence of the optimum economizer pressure is mainly on account of the changing slope of the saturation curve"* [18]. Increasing the economizer pressure, the vapour quality at the evaporator entrance decreases. Thus, both the refrigeration effect and secondary compressor work decreases. The impact on the refrigeration effect is due to the increasing enthalpy of the liquid, the impact on the compressor work is connected with the decreased quality. It is clear that the optimum is reached by means of a trade-off between these two opposite contributions. In another paper, Sarkar shows how the optimum economizer pressure is sensitive to the chosen working conditions, even the chosen gas cooler pressure influence the position of this optimum [17].
As remembered by Chesi et al., the optimization of the system as performed here, varying gas cooler and economizer pressure, requires that the compressors are able to vary their volumetric flow. This can be achieved or selecting different compressors or using ones that allow for a rotational speed control. Also a combination of the two cited solutions can be used [19]. It is worthy to remember that the economizer is here assumed to separate completely the liquid and vapour phase. Of course this is not a realistic assumptions and it can affect considerably the performances of the system. As reported by Chesi et al., *"the lack of complete separation within the flash tank is very detrimental for the performances, in some conditions it is even possible to lose completely the positive effect of the parallel compression"* [19]. They conclude their work saying that the separator design is probably the most important aspect to achieve an improvement of the performances respect the reference cycle.
Parallel compression economization is an useful technique to reduce the gas cooler pressure. This reduction brings important system's improvements well summarized by an increase of the cycle's COP [21]. These remarks are confirmed by the computational simulations made during this project. Compared to the 1-stage transcritical reference cycle working at the same operating conditions, the discharge pressure in the PCE cycle is reduced of more than 7%. The COP seems to benefit considerably from this reduction, as it will be presented in the last section of this chapter.
From the same work of Sarkar and Agrawal already cited, it comes an important validation of

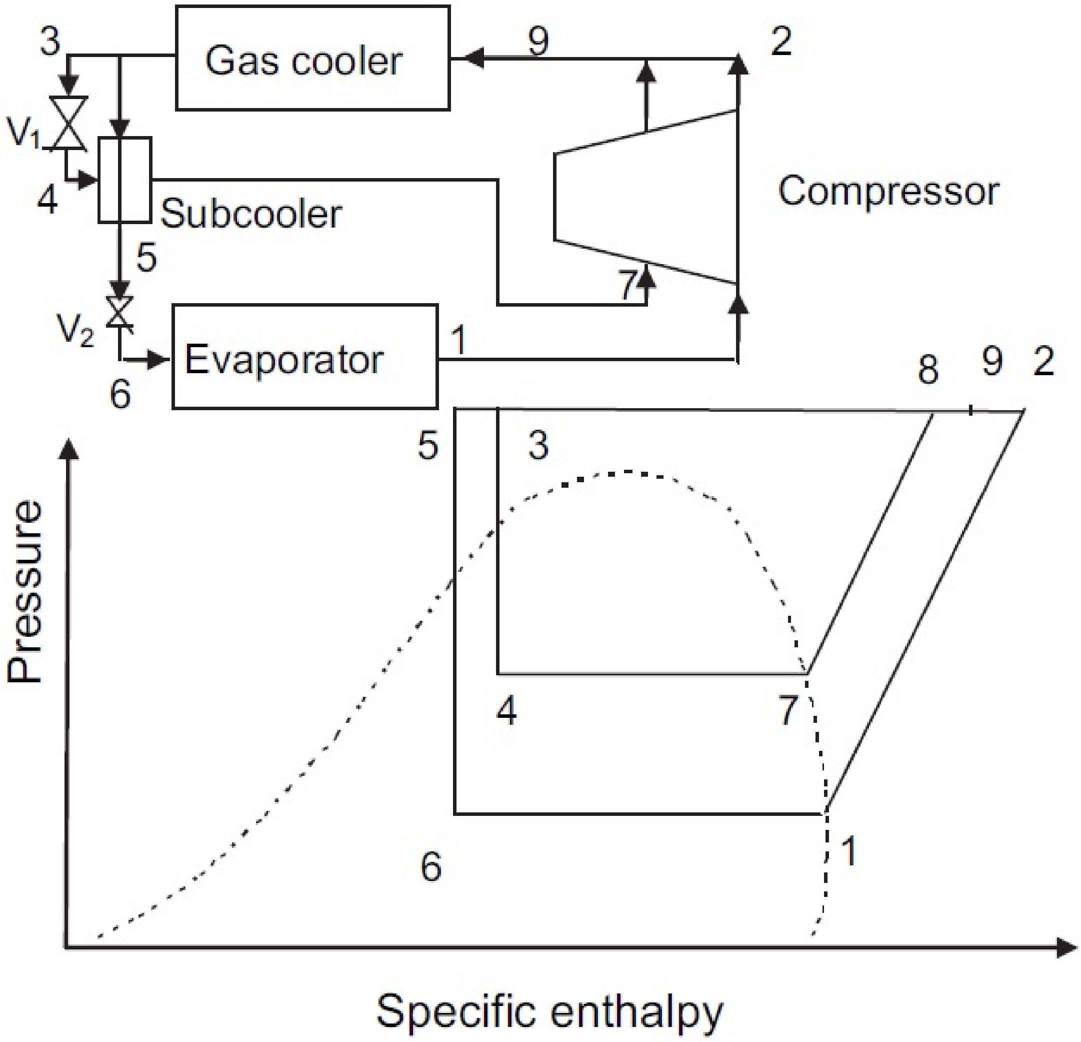

Figure 6.5 – PI diagram and (log)p-h diagram for $CO2$ parallel compression cycle with recooler [18]

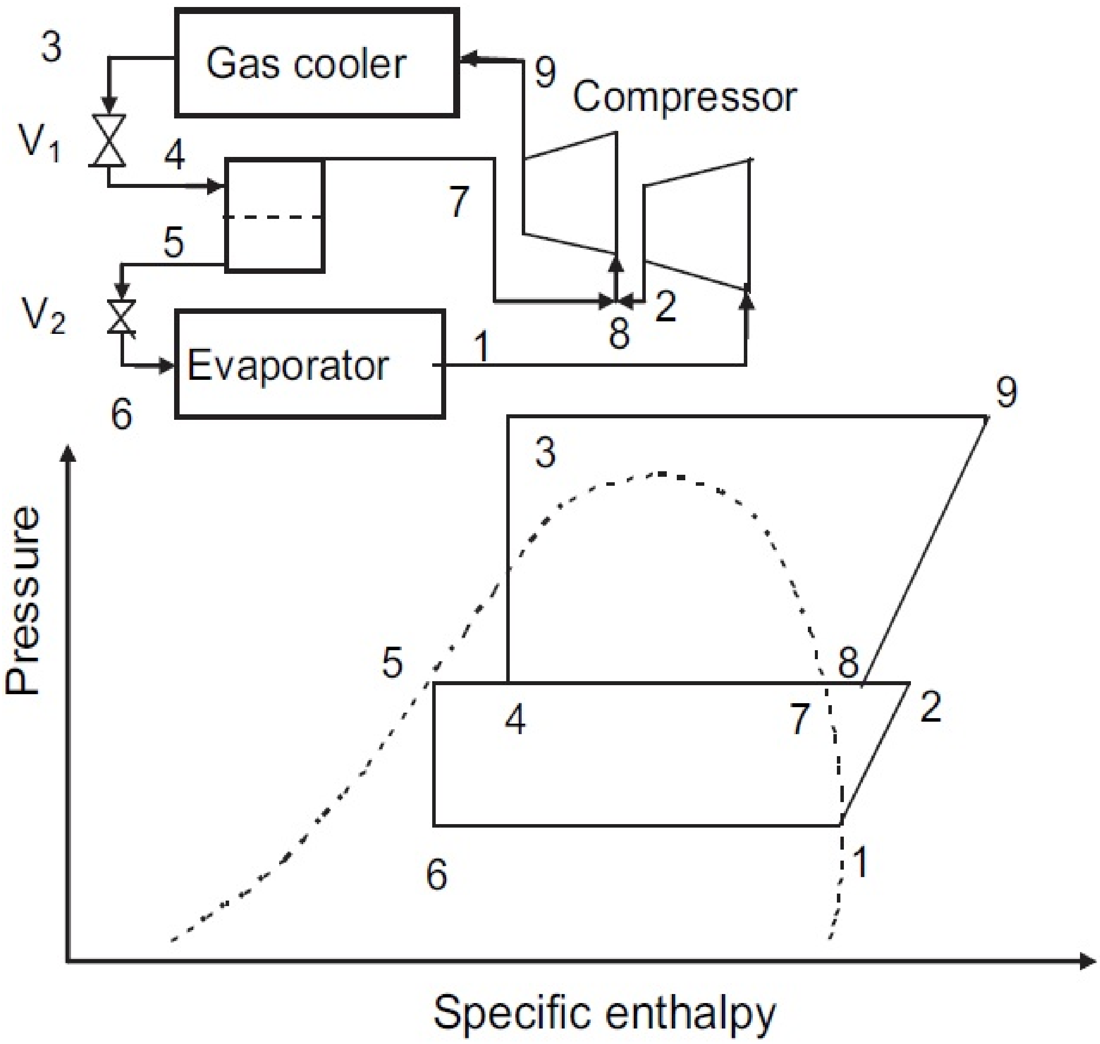

Figure 6.6 – PI diagram and (log)p-h diagram for $CO2$ two-stage cycle with flash gas bypass [18]

the method used in this project. They specify that their investigation was made keeping the isentropic efficiency of the two compressors constant, due to the fact that it does not affect the gas cooler optimum pressure. The same assumption was made during this work and some computational simulations were made to support the choice. Keeping constant all the cycle's parameters and changing only the isentropic efficiency of the two compressors, the optimal gas cooler pressure and the optimal economizer pressure remain constant. On the other hand, obviously, the COP is strongly affected by the different values of isentropic efficiency. For this results see figure 6.3.
The impact of different evaporation pressures on the system's performances was also studied. Figure 6.4 present the system's behaviour as a function of the evaporation pressure. Increasing the evaporation pressure, the optimal gas cooler pressure slightly decreases. The combination of these two effects bring a beneficial reduction of the power input required by the system consequently the COP increases. For higher evaporation pressure, the optimal economizer pressure increases.
"By parallel compression, quality of the refrigerant decreases in evaporator and both refrigeration effect and compressor work increase, but increase in compressor work is less than increase in refrigeration effect, hence COP increases" [17]. Concerning the refrigerant quality at the evaporator inlet, a computational investigation was performed. Compared with 1-stage reference system, the PCE cycle allows for a reduction of the quality around 14%. Improvements regarding COP and compressor work are presented in the next section.

The section about the system analysis can be concluded reporting some interesting considerations made by Sarkar and Agrawal. Figure 6.5 and 6.6 presents systems having, at least apparently, a similar thermodynamic behaviour as the PCE cycles. Consequently, it is reasonable to wonder why only the PCE cycle earned such an high consideration among these possible system's improvements. This is the same question that Agrawal et al. asked themselves in their paper [18]. They found that PCE cycles and the ones with recooler have similar performances from the COP point of view but PCE cycles becomes more and more beneficial the lower are the temperature of application (T_{evap}). Moreover, also from the costs point of view the PCE cycles seem to be preferable. The lower cost of the separator compared to the one of the recooler is something to take into account when designing a cycle [18]. Two-stage cycles with flash gas bypass present always lower performances than the other two systems.

6.3 System performances

The results coming from the computational simulation of the CO_2 PCE cycle are presented in table 6.1. The optimal gas cooler and economizer pressure are shown, along with the resulting maximum COP. Compressor work and maximum temperature in the cycle are also presented. Figure 6.7 offers a comparison between the performances of the CO_2 parallel compression economization cycle and the CO_2 1-stage reference cycle. Optimal gas cooler pressure and

COP	$p_{gc,opt}$	$p_{eco,opt}$	T_{max}	W_{compr}
[-]	[bar]	[bar]	[°C]	[kW]
2.643	96.04	56.59	87.23	22.7

Table 6.1 – System performances for CO_2 transcritical cycle with parallel compression at design conditions

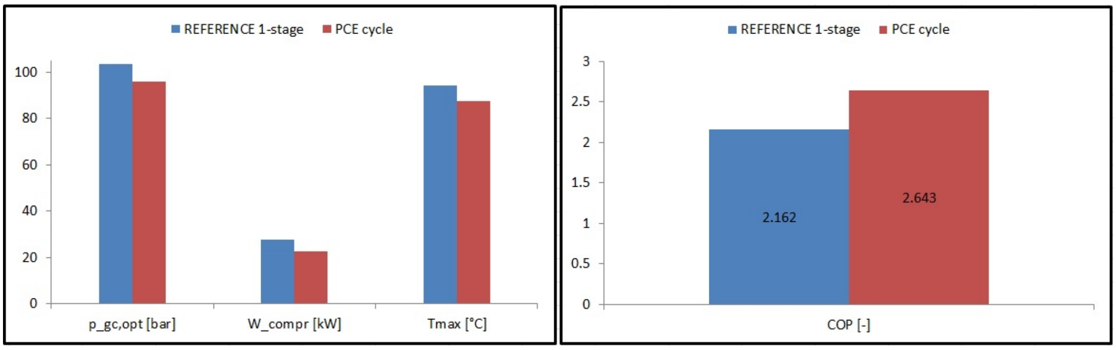

Figure 6.7 – Performances comparison: PCE cycle vs. 1-stage reference cycle

compressor work are reduced, respectively, of 7% and 18% respect the 1-stage reference cycle. The COP faces an increase around 22%.

As reported by Chesi et al., the improvements that the PCE imply respect the 1-stage reference cycle are more valuable when the outdoor ambient temperature are high [19]. This will be even more clear when performing the yearly energy consumptions investigation (part III of this project). At that point a comparison of the cycle's performances in summer and winter will be carried out.

7 CO_2 transcritical cycle with integrated mechanical subcooling, design conditions

"The Committee on the Environment, Public Health and Food Safety of the European Union just (January 2014) voted on restrictive amendments to the F-gas regulation, including both limit to the HFC quantity that can be launched in the market (phase-down) and a ban of the use of HFC for some applications, with different gradualness according to the specific application and the used refrigeration GWP value" [3]. Even the HFCs, expected to be the perfected replacement for the CFCs and HCFCs, are facing their phase out from the market. Their high GWP is as worrying as the CFCs' and HCFCs' ozone depletion potential. Based on data from Legambiente, only in 2012, 10.600 tons of HFCs entered the Italian market. Moreover, around 100.000 tons of HFCs are quantified for the already existing plants. The resulting GWP is impressive: 250 millions tons of CO_2 equivalent. This data, *"together with the fact that the refrigerant recovery rate from existing plants is still very limited in Italy, gives an idea about the phenomenon dimensions and importance and pushes towards the introduction of natural refrigerants, in Italy and more in general in South Europe."* [3].
The harmful effect of HFCs is real and the awareness of this is spreading around the world. The solution is represented by the low-impact "natural refrigerants" just cited, CO_2 first of all. The problem of $CO_2 - only$ systems is the lower efficiency for a given application respect cycles using traditional refrigerant, especially if compared for hot climates applications. Considerable efforts are focused on the performance improvements of carbon dioxide transcritical systems. Mechanical subcooling is one of the solutions that can be adopted for vapour-compression CO_2 cycles in order to increase their efficiency.
Actually, mechanical subcooling is only one of the so called "subcooling technologies". A brief description also of the other solutions is here presented [22]:

- Ambient subcooling: additional heat exchanger area is used to allow the liquid refrigerant to exchange with the ambient, performing the desired subcooling. Subcooler and condenser can be part of the same heat exchanger or the subcooler can be a different device positioned after the condenser. This solution is not always beneficial for the system: the cooling capacity increases along with the refrigerant charge in the system (two opposite effects, one beneficial and one not, are faced by the system).

- Subcooling with liquid-suction heat: the refrigerant coming out the condenser is subcooled exchanging heat with the vapour coming out the evaporator. This solution is beneficial for the system's operation because the absence of gas at the expansion valve inlet and of liquid at the compressor inlet is ensured. On the other hand, this device is not always beneficial for the COP. The refrigerant charge is reduced but the specific volume inside the compressor is increased. Again, two opposite contributions are affecting the system, the resulting COP could be higher or lower depending on the specific application.

- Subcooling with external heat exchange: the refrigerant after the condenser is subcooled in an external heat exchanger. The secondary fluid could be water flowing in a closed loop including a cooling tower. Of course the additional fan and pump work must be taken into account when establishing which solution is the optimal for the given application. *"The investment is high but the savings are also large. The ambient conditions will determine the amount of energy saved. Saving potential is maximum when ambient temperatures are high and is related to compressor run time"* [22].

- Mechanical subcooling: *"the most significant consequence of applying mechanical subcooling is the cooling capacity increases due to the lower quality of the refrigerant flowing into the evaporator"* [22]. Moreover, the discharge pressure is reduced, along with the outlet temperature of the refrigerant from the compressor. This improves the cycle's COP and ensures longer compressor's life. The literature is full of promising results about mechanical subcooling, for this reason this solution is here taken into account to improve the CO_2 cycles performances. Two types of mechanical subcooling are possible: integrated mechanical subcooling and dedicated mechanical subcooling. In the following two chapters they both will be widely discussed.

7.1 System description

Figure 7.1 and 7.2 presents the PI diagram and the (log)p-h diagram for subcritical operation of a vapour-compression cycle with integrated mechanical subcooling. For the transcritical operation the cycle has the same shape, the only difference is the discharge pressure that is higher than the critical point. In the literature was not found an accurate investigation of this kind of cycles for transcritical operation. Anyway, the "subcritical literature" will be used because many thermodynamic behaviours are exactly the same.
As reported from figure 7.1 two "integrated" loops are visible: one is called main cycle, the other is known as subcooler cycle. The main cycle is simply a 1-stage vapour-compression cycle with the addition, after the receiver, of an heat exchanger called subcooler (states from 5 to 6). The subcooler loop is represented by the additional path going from the receiver through the secondary expansion valve (states 5 to 8), subcooler (states from 8 to 9) and secondary compressor (states from 9 to 12). In integrated MS cycles, the condenser/gas cooler

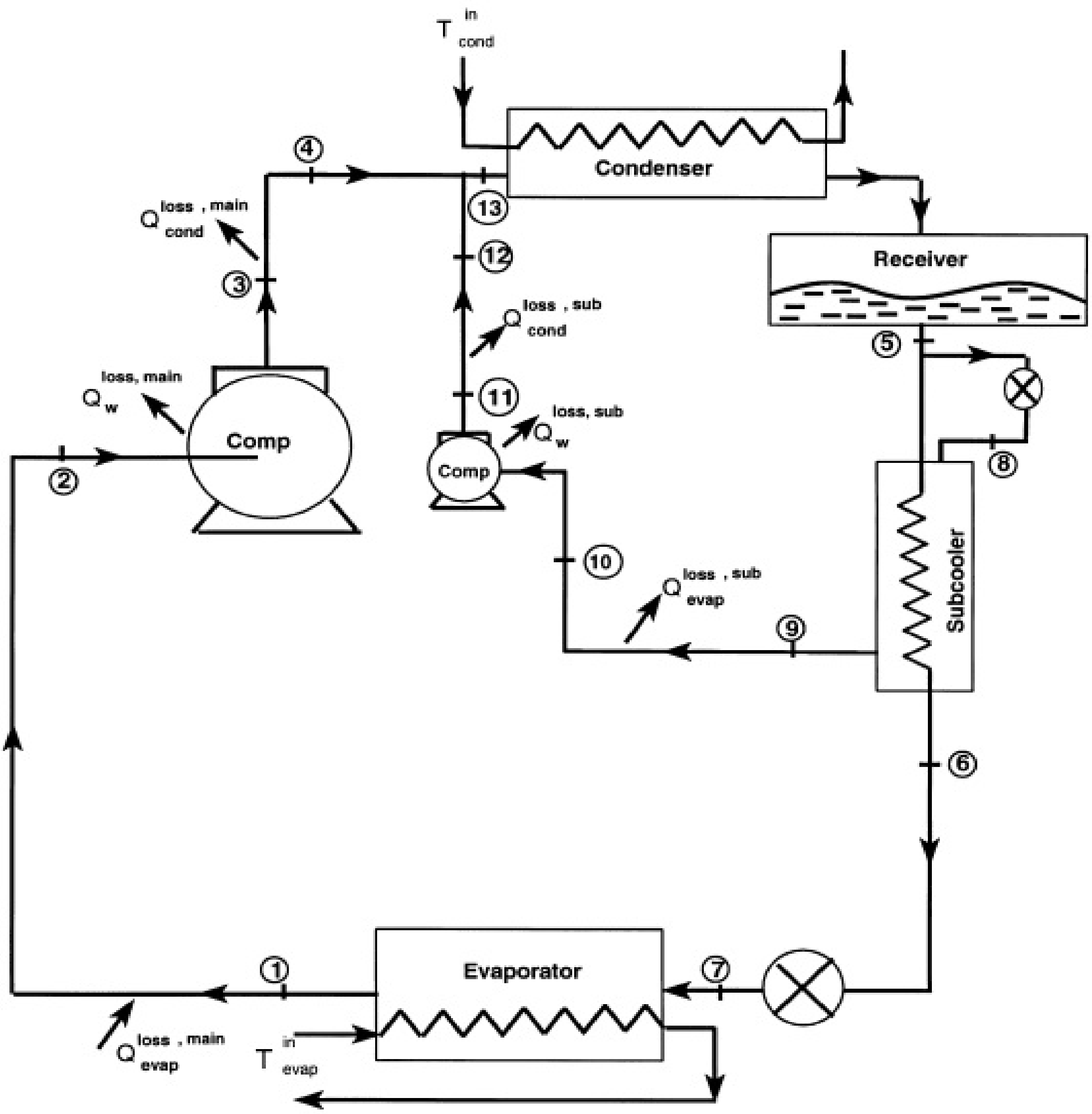

Figure 7.1 – PI diagram, transcritical CO_2 cycle with integrated mechanical subcooling [23]

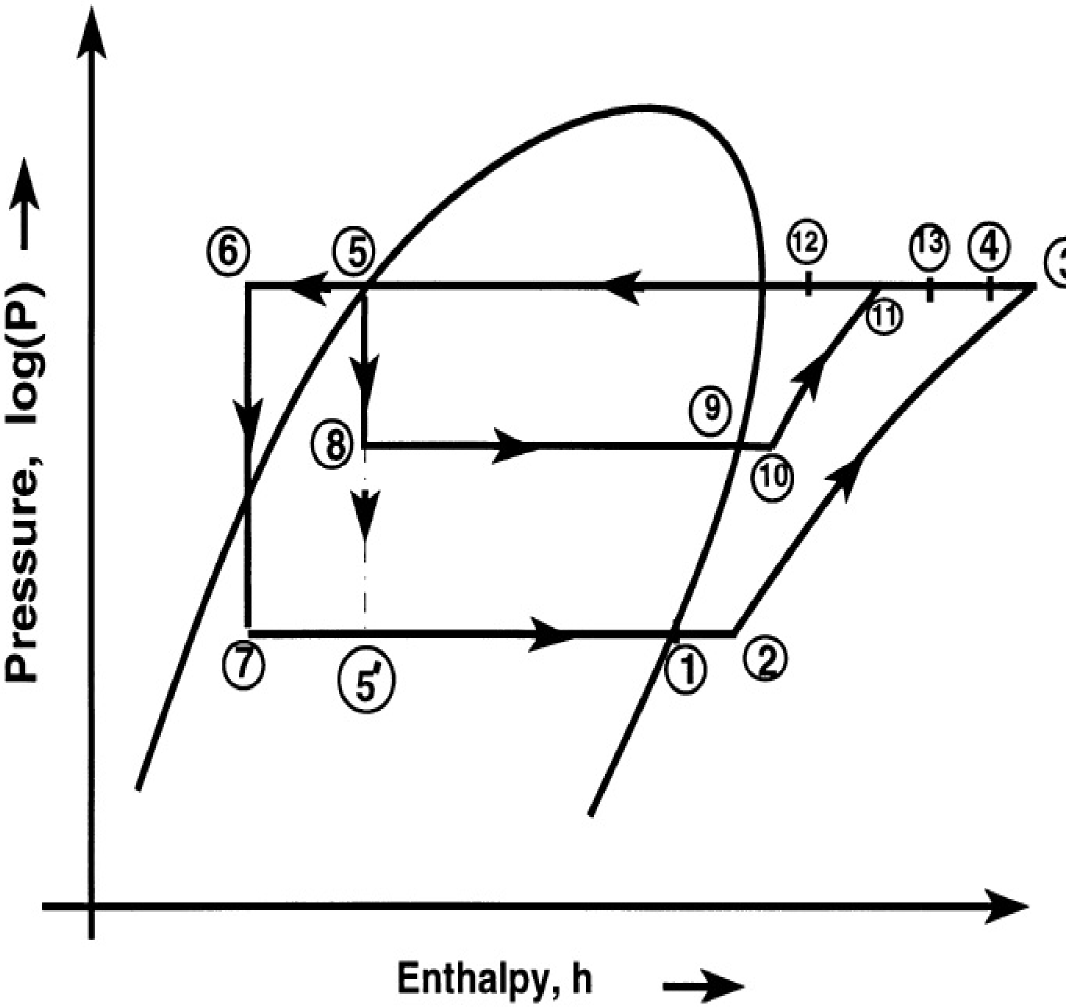

Figure 7.2 – (log)p-h diagram, transcritical CO_2 cycle with integrated mechanical subcooling [23]

is shared by the main and subcooler cycle. In dedicated MS systems, each loop has its own condenser/gas cooler. This is the main difference between the two mechanical subcooling solutions.

The literature about integrated mechanical subcooling cycles agrees that the additional complexity of the system brings improvements to the COP. In particular, the improvements are greater when the difference between condensing and evaporating pressure is high [23]. This result is particularly promising for the study here conducted. For hot climates operation, the pressure difference between evaporator and condenser/gas cooler is really high and the mechanical subcooling cycle performs even better.

Coming back to how the system works, after the receiver (state 5), the refrigerant is divided into two streams. The secondary stream is throttled (state from 5 to 8) and later evaporates inside the following heat exchanger (states from 8 to 9). The main stream enters directly in the subcooler at the receiver pressure. Here it is further cooled down due to the heat flux required by the evaporating secondary stream of CO_2. In this way a reduction of the vapour quality at the evaporator inlet is achieved and the specific enthalpy difference across the evaporator increase. Figure 7.2 presents the comparison between the (higher) vapour quality at the evaporator inlet for the 1-stage reference cycle (state 5') and the (lower) one provided by the integrated mechanical subcooling cycle (state 7). The increase in refrigeration effect is represented by $h_{5'} - h_7$ of figure 7.2. In the models used along this project, the cooling capacity of the system is fixed: an improvement of the specific enthalpy difference across the evaporator allows for a reduction of the mass flow rate circulating in the cycle! Moreover, due to the mechanical subcooling effect, the optimal gas cooler pressure is reduced along with the pressure drop in the expansion valve, bringing further improvements to the system COP. After the subcooler, the secondary flow is directly compressed up to the discharge pressure (states from 9 to 12), while the main flow is throttled and goes through the evaporator (states from 7 to 1). After that, it is finally compressed up to the condenser/gas cooler pressure (states from 1 to 4). The two streams join each other before the condenser/gas cooler (state 13).

Anyway, the benefits of the integrated MS do not come without price. *"The amount of subcooling provided to the main cycle must equal the heat addition to the subcooling cycle evaporator. The heat addition to the subcooler must be rejected in the subcooling cycle condenser at the cost of the work consumed in the subcooling cycle compressor"* [23]. It is clear that an optimal configuration of the system exist and it is the one that ensure the proper balancing of negative and positive effects of the integrated mechanical subcooling. The study of this trade-off is left for the next sections of the chapter.

When performing the computational simulations, the two streams in the subcooler were considered counter-flow (so the draw in figure 7.1 is not precisely the cycle that is here studied but the difference is very small). In order to implement a well-designed heat exchanger, the temperature difference at one side of the subcooler was set. More precisely, considering counter-flow in the subcooler of figure 7.1, it was set $T_6 - T_8 = 5$ °C. Moreover, it was assumed that no heat losses to the environment are faced during the subcooling process.

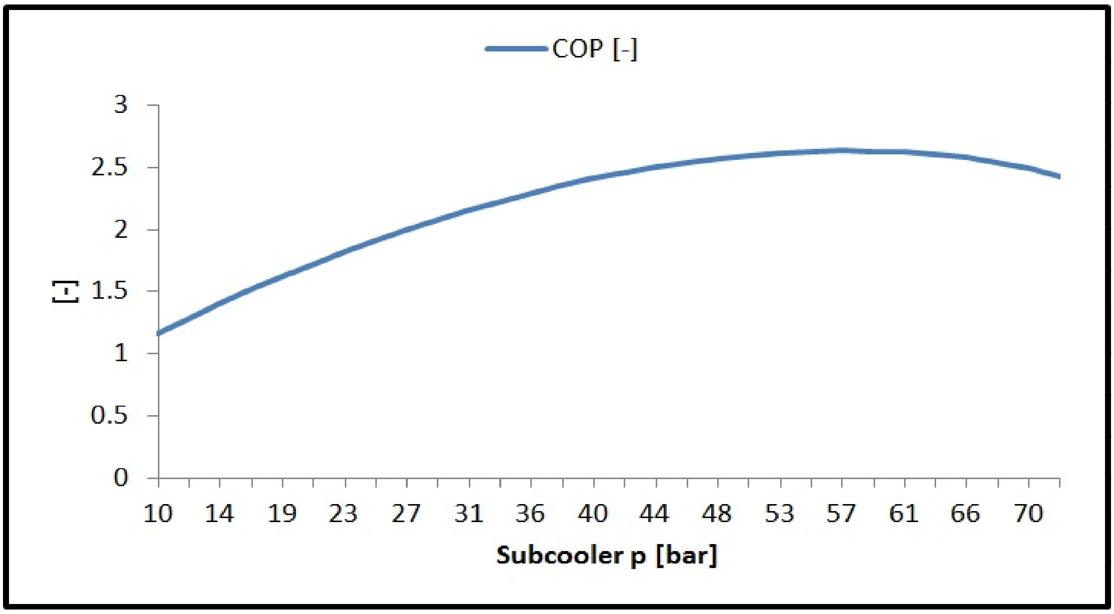

Figure 7.3 – COP vs. subcooler pressure, CO_2 integrated mechanical subcooling cycle

7.2 System analysis

The studied CO_2 vapour-compression cycle with integrated mechanical subcooling operates in transcritical conditions. Consequently, for the given boundary conditions, the optimal gas cooler pressure must be determined.
It was found that also the subcooler pressure (pressure of state 8 or 9 of figure 7.2) has its own optimal value that maximizes the system's COP. Figure 7.3 presents the COP as a function of the subcooler pressure p_{sub}, all the remaining cycle's parameters are kept constant (the gas cooler pressure was set to its optimum value). The inverted U-shape presented in the figure shows the possibility to find an optimal subcooler pressure from the COP point of view. For this reason, the optimization of an integrated MS cycle must be performed looking for the gas cooler and subcooler pressure that, simultaneously, gives the maximum COP for the given working conditions. Anyway, as reported by Zubair [24], the COP curve as a function of the subcooler pressure is quite flat at its top. This means that a small difference of the subcooler pressure from its optimum does not change that much the resulting COP.
The subcooler pressure is the main parameters that affects the performances of integrated mechanical subcooling systems [25]. As reported by Khan et al., *"the irreversibility losses in the expansion device (the major source of irreversibility) can be significantly reduced by operating the system at the optimum subcooling conditions"* [23]. Changing only the gas cooler pressure, the subcooling degree does not change, while it is only affected by the subcooler pressure. It is worthy to specify that, even if ΔT_{sc} does not vary with the gas cooler pressure, the heat exchange in the subcooler does. As a consequence of the subcooling, the quality of the refrigerant at the evaporator inlet is lower, if compared to the 1-stage reference cycle. From

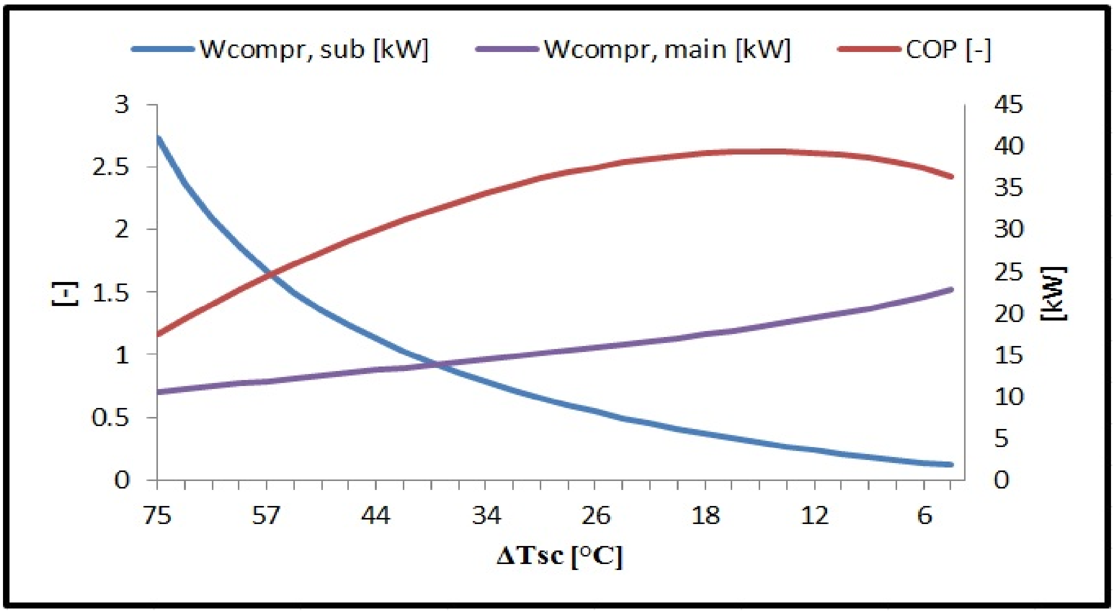

Figure 7.4 – COP and compressor work vs. ΔT_{sc}, integrated mechanical subcooling cycle

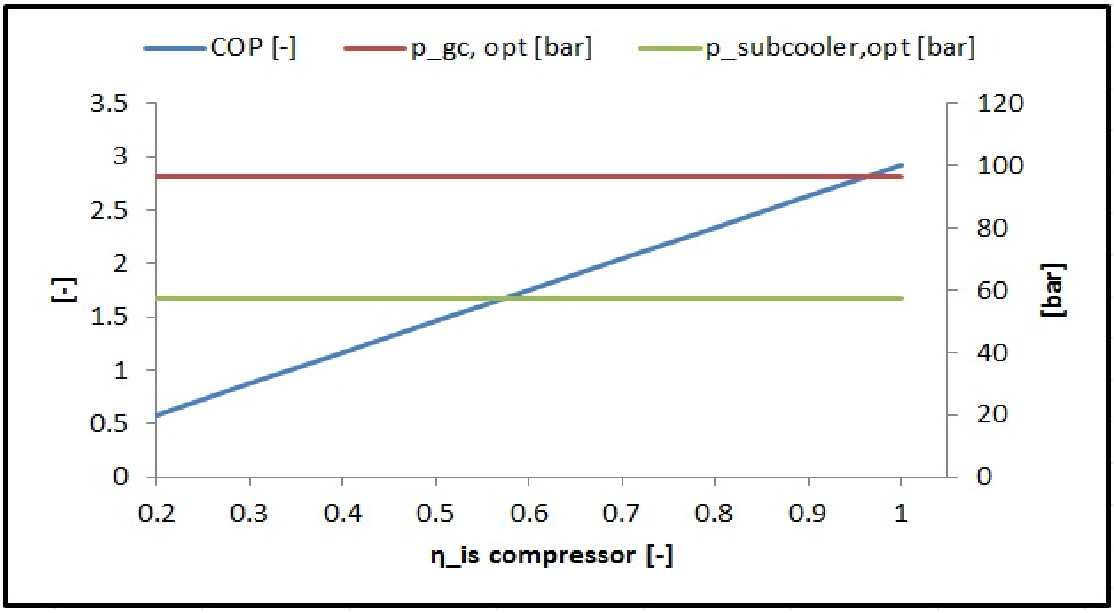

Figure 7.5 – COP, optimal gas cooler pressure and optimal subcooler pressure vs. compressor's isentropic efficiency, integrated mechanical subcooling cycle

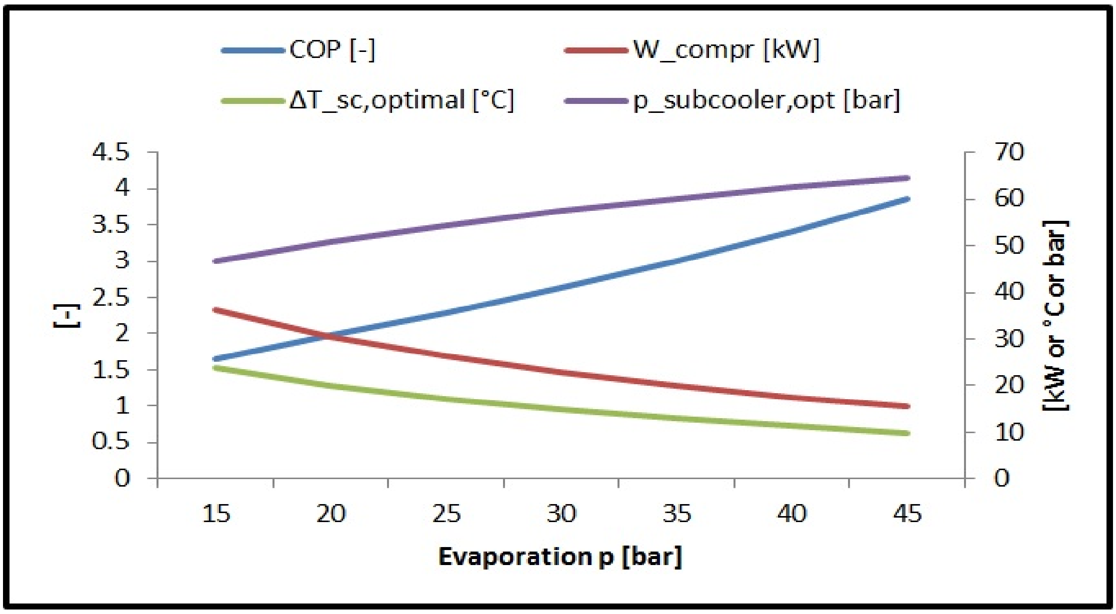

Figure 7.6 – COP, compressor work, optimal subcooler pressure and optimal ΔT_{sc} vs. evaporation pressure, integrated mechanical subcooling cycle

the computational simulations performed, the evaporator inlet quality of the refrigerant is reduced of 42% respect the reference case. *"The low quality of the refrigerant at the evaporator inlet corresponds to an increase in the refrigeration capacity per unit mass of refrigerant circulating in the main cycle"* [23]. Moreover, as it is shown in the last section of this chapter, the optimal gas cooler pressure is reduce respect the 1-stage reference cycle and the system considerably benefits from the reduced compression ratio.

Figure 7.4 deals again with the optimal subcooler pressure. For different subcooler pressure, it is possible to obtain different ΔT_{sc}. Increasing this temperature difference the quality at the evaporator inlet decrease with beneficial effects on the system's performances. Simultaneously also the subcooler compressor's work increases with negative effect on the system. It is clear that is not the highest subcooling the preferable and it is the trade-off between the two contributions that gives the optimal subcooler pressure that brings the maximum COP (for the given working conditions). Of course the additional investment cost that is faced must be mentioned, more components are required. This is a drawback of the integrated MS system that it is not included in the COP.

The impact of the compressors' isentropic efficiency on the system's behaviour is reported in figure 7.5. The increasing value of η_{is} brings improvements to the COP, due to the lower compressor's consumptions. Anyway, the isentropic efficiency does not affect the optimum values of the subcooler and gas cooler pressure.

As reported by figure 7.6, increasing the evaporation pressure the COP increases and the total compressor work decreases. The COP improvement is due to the fact that the optimal gas cooler pressure decreases when the evaporation pressure is raised. Together, these two effects

COP	$p_{gc,opt}$	$p_{sub,opt}$	T_{max}	W_{compr}
[-]	[bar]	[bar]	[°C]	[kW]
2.632	96.34	57.41	87.51	22.79

Table 7.1 – System performances for CO_2 transcritical cycle with integrated mechanical subcooling at design conditions

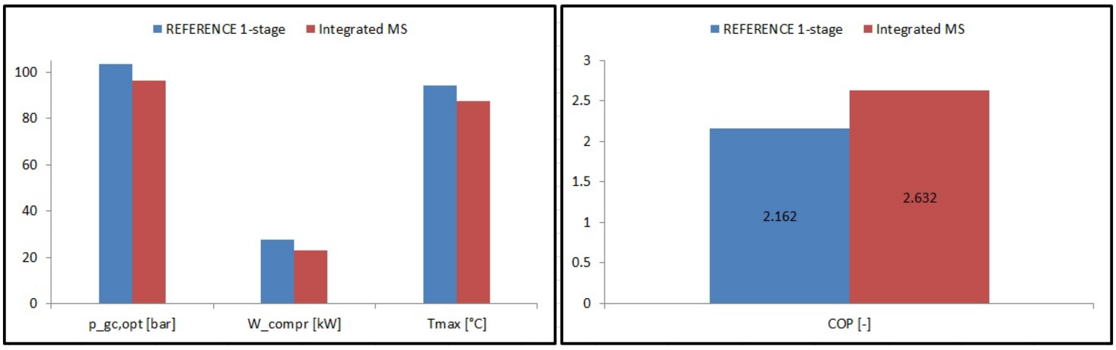

Figure 7.7 – Performances comparison: integrated mechanical subcooling cycle vs. 1-stage reference cycle

provide benefits to the system's performances. The trend of the gas cooler pressure was not reported in figure 7.6 because it would be to chaotic. It is worthy to notice that increasing the evaporation pressure, the optimal ΔT_{sc} decreases and the optimal subcooler pressure increases.

7.3 System performances

The results coming from the computational simulation are presented in table 7.1. The optimal gas cooler and subcooler pressure are shown, along with the resulting maximum COP. Total compressor work and maximum temperature in the cycle are also presented.
Figure 7.7 offers a comparison between the performances of the integrated mechanical subcooling cycle and the 1-stage reference cycle. The COP faces an important improvement, around 22% respect the basic system. Optimal gas cooler pressure and compression power are reduced, respectively, of 7% and 18%.

8 CO_2 transcritical cycle with dedicated mechanical subcooling, design conditions

"Nowadays, cold conditions are mostly produced by vapour-compression, mainly due to the inherent low cost and high efficiency of this principle" [27]. The synthetic refrigerants used in these cycles for the last century are on their way out of the market, due to their devastating impact on the environment. "Natural refrigerants", CO_2 particularly, seem to be the solution to move on, due to their reduced environmental direct impact. *"Below an annual average ambient temperature of 15°C the regular vapour compression cycle with synthetic refrigerants is outperformed by the CO_2 cycle, which makes the latter an attractive solution for some region of the globe"* [27]. Considering as an example the supermarket refrigeration sector, CO_2 transcritical booster cycles are the state-of-art in many countries of the North Europe. On the other hand, CO_2 systems are much less efficient than "traditional" ones if compared for high ambient temperatures applications. Coming back to the supermarket refrigeration example, for applications in hot climates, the preferred solution that includes CO_2 is represented by the cascade systems, due to the low performances of the transcritical one [28].
Lower COP means high environmental indirect effect, represented by an high value of the index TEWI (Total Equivalent Warming Impact). *"Performance enhancement of refrigeration and heat pump systems by cycle modification is an emerging research topic nowadays to reduce the electricity consumed leading to mitigating the problems related to environmental pollution by utility power plants"* [29]. This is particularly true for CO_2 transcritical systems due to their low efficiency in high ambient temperatures applications.
The performances of a vapour-compression refrigeration cycle, exploited in hot climates, can be enhanced using a mechanical subcooling process of the refrigerant after the condenser/-gas cooler. There are two ways to perform the mechanical subcooling, one of them is the integrated mechanical subcooling seen in the previous chapter, the second is the so-called dedicated mechanical subcooling. The main difference between them is the "integrated" subcooler loop used for the integrated MS cycle and the secondary loop, separate from the main one, used in dedicated MS cycle. *"In a dedicated mechanical subcooling system there are two condensers/gas coolers, one for the main cycle one for the subcooler cycle, whereas for an integrated mechanical subcooling system there is only one condenser/gas cooler serving both the main cycle and the subcooler cycle"* [26]. It is worthy to repeat that the literature gives

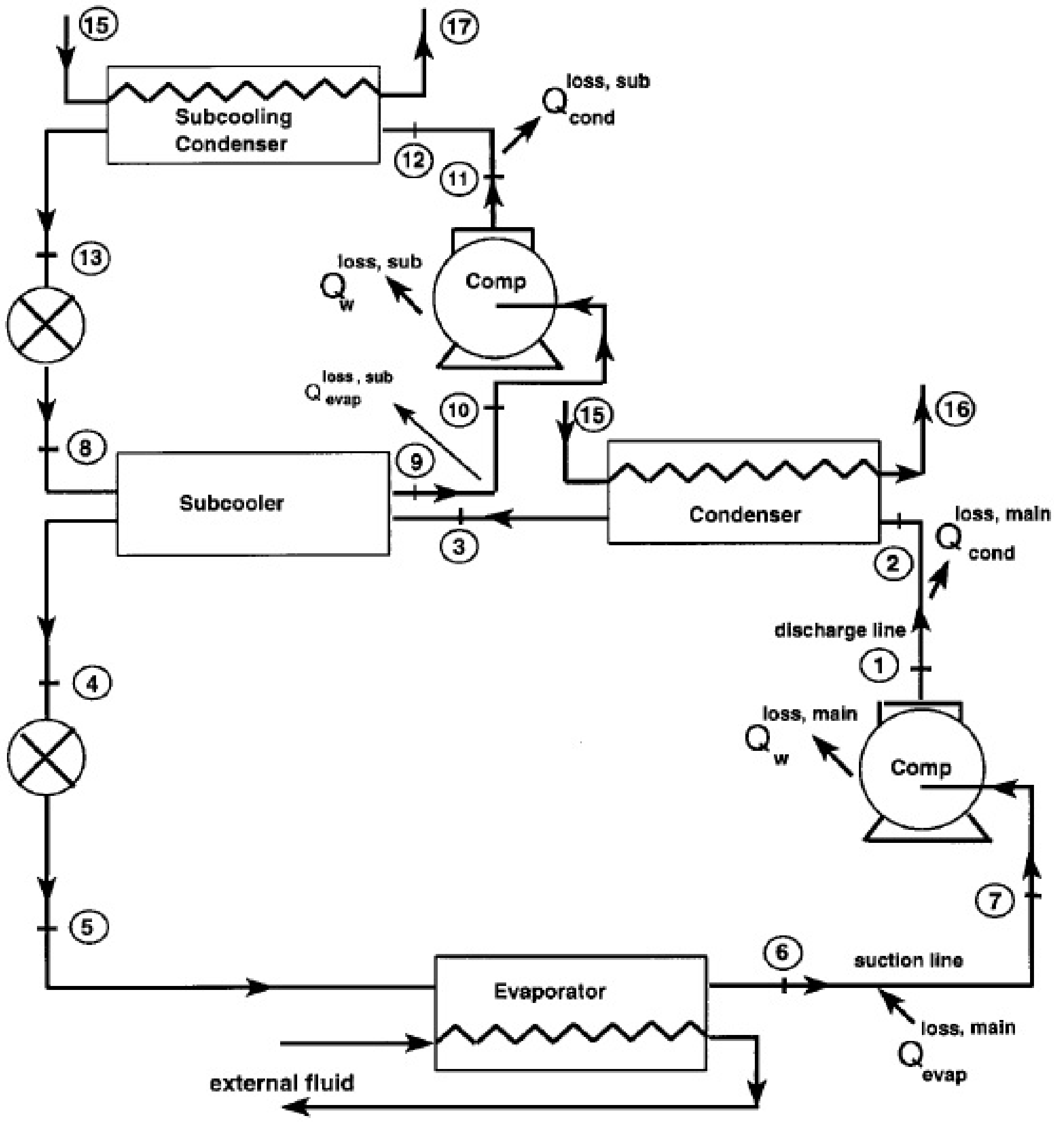

Figure 8.1 – PI diagram, transcritical CO_2 cycle with dedicated mechanical subcooling [26]

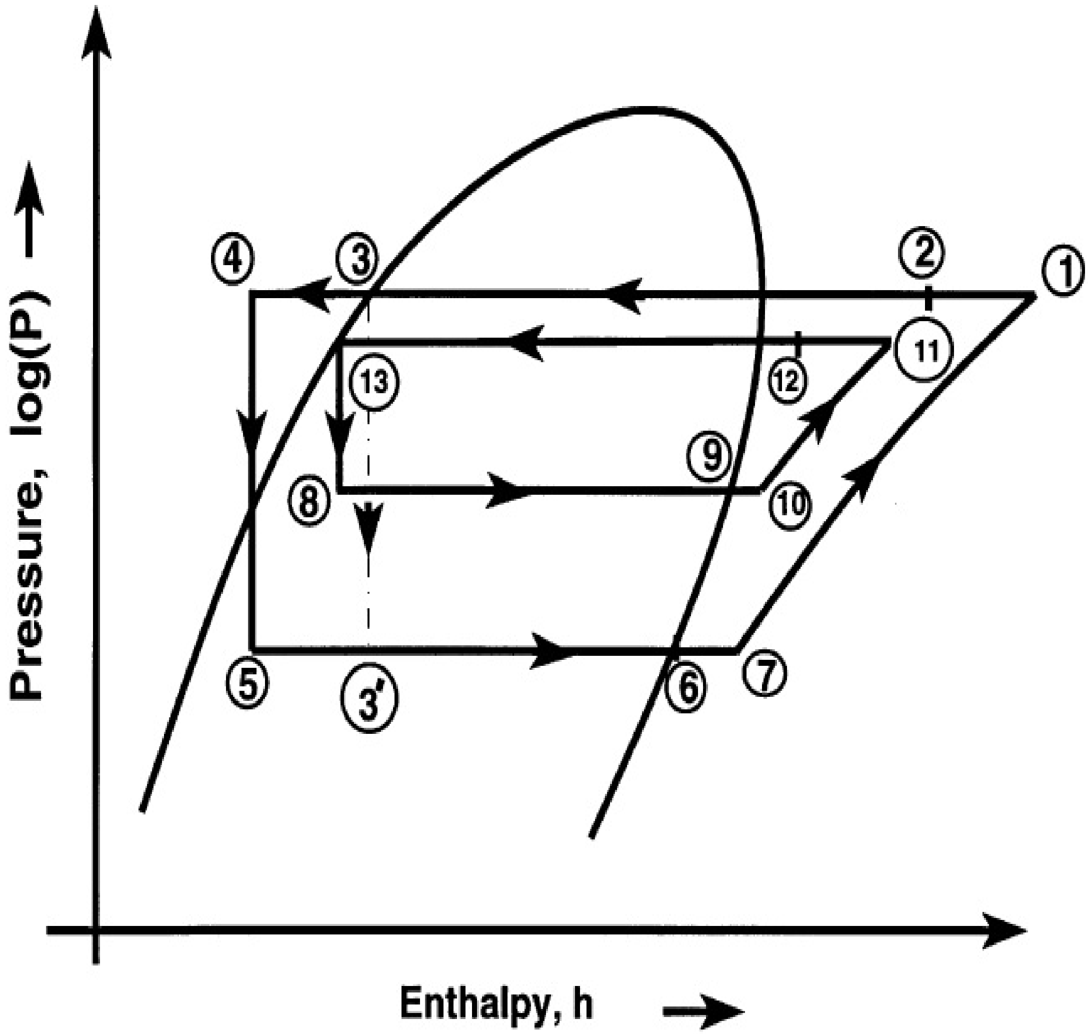

Figure 8.2 – (log)p-h diagram, transcritical CO_2 cycle with dedicated mechanical subcooling [26]

promising results about different applications of mechanical subcooling. Like the integrated mechanical subcooling, the dedicated one gives even better performances when the difference between the evaporation and condensation/gas cooler pressure is high [26].

8.1 System description

Figure 8.1 and 8.2 presents the PI and (log)p-h diagram for subcritical operation of a vapour-compression refrigeration cycle with dedicated mechanical subcooling. The cycle does not have any difference when the systems runs transcritical. Of course the discharge pressure would be over the critical point but the shape of the (log)p-h diagram would be the same. The literature seems to be quite lacking regarding transcritical operation of dedicated mechanical subcooling cycle. Anyway, the "subcritical literature" will be used due to the fact that many thermodynamic behaviours are exactly the same for transcritical and subcritical operation.
Two separated cycles are composing the system, the secondary loop that allows the additional subcooling is called subcooler cycle, while the other is called main cycle. The two systems are coupled by means of an heat exchanger, called subcooler. The subcooler cycle requires its own compressor, condenser/gas cooler, expansion valve and, of course, the subcooler. *"In practise, the components of the subcooler cycle are smaller than those of the main cycle"* [26]. Different refrigerants can be used in the two cycles. Regarding this topic, it is interesting the appendix C or the paper of Qureshi and Zubair [30]. This project is specifically focused on CO_2 systems, for this reason in the two cycles was assumed carbon dioxide. The effect of other refrigerants was not considered during the main studies of this work.
The main cycle works as the 1-stage reference cycle already introduced (chapter 5), the only difference is the additional subcooling after the condenser (state from 3 to 4, figure 8.2). The subcooling is allowed by the evaporating refrigerant in the subcooler loop (state from 8 to 9). During the evaporation an heat flux is required by the secondary CO_2, in this way the refrigerant in the main cycle is further cooled after the condenser/gas cooler. The subcooler cycle works exactly like the reference 1-stage cycle, the subcooler is actually the evaporator of the cycle.
Due to the additional subcooling, the quality of the refrigerant at the evaporator inlet is lower and, consequently, the refrigeration capacity per unit mass of refrigerant circulating is higher [26]. Returning to figure 8.2, state 5 represents the (lower) quality at the evaporator inlet provided by the MS system, respect the (higher) quality faced by the reference 1-stage cycle (state 3'). Beyond the specific cooling capacity gain, mechanical subcooling ensures also a reduction of the optimal gas cooler pressure for the main cycle. Comparing all the cycles for the same evaporation pressure (see the assumptions made in chapter 4), it is clear that a reduction of the discharge pressure brings a COP improvement. In general, a reduction of the compression ratio is always beneficial for the performances of a system [28]. Compressors, usually, work better with smaller compression ratio, moreover the temperatures at the compressors' outlet are lower, increasing the lifetime of these components [31].

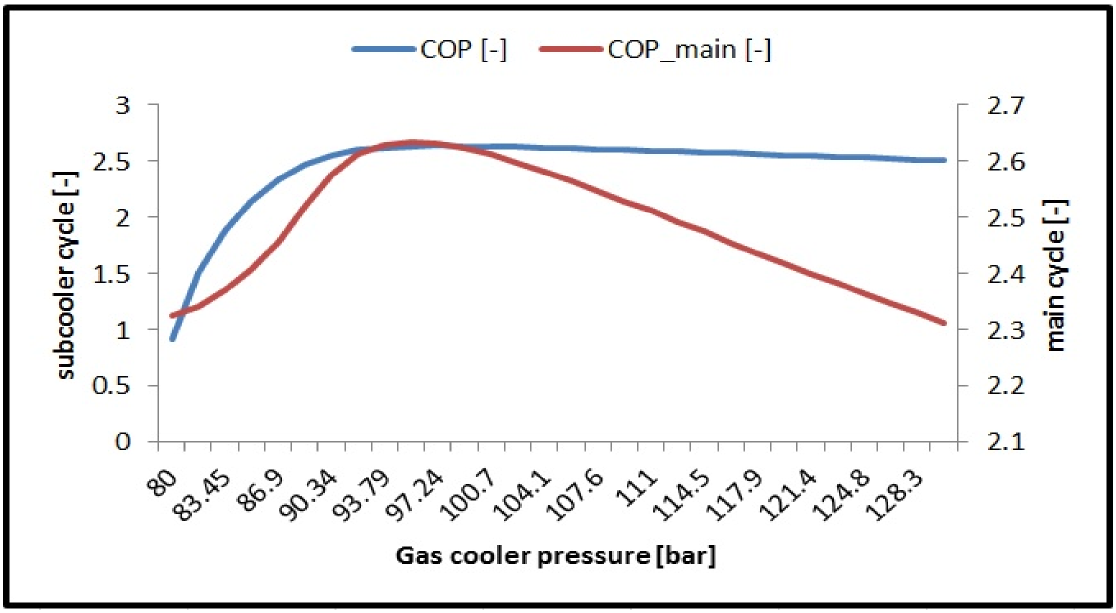

Figure 8.3 – COP vs gas cooler pressure of the main cycle (the COP is addressed as COP_{main}) and the subcooler one (the COP is addressed as COP_{sub}), transcritical CO_2 cycle with dedicated mechanical subcooling

Anyway, these improvements do not come without a price. *"The amount of subcooling provided to the main cycle must equal the heat addition to the subcooling cycle evaporator. The heat addition to the subcooling cycle must be rejected in the subcooling cycle condenser/gas cooler at the cost of the work of the subcooling cycle compressor"* [25]. It is clear that the optimal cycle configuration is the result of a trade-off between al these positive and negative contributions to the system's COP. This trade-off is widely discussed in the next sections. Another price that the cycle modification have regards the higher investment cost that must be faced due to the higher number of components respect the 1-stage reference cycle. The aspect is not reflected in the COP but must be taken into account.
When performing the computational simulations, in order to implement a well-designed heat exchanger, the temperature difference at one side of the subcooler was set. More precisely, it was set $T_4 - T_8 = 5°C$. Finally, it is assumed that no energy losses to the environment are faced in the subcooler.

8.2 System analysis

During transcritical operation of the cycle, the gas cooler pressure must be optimized. The system is provided of two gas coolers, both these two pressures must be optimized for the given operating conditions. Figure 8.3 presents the COP trends changing only the main gas cooler pressure (COP_{main}) and subcooler cycle's gas cooler pressure (COP_{sub}). During these

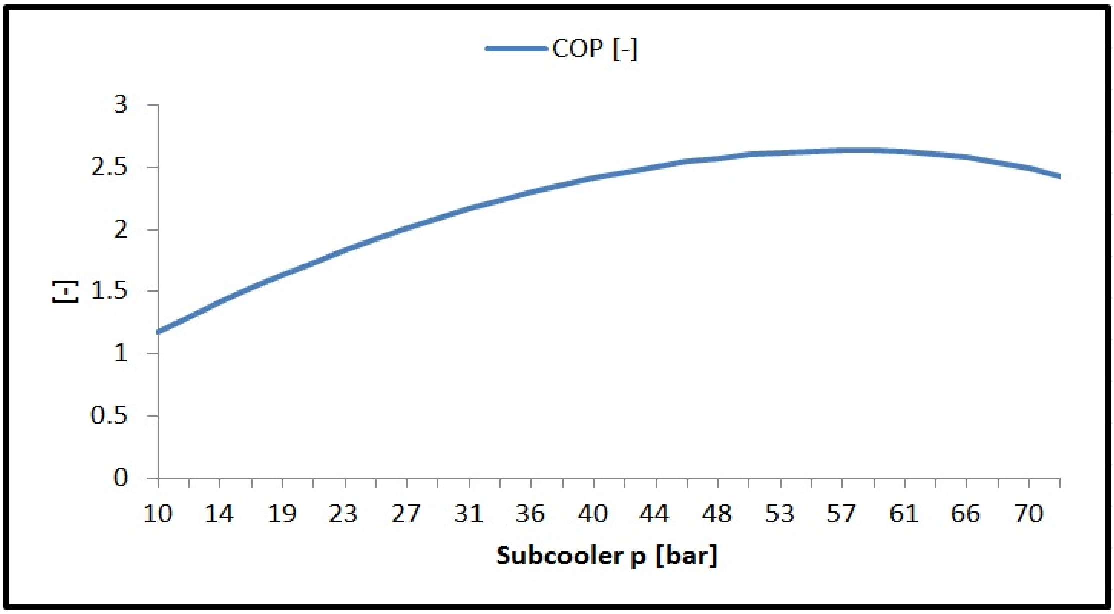

Figure 8.4 – COP vs. subcooler pressure, transcritical CO_2 cycle with dedicated mechanical subcooling

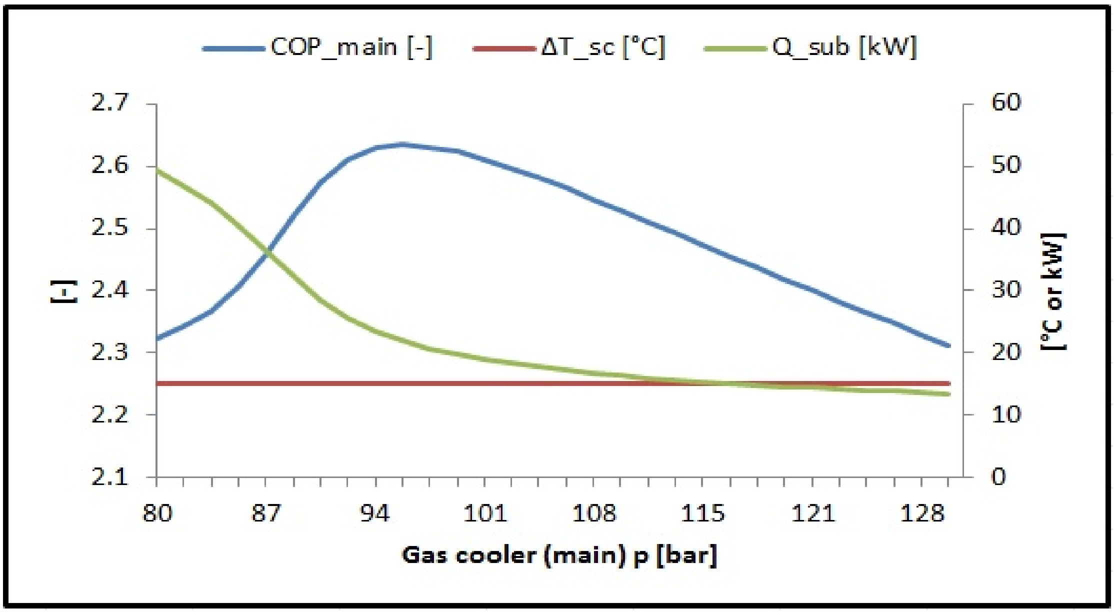

Figure 8.5 – COP, subcooling degree and subcooler heat exchange vs. main cycle gas cooler pressure, transcritical CO_2 cycle with dedicated mechanical subcooling

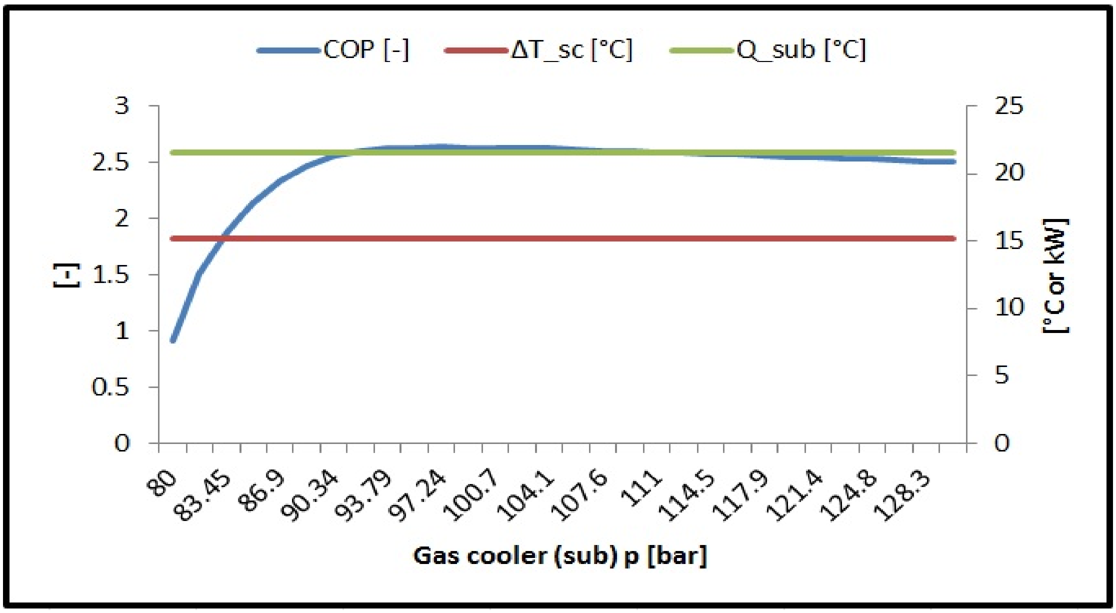

Figure 8.6 – COP, subcooling degree and subcooler heat exchange vs. subcooler cycle gas cooler pressure pressure, transcritical CO_2 cycle with dedicated mechanical subcooling

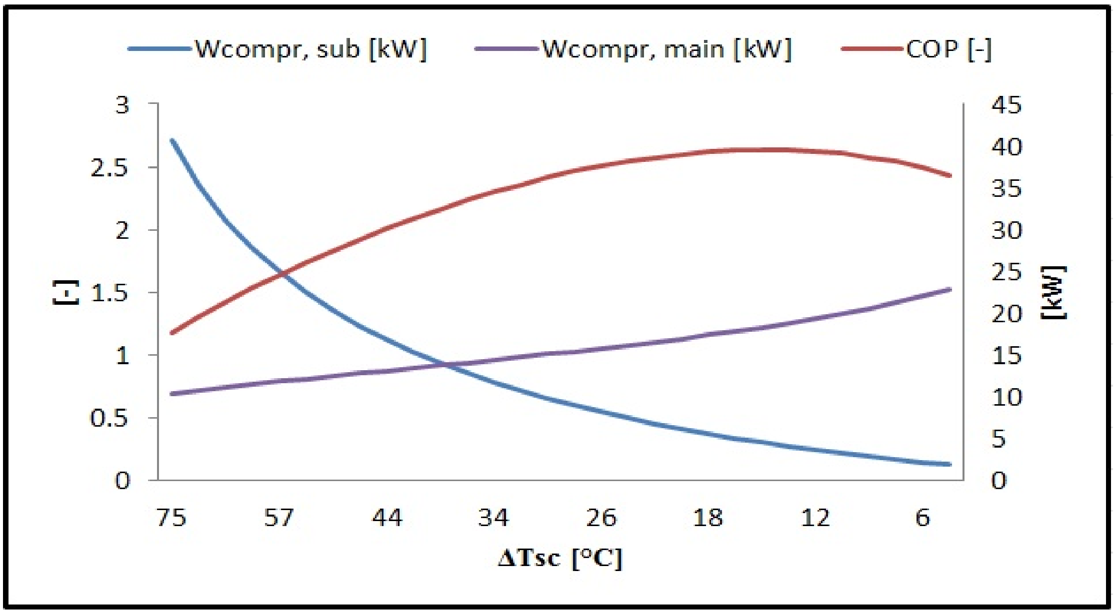

Figure 8.7 – COP and compressors work vs. subcooling temperature difference, transcritical CO_2 cycle with dedicated mechanical subcooling

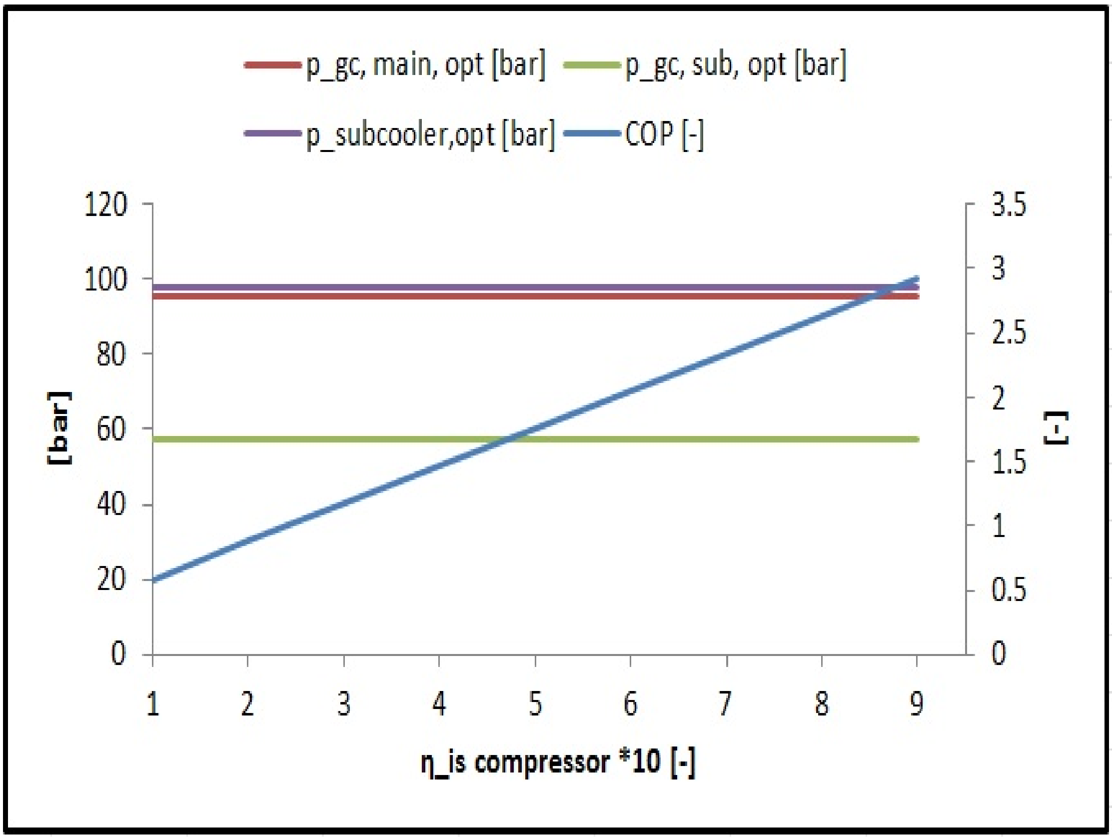

Figure 8.8 – Gas coolers optimal pressure, optimal subcooler pressure and COP vs. isentropic efficiency of the compressor, transcritical CO_2 cycle with dedicated mechanical subcooling

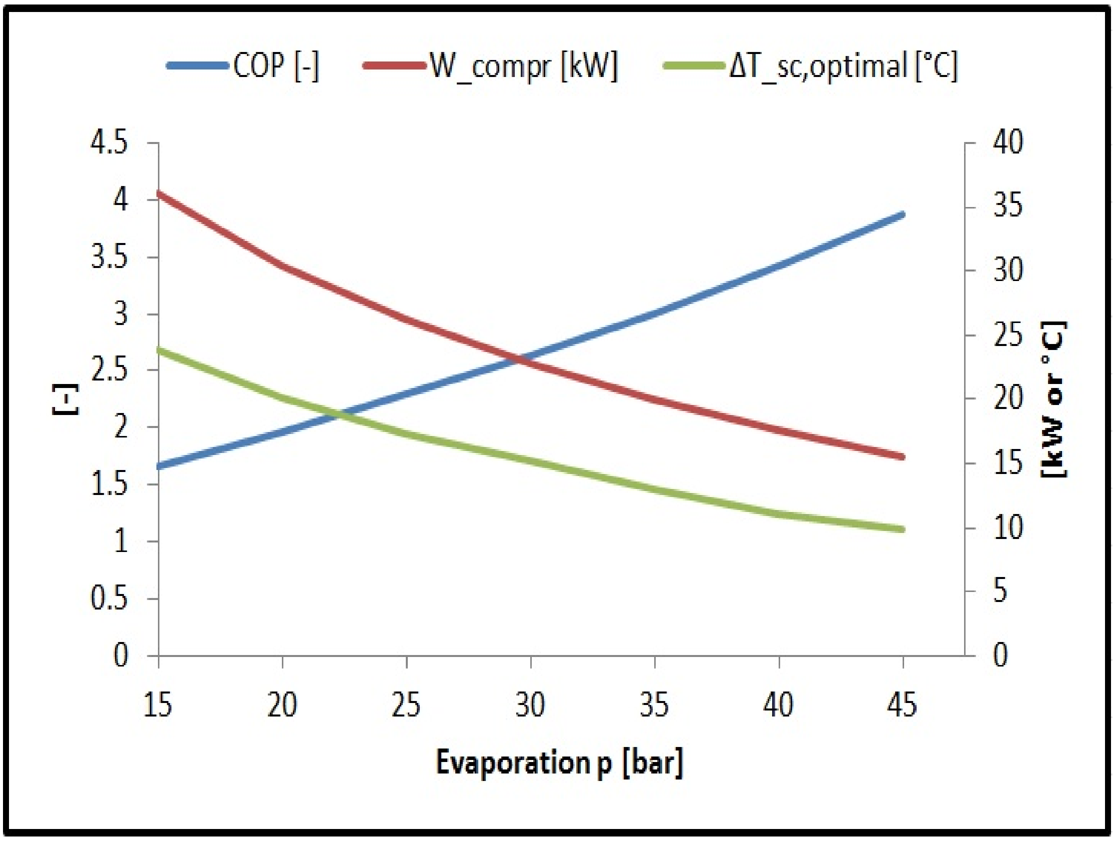

Figure 8.9 – Compressors work, subcooling degree and COP vs. evaporation pressure, trans-critical CO_2 cycle with dedicated mechanical subcooling

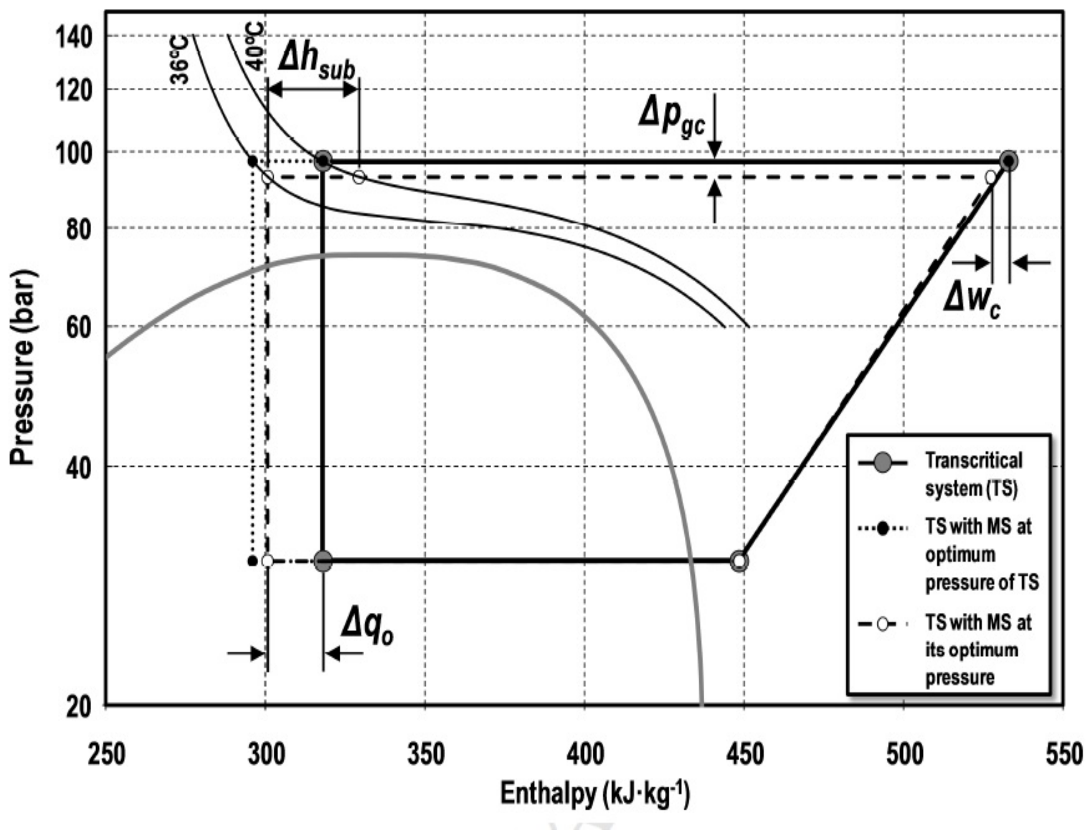

Figure 8.10 – (log)p-h diagram for the reference cycle and the MS transcritical CO_2 one [28]

investigations all the parameters except for one gas cooler pressure were kept constant. When one gas cooler pressure was varied, the other one was kept to its optimal value. As visible, an optimal value exists for both the two gas coolers pressures. This optimum is also quite similar for the two cycles (subcooler and main one).

Actually a three-dimensional optimization of the system is required due to the fact that an optimum exists also for the evaporation pressure in the subcooler cycle. It will be referred to this pressure as "subcooler pressure". Figure 8.4 shows the COP of the system as a function of the subcooler pressure, all the remaining parameters of the systems were kept constant to accomplish this analysis. The inverted U-shape of the graph suggests the existence of an optimal subcooler pressure that maximizes the COP. When optimizing the system, it was looked for the two gas cooler pressures and the subcooler one that, simultaneously, give the maximum COP for the given boundary conditions.

The existence of an optimal subcooler pressure is confirmed by the literature: *"It should be emphasized that there is an optimum temperature for the subcooler at which the COP of the cycle is maximized"* [26]. Moreover, as reported by Thorton et al., *"the performance of a mechanical subcooling system is controlled mostly by the subcooler temperature - and thus the subcooler pressure, not declared by the authors -"* [25].

The subcooling degree is completely controlled by the subcooler pressure. Changes of the main cycle gas cooler pressure, does not affect the ΔT_{sc}, as visible from fig 8.5. Anyway, even if the subcooling degree does not vary, the heat exchange in the subcooler changes along with the gas cooler pressure. The subcooler cycle gas cooler pressure does not have absolutely impact on the heat exchange in the subcooler. As shown by figure 8.6, varying this pressure, neither the subcooling degree nor the subcooler heat exchange vary. Like it was for the integrated mechanical subcooling cycle, the increasing ΔT_{sc} generates opposite trend of the two compressors' works, as visible in figure 8.7. Little subcooling means smaller heat rejection in the subcooler cycle's gas cooler/condenser and, consequently, small work for the secondary compressor. Increasing the subcooling degree, the optimal high pressure of the main cycle is reduced and the compressor work of the main cycle decreases. Moreover, the quality at the evaporator inlet is higher and, consequently, the enthalpy difference across the evaporator is higher. On the other hand, the work of the subcooler compressor increases (because of the higher heat rejection required) and this destroys the benefit of using mechanical subcooling. The trade-off between these opposite contributions gives the optimum subcooler pressure (and consequently the optimum ΔT_{sc}). Of course this optimisation is simultaneous with the ones of the gas coolers pressures. It is actually the optimization of the three all together that gives the perfect balancing of all the opposite forces in the system.

From figure 8.7 it is visible that the optimal ΔT_{sc} is around 12°C. It must be said that for some applications this value could be too high, as reported by Llopis et al. [28]. The practical limitation of this temperature difference is due to the fact that *"high subcooling degrees will be lost due to the heat transfer to the environment during the distribution of the refrigerant"* [28].

The system behaviour for different isentropic efficiencies of the compressors is represented in figure 8.8. The increasing isentropic efficiency brings considerable benefits to the systems performances due to the reduction of the consumptions. Anyway, these changes do not affect

COP	$p_{gc,main,opt}$	$p_{gc,sub,opt}$	$p_{sub,opt}$	T_{max}	W_{compr}
[-]	[bar]	[bar]	[bar]	[°C]	[kW]
2.634	95.55	97.71	57.46	86.78	22.77

Table 8.1 – System performances for CO_2 transcritical cycle with dedicated mechanical subcooling at design conditions

any of the optimized pressure in the system.
The impact of the evaporation pressure on the system's performances is reported in figure 8.9. Along with the increasing evaporation pressure, the two optimal gas cooler pressures decrease too, bringing an important reduction of the compressors work and, consequently, an improvement of the system COP. The optimal subcooler pressure increases with p_{evap}: it is interesting to report that the optimal subcooling degree decreases with the increasing evaporation pressure.
The optimal gas cooler pressure in a MS cycle is lower than the one of the reference 1-stage cycle for the same operating conditions. For this reason, as reported by Llopis et al., the control strategy must be updated when adopting the mechanical subcooling [28]. The highest benefits are obtained if the CO_2 is subcooled at the reduced discharge pressure, instead the higher value of the transcritical reference cycle. This improved control *"allows increasing the COP and the capacity of the system in all the operating conditions"* [28]. Figure 8.10 summarizes all the beneficial effects of the MS cycle respect the reference one and respect the cycle with subcooling at the optimal gas cooler pressure for the reference cycle. If the carbon dioxide's subcooling is performed at the reduced optimal gas cooler pressure, the compression work is reduced and a considerable gain in specific cooling capacity is faced. If the subcooling is performed at the gas cooler pressure of the reference cycle, the COP still increases because of the higher specific cooling capacity but the highest possible benefits are not achieved.

8.3 System performances

The results coming from the computational simulation are presented in table 8.1. The optimal gas coolers and subcooler pressures are shown, along with the resulting maximum COP. Total compressor work and maximum temperature in the cycle are also presented.
Figure 8.11 offers a comparison between the performances of the dedicated mechanical subcooling cycle and the 1-stage reference cycle. Optimal gas cooler pressure and compressor work are reduced, respectively, of 7% and 18% respect the base cycle. The COP faces an improvement around 22%. Additionally, it is worthy to report that the evaporator inlet quality is reduced of 42% respect the 1-stage reference cycle. As already mentioned, this is one of the main positive contribution brought by the mechanical subcooling, along with compressor work and gas cooler pressure reduction.

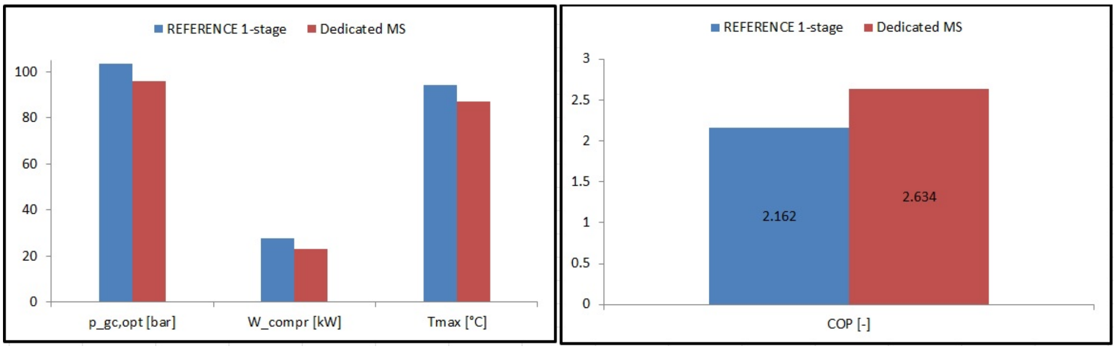

Figure 8.11 – Performances comparison: dedicated mechanical subcooling cycle vs. 1-stage reference cycle

9 CO_2 transcritical cycle with ejector-expansion device, design conditions

In all the computational simulations performed, the process inside the expansion valve was considered isenthalpic. In reality, this process is not isenthalpic and the refrigeration effect is reduced by the losses in this device. *"During the expansion of a refrigerant in a throttling process, much friction heat is dissipated to the refrigerant due to the large kinetic energy increase as the refrigerant pressure decreases"* [33]. Moreover, in transcritical cycles, the throttling losses are even higher than for subcritical cycles. This is due to the big pressure difference across the valve, in fact the refrigerant is expanded from supercritical conditions to subcritical ones [33]. Even if the high discharge pressure allows for more compact systems, it represents one of the main drawbacks of CO_2 systems. Nowadays, as reported by Groll et al., three are the more discussed solutions to recover part of the expansion work and, consequently, to reduce the throttling losses [12]:

- an expansion machine can be used in the CO_2 transcritical cycle. In the opinion of Groll et al., this solution in particularly suitable for CO_2 due to its properties. Citing the authors, *"with a conventional refrigerant such as R-134a, most of the theoretical expansion work comes from the flash gas, and the p-V diagram for the expansion process becomes very narrow with a low mean pressure. For CO_2 the situation is quite different, with most of the work in the liquid phase, a high mean pressure and a small volume requirement"* [12]. Anyway, as reported by She et al., this hardware is quite expensive and many times not feasible, especially for small size applications [34]. Kornhauser, speaking about expanders, said *"such a device would be expensive and prone to damage by low quality two-phase flow"* [35]. Additionally to the costs issue, the use of expanders for commercial refrigeration application is not a mature solution yet, both from the hardware and control point of view [36]. As reported by Fukuta et al., the expansion from transcritical conditions to two-phase ones is complicated and still not clear [37].
- another solution is the replacement of the expansion valve with a vortex tube expansion device. Even if the mechanism inside the tube is still not clear, the performances increase that can be achieved reach almost 40%, based on the data provided by Groll et al. [12]. To obtain this result they assumed a vortex tube's efficiencies of 100%. Anyway, a wide

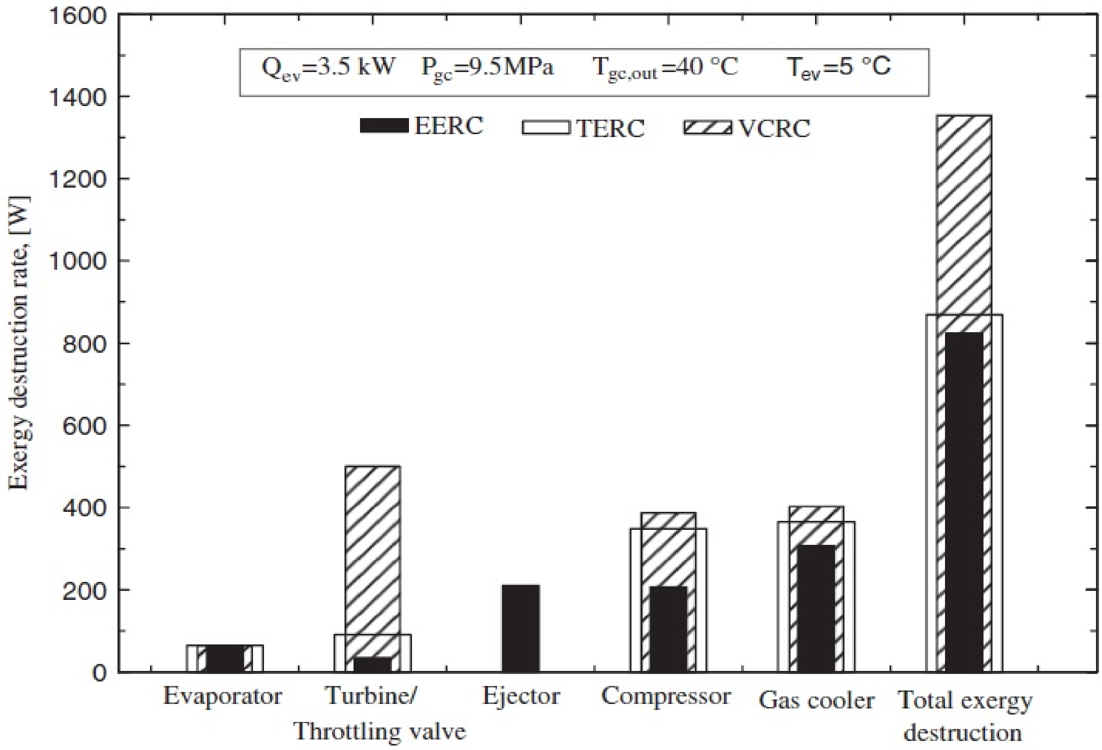

Figure 9.1 – Components and total exergy destruction comparison for different CO_2 vapour-compression cycles [32]

discussion is still open regarding the efficiencies that it is possible to reach with this device. Not enough experimental data are available about this solution that is still at its early stages.

- a widely discussed idea to reduce the expansion losses is the use of an ejector instead of the expansion valve. This solution seems the most promising among the one here proposed. Fixed size ejectors do not have moving parts, so they ensure durability and reliability, moreover their cost is considerably reduced respect the expansion machine. The literature about CO_2 transcritical cycles using an ejector is already quite vast and all the authors agree on the beneficial effect of the ejector as expansion device.

In EERC (Ejector Expansion Refrigeration Cycles), the expansion of the refrigerant is performed by the ejector. This device is able to recover the expansion work that goes all lost using the traditional expansion valves. This work recovered is used to pre-compress the vapour at the outlet of the evaporator. In this way, the compressor's suction inlet pressure is increased, providing a reduction of the compressor work. Moreover, generally the compressors' efficiencies are improved by reduction of the pressure ratio. Also the cooling capacity benefits from the use of an ejector because the specific enthalpy difference across the evaporator is bigger respect the systems using a traditional expansion valve. All the literature agrees on the possibility to achieve improved performances of the CO_2 transcritical 1-stage cycle using ejectors as expansion device. Ersoy et al. made an important literature review regarding CO_2 systems with ejector. They found out that *"in all the literature and experimental results conducted on ejector expander refrigeration/heat pump cycles, the COP has been shown to be higher than that in the basic system."* [32]. Here some emblematic examples from the literature are reported. The Japanese company Denso Corporation developed in 2004 what they claimed to be *"the world's first passenger vehicle air conditioning system using an ejector"* [38]. Their ejector CO_2 system provided a system's performances improvements of 25% respect the conventional vapour-compression car air conditioning. Quite effective is also the study of Ersoy et al. [32] about exergy destruction in three different CO_2 cycles. Figure 9.1 comes from their study and compares a vapour-compression traditional cycle (VCRC), an EERC and a turbine expander vapour-compression refrigeration cycle (TERC). A considerable reduction of the irreversibility is achieved passing from the traditional system to EERC. Referring to figure, ejector cycles seems to perform even better than TERC from the exergy destruction point of view, for the studied operating conditions.
EERC are provided of a liquid-vapour separator at the ejector outlet. This brings a second important advantage: the flash gas bypass allows for a reduction of the evaporator size [29]. The use of an ejector to enhance the performances of CO_2 vapour-compression refrigeration systems seems to be a quite attractive solution. In addition to the cited positive effects on the system's operation, ejector are remarkable *"due to no moving parts, low costs, simple structure and low maintenance requirements"* [29].

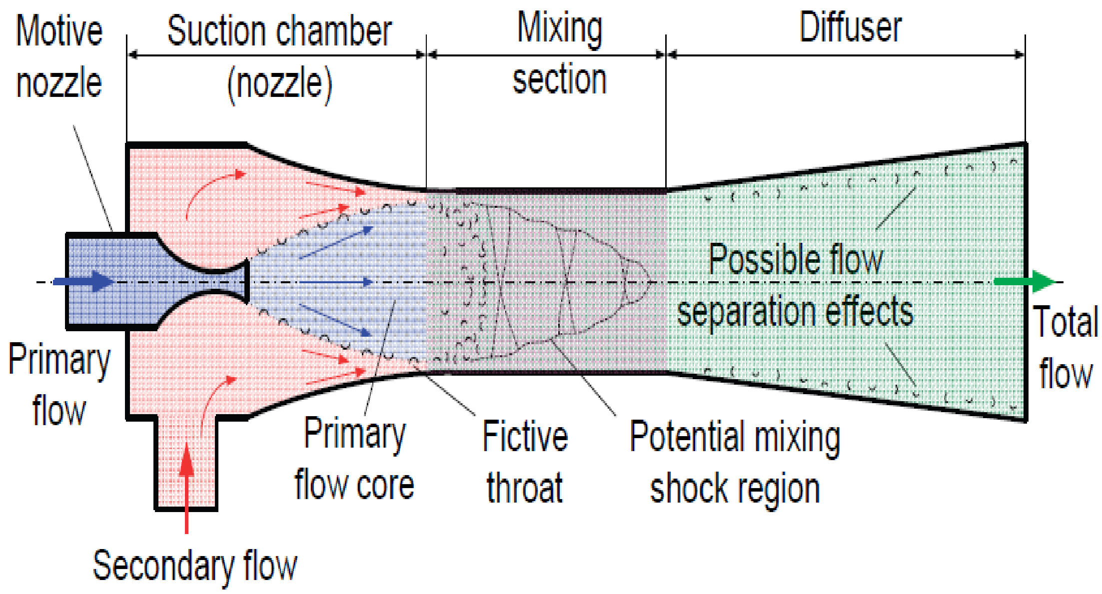

Figure 9.2 – Representation of a two-phase ejector [39]

9.1 The ejector model

A typical two-phase ejector used for performances enhancement of CO_2 vapour-compression refrigeration cycles is presented in figure 9.2. *"The working principle of the ejector is based on converting internal energy and pressure related flow work contained in the motive fluid stream into kinetic energy"* [39]. Internally, the ejector can be divided into four parts:

- motive nozzle: converging-diverging nozzle that converts the high pressure of the refrigerant coming from the gas cooler into high velocity of the flow going out from the nozzle. Supersonic speeds are reached at the outlet of this first part of the ejector. During typical operation in CO_2 vapour-compression cycles, the outlet of the motive nozzle is refrigerant in two-phase conditions.
- suction chamber: the primary flow interacts with the secondary flow due to the high velocity and low pressure of the former. The pressure at the end of the motive nozzle is lower than the evaporation one, thus, the secondary flow is sucked inside the suction chamber. The momentum of the primary flow accelerates the secondary flow that enters the chamber. Still no mixing of the two flows happens until the mixing chamber. As reported by Elebel et al., *"an additional suction nozzle can be used to pre-accelerate the relatively stagnant suction flow"* [39]. In this way the losses inside the ejector are reduced due to the small velocity difference of the two fluids.
- mixing chamber: primary and secondary flow completely mix in this part of the ejector. At the mixing chamber outlet the resulting stream has still high velocity.

- diffuser: it is used to convert the still high velocity of the stream into a pressure increase of the refrigerant going to the compressor. In this way the compressor's suction pressure is raised and the system's COP benefits from it.

Two are the most important parameters when evaluating an ejector from the refrigeration point of view: the entrainment ratio, related to the cooling capacity, and the pressure lift, connected to the compressor work. The entrainment ratio ω is defined as:

$$\omega = \frac{suction\, mass\, flow\, rate}{motive\, mass\, flow\, rate} \tag{9.1}$$

The ejector's pressure lift p_{lift} is defined as:

$$p_{lift} = diffuser\ outlet\ pressure - suction\ inlet\ pressure \tag{9.2}$$

An ejector properly designed should be able to provide simultaneously, high p_{lift} and ω [39]. Anyway, these parameters do not depend only on the ejector geometry, but also on the working fluid's properties [29].
To deal with off-design operating conditions (different working mass flows), two approaches are possible: or multiple ejectors with different geometry are used or the ejector geometry has to be assumed as not fixed. In this case, the ejector control can be performed as reported by Liu et al. . *"The ejector expansion device has two control options. First, the motive nozzle exit position relative to the mixing section inlet and thus, the suction nozzle throat area is adjustable through a thread mechanism. Second, the throat area of the motive nozzle is adjustable by positioning a needle in the nozzle via a thread mechanism"* [40]. To allow the continuous optimization of the mixing pressure, variable geometry ejector must be considered.

Computational simulation of ejector systems is still a widely discussed topic. Going through the literature, a lot of different models can be found. These models mainly refer to two ideal cases: the constant pressure mixing model and the constant area mixing model. Some more specific models were also realized but their application is very limited and their results were often questioned due to the very complicated modelling [41].
Neither the constant area nor the constant pressure model represents properly what happens inside the ejector. Anyway, they offer an interesting tool to study the overall behaviour of CO_2 vapour-compression refrigeration systems with ejector expansion devices. See Appendix A for a more accurate comparison of the main two ejector models.
Here, the chosen ejector model is the one proposed by Kornhauser in 1990 [35], one of the most used in the literature [39]. His model is a one-dimensional constant pressure mixing one. More precisely, he assumed an "homogeneous equilibrium model" for the two phase flow, meaning that the model is 1-D and the thermodynamic quasi-equilibrium is assumed always for the refrigerant. Isentropic efficiencies for the two nozzles and the diffuser are considered given and constant. They represent the "distance" between the ideal reversible process and the real one, any shock effects are included in the ejector's isentropic efficiencies. For this

study, it was assumed $\eta_m = \eta_s = 0.8$ and $\eta_d = 0.75$. Liu and Groll made an important literature review about the ejector efficiencies, discovering that *"in most of the literature studies, values of 0.7-0.95 were assumed for the individual ejector component efficiencies"* [42]. The ejector efficiencies here assumed were just chosen among the most used in the literature.
In the following lines, the Kornhauser's ejector model will be explained. The mixing pressure p_{mix} must be assumed, it has to be a value lower that the evaporation pressure. It is supposed that the motive and secondary flow reach the same pressure (p_{mix}) at the mixing section inlet. Another initial assumption must be done for the ejector ratio r, defined as:

$$r = \frac{motive\,mass\,flow}{total\,ejector\,mass\,flow} \tag{9.3}$$

At the outlet of the motive nozzle, enthalpy and velocity are found as:

$$h_{m,o} = (1-\eta_m)\cdot h_{m,i} + \eta_m \cdot h_{m,o,id} \tag{9.4}$$

$$u_{m,o} = \sqrt{2\cdot(h_{m,i} - h_{m,o})} \tag{9.5}$$

The conservation of energy establishes that, at the suction nozzle outlet, velocity and enthalpy are defined as:

$$h_{s,o} = (1-\eta_s)\cdot h_{s,i} + \eta_s \cdot h_{s,o,id} \tag{9.6}$$

$$u_{s,o} = \sqrt{2\cdot(h_{s,i} - h_{s,o})} \tag{9.7}$$

It is assumed that no mixing happens before the mixing chamber. The conservation of energy and momentum, at the mixing section outlet gives:

$$u_{mix,o} = (1-r)\cdot u_{s,o} + r\cdot u_{m,o} \tag{9.8}$$

$$h_{mix,o} = (1-r)\cdot h_{s,i} + r\cdot h_{m,i} - \frac{u_{mix,o}^2}{2} \tag{9.9}$$

$$s_{mix,o} = s(h_{mix,o}, p_m ix) \tag{9.10}$$

At the outlet of the diffuser:

$$h_{d,o} = h_{mix,o} + \frac{u_{mix,o}^2}{2} \tag{9.11}$$

$$h_{d,o,is} = h_{mix,o} + \eta_d \cdot \frac{u^2_{mix,o}}{2} \tag{9.12}$$

$$p_{d,o} = p(s_{mix,o}, h_{d,o,is)}) \tag{9.13}$$

$$r = x_{d,o} = x(p_{d,o}, h_{d,o}) \tag{9.14}$$

The last equation represents a constraint on the quality at the diffuser outlet, it is used to ensure compressor constant mass flux. This conditions ensures that the system is working at steady state conditions.

Some iterations are required to find the solution of the system. The initially assumed value for r must be compared to the one found using equation 9.14. If they are not identical, a new value of r must be assumed. Kornhauser suggests the use of the arithmetic mean of the two values to reach a fast convergence [35]. The EES model used for the computational simulation of the ejector cycle was based on the reported equations. An iterative procedure with an internal check on the two r values was built.

To complete the ejector's description, some limitations of the model are discussed. First of all, the 1-D analysis does not allow for an accurate investigation of temperature and velocity gradients inside the ejector. Anyway, this project is not focused on inside phenomena of the ejector thus it is not a big issue. More important, cross-sectional homogeneity and thermodynamic equilibrium are incorrect assumptions. This can lead to important differences from the real ejector behaviour.

9.2 System description

Now that the ejector has been introduced, the overall EERC can be described. Figure 9.3 presents PI and (log)p-h diagram for a CO_2 transcritical vapour-compression refrigeration cycle with ejector expansion device.

The refrigerant, after the gas cooler is expanded (state from 3 to 4) and consequently compressed (state from 6 to 7) in the ejector. After the ejector the stream is divided into two saturated flows by the liquid vapour separator. The saturated vapour flow (state 1) is sent to the compressor. The saturated liquid flow (state 8) is throttled and after that goes into the evaporator. The evaporator works in a liquid recirculation system due to the additional two-phase ejector. *"In the ejector cycle, an expansion valve is positioned at the liquid outlet of the separator. It is used to maintain a pressure difference between the ejector suction inlet and the ejector outlet by controlling the suction mass flow rate"* [43]. The refrigerant coming out

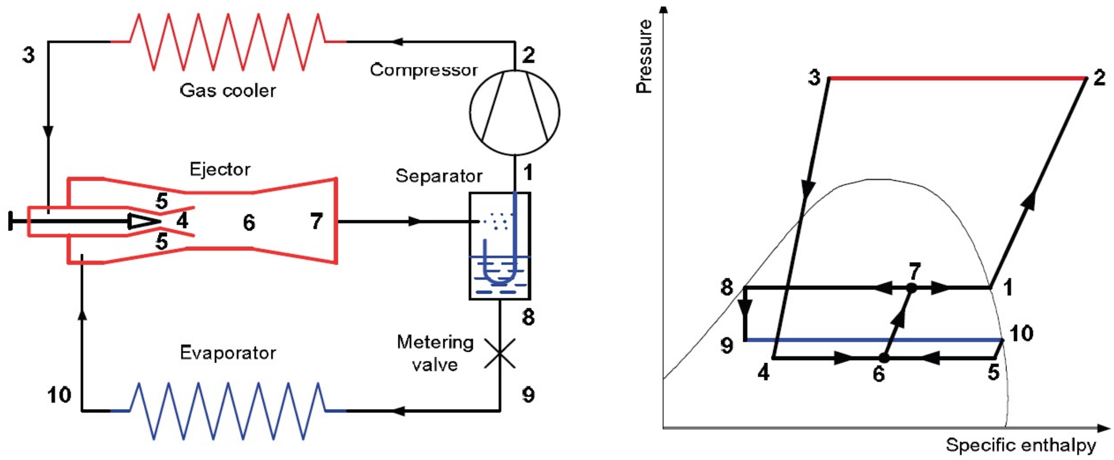

Figure 9.3 – PI diagram and (log)p-h diagram, transcritical CO_2 cycle with ejector expansion device [39]

from the evaporator (state 10) represents the secondary flow going inside the ejector.
The system reduces considerably the expansion losses due to the fact that the expansion through the valve is now across a much smaller pressure difference. In this way the irreversibility in the system are downsized.

9.3 System analysis

The EERC is operating under the common conditions defined at the beginning of part I of the project. Consequently, the system is running transcritical and the optimal gas cooler pressure must be determined, for the given cycle and working conditions. Figure 9.4b) presents the inverted U-shape for the gas cooler pressure. During off-design operation, the ejector actively controls the discharge pressure to achieve the highest possible performances. As reported by Sarkar, *"the gas cooler pressure can be controlled by changing the throat area of the nozzle since the flow rate is proportional to throat area of the nozzle"* [29].
It was discovered that even an optimal mixing pressure can be determined. Figure 9.4a) presents the COP of the system as a function of the mixing pressure. The inverted U-shape of the COP's graph indicates the existence of an optimal value for p_{mix}. The optimization of the system was performed determining the gas cooler and mixing pressure that simultaneously give the maximum system's COP for the given conditions.
Kornhauser noticed the same behaviour in his studied. He explained that *"the selection of optimum mixing pressure becomes a trade-off between mixing loss and nozzle and diffuser losses"* [35]. In other words, optimizing the mixing pressure, the velocity of the two streams (motive and secondary) are regulated in order to minimize the irreversibility.
The performances of the system are not that much affected by the isentropic efficiencies of nozzles and diffuser. Figure 9.5 was built considering different values for the ejector's isentropic

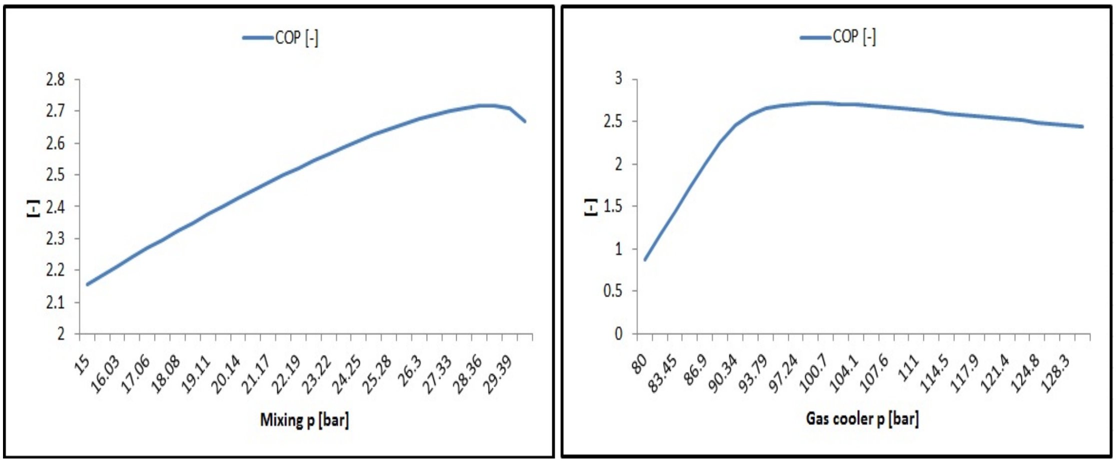

Figure 9.4 – a) COP as a function of the mixing pressure, b) COP as a function of the gas cooler pressure, transcritical CO_2 cycle with ejector expansion device

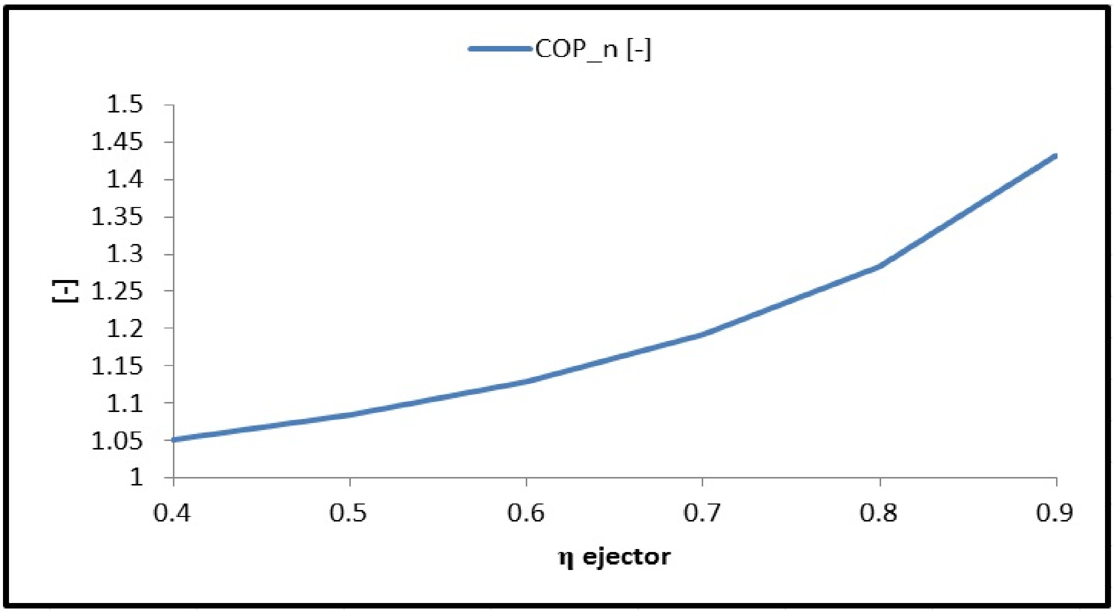

Figure 9.5 – Normalized COP vs. isentropic efficiency of motive nozzle, suction nozzle and diffuser of the ejector, transcritical CO_2 cycle with ejector expansion device

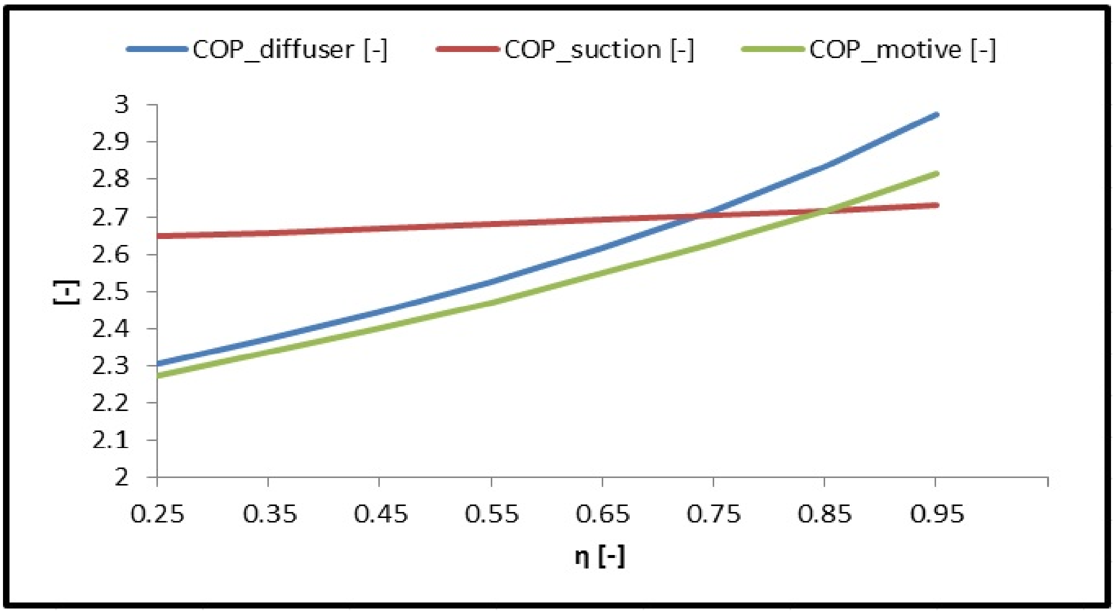

Figure 9.6 – COP vs. ejector isentropic efficiencies, transcritical CO_2 cycle with ejector expansion device

efficiencies, assuming $\eta_m = \eta_s = \eta_d$. COP_n is the normalized COP: the COP of the ejector cycle divided by the one of the 1-stage reference cycle. In this way, it is possible to analyse the impact of these efficiencies respect to the performances of the reference cycle. It is visible from the figure that, even for really low values of efficiency, such as 0.4, the performances of the ejector cycle are higher than the one of the reference cycle (for the given operating conditions).

The impact of each single ejector efficiency can be observed from figure 9.6. In the figure, each COP line was created changing, at once, only one of the isentropic efficiency inside the ejector model. For instance, the curve $COP_{suction}$ was built keeping constant η_m and η_d to their reference values and changing only η_s. In the same way, COP_{motive} was built changing only η_m and $COP_{diffuser}$ was created changing only η_d. As visible from figure 9.6, the diffuser isentropic efficiency has the greatest impact on the system's COP, followed by the motive nozzle efficiency. The suction nozzle efficiency does not affect that much the system's COP, the red line is almost flat.

If the optimal mixing pressure is only slightly affected by changes in the ejector's efficiencies, this is not true for the gas cooler optimal pressure. An appreciable reduction of the mixing pressure is faced when improving the efficiency of diffuser and/or motive nozzle. Due to the small pressure drop realized, the suction nozzle does not affect considerably the optimal gas cooler pressure.

As for the already studied cycles, increasing evaporation pressure and increasing isentropic efficiency of the compressor brings higher system's COP. Changing η_{is} of the compressors the optimal gas cooler and mixing pressure do not vary. On the other hand, higher evaporation

pressures lower down the optimal gas cooler pressure. Interesting was the comparison of the performances of the ejector cycle with the reference one for different evaporator pressures. The lower is p_{evap} the greater improvements are provided by the former respect the 1-stage cycle. This is due to the increasing pressure difference $p_{gc} - p_{evap}$ that increases the pressure recovery of the ejector.
The EERC seems to be a promising solution from the performances point of view. Respect the 1-stage reference cycle, the COP increases while compressor's displacement and pressure ratio decreases. The COP improvements are particularly important from the environmental point of view. The higher is the COP, the lower are the consumptions (electricity) to provide the same cooling effect. From a slightly different perspective, COP improvements reduces the indirect greenhouse gasses emissions associated to the electricity production.
If the COP based analysis represents only a first-law investigation, the beneficial effects of additional ejector are confirmed even by second-law analysis. This investigation can be performed on exergy base and reflects the behaviour of each single component. In fact, the COP analysis allows only for an overview of the overall system. Interesting are the results obtained by Fangtian et al., *"on average, the exergy loss of the ejector cycle reduces about 23%"* [44]. Regardless the value of the exergy destruction reduction, all the literature agrees on the increase exergetic efficiency of the ejector cycle respect the reference one. Moreover, Fangtian et al. observed that, with increasing ambient temperature, the exergy loss of the ejector cycle increases slowly than the one of the 1-stage reference cycle. The same behaviour was found for the system's COP by Liu et al.: *"the enhancement of COP becomes more significant as the outdoor temperature increases"* [40]. This result is promising for the investigation proposed in this project. The ejector cycle seems to be one solution to improve the CO_2 transcritical cycles performances when applied in hot climates. Similar observations are reported also in this report when talking about the yearly behaviour of the system (part III). An interesting second-law overview of the components behaviour is offered by Ahammed et al. [45]. Figure 9.7 presents the exergy destruction (irreversibility) for each component in an ejector cycle (RCE) and in the 1-stage reference cycle (CRC). *"It may be noted that total exergy destruction in the entire ejector (nozzles, mixing and diffuser) is around half of that in an expansion valve of the conventional cycle"* [45].

9.4 System performances

The results coming from the computational simulations are presented in table 9.1. The optimal gas cooler and mixing pressures are shown, along with the resulting maximum COP. Total compressor work and maximum temperature in the cycle are also presented.
Figure 9.8 offers a comparison between the performances of the EERC and the 1-stage reference cycle. The gas cooler pressure, respect the reference cycle, is reduce of 4%, while the compressor work required is lower down of about 20%. The resulting COP increase respect the 1-stage reference cycle is around 26%, for the given working conditions. This result is

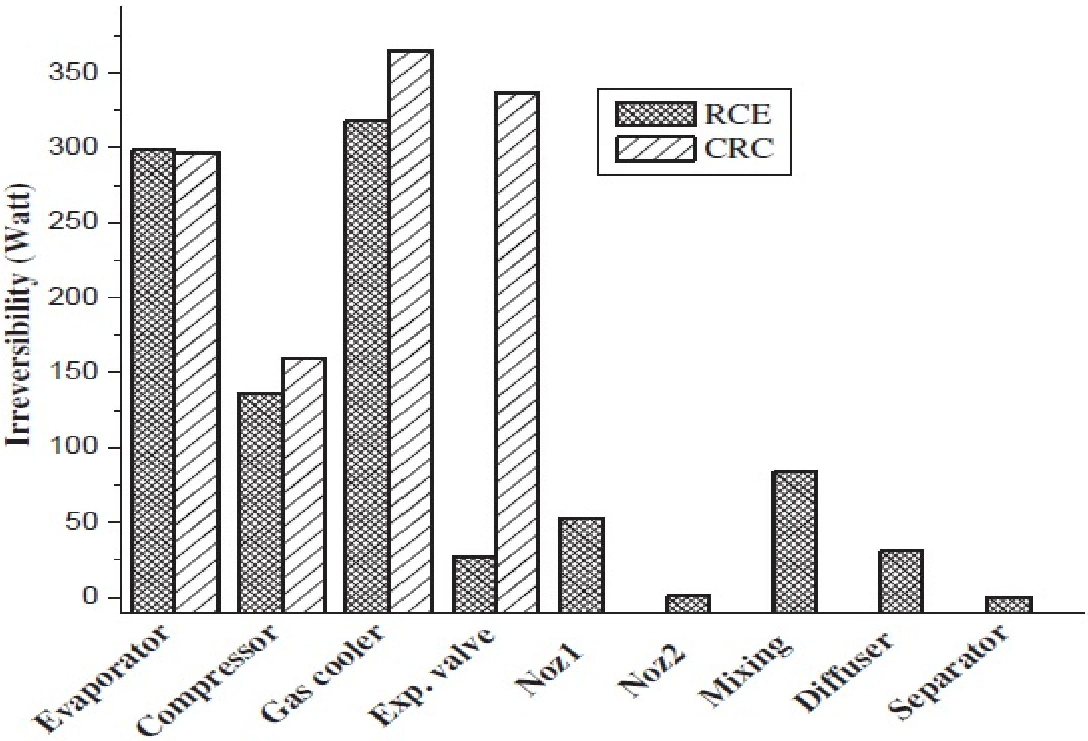

Figure 9.7 – Exergy destruction for each component of the ejector cycle (RCE) and the 1-stage reference one (CRC) [45]

COP	$p_{gc,opt}$	$p_{mix,opt}$	T_{max}	W_{compr}
[-]	[bar]	[bar]	[°C]	[kW]
2.717	99.43	28.66	79.95	22.08

Table 9.1 – System performances for CO_2 transcritical cycle with ejector expansion device at design conditions

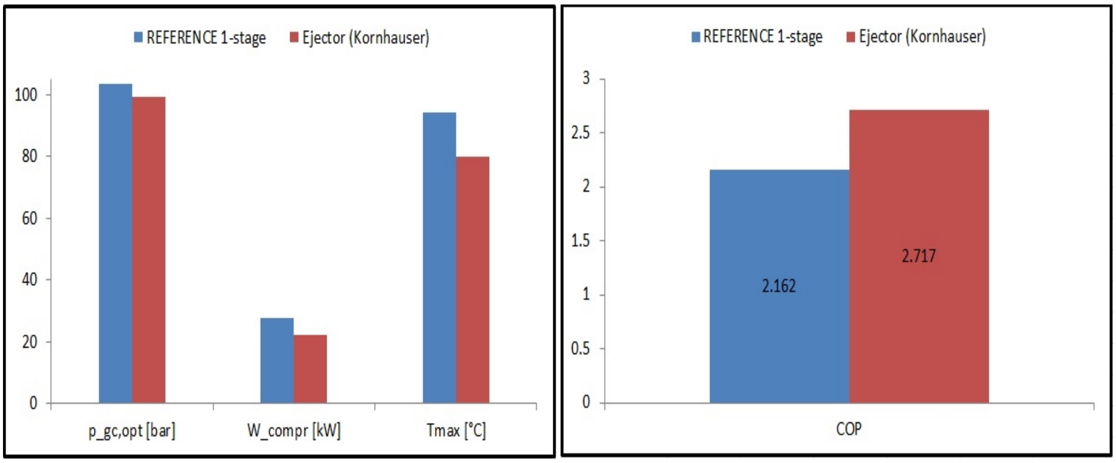

Figure 9.8 – Performances comparison: ejector expansion cycle vs. 1-stage reference cycle

aligned with some of the literature found about ejector cycles (such as [44], [45], [46], [33]). Anyway, even if all the literature agrees on the beneficial effect of the ejector expansion device, it must be said that the improvements reported in the literature are quite different. The entity of the performances increase is strongly related to the considered application and the ejector modelling (for computational investigation) and/or design (for experimental analysis). Great attention must be paid to the inner geometry of this device to achieve the hoped improvements in real-life systems.

10 Comparison of the studied cycles, design conditions

To conclude the first part of the project, a comparison of the studied solutions is presented. Mechanical subcooling cycles, PCE cycle and EERC are here compared with 1-stage CO_2 vapour-compression reference cycle from a different point of view. The choice was not to base this "cycles' contest" on the absolute value coming from the computational simulations. For sake of simplicity and for better understanding, all the values are here presented as normalized. All the parameters are divided by the corresponding values for the reference cycle. In this way, if the value of one parameter is higher than 1, it means that the absolute value of that parameter for that specific cycle is higher than the corresponding one for the 1-stage reference.
Figure 10.1 presents the normalized optimal gas cooler pressure for the different cycles. For the given operating conditions, the higher optimal gas cooler pressure is the one of the reference 1-stage cycle. The lowest optimal gas cooler pressure is faced in the dedicated mechanical subcooling cycle. This is actually not visible in the graph because the difference from integrated mechanical subcoolig and PCE cycle is very small. Anyway, the MS cycle and the PCE one generates a considerable and beneficial reduction of the optimal gas cooler pressure. The optimal gas cooler pressure in the ejector cycle is lower than the reference one, but really close.
Figure 10.2 depicts the maximum temperature in the cycle for the different system configurations. The higher temperature is faced in the 1-stage reference cycle. A great reduction of this temperature is obtained in all the cycles but the biggest change was found in the ejector cycle.
Figure 10.3 presents the different consumptions for the investigated cycles. The system with the highest consumption is certainly the 1-stage one. On the other hand, the other cycles generate a considerable reduction of the compressor work. The reduction is almost of the same magnitude but the ejector system has again the best performances.
Finally, figure 10.4 illustrates the normalized COP for the different cycles. As a consequence of what already said about the systems' consumptions, the highest COP is faced in the 1-stage reference cycle. The other cycles ensure a COP improvement quite similar but it is again the ejector cycle to perform better.
From this comparison emerges that all the four considered solutions give the expected enhancement of the system performances, if compared to the 1-stage vapour-compression system. Anyway, the ejector cycle generates slightly lower consumptions and higher COP. For

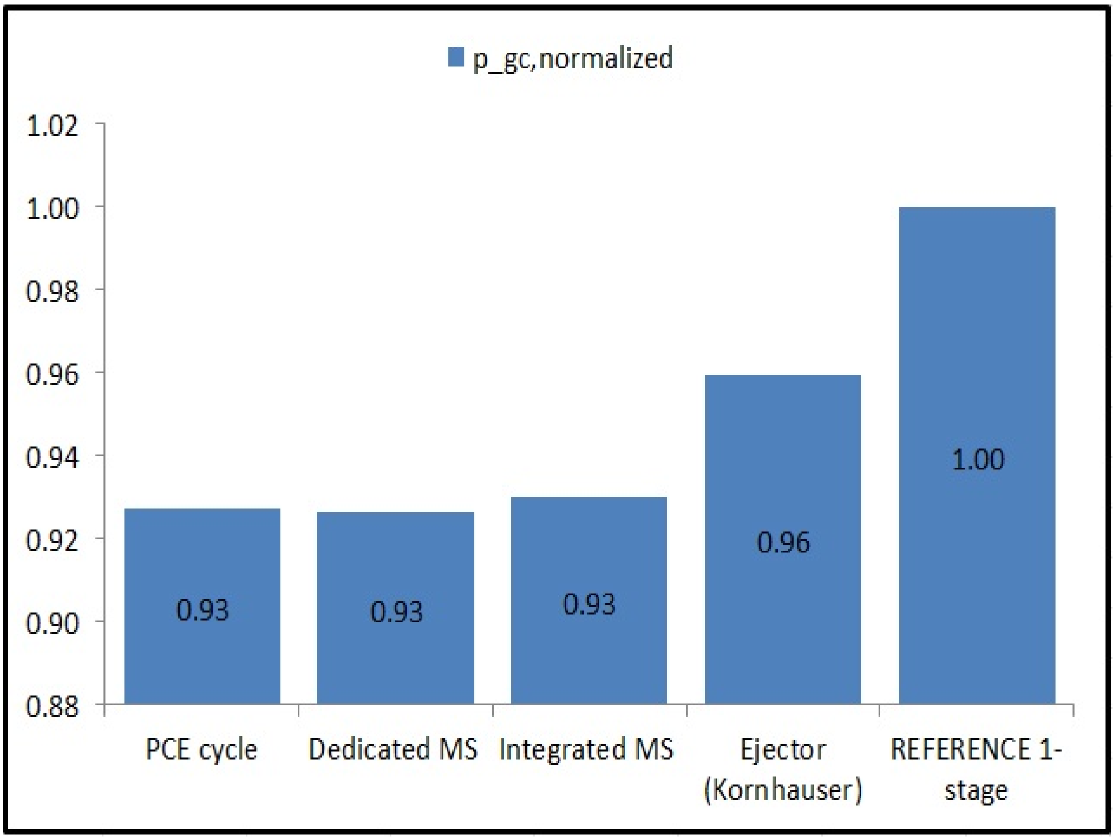

Figure 10.1 – Normalized optimal gas cooler pressure for the different cycles

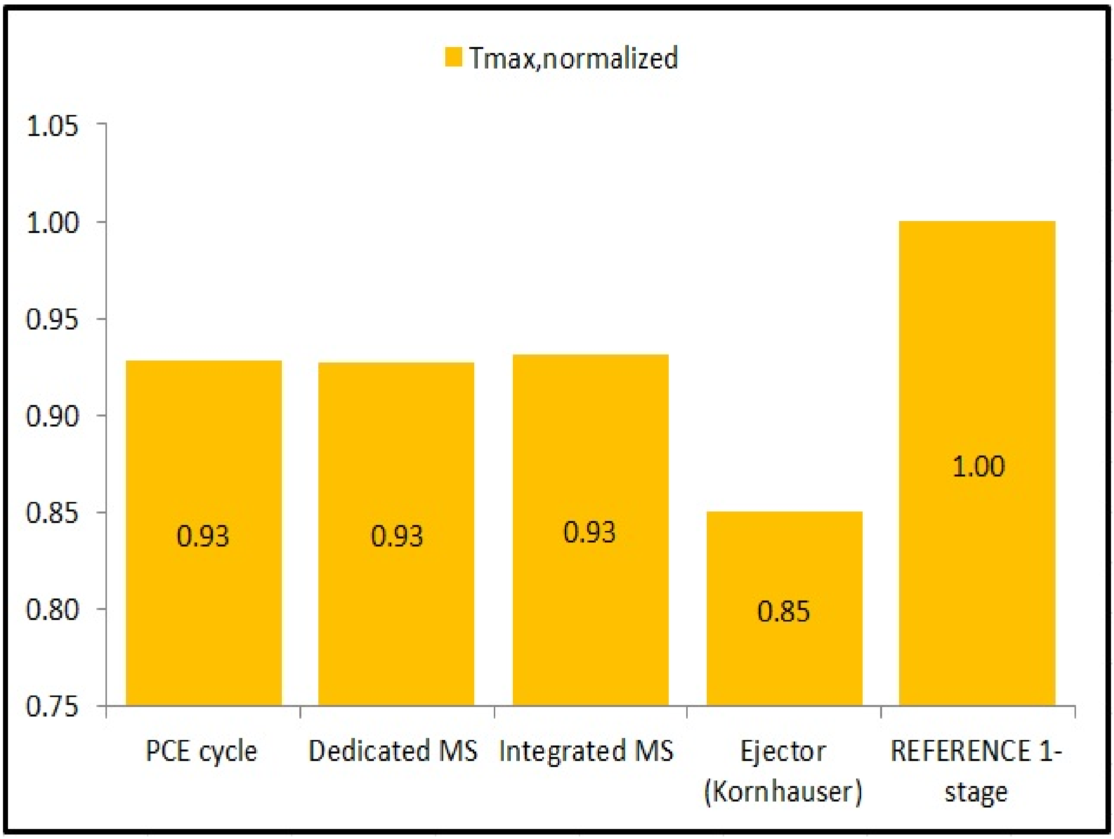

Figure 10.2 – Normalized maximum temperature for the different cycles

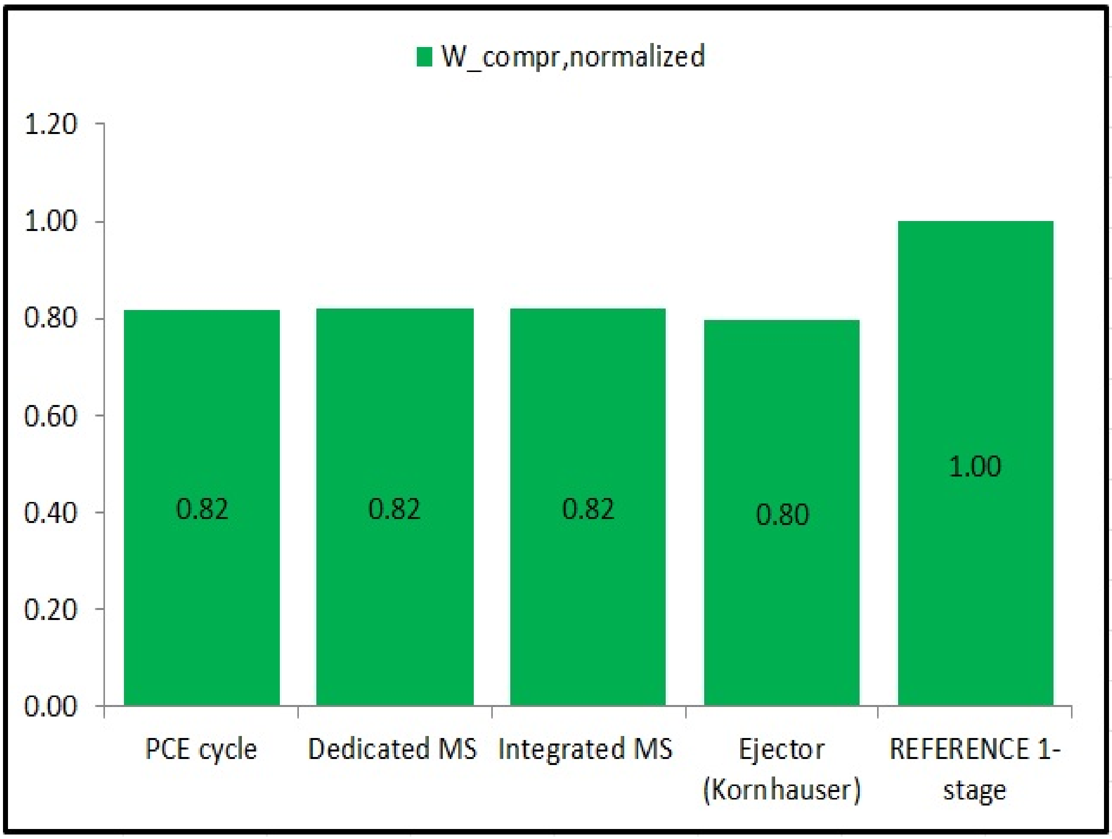

Figure 10.3 – Normalized compressor work for the different cycles

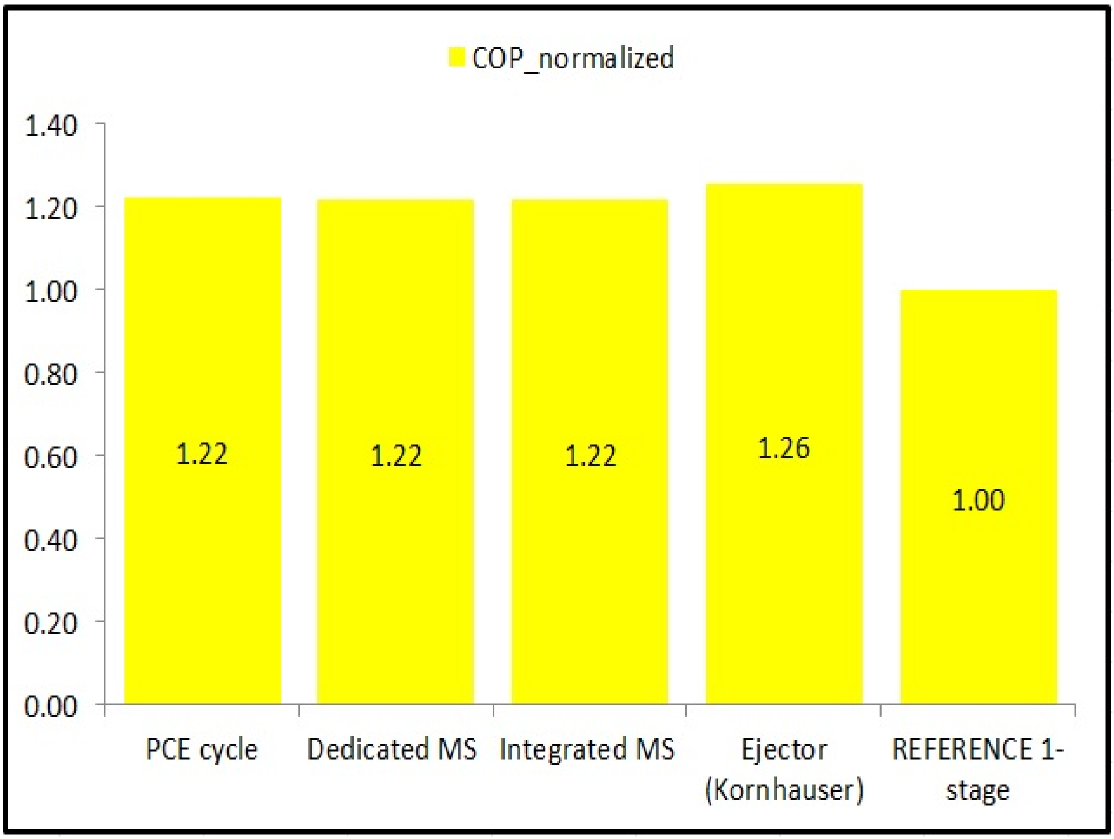

Figure 10.4 – Normalized COP for the different cycles

the given operating conditions, it seems the most promising solution between the illustrated one, in particular considering that quite low ejector efficiencies were assumed.

Part II

Design operation of further improved cycles

11 Further improvements of the studied cycles

In the first part of the project, four different solutions (mechanical subcooling, PCE and use of ejector) were presented, with the purpose to enhance the performances of CO_2 vapour-compression refrigeration cycles. The results were already quite satisfactory due to the considerable performance improvements achieved. Anyway, further improvements of the four studied cycles are possible and some of these solutions were already explored in the literature. Even different CO_2 cycles boost (from the already studied four) are here mentioned. The literature presents plenty of different solutions to improve the CO_2 transcritical cycle, moreover, also the author used a bit his imagination in this part of the project.
The study of the cited four solutions suggested the idea to combine them together, where possible. This, theoretically, should increase even more the performances of the system. What was unknown at the time of the investigation was if this increased complexity of the cycles was worthy or not. For instance, applying mechanical subcooling and using an ejector together in a vapour-compression CO_2 can give a big or small improvement to the system, if compared to EERC or mechanical subcooling "traditional" cycles. Sometime, the additional cycle's complexity does not give the expected magnitude of improvement. Sometime, the enhancement of performances are even higher than expected.
The second part of the report is about these further improved cycles and their performances enhancements. As said, not only the already studied devices are considered. Also some new alternative solutions are presented and, hopefully, they will give even wider possibility of choice to the future CO_2 cycle's designers. In this part of the project, each cycle is described and briefly investigated to finally compare it with the cycles studied in the first part of the project. To perform a "fair" comparison, the same operating conditions applied in part I of the project were used for the following cycles.

12 Further improvements of the CO_2 transcritical cycle with PCE, design conditions

This chapter is about possible further improvements of PCE cycles operating with CO_2. The investigation is still focused on transcritical operation due to the unchanged boundary conditions.

PCE cycles offer the interesting possibility to be combined with mechanical subcooling, both integrated and dedicated. Moreover, the effect of two improved mechanical subcooling loops applied to the PCE cycle are studied.

12.1 PCE cycle with dedicated mechanical subcooling

The idea is to apply dedicated mechanical subcooling to the already studied CO_2 transcritical PCE cycle. The additional heat exchanger (subcooler) and the subcooler loop ensure a further cooling of the refrigerant after the gas cooler, improving the systems performances.

The optimization of the system is a 4-dimensional process. It must be simultaneously optimized the two gas coolers pressures, the subcooler pressure and the separator one. The min/max function provided by the software EES was used. The main results from this optimization are presented in figure 12.1. The optimal gas cooler pressure is reduced around 2%, while the COP faces an improvement of more than 6%.

12.2 PCE cycle with integrated mechanical subcooling

Similarly to what it was done before, the integrated mechanical subcooling is applied to the transcritical CO_2 vapour-compression cycle with PCE. The subcooler loop provides subcooling of the refrigerant at the gas cooler outlet, further improving the performances of the system.

The optimization of the system is now a 3-dimensional process. Gas cooler, subcooler and separator pressures must be simultaneously optimized to reach the maximum system's COP for the given operating conditions. The main results from this optimization are presented in figure 12.2. The figure offers also a comparison between the improved cycle and the "traditional" PCE one. The improvements achieved with the additional integrated MS loop are similar to the one obtained using dedicated MS. The optimal gas cooler is reduced around 1%, the COP is improved of 6%.

12.3 PCE cycle with ejector-dedicated mechanical subcooling

This section tries to study the impact on the PCE cycle's performances of an enhanced dedicated mechanical subcooling system. The dedicated subcooler loop is improved by the use of an ejector used to recover the expansion work. In this way the subcooler loop is actually an EERC.

The optimization of the system requires the simultaneous research of the optimal gas coolers pressures, separator pressure, subcooler pressure and ejector's mixing one. Thus, the optimization requires a 5-dimensional function.

A small investigation was performed on the influence of the ejector efficiencies on the system performances. The reference case, using $\eta_m = \eta_s = 0.8$ and $\eta_d = 0.75$, was compared to a situation in which $\eta_m = \eta_s = 0.9$ and $\eta_d = 0.85$. Figure 12.3 presents the normalized COP and normalized compressor work obtained dividing the resulting values of the second case with the corresponding values of the reference case. In this way it is easier to visualize the changes that the second case offers respect to the reference. Figure 12.3 shows that the ejector's efficiencies do not have a great impact on the system performances. A considerable change of them leaves almost unchanged COP and compressor work of the system.

A comparison of the performances between the improved cycle and the PCE one is presented in figure 12.4. The optimal gas cooler pressure is reduced around 3% respect the "traditional" PCE cycle, while the COP faces an important improvement of 9%.

12.4 PCE cycle with PCE-dedicated mechanical subcooling

An enhanced dedicated mechanical subcooling loop is applied to the CO_2 transcritical PCE cycle. The parallel compression economization solution is used also in the subcooler cycle in order to improve the overall system's performances. In this way, even the subcooler cycle is actually a PCE cycle.

The system can be optimized finding the best gas coolers, separators and subcooler pressures. Once again, the optimization function requires as input 5 parameters.

Figure 12.5 presents the comparison between the improved PCE cycle with PCE-dedicated mechanical subcooling and the "traditional" PCE cycle. The optimal gas cooler pressure is reduced only around 1% respect the PCE system. The same small improvement is faced by the maximum temperature in the cycle. A more interesting improvement is faced in terms of compressor work and COP. The COP of the PCE cycle with dedicated mechanical subcooling is

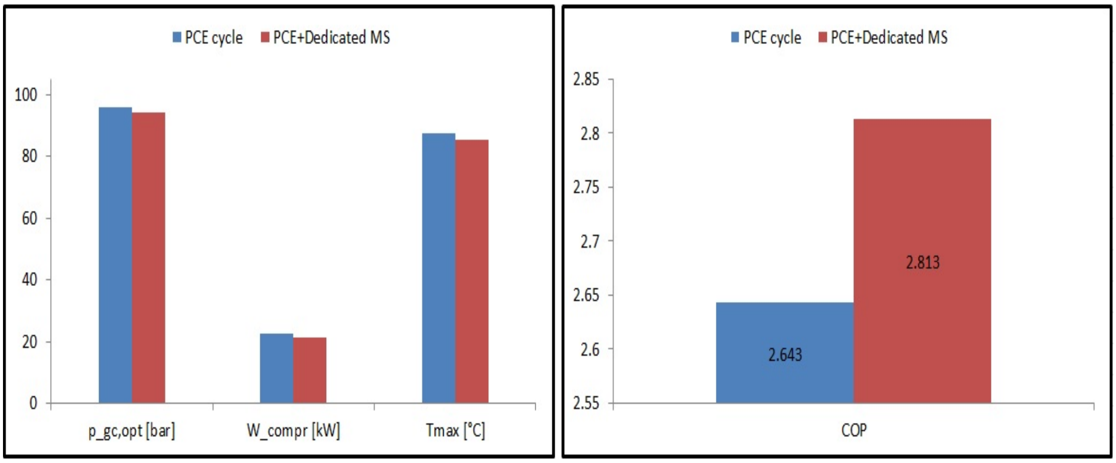

Figure 12.1 – Performance comparison: PCE cycle with dedicated mechanical subcoolig vs. PCE cycle

around 7% higher than the "traditional" PCE system.

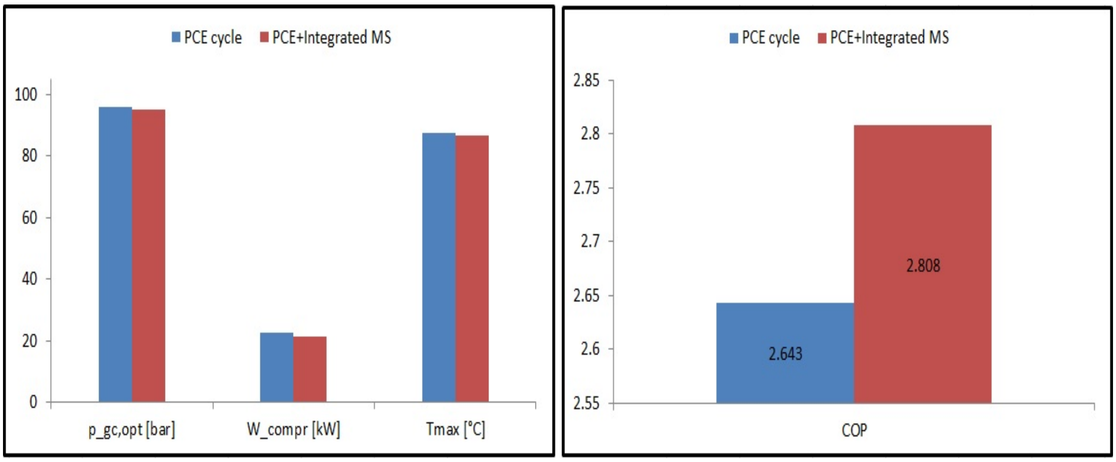

Figure 12.2 – Performance comparison: PCE cycle with integrated mechanical subcoolig vs. PCE cycle

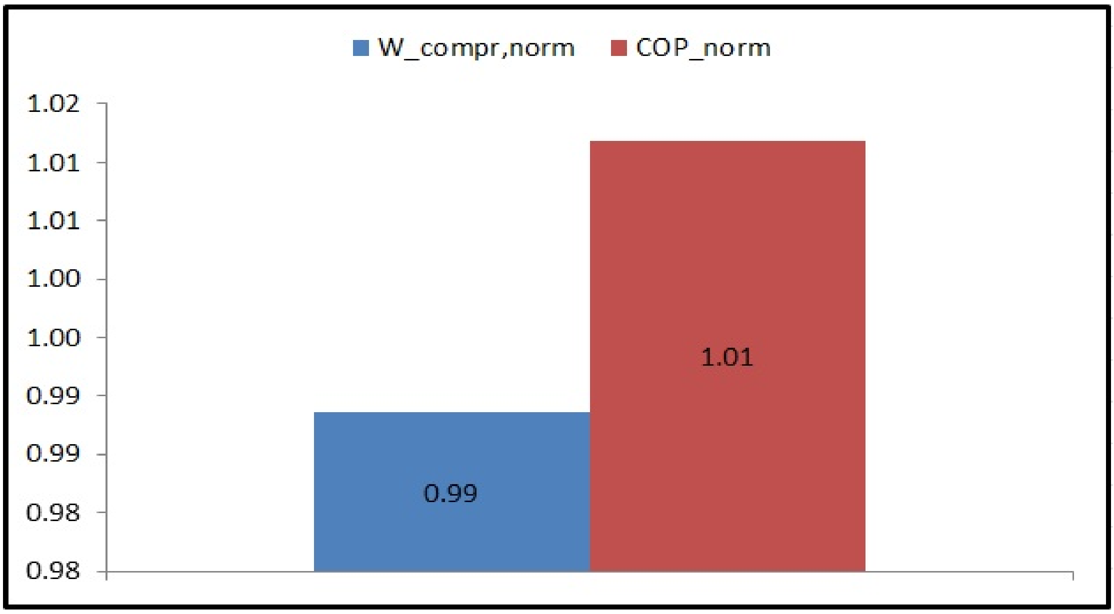

Figure 12.3 – Normalized COP and compressor work: comparison of the systems performances for two different sets of ejector efficiencies, PCE cycle with ejector-dedicated mechanical subcooling

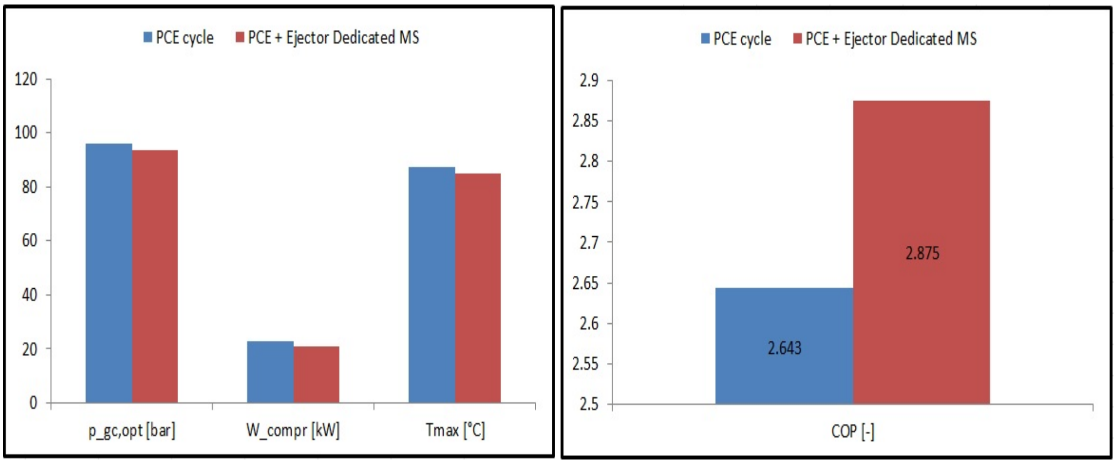

Figure 12.4 – Performance comparison: PCE cycle with ejector-dedicated mechanical subcoolig vs. PCE cycle

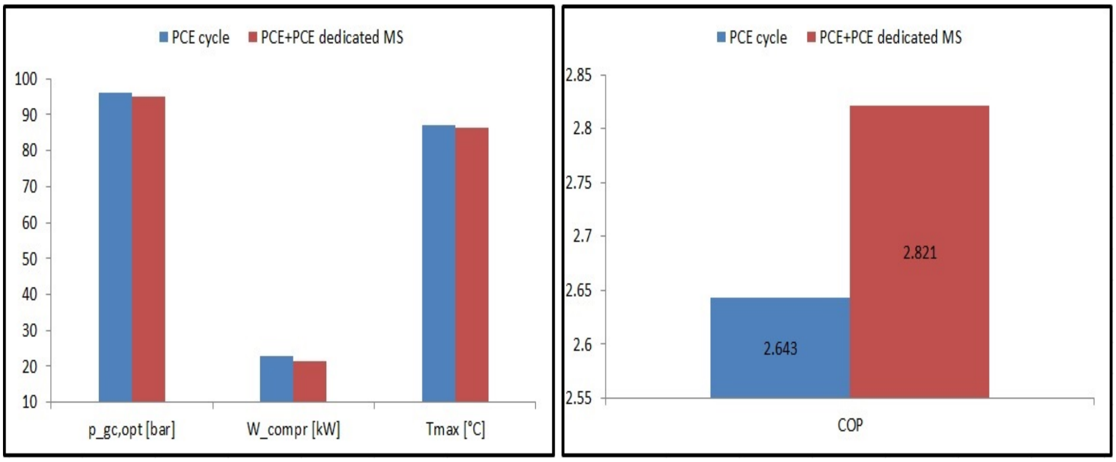

Figure 12.5 – Performance comparison: PCE cycle with PCE-dedicated mechanical subcoolig vs. PCE cycle

13 Comparison of the further improved CO_2 transcritical cycles with PCE, design conditions

In this chapter, a comparison between the improved CO_2 transcritical vapour-compression cycles using PCE is presented. All the observations presented in this chapter refers to the chosen operating conditions of the systems. Only normalized values are presented in the following graphs. The CO_2 vapour-compression 1-stage cycle is taken as reference for the normalization of the results.

Figure 13.1 represents the normalized optimal gas cooler pressure for the different PCE cycles investigated. In all the improved cycles, the reduction of the gas cooler pressure respect the reference cycle is almost of the same magnitude. Anyway the minimum optimal gas cooler pressure was found in the PCE cycle with ejector-dedicated mechanical subcooling.

Figure 13.2 depicts the maximum cycle's temperature for the PCE cycles investigated. The enhanced cycles present similar maximum temperatures with the lowest of these faced by the PCE cycle with ejector-dedicated mechanical subcooling. Anyway, the reductions that this cycle provides in terms of maximum cycle's temperature and optimal gas cooler pressure do not justify the increased system's complexity.

A more appreciable improvements of the performances (respect the "traditional" PCE cycle) can be noticed if looking at the cycle's power requirements. Figure 13.3 presents the compressor's work for the investigated PCE cycles. Of course, the highest compressor work is the one of the reference cycle. The "traditional" PCE cycle already offers a great reduction of the consumptions, but a more interesting achievement is obtained with the new cycles. The lowest consumptions are ensured by the PCE system with ejector-dedicated mechanical subcooling. Anyway, the magnitude of the work's reduction achieved by all the further improved cycles is quite similar.

Finally, the COPs of the cycles are compared for the given operating conditions. Figure 13.4 depicts the normalized system's COPs for the different PCE cycle under consideration. The PCE "traditional" system already offers a good improvement respects the reference cycle. Anyway, even higher performances can be achieved with the improved cycles. Again the best performer is the PCE cycle with ejector-dedicated mechanical subcooling, but only with slightly better performances respect the other improved cycles.

The further improved PCE cycles seem to offer interesting gains, especially from the compres-

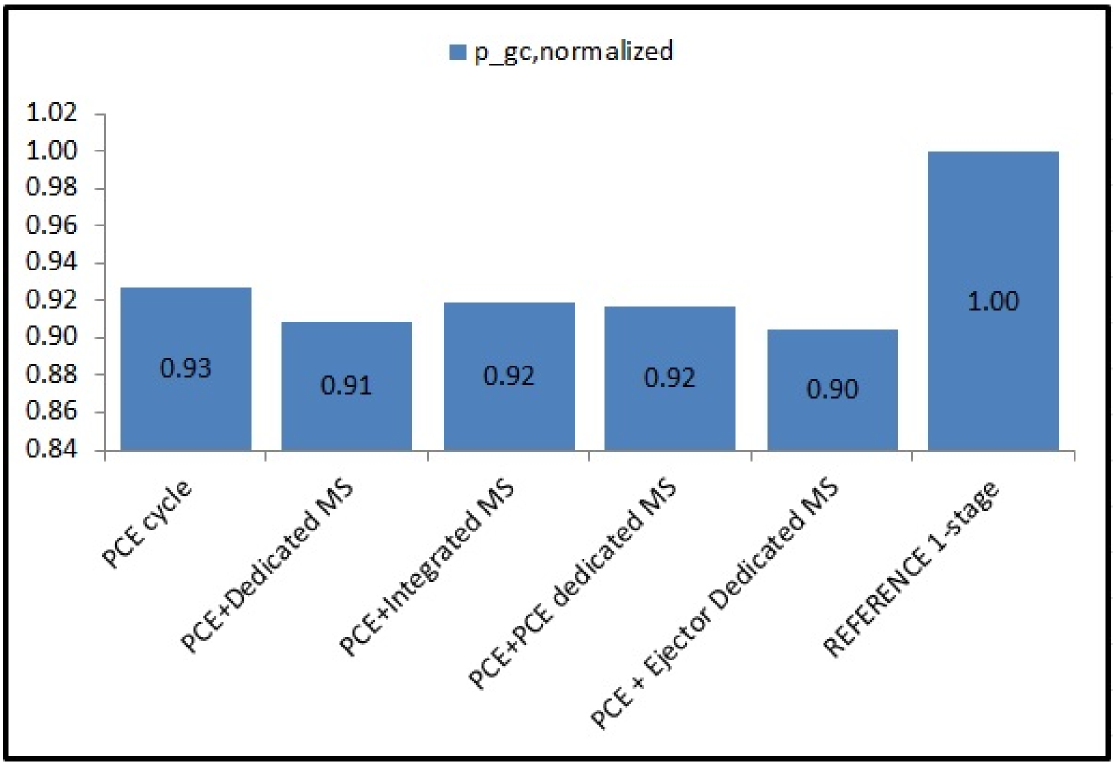

Figure 13.1 – Normalized optimal gas cooler pressure for the different PCE cycles

Figure 13.2 – Normalized maximum cycle temperature for the different PCE cycles

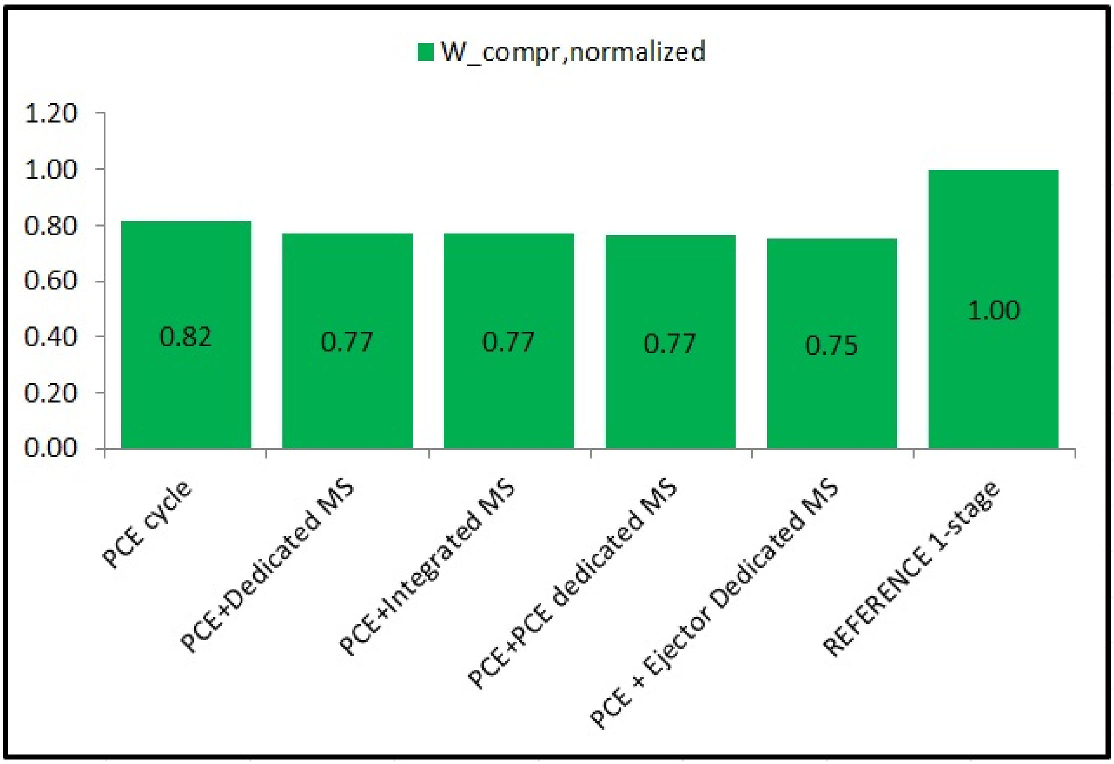

Figure 13.3 – Normalized compressor's work for the different PCE cycles

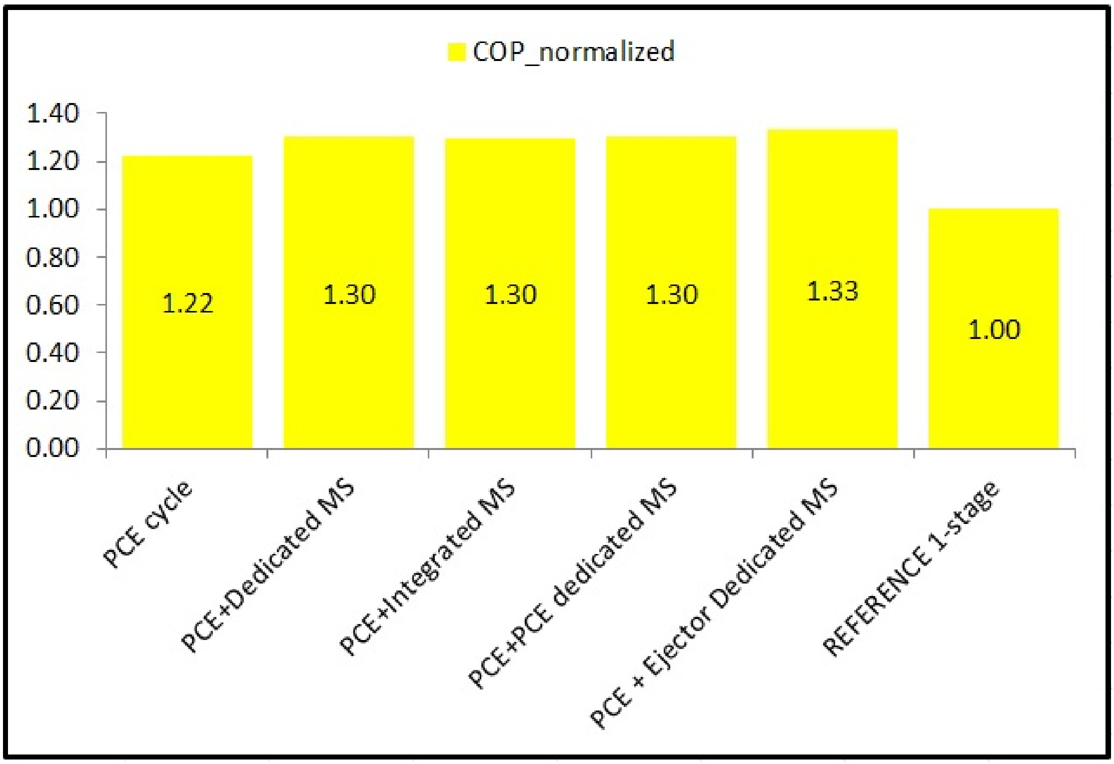

Figure 13.4 – Normalized system's COP for the different PCE cycles

sor's work and COP point of view. Even if the PCE cycle with ejector-dedicated mechanical subcooling seems to be the most effective one, the increased complexity is not justified by the improvements that it brings. The PCE cycle with dedicated mechanical subcooling offers quite similar performances from all the considered points of view, bringing even a more simple cycle's structure.

14 Further improvements of the CO_2 transcritical cycle with dedicated MS, design conditions

The chapter is about possible improvements of the mechanical subcooling loop for CO_2 transcritical vapour-compression systems.
The already studied mechanical subcooling is improved with the help of an ejector in the subcooler loop. The possibility to improve the secondary loop with PCE is discarded due to the non-satisfactory results obtained and presented in the previous chapter (chapter 13). Additionally, two new proposals for the subcooler loop are also here introduced and investigated.

14.1 Ejector-Dedicated mechanical subcooling

The first investigated improvement of the dedicated mechanical subcooling loop is the possibility to use an ejector in the subcooler cycle. As already seen during the study of the improved PCE cycles (chapter 13), it gives an interesting boost to the cycle. Using the experience gained from the investigations proposed in the previous chapter, the use of PCE-dedicated mechanical subcooling was abandoned. It was clear from those analyses that the most efficient of these two solution is the one with the ejector in the subcooler cycle.
An ejector is added to the subcooler cycle while the main cycle is simple 1-stage vapour-compression transcritical CO_2 system. The subcooler loop is now an EERC. In the previous chapter the ejector-dedicated mechanical subcooling loop was applied to a PCE cycle. Here, it is applied to the 1-stage reference cycle. In this way the gains provided by the improved secondary loop can be directly compared to the performances of the "traditional" mechanical subcooling system studied in the first part of this project. In other words, in this way it is possible a direct comparison of the two subcooler loops.
The optimal cycle's configuration was found by optimizing simultaneously the two gas coolers' pressures and the ejector's mixing pressure. Figure 14.3 proposes a comparison of the ejector-dedicated mechanical subcooling cycle and the dedicated mechanical subcooling one. The optimal gas cooler pressure is unchanged when comparing the two systems. The COP faces a very small improvement around 3%.

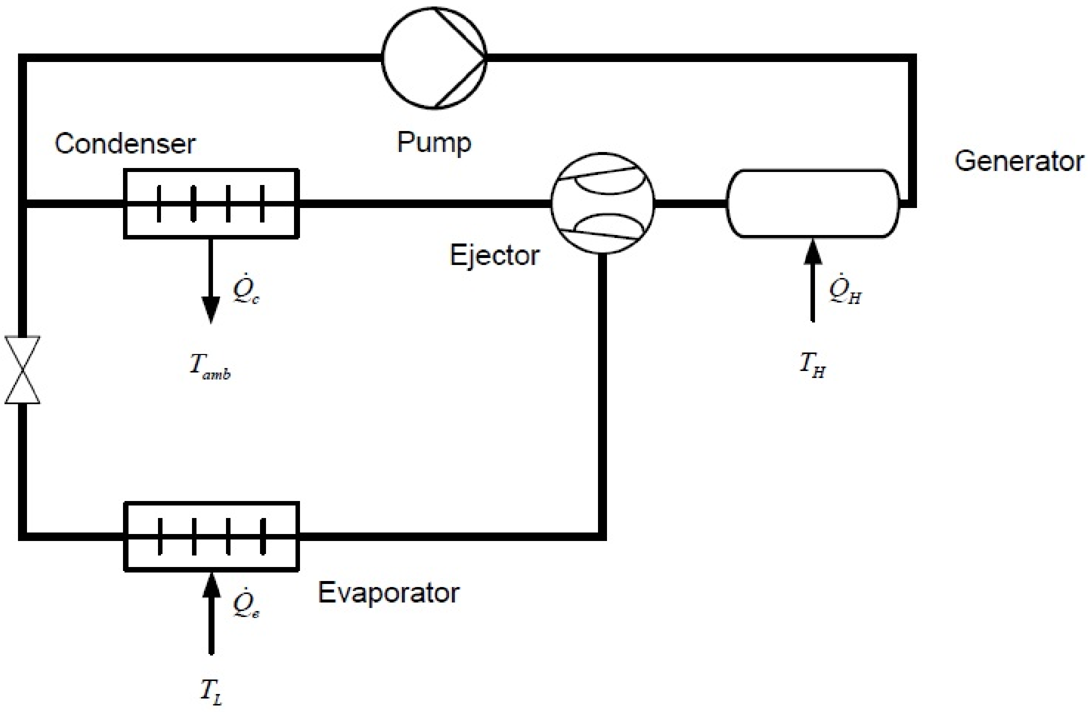

Figure 14.1 – PI diagram of a steam-ejector refrigeration system

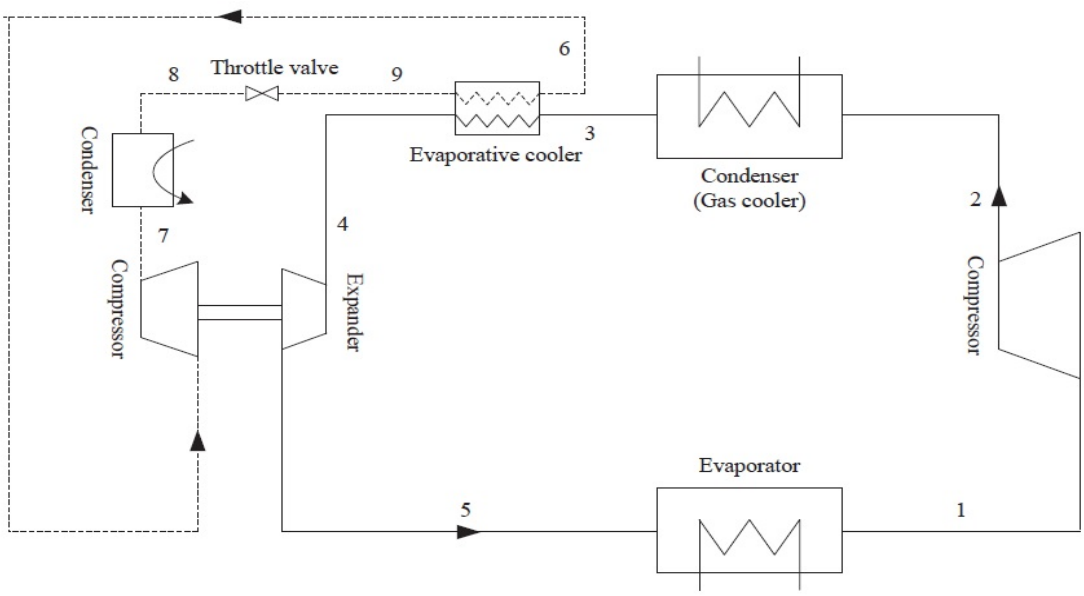

Figure 14.2 – PI diagram of an EPR-dedicated mechanical subcooling system refrigeration system [34]

14.2 Dedicated heat-driven subcooling

A new proposal for the subcooler cycle is here pointed out. The idea comes from another important application of the ejectors in the refrigeration industry: the steam-ejector refrigeration cycles. The PI diagram of one of these systems is presented in figure 14.1. The working fluid is water, another "natural refrigerant" environmentally friendly.
The cycle is composed by condenser, expansion valve and evaporator like the traditional vapour-compression cycles. The compressor is replaced by the combination of pump, generator and ejector. The main difference from the traditional plants is the reduction in mechanical work required. The pump has really small consumptions respect the use of a compressor and the main requirement of the cycle is heat at the generator.
The idea is to use an heat-driven steam ejector cycle as subcooler loop, instead of the traditional vapour-compression subcooler cycle. The main cycle is still a simple 1-stage CO_2 vapour compression cycle with the addition of the subcooler after the gas cooler. The working fluid is not any more only CO_2 in this system, in fact the subcooler loop uses water. Anyway, the choice to study CO_2 cycles was dictated by the necessity to switch to environmentally friendly refrigerants. Water is, as well as carbon dioxide, a low impact refrigerant.
Some assumptions were made when simulating this system. First of all the ejector model is the same used for the enhancement of the CO_2 vapour-compression cycles. The task required to the ejector is different, as well as the working fluid. Anyway, this model was considered good enough to roughly represents the behaviour of a steam-ejector. The ejector's efficiencies were not changed from the previously assumed ones: $\eta_m = \eta_s = 0.8$ and $\eta_d = 0.75$. The heat used inside the generator was assumed waste heat recovered from other application. It was consequently considered "gratis" and not included in the system's COP calculations. The pumping process was assumed isenthalpic and the pumping work was considered negligible. Combining all these assumptions, the subcooling provided to the cycle is for free and the bigger it is the better. This can be observed from figure 14.4: the smaller is the subcooler pressure, the higher is the subcooling provided and the system's COP. Of course, there are technical limits to the minimum pressures in a system. For this reason, a subcooler pressure of 0.04 bar was assumed. In figure 14.4 the normalized COP is presented as a function of the subcooler pressure. It refers to the cycle's COP divided by the COP of a "traditional" dedicated mechanical subcooling system operating at the same conditions. In this way it is possible to see that, for the given conditions, a subcooler pressure lower that 0.042 bar always guarantees that the dedicated heat-driven subcooling cycle performs better that the traditional one. The generator pressure was set to 50 bar. Again, at the subcooler the temperature difference was fixed to 5°C.
The assumptions made about pump power and generator heat can be considered quite realistic. In steam-ejector cycles the pump power is usually neglected in the calculations. Moreover, the possibility to recover heat from other industrial processes is quite realistic too. For this investigation, it was assumed available a water mass flow of 0.15 kg/s at 100°C.
For the given operating conditions and assumptions, the ejector mixing pressure and the gas cooler pressure were optimized to achieve the maximum system's COP. Figure 14.5 presents

a comparison of the dedicated heat-driven subcooling cycle and the dedicated mechanical subcooling one. The optimal gas cooler pressure is reduced of more than 10%, while the COP faces an improvement of 6%.

14.3 Expansion power recovery dedicated mechanical subcooling

The last section of the chapter is about another alternative solution to improve the performances of the subcooler loop. The idea came after the reading of a recent work of She et al. [34].
To improve the performances of CO_2 transcritical cycles, the reduction of the expansion losses in the throttling valve is always a good starting point. This is the purpose of the ejector, as already seen, but also a turbine can be used instead of the traditional expansion valve. This provides an improvement of the system's COP but from the economical point of view this solution is rarely feasible, in particular if considering small CO_2 systems [47]. For this reason She et al. proposed the use of a power recovery device, instead the expansion valve, used to power the subcooler cycle: the result is an expansion power recovery (EPR) dedicated mechanical subcooling system. Citing from their work *"in the main refrigeration cycle, the expander output power is employed to drive a compressor in an auxiliary subcooling cycle, and the refrigerant at the outlet of the condenser (or gas cooler) is subcooled by the evaporative cooler"* [34]. The results of their work is very interesting due to the fact that they claim this system to be even a best performer than the conventional expansion recovery system (the expander recovers work that is used in the compressor of the main cycle).
Also the solution proposed by She et al. is surely costly. Anyway, it was chosen to investigate this system because of the really good performances. Moreover, it was considered worthy to present in this project at least one cycle using an expander because it represents another widely discussed alternative in the literature about CO_2 transcritical cycles' improvements.
The PI diagram of the system is presented in figure 14.2. As visible, the compressor of the subcooler cycle and the expander are mechanically coupled. The remaining components of the main and subcooler cycle are exactly the same that are used in a 1-stage vapour-compression cycle with dedicated mechanical subcooling.
The operating conditions assumed are the same as the other cycles, the only additional assumption required was about the isentropic efficiency of the expander. She et al. [34] reported a lot of literature about the isentropic efficiency of the expander, on the base of that a value of 0.7 was assumed.
The impact of the expander efficiency on the system's performances was analysed. Values of turbine's efficiency of 0.3, 0.5, 0.7 and 0.9 were taken into account. Figure 14.6 presents the results of this investigation. The COP is presented as normalized, the reference COP value is the one of the "traditional" mechanical subcooling cycle. The performances of the EPR system increase with the turbine's efficiency and, as reported by She et al., the impact of the turbine's efficiency on the system COP is considerable [34]. Anyway, even for really low

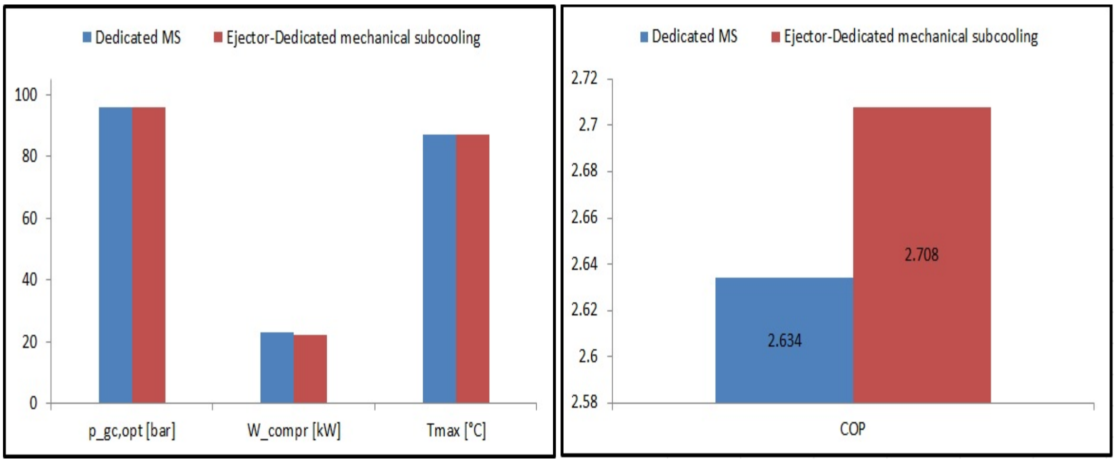

Figure 14.3 – Performance comparison: Ejector-dedicated subcooling cycle vs. dedicated mechanical subcoolinge cycle

expander performances, the system performs better than the mechanical subcooling cycle. With the given assumptions and operating conditions the system was optimized. It was found that an optimal gas cooler and subcooler pressure exists even for this cycle. The optimization process requires three inputs: the two gas coolers pressures and the subcooler one. Figure 14.7 presents the optimization results got from the computational simulations. Comparing this improved cycle with the "traditional" dedicated MS one, the optimal gas cooler pressure is reduced around 2%. Very interesting improvements are faced in terms of compressor work (-17%) and COP (+21%).

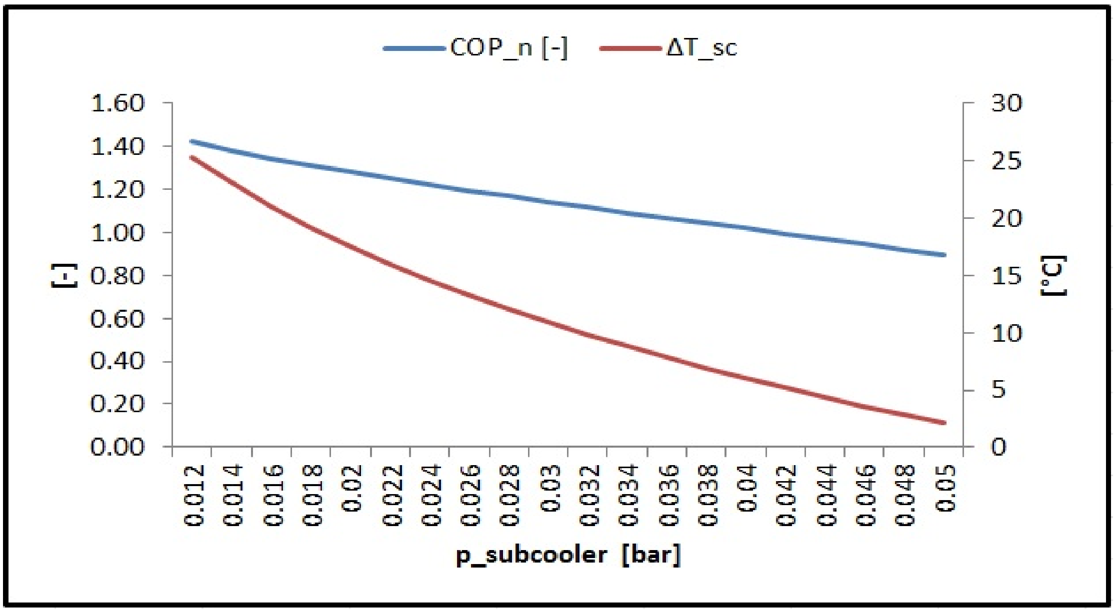

Figure 14.4 – Normalized COP and ΔT_{sc} vs. subcooler pressure, dedicated heat-driven subcooling cycle

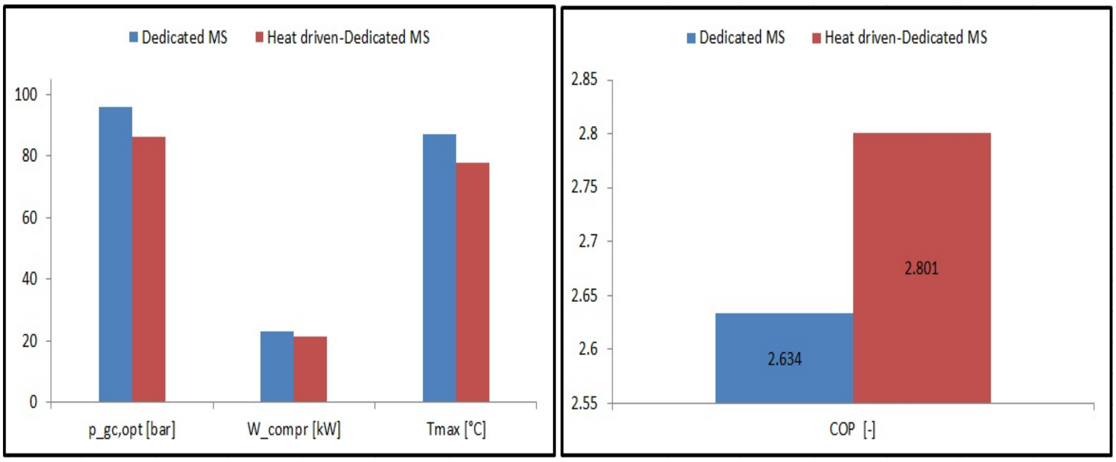

Figure 14.5 – Performance comparison: Dedicated heat-driven subcooling cycle vs. dedicated mechanical subcooling cycle

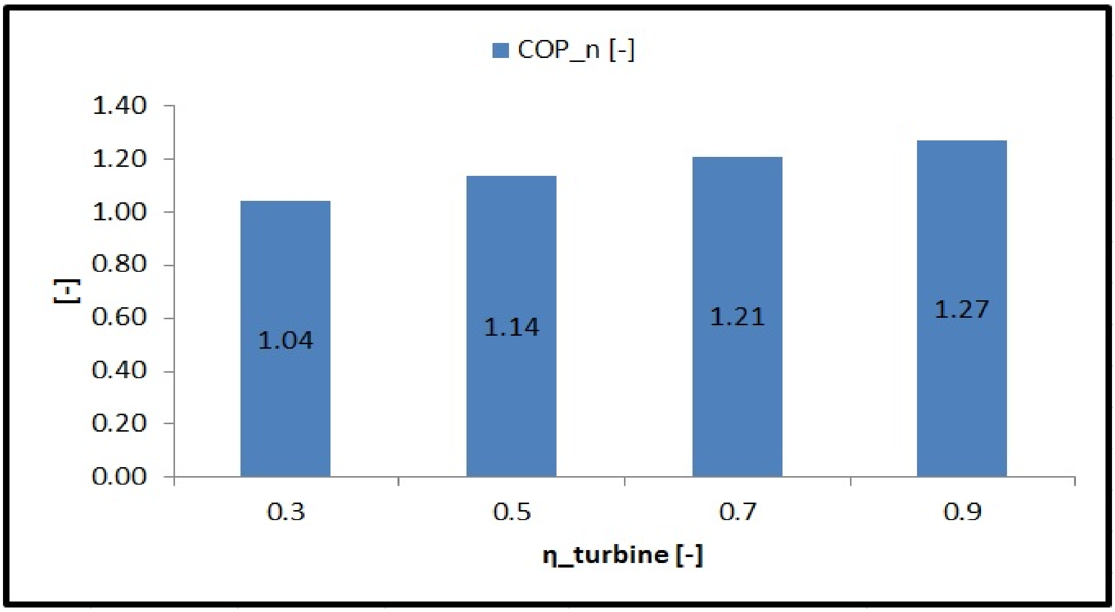

Figure 14.6 – Normalized COP vs. expander's isentropic efficiency, EPR dedicated mechanical subcooling cycle

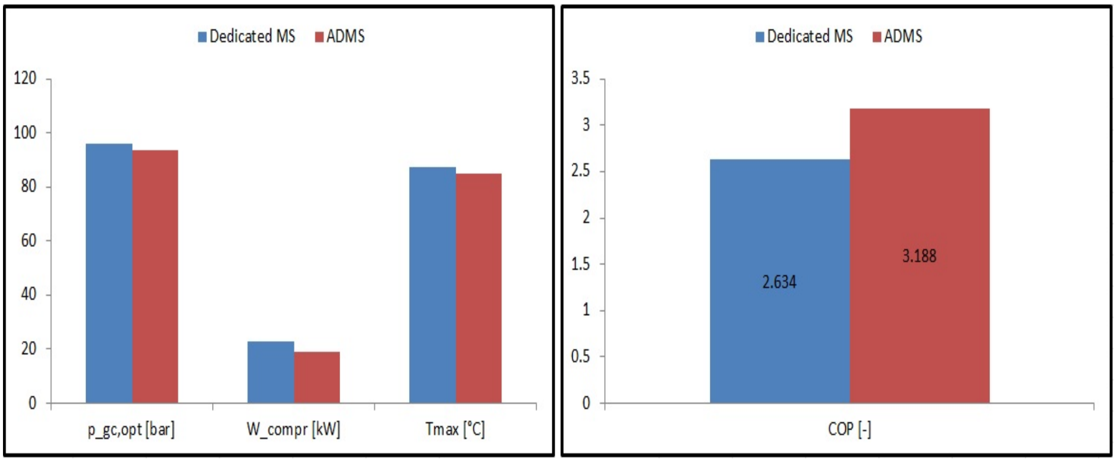

Figure 14.7 – Performance comparison: EPR-dedicated subcooling cycle vs. dedicated mechanical subcooling cycle

15 Comparison of the further improved CO_2 transcritical cycles with dedicated MS, design conditions

In this chapter, the studied different solutions to improve the dedicated mechanical subcooling loop are compared (see chapter 14 for the cycles' descriptions). One solution adopted was the addition of an ejector to the traditional subcooler loop to enhance its performances. Also new ideas for the dedicated subcooling were advanced. The possibility to recover the expansion work of the main cycle and use it to run the compressor of the subcooler loop was taken into consideration with promising results. A very interesting solution regards the possibility to abandon the mechanical subcooling to switch to the heat-driven subcooling. This solution offers the interesting possibility to obtain a "free subcooling" using waste heat recovery. In order to make a fair comparison, all the cycles were compared and optimized under the same operating conditions. All the results are presented as normalized values. As a reference, the CO_2 1-stage vapour-compression cycle was considered.

Figure 15.1 presents the normalized optimal gas cooler pressure for the investigated cycles. Speaking about gas cooler pressure, it is always referred to the one of the main cycle. All the solutions concerning mechanical subcooling systems give the same magnitude of the gas cooler pressure's reduction. The improvements are considerable if compared with the 1-stage reference cycle. Anyway, even higher reductions can be achieved applying the dedicated heat-driven subcooling system to a vapour-compression CO_2 transcritical cycle (at least for the given operating conditions).

Figure 15.2 depicts the normalized maximum cycle's temperature for the studied systems. Again a considerable reduction is allowed applying the mechanical subcooling. Adding to the 1-stage cycle one of the enhanced mechanical subcooling loops, the maximum temperature reduction provided is almost the same. Anyway, an even higher reduction can be achieved applying the dedicated heat-driven subcooling loop.

Even more interesting results are shown by figure 15.3, that depicts the normalized compressor work for the investigated cycles. The compressor work reduction provided applying "traditional" dedicated mechanical subcooling and ejector-dedicated mechanical subcooling is similar. A small further reduction is achieved applying the dedicated heat-driven subcooling, but the most surprising results are given by the EPR-dedicated mechanical subcooling. The addition of this dedicated subcooling loop to the 1-stage transcritical cycle brings a really big

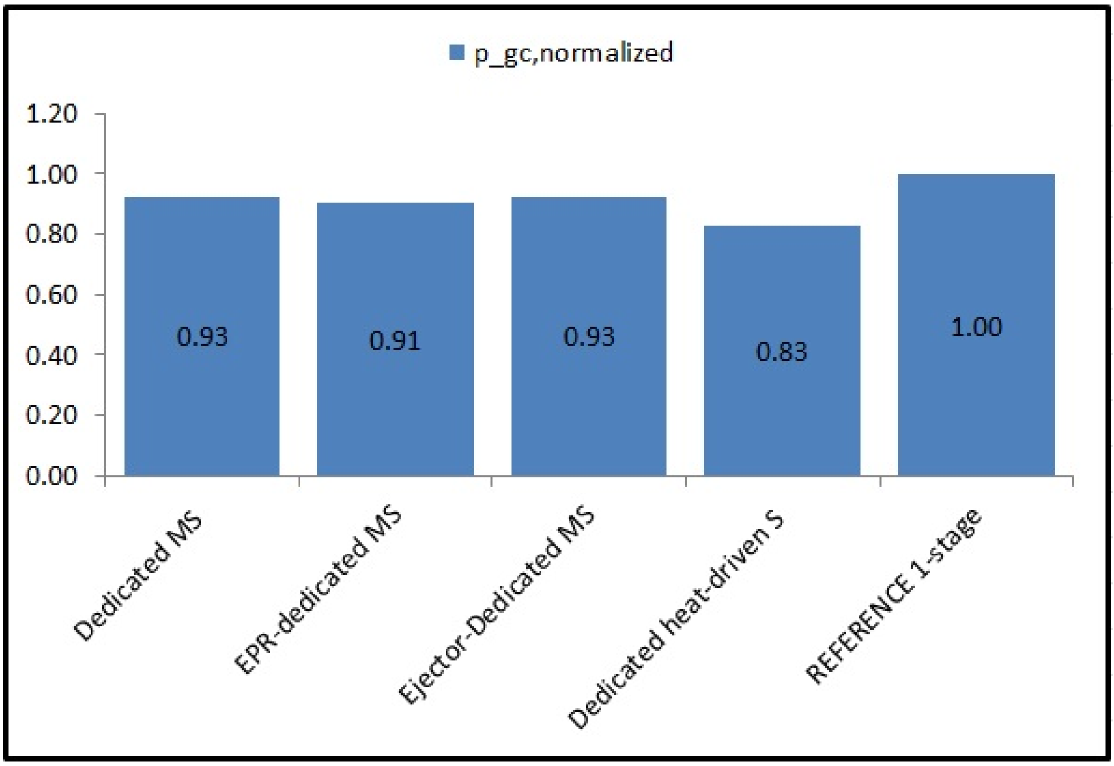

Figure 15.1 – Normalized optimal gas cooler pressure for the different CO_2 transcritical vapour compression cycles with dedicated subcooling system

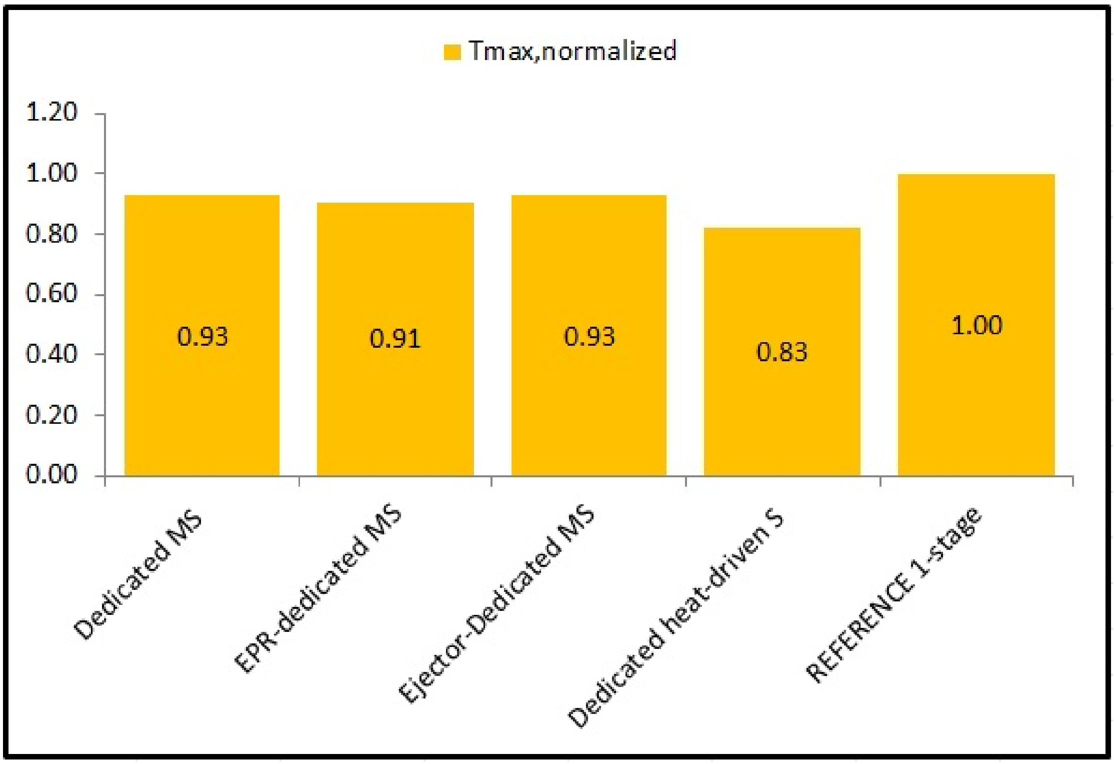

Figure 15.2 – Normalized maximum cycle's temperature for the different CO_2 transcritical vapour compression cycles with dedicated subcooling system

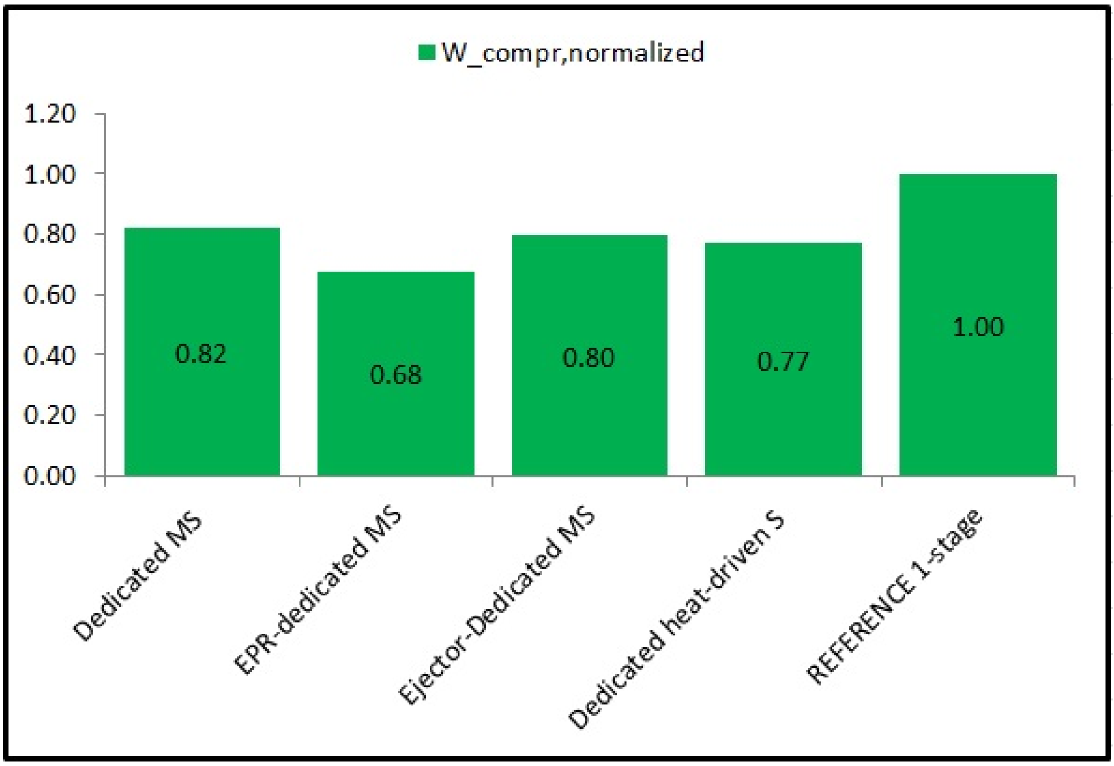

Figure 15.3 – Normalized compressor work for the different CO_2 transcritical vapour compression cycles with dedicated subcooling system

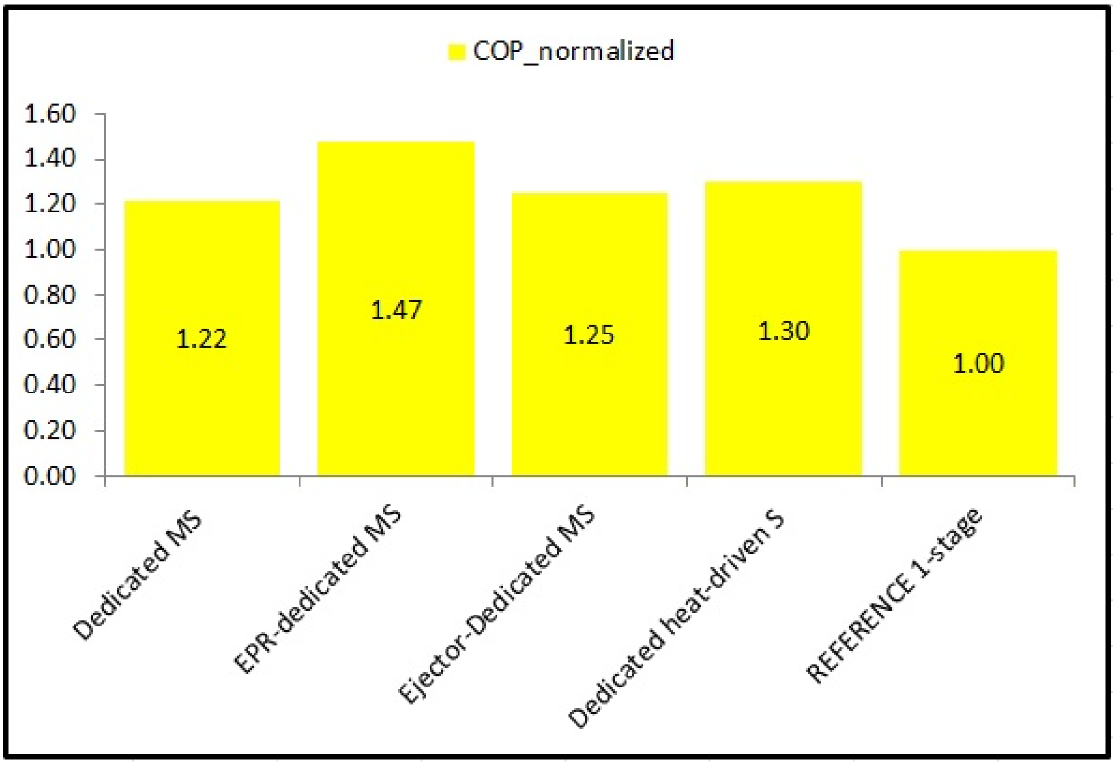

Figure 15.4 – Normalized system's COP for the different CO_2 transcritical vapour compression cycles with dedicated subcooling system

reduction of the compressor work. This is due to the fact that the subcooler compressor is working "for free". This compressor is actually conducted using the work recovered by the expander.

Finally, figure 15.4 compares the COPs of the cycles under consideration. The ejector-dedicated mechanical subcooling performs slightly better than the "traditional" dedicated mechanical subcooling cycle. For this reason, the increased complexity of the former is not justified. The dedicated heat-driven subcooling solution is very interesting because offers very good overall cycles performances and gives the possibility to use "free" waste heat. The COP improvement achieved is very high for the considered operating conditions. Anyway, the best performer is the EPR-dedicated mechanical subcooling system. This cycle offers a COP almost 50% higher than the one of the reference 1-stage cycle.

16 Further improvements of the CO_2 transcritical cycle using an ejector, design conditions

This chapter concerns the possibility to increase the performances of CO_2 transcritical vapour-compression cycles using an ejector. All the improvements are obtained combining the ejector cycle with already studied alternative solutions, more precisely dedicated and integrated mechanical subcooling. Also the possibility to use two ejectors in the same cycle was investigated. This should lead to an even higher reduction of the expansion losses and compressor work.
To have a fair comparison of the system's performances, all the cycle were running under the same working conditions. This operating conditions are the same used for all the investigation performed so far in this project.

16.1 Ejector cycle with integrated mechanical subcooling

The CO_2 transcritical vapour-compression cycle using an ejector can be improved applying to it integrated mechanical subcooling.
The system was optimized searching simultaneously for the optimal gas cooler, subcooler and ejector's mixing pressure. The results from the computational simulations are presented in figure 16.3. The figure offers a comparison of the improved cycle with the "original" ejector cycle. The cycle's enhancement brings interesting performances' improvements, except for the maximum cycle's temperature that is now slightly higher (actually, only less than 1°C higher!). The COP faces an interesting increase of almost 9%. The addition of mechanical subcooling to transcritical cycles offers important enhancement of the performances, this is a lesson that we all learn from this report.

16.2 Ejector cycle with dedicated mechanical subcooling

In the same way as the integrated mechanical subcooling was applied to the ejector cycle, it appears clear the possibility to analyse the system performances when applying a dedicated

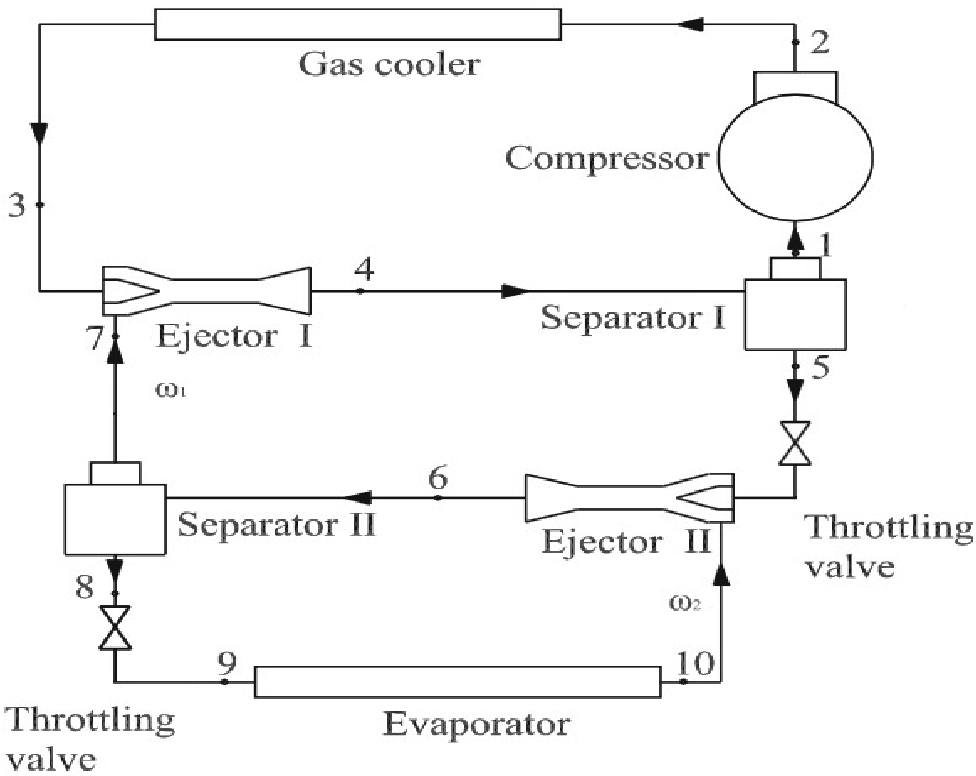

Figure 16.1 – CO_2 refrigeration cycle with two ejectors with expansion power recovery [48]

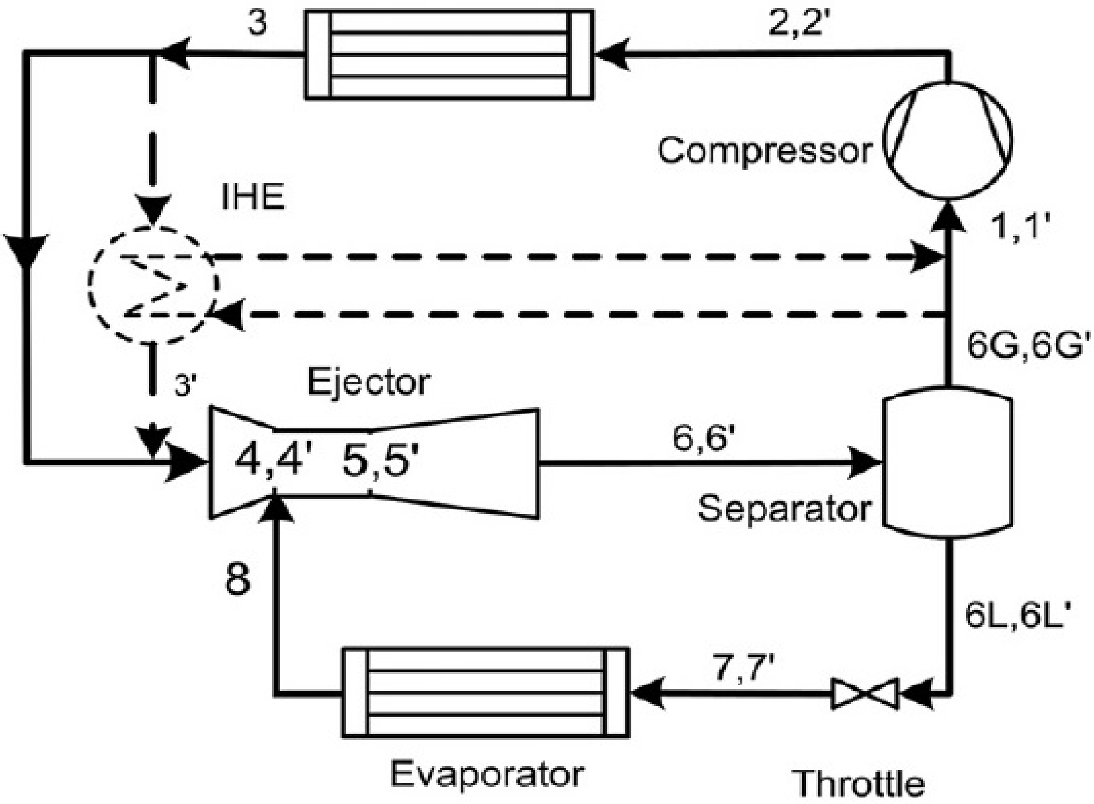

Figure 16.2 – PI diagram of the ejector cycle with IHE [49]

mechanical. The subcooler cycle is simply a 1-stage CO_2 vapour-compression loop.
The optimization of the system requires a 4-dimensional function. Simultaneously, the system must be investigated to find the optimal gas coolers, subcooler and ejector's mixing pressure. The results coming from the computational simulations are presented in figure 16.4. The performances' boost offered by the additional dedicated mechanical subcooling loop are similar to the ones offered by the addition of an integrated mechanical subcooling loop. Again the maximum cycle's temperature is slightly higher in the improved cycle but the difference respect the "traditional" ejector cycle is very small. On the other hand, the COP faces an important increase, it is almost 9% higher than for the ejector cycle.

16.3 Ejector cycle with ejector-dedicated mechanical subcooling

The performances of the ejector cycle are improved applying to it the ejector-dedicated mechanical subcooling loop. The ejector-dedicated mechanical subcooling was already introduced and studied in the previous chapters. The subcooler loop is further improved using an additional ejector with expansion power recovery function. The main cycle is a "traditional" ejector cycle with the addition of the subcooler, device in which the refrigerant is further cooled down after the gas cooler.
The system was optimized before running the simulation. The optimization requires a 5-dimensional function, the needed inputs are the two gas coolers pressures, the subcooler one and the two ejectors' mixing pressures. The performances of the system are presented in figure 16.5. A comparison of the improved cycle with the "traditional" ejector cycle is provided in that figure. As expected, the improved system works better than the traditional one. The use of a mechanical subcooling loop for the ejector cycle already gave appreciable improvements. The use of an ejector-dedicated mechanical subcooling loop allows for even higher performances, due to the improved behaviour of the subcooler loop. In fact, the compressor work in the subcooler loop can be reduced due to the expansion power recovery operated by the ejector. Even if not so clear from figure 16.5, the maximum cycle's temperature is slightly higher for the improved cycle. Anyway the difference is around 1°C and, thus, not really remarkable. The system's COP faces an important improvement using the ejector-dedicated mechanical subcoolig. Respect the "traditional" ejector cycle, it grows about 12%.

16.4 Transcritical CO_2 vapour-compression cycle using two ejectors

The expansion losses in the throttling valve can be reduced even more using simultaneously two ejector in a vapour-compression CO_2 cycle. This idea was provided by the work of Cen et al. [48]. In their work, they found out this cycle to present higher performances than the "traditional" ejector system. The PI digram of the resulting system is presented in figure 16.1. Compared to the "traditional" ejector cycle, the system requires an additional ejector, separa-

tor and expansion valve. The double expansion power recovery ensured by the two ejectors further decreases the expansion losses and compressor work, leaving room for system's COP improvements.
Actually, the investigations on this cycle proved that the improvements provided by the second ejector are quite small. The pressure difference throttled in the expansion valves of the improved cycle is reduced of almost 90%. Anyway, the pressure difference across the expansion valve of the "traditional" cycle was already very small (considering the absolute value) and so the system does not benefits that much from this improvement. The inlet pressure to the compressor is almost the same in the two cycles, as well as the optimal gas cooler pressure. This results in compressor work almost identical for the two cycle, it is only slightly smaller for the improved cycle (the compressor work is reduced less than 1%!). Figure 16.6 presents the results obtained from the computational simulations. As visible, the considerable complexity added to the system only allows for a COP increase of less than 1%.
Figure 16.7 offers an interesting investigation on the ejectors' efficiencies impact. The results presented in figure 16.6 were obtained assuming $\eta_m = \eta_s = 0.8$ and $\eta_d = 0.75$ for both the two ejectors. Now the two ejectors' efficiencies are varied to study their impact on the system. The case taken as a reference for the normalization of the COPs is represented by the single-ejector cycle studied in chapter 9. As visible from figure 16.7, when changing only the efficiencies of ejector 2 the COP does not vary that much, respect the single-ejector case. A greater impact on the COP is given by the efficiencies of ejector 1. This is of course a reasonable result. Ejector 1 is the one that deals with the biggest pressure recovery. Mainly from its performances depends the inlet pressure of the compressor and so the energy consumptions of the system.

16.5 Internal heat exchange in ejector cycles

The use of internal heat exchange (IHE) in ejector cycles is a solution widely discussed in the literature. Few word are here addressed to this solution. The PI diagram of the system is presented in figure 16.2. The dotted line represents the cycle's modification required to introduce the IHE.
The additional IHE is a solution not taken into account in this project. The reason is simple, for the given operating conditions, the addition of IHE does not give any COP's improvement neither for the 1-stage reference cycle nor the ejector cycle. Actually, the system performances of these cycles are reduced adding an internal heat exchanger to the cycle.
This is not a surprising result. The literature agrees on the fact that the IHE does not always ensure improvement of the ejector cycle's COP. For instance, this is stated in the work of Zhang et al. . In their work they also add their own interpretation of this behaviour, *"whether the energy efficiency of the ejector cycle by IHE can be improved depends on the isentropic efficiency level of the ejector. The utilization of IHE is only applicable in the cases of lower ejector isentropic efficiencies or high gas cooler exit/vapour temperatures for the ejector expansion system from the view of energy efficiency"* [49].

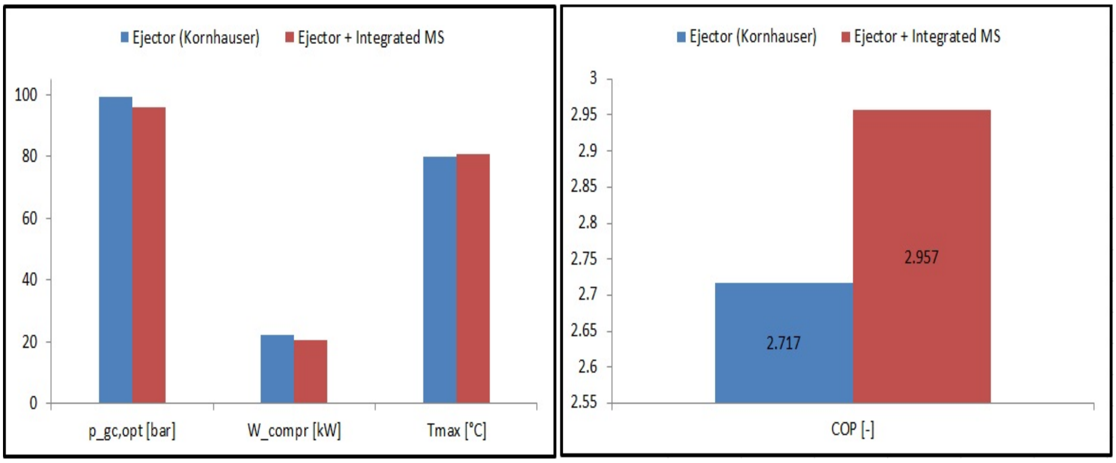

Figure 16.3 – Performance comparison: Ejector cycle with integrated mechanical subcooling vs. CO_2 transcritical vapour-compression cycle with ejector

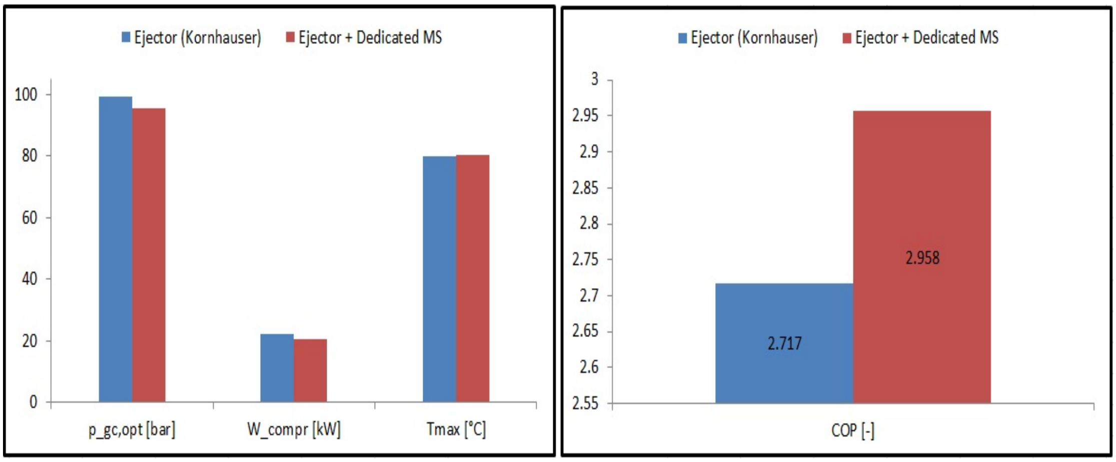

Figure 16.4 – Performance comparison: Ejector cycle with dedicated mechanical subcooling vs. CO_2 transcritical vapour-compression cycle with ejector

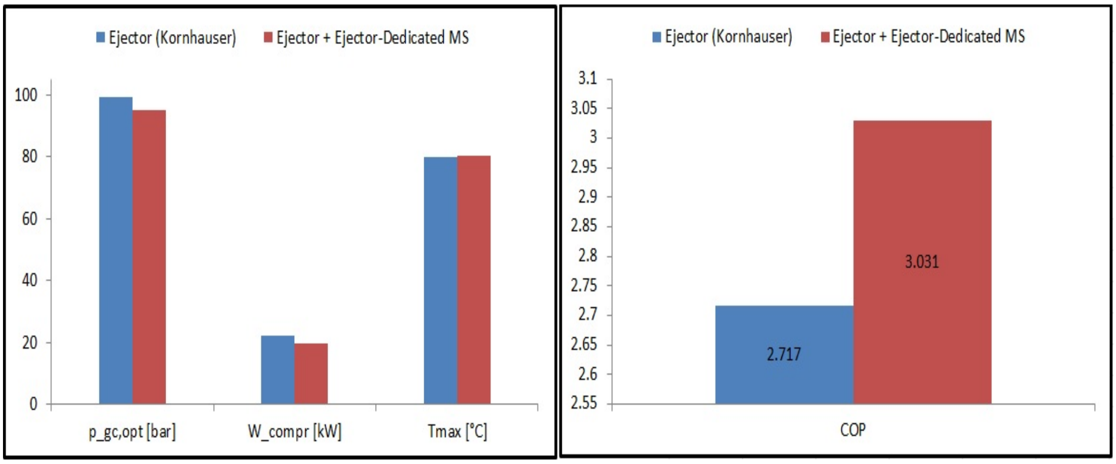

Figure 16.5 – Performance comparison: CO_2 Ejector cycle with ejector-dedicated mechanical subcooling vs. CO_2 transcritical vapour-compression cycle with ejector

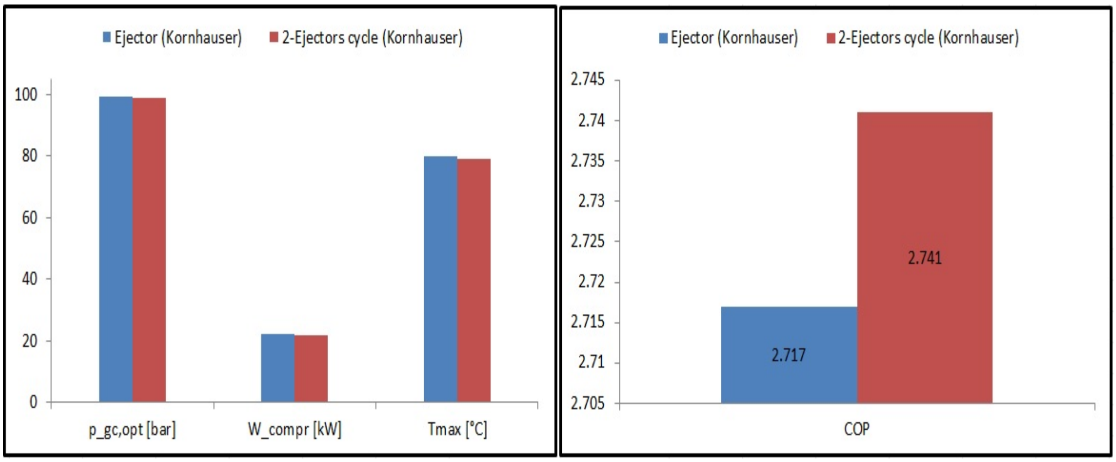

Figure 16.6 – Performance comparison: CO_2 double-ejector cycle vs. CO_2 transcritical vapour-compression cycle with ejector

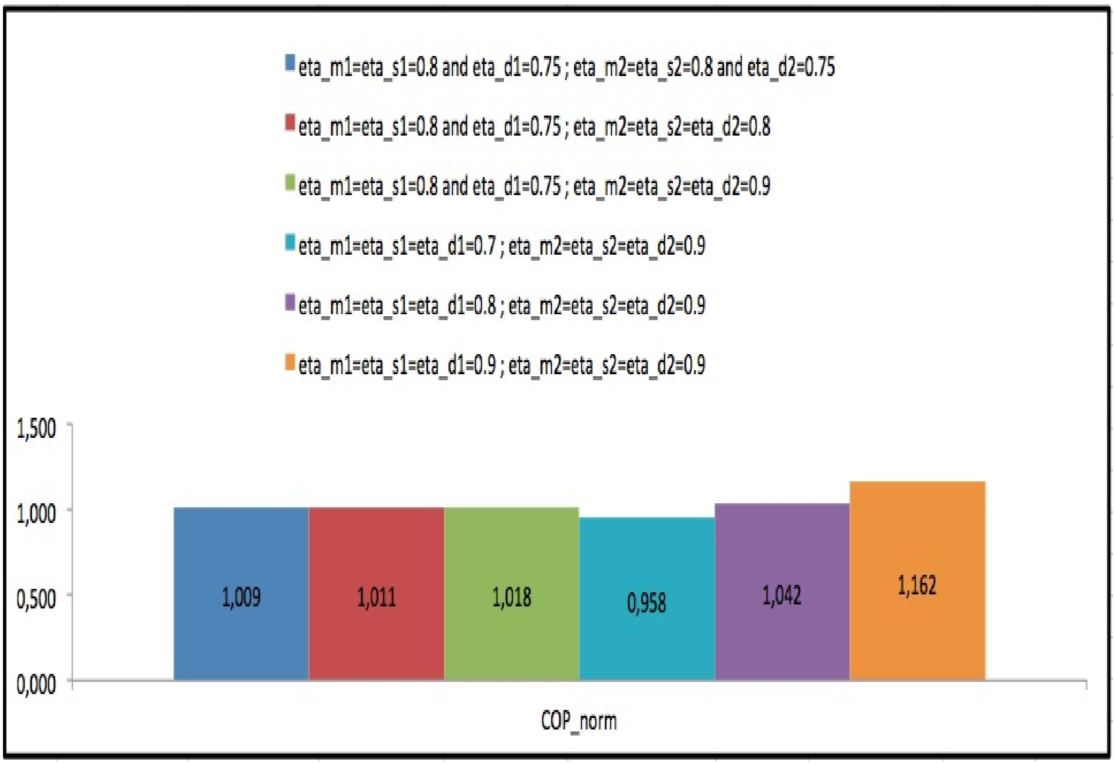

Figure 16.7 – Normalized COP for the CO_2 double-ejector cycle at different ejectors' efficiencies

17 Comparison of the further improved CO_2 transcritical cycles with ejector, design conditions

This chapter deals with the comparison of different improved ejector cycles (already introduced in chapter 16). Their performances are analysed respect the ones of the other systems in order to find out which one has the highest performer, for the given working conditions. All the results are presented as normalized values, the 1-stage CO_2 vapour compression cycle is taken as reference.

Figure 17.1 presents the normalized optimal gas cooler pressure for the studied cycles. The reduction in the optimal pressure is quite similar for all the ejector cycles. The additional mechanical subcooling loop (both dedicated or integrated) seems to be the best solution from this point of view. On the other hand, the improvement provided by the use of a secondary ejector is negligible.

Figure 17.2 shows the normalized maximum cycle temperature for the different cycles under investigation. The temperature reduction respect the 1-stage reference cycle is considerable for all the improved cycles. They all give almost the same good behaviour from this point of view, the differences between them are negligible.

As well as for the maximum cycle's temperature, also the compressor work is remarkably reduced, respect the reference cycle, using ejector cycles. Figure 17.3 presents the normalized compressor work for all the cycles. In general, all the systems give amazing performances from the compressor work reduction point of view. Again, the additional complexity of the 2-ejectors cycle is not justified by its performances. The greatest consumptions reduction is provided by the ejector cycle with ejector-dedicated subcooling. Anyway, it is worthy to say that also the "traditional" dedicated or integrated mechanical subcooling loop gives very good performances.

Finally, the normalized COP of the studied cycles is compared in figure 17.4. The cycle with the highest performances is the ejector cycle with ejector-dedicated mechanical subcooling. Anyway, its COP is only slightly higher than the one of the ejector cycle with dedicated or integrated mechanical subcooling. For this reason, the more complex ejector-dedicated mechanical subcooling loop can be avoided and instead integrated or dedicated mechanical subcooling loop can be adopted. Once again, the 2-ejectors cycle does not show particular improvements. Its complexity does not bring important benefit respect the "traditional"

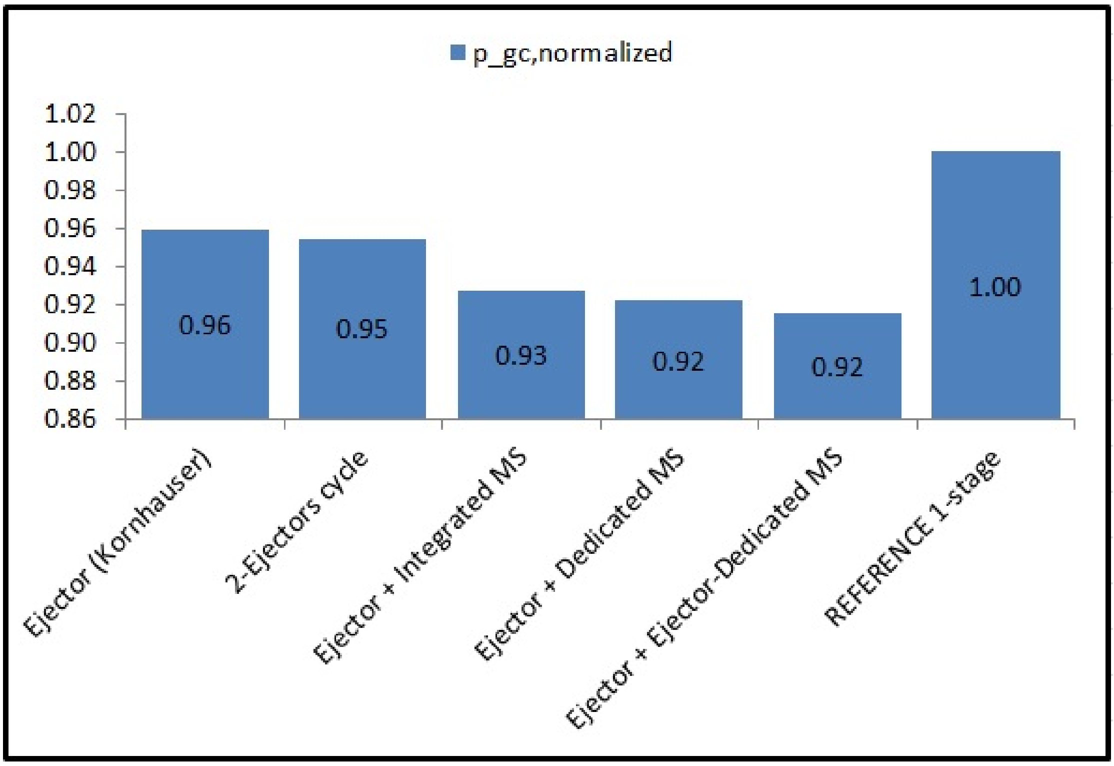

Figure 17.1 – Normalized optimal gas cooler pressure for the different CO_2 transcritical vapour compression cycles with ejector

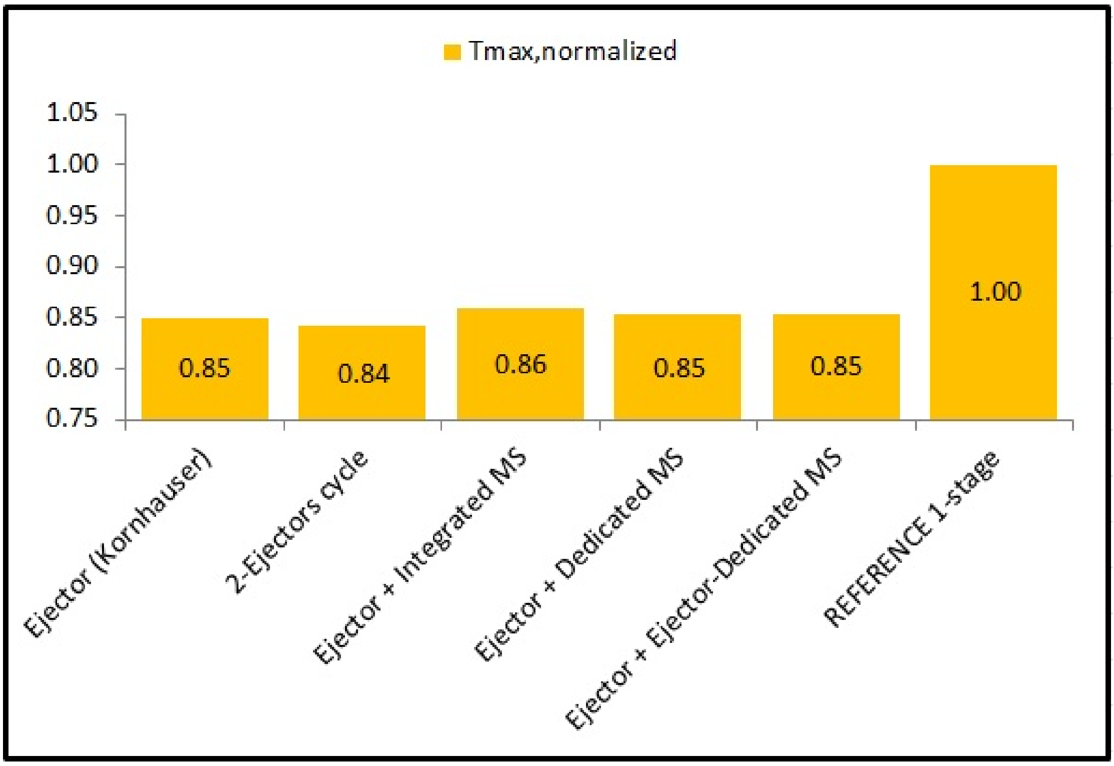

Figure 17.2 – Normalized maximum cycle's temperature for the different CO_2 transcritical vapour compression cycles with ejector

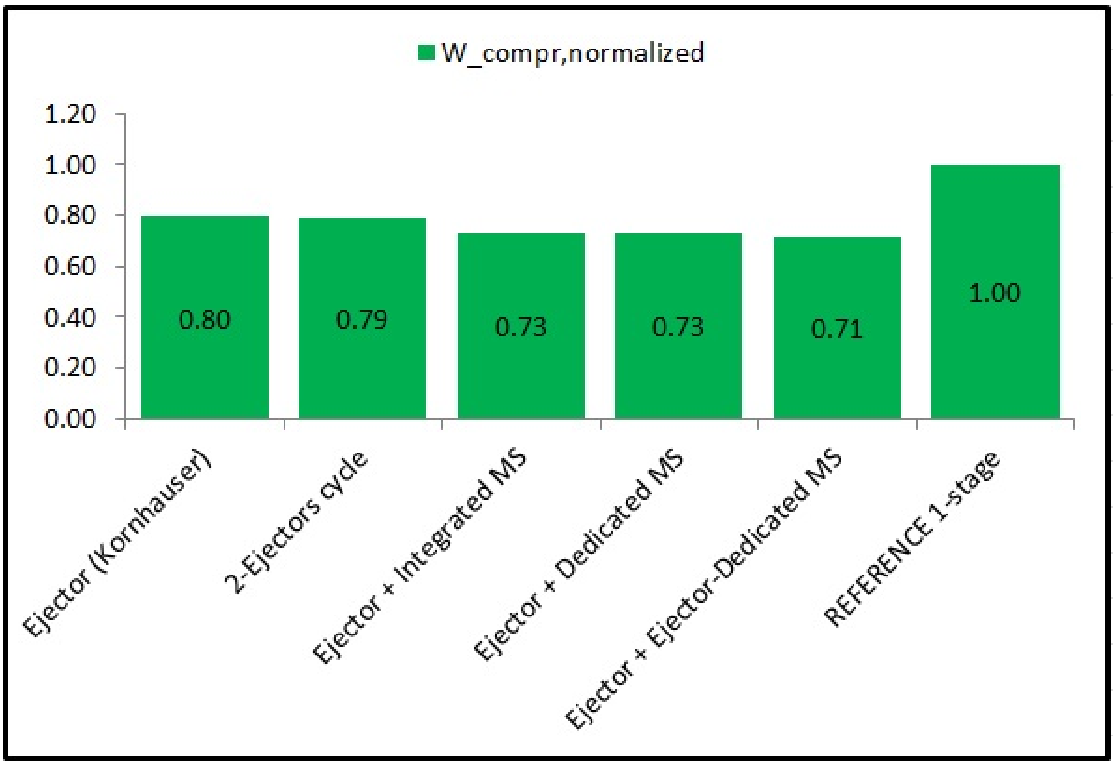

Figure 17.3 – Normalized compressor work for the different CO_2 transcritical vapour compression cycles with ejector

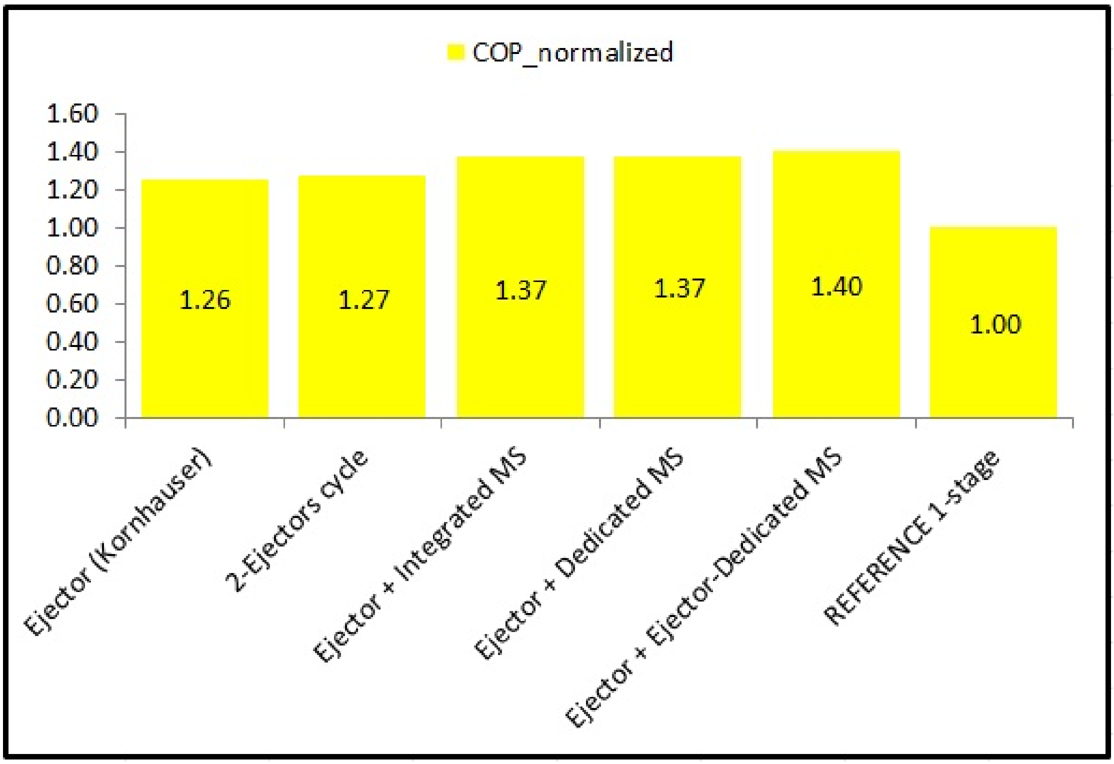

Figure 17.4 – Normalized system's COP for the different CO_2 transcritical vapour compression cycles with ejector

ejector cycle.

Part III

Part-load operation

18 Part-load operation of the studied cycles

In the first two parts of this report, the cycles were analysed and compared for the same, fixed, operating conditions. The design operation of a cycle is surely interesting to have a preliminary idea of its behaviour. Anyway, it is hard to imagine a refrigeration cycle that operates under the same conditions all along the year.
The usual heat sink for refrigeration systems is the ambient air, consequently, these cycles are strongly affected by the ambient conditions, in particular the outdoor temperature. This parameter influence considerably the working conditions of the system. Commonly, a refrigeration system must be able deal with cold winter's operation and high temperature summer's operation. When far away from the design conditions, the performances of one system can change considerably. It is this aspect that is under investigation in this third part of the project.
The performance enhancements of CO_2 vapour-compression refrigeration cycles are here studied applying solutions like mechanical subcooling (both integrated and dedicated), parallel compression economization and the use of an ejector. All these solutions were discovered really promising during the performed investigation at design conditions. Now, it is interesting to analyse if the considered solutions give a performances boost all along the year, under all the very different ambient conditions faced by the system.
Part-load analysis means that the behaviour of one system is investigated when away from its design conditions. The part-load of the four considered solutions was analysed for realistic weather conditions, in order to asses which one performs better in a real-year operation.
The following chapters are about the off-design operation of the improved CO_2 cycles. The results obtained for each one of them are presented and compared. Before this, some words are spent in this chapter to describe the method followed during the performed analysis.

18.1 Cycles modelling

The way it was dealt with the part-load operation is quite similar for all the studied cycles. Here the common modelling part for all the cycles is presented. Particular modelling of some components or specific assumptions are presented when it is required.

The ambient temperature is considered a given input for all the cycles. Realistic temperature profiles were considered, this topic will be discussed more in details in the last section of the chapter. For the preliminary investigation, the isentropic efficiency of all the compressors was kept constant (to a value of 0.9) all along the year. This is similar to what it was done by Sharma et al. in their CO_2 systems yearly consumptions investigation [1]. The cooling capacity of the system was treated similarly, a fixed value of 60kW was assumed for all the hours of operation of the system. Anyway, in the following chapters more realistic behaviour of the compressor and proper load variations are also considered. The modelling for that conclusive analysis is introduced later in this report. During transcritical operation of the systems, the gas cooler pressure was always optimized, for every hour of the year.

After this introductory assumptions, more details are presented about the heat exchangers' models: they represent the real key for a proper simulation of the part-load operation. First of all, it was assumed that all the transcritical system's switch to subcooler operation when the ambient temperature is below 22°C. Of course this assumption is not very realistic due to the fact that real systems pass through a transition phase between the transcritical and subcritical operation. The assumed abrupt change at 22°C is used only to simplify the investigation. Moreover, using a common transition temperature, the comparison of the cycles behaviours was eased.

For evaporator and gas cooler, it was assumed that no control of the air mass flow was performed. The air mass flow through these heat exchangers was defined on the base of the chosen design conditions and it was kept constant for all the different operating conditions faced along one year. This is not what happens for the condenser: in this component the air mass flow rate is controlled and variable speed fans are assumed.

It is demonstrable that the air-side UA-value and the refrigerant-side UA-value of the cycles' condenser, gas cooler and evaporator varies with the air/refrigerant mass flow rate following this equations:

$$\frac{UA}{UA_0} = (\frac{m}{m_0})^{0.6} \tag{18.1}$$

This is of course an approximation. For simple part-load investigations, without involving complicated empirical correlation for the heat transfer coefficients, the UA-value can be considered a function of the mass flow. The subscript "0" means that it is referred to the chosen design conditions. In this way, knowing mass flow (air and refrigerant side) and UA-value (air and refrigerant side) at the chosen design conditions, it is possible to determine the heat exchanger UA-value for each one of the system's operating conditions along the year.

In this way, part-load operation of gas cooler, condenser and evaporator was modelled. Starting from the gas cooler, input parameters are the design mass flow rates, the design UA-values and the ambient temperature. At the design conditions, it was assumed that the gas cooler outlet refrigerant temperature is 8°C higher than the ambient temperature. Actually the literature suggests even lower approach temperatures at the gas cooler outlet([3], [1], [4] and [28]). Anyway, a temperature difference of 8°C was assumed to be on the safety side. Using the described simple correlation (equation 18.1), the gas cooler UA-value for each working conditions can

be found. Knowing the heat exchanged at the gas cooler and using the logarithmic mean temperature difference combined with the known mass flow rates, all the required outputs are got from the model. The outlet gas cooler temperatures, both on air and refrigerant side, are calculated.
The same goes for the evaporator. For each working condition, cooling capacity, air mass flow rate and inlet and outlet air conditions are fixed. Consequently, the system adjusts slightly the evaporation pressure to satisfy the requirements.
Also for the condenser, the way the model works is similar. The main difference is the non-fixed air mass flow rate. The different air mass flow control for evaporator, gas cooler and condenser was suggested when speaking with DANFOSS about the way they control the operation of their systems. For the condenser model, a temperature difference was fixed. To ensure a reasonable and realistic heat exchange between the two fluid, the condensation temperature was assumed to be 8°C higher than the inlet air temperature (that it is actually the ambient temperature). This is similar to what it is done by Minetto et al. in their yearly consumption investigation of CO_2 systems [3]. On the other hand the outlet air temperature is free to vary on the base of the working conditions. The UA-value of the condenser is determined on the base of the simple correlation described (equation 18.1) and the design conditions, exactly as for gas cooler and evaporator.
Also during the part-load investigation, subcooling and superheating of the refrigerant after condenser and evaporator were considered nil. Moreover, the throttling process in the expansion valve is still considered an isentropic phenomena.
The main aim of this investigation is to analyse the different performances of the cycles for the variable ambient conditions faced along the year. Special attention is directed to the consumptions of the cycles but the fan power was not included in this analysis. As a lot of literature report (as [1] for instance), the fan consumptions are a very small percentage of the compressor work. This was also personally checked by the author. Some simulations were performed using the software Pack Calculations Pro [50]. This is a specific software that allows detailed investigation of the energy consumptions of refrigeration cycles. The results always suggested that the fan power is a negligible percentage of the compressor work.

18.2 Location

The part-load analysis was performed using real ambient temperature profile. This gives a more realistic mark to the results obtained and presented in this section of the report. The chosen location for the systems is Rome, Italy. The purpose of the study is an investigation of the CO_2 vapour-compression cycles used in hot climates. As already explained, the reason are the particularly low performances of this systems for high ambient temperature. Rome is the perfect test to investigate the thermodynamic behaviour of these improved cycles. In a typical winter day, the temperature hardly goes below zero. On the other hand, summer is really hot, with typical August temperature that easily reach 30°C.
As already said in this report, for average ambient temperatures below 15°C the CO_2 cycles out-

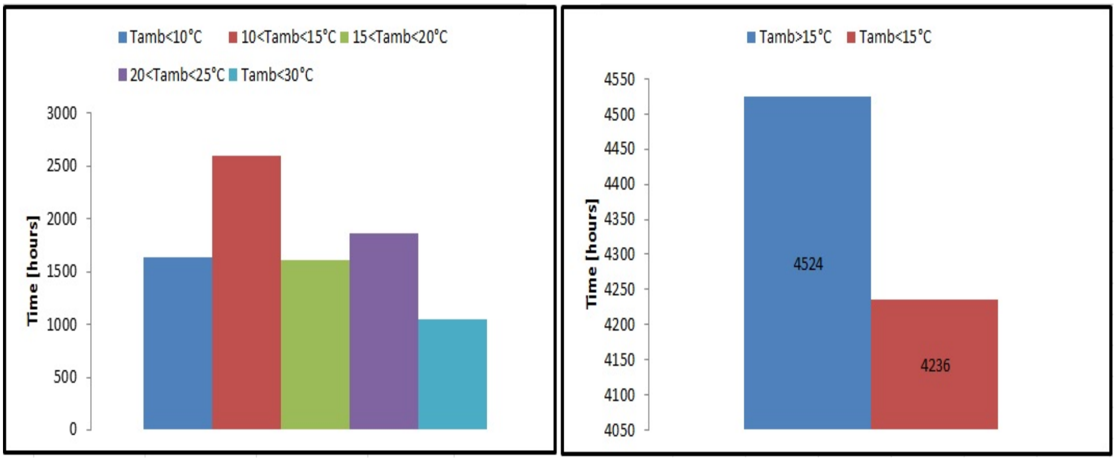

Figure 18.1 – Ambient temperatures distribution for Rome, Italy

perform the systems using "traditional" refrigerants. On the other hand, their performances drop considerably when the working conditions are, on average, over 15°C [27]. The ambient temperatures distribution for Rome is presented in figure 18.1. As visible, the hours with ambient temperatures above and below 15°C are almost the same amount along the year. Of course, this 15°C are just a representative temperature but it is clear that Rome is a location that requires many hours of transcritical operation. For this reason it is particularly suitable for the investigation here conduct.

The yearly temperature profiles were provided by the U.S. department of energy [51]. Actually, they do not provide temperature profiles for each day of the year, but only the temperature of a typical day for each month. Anyway, these data were considered more than enough for the kind of investigation that had to be performed. The profiles come from an accurate investigation of the daily temperature for 2014, so these representative days are reliable and realistic enough profiles.

When calculating the yearly energy consumptions, each month was considered as composed by all days having the same temperature profile, the one of the representative day of the considered month.

19 One-stage vapour-compression CO_2 transcritical cycle, part-load operation

The CO_2 1-stage vapour-compression refrigeration cycle was already introduced in chapter 5. This cycle was chosen as reference also for the part-load investigation of the improved cycles. The system was modelled as described in chapter 18, no other particular expedients were required. Due to the fact that the modelling of the part-load operation of compressor, gas cooler/condenser and evaporator was already introduced, this part is here considered as known to the reader. As already introduced, the cycle switches to subcritical operation for temperature lower or equal than 22°C. Due to the fact that all the cycles have the same switch temperature and use the same temperature profiles, they all work in transcritical mode around 20% of the hours of the year. The majority of the hours of the year, the systems run subcritical. Anyway, the transcritical operation hours are considerable and can penalize consistently the yearly system performances if no improvements to the cycle are adopted.

When running transcritical, the cycle is provided of one gas cooler, one evaporator, one compressor and one expansion valve. As already explained in previous chapters, the throttling process is assumed to be isenthalpic while the compressor's behaviour is determined only by its constant isentropic efficiency. Gas cooler and evaporator are modelled as described in the previous chapter (chapter 18), as well as the condenser used in subcritical operation of the system. During transcritical operation, the gas cooler pressure is optimized every time that the ambient temperature (and consequently the operating conditions) changes.

Some troubles were found when running the part-load simulations using the software EES. The program was not able to automatically optimize the gas cooler pressure for each value of ambient temperature. Some help to the software was required, limiting the intervals in which EES was required to find the solution. All the results for the system's operation at different ambient temperature were collected in spread sheets. After that, a look-up table was built in order to easily recreate the hourly behaviour of the system along the year.

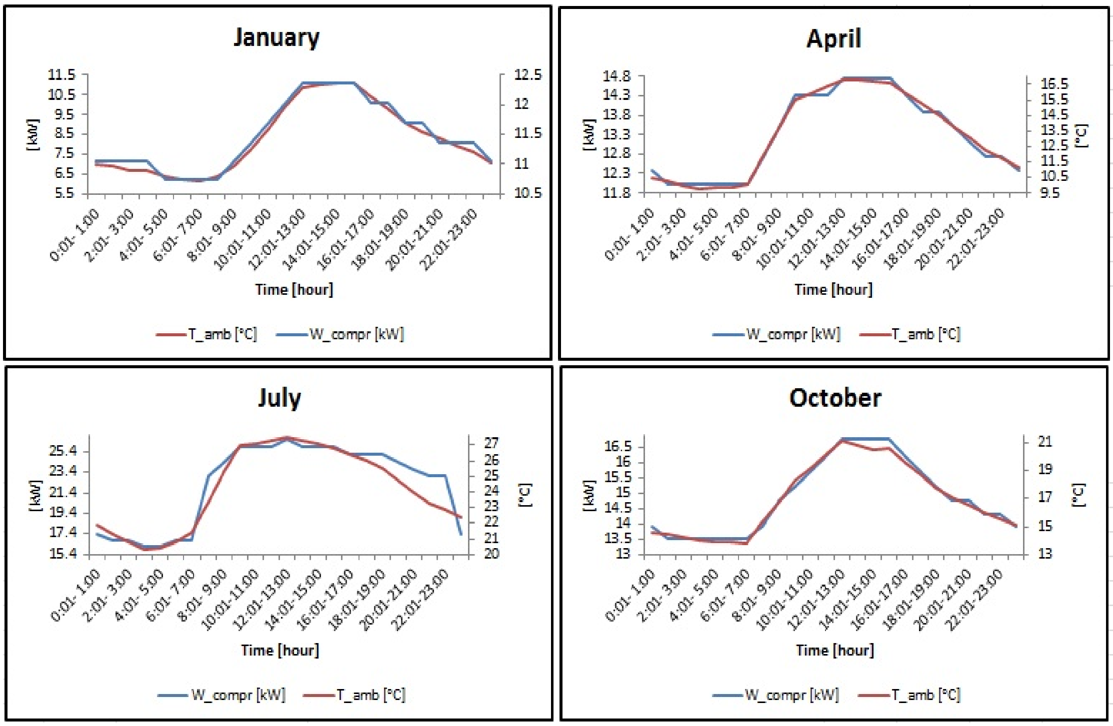

Figure 19.1 – Daily compressor work profiles for one day from each season, 1-stage reference cycle

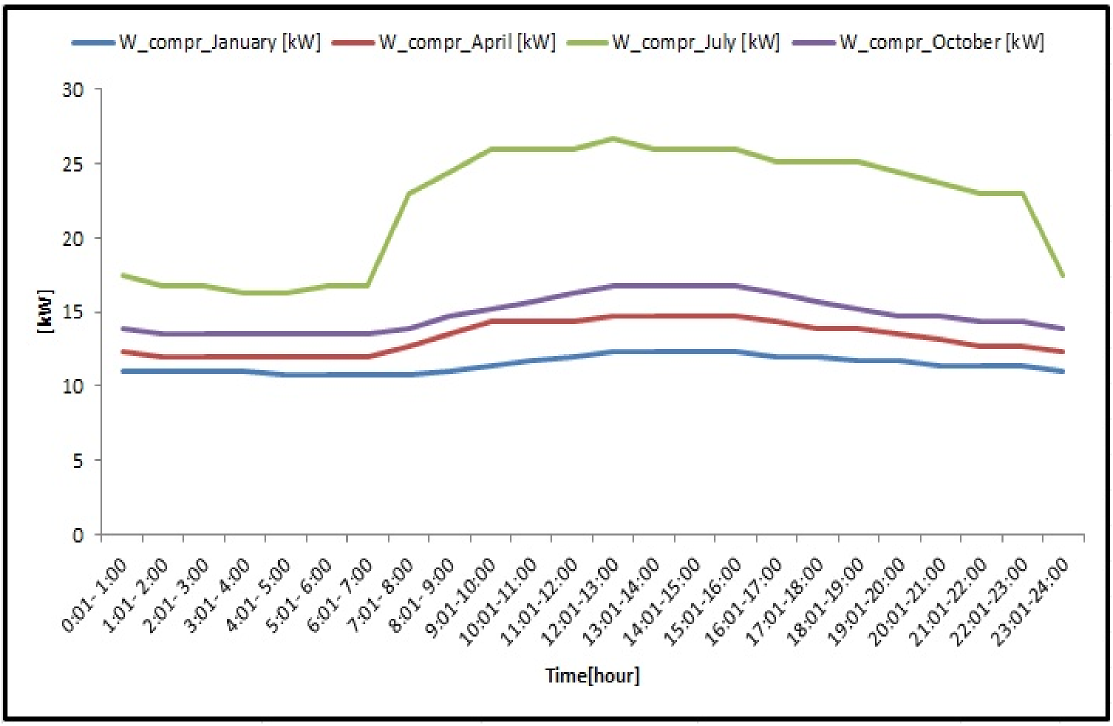

Figure 19.2 – Comparison of the daily compressor work profiles of figure 19.1, 1-stage reference cycle

19.1 Part-load behaviour of the system

Figure 19.1 presents the daily compressor power profiles for one representative day from each season. It was chosen to show the month immediately after the one in which the season "officially" begins. For instance, winter starts the 21st of December, so it is shown a day from January as representative of the Winter season. In this way it is ensured that the "fully developed" weather of that specific season is considered. As expected, all the compressors' power profiles follow almost perfectly the daily temperature profile. In fact, the ambient temperature is the only parameter that determines the change in the operating conditions of the system. When the ambient temperature increases, the gas cooler mean pressure/condenser pressure increases too, in order to make feasible the heat release to the ambient air. On the other hand, the evaporation pressure only slightly changes because the cooling capacity is independent from the outdoor conditions (see the assumptions made in chapter 18). The result is that higher ambient temperatures bring higher compressors' power requirements. For this reason, it is logic that the compressor work follows the outdoor temperature trend.
Of course, *"the COP variation is a mirror image of the ambient temperature pattern"* [31].
Figure 19.2 is probably even more interesting. The daily compressor work profiles from figure 19.1 are now in the same graph. In this way it is possible to understand which is the period of the year more "stressful" for the system, from the compressors' work point of view. The highest consumptions are faced in the summer months. The high ambient temperatures force the system to work most of the time transcritical with high pressure differences between gas cooler and evaporator. This increase considerably the required input to the system (in other words, the consumptions of the compressors). On the other hand, the coldest months are the ones with the lowest energy requirements. The pressure difference evaporator/condenser is considerably reduced and so the compressors' work.
As already introduced the transcritical operation of CO_2 cycle at high ambient temperature is their real week point. Figure 19.2 shows that the consumptions during summer months are much higher than the ones for the other seasons. It is clear the necessity to work on this cycle to improve their performances especially for high ambient temperature operation.
Table 19.1 shows some representative parameters that can give an approximative idea about the system performances along the year. The average COP and the average compressor work is presented. These two parameters give a taste of how good, on average, is behaving the system during the year. Moreover, table 19.1 presents the maximum cycle's temperature and the maximum optimal gas cooler pressure faced by the cycle during a typical year operation. These data will be even more meaningful when compared to the ones of the improved cycles.

COP_{av}	$p_{gc,opt,max}$	T_{max}	$W_{compr,av}$
[-]	[bar]	[°C]	[kW]
4.14	97.76	93.24	15.58

Table 19.1 – Average and maximum system performances for part-load operation of CO_2 1-stage vapour-compression refrigeration cycle

20 Vapour-compression CO_2 cycle with parallel compression economization, part-load operation

The CO_2 vapour-compression cycle with parallel compression economization (PCE) was already introduced in chapter 6. The cycle showed very interesting performances during the design operation analysis. If the improved cycles are ranked based on their COPs at the studied design conditions, the improvements the PCE cycle brings is second only to the ones provided by the ejector cycle. For this reason the PCE cycle is a quite promising solution to improve the performances of CO_2 cycles, especially when operating in hot climates.
The cycle, respect the known 1-stage reference one, presents an additional separator, compressor and expansion valve. The isentropic efficiency of the secondary compressor is still assumed to be 0.9 all along the year and the expansion valve operates always as an isenthalpic process. The separator is assumed to work ideally: two streams are coming from it, one is saturated vapour and the other saturated liquid. For the part-load operation, gas cooler, condenser and evaporator are modelled as already explained in chapter 18.
During the hours of transcritical operation, the gas cooler pressure was optimized every time the ambient conditions change. Obviously, no optimization is required for the condenser pressure when the system runs subcritical. Regardless the operating conditions (subcritical or transcritical), the separator pressure was optimized for every operating condition. The computational simulations were performed assuming that the separator pressure is free to vary, until it remains at least 4 bar higher the evaporation pressure. A minimum pressure difference across the expansion valve must be ensured for a proper functioning of this device.

20.1 Part-load behaviour of the system

As it was found for the 1-stage reference cycle, the compressors' work profiles follow almost perfectly the ambient temperature. They are not presented here because they look like the ones shown for the reference cycle, except for the absolute value assumed by the compressor work that is lower considering the same hour of the year.
Figure 20.1 presents the compressor work daily profiles for one representative day of each season. Moreover, the figure shows the July compressor work profile of the reference 1-stage

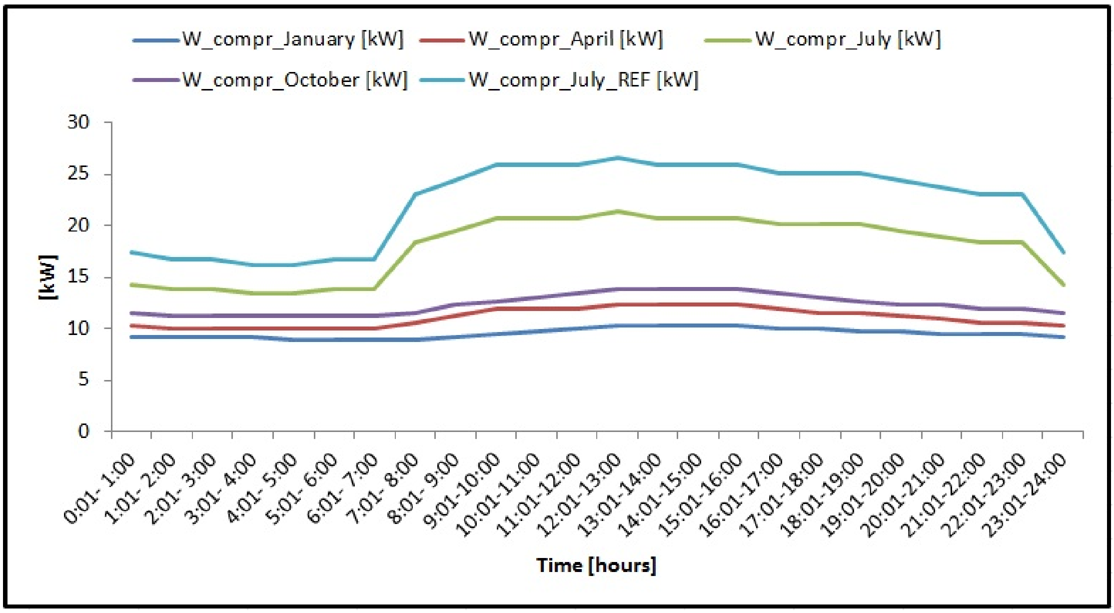

Figure 20.1 – Comparison of the daily compressor work profiles for a representative day from each season, PCE cycle

COP_{av}	$p_{gc,opt,max}$	T_{max}	$W_{compr,av}$
[-]	[bar]	[°C]	[kW]
4.99	92.36	87.97	12.82

Table 20.1 – Average and maximum system performances for part-load operation of CO_2 PCE cycle

cycle, in order to make possible a comparison of the two cycle. Still, the warmest months are the one that give the highest energy consumptions. Anyway, the figure shows a considerable reduction of the consumptions in July respect the 1-stage reference cycle.
Table 20.1 shows some representative parameters of the cycle. The average COP and compressor work are presented, along with the maximum gas cooler optimal pressure and the maximum cycle's temperature. They allow the reader to have a taste of the system's performances along the year.

21 Vapour-compression CO_2 cycle with integrated mechanical subcooling, part-load operation

In chapter 7, the integrated mechanical subcooling cycle was already introduced. The additional subcooling loop provides a further cooling of the refrigerant after gas cooler/condenser. During the design operation analysis, this was proved to be beneficial for the cycle. The CO_2 vapour-compression cycle with integrated mechanical subcooling was found to provided a great increase of the system's performances, comparable to the boost provided by the PCE cycle. In this chapter, the behaviour of this system in part-load operation is analysed.
The way the system was modelled is not different from what it was said in the previous chapter for the other cycles. The components exploited in this cycle are the same used in the 1-stage reference cycle, except for the subcooler. The subcooler model was object of a detailed investigation. Two different models were analysed: one based on the UA-value correlation used for gas cooler, evaporators and condenser and one based on a fixed temperature difference between the two fluids. The former model is surely more complicated and sometimes this additional complexity made difficult the convergence of the simulations. On the other hand, the fixed ΔT model is much more simple. Anyway, the two systems resulting from the use of the two different models for the subcooler gave almost the same results in term of performances (see appendix B). For this reason, the author felt free to choose the model on the base of the facility to perform the investigation. The fixed ΔT model was chosen because it made much more fluent the achievement of the simulations' convergence. Similarly to what it was done during the design simulations, the $\Delta T = T_5 - T_7$ was fixed to 5°C, for each operating condition faced along the year.
Every time that the ambient conditions change, the system was optimized finding the optimal gas cooler and subcooler pressure. Of course, no optimization of the condenser pressure was performed when the system runs subcritical. On the other hand, subcooler pressure was always optimized, regardless the operating conditions of the cycle.

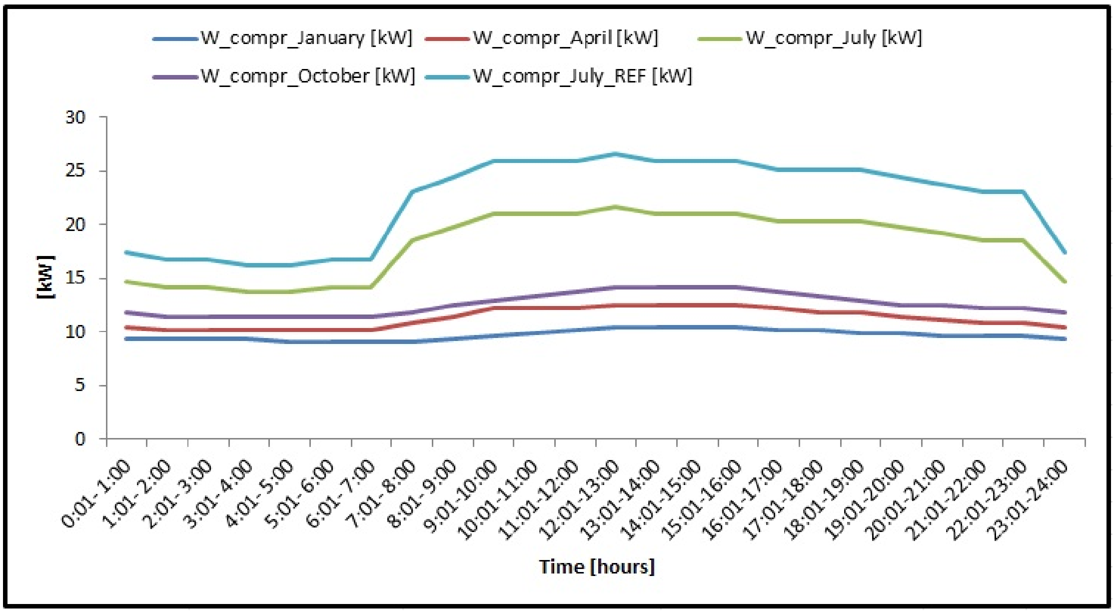

Figure 21.1 – Comparison of the daily compressor work profiles for a representative day from each season, integrated mechanical subcooling cycle

COP_{av}	$p_{gc,opt,max}$	T_{max}	$W_{compr,av}$
[-]	[bar]	[°C]	[kW]
4.89	92.48	88.09	13.05

Table 21.1 – Average and maximum system performances for part-load operation of CO_2 integrated mechanical subcooling cycle

21.1 Part-load behaviour of the system

Figure 21.1 presents the daily compressor work profiles for one representative day from each season. If the temperature profile for each one of those days was added, it would be possible to see how the compressor work profiles follow accurately the temperature profile. As a benchmark, the daily compressor work profile of July for the 1-stage reference cycle is also added in figure 21.1. The reference profile allows the reader to notice the considerable consumptions reduction performed by the integrated mechanical subcooling loop during transcritical operation of the system. During the summer months, the one in which the system's performances are lower, the integrated mechanical subcooling cycle allows for considerable energy savings. Table 21.1 shows some representative parameters of the cycle. The average COP and compressor work are presented, along with the maximum gas cooler pressure and cycle's temperature. They allow the reader to have a taste of the system's performances along the year.

22 Vapour-compression CO_2 cycle with dedicated mechanical subcooling, part-load operation

Chapter 8 makes available a very thorough description of the dedicated mechanical subcooling cycle. Differently from the integrated one, the dedicated mechanical subcooling loop is formed by a separated CO_2 vapour-compression refrigeration cycle. The evaporator of the secondary loop is actually the subcooler of the main cycle. In this component, the refrigerant coming out from the gas cooler/condenser is further cooled down due to the evaporating CO_2 of the subcooler loop. The design operation analysis of the cycle gave promising results. The performances of the CO_2 system using dedicated mechanical subcooling were found to be more than 20% higher than the ones for the 1-stage vapour-compression refrigeration cycle.
To simulate the part-load operation of the system, two 1-stage vapour-compression cycles must be modelled, one for the main cycle and one for the subcooler cycle. How evaporator, gas cooler, condenser, compressor and expansion valve are modelled was already discussed in chapter 18. The two separate loops are coupled due to the heat exchanger called subcooler. This device was modelled in the same way explained for the integrated mechanical subcooling cycle and the reasons behind the choice of the model are identical. See appendix B for a more detailed comparison of the subcooler models applied to the dedicated mechanical subcooling loop.
The yearly part-load investigation of the cycle was performed optimizing the system for all the ambient conditions faced along the year. During transcritical operation, the two gas coolers pressures were optimized along with the subcooler one. During hours of subcritical operation, only the subcooler pressure required to be optimized on the base of the outdoor conditions.

22.1 Part-load behaviour of the system

The daily compressors' work profiles for one representative day from each season are presented in figure 22.1. As expected for all the CO_2 vapour-compression refrigeration cycles, the summer months, with very high ambient temperatures, are the ones with the lowest system's performances. The cycle is forced to work for the most part of the hours as transcritical, with high energy consumptions. Figure 22.1 present also the work profile for July of the 1-stage

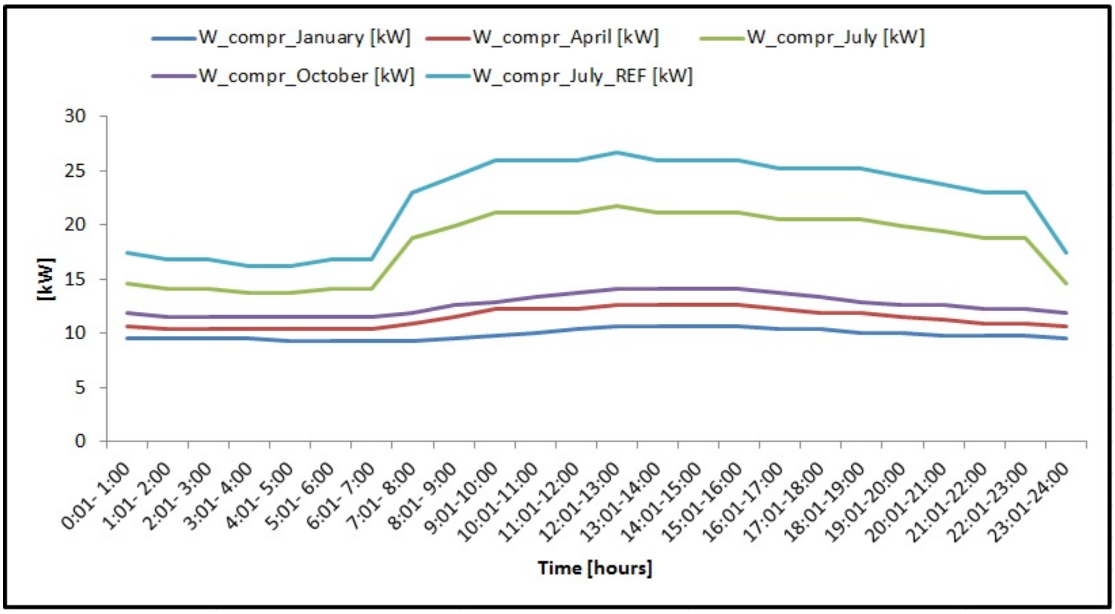

Figure 22.1 – Comparison of the daily compressor work profiles for a representative day from each season, dedicated mechanical subcooling cycle

COP_{av}	$p_{gc,opt,max}$	T_{max}	$W_{compr,av}$
[-]	[bar]	[°C]	[kW]
4.86	90.84	86.33	13.13

Table 22.1 – Average and maximum system performances for part-load operation of CO_2 dedicated mechanical subcooling cycle

reference cycle. This benchmark allows the reader to evaluate the considerable reduction of the compressor work that the mechanical subcooling cycle provides. It seems that even when evaluating the system in part-load operation (not only at design conditions), the mechanical subcooling allows for great system performances improvement. The same results were obtained by Llopis et al. during their yearly investigation on mechanical subcooling cycles. *"For environmental temperature higher than 25°C approximately, the MS cycle produces high increments of the COP of the system, being thus recommended for warm and hot countries. For high environment temperatures, the use of MS cycle allows the optimum pressure of the transcritical cycle to be reduced, that producing an additional reduction of the power consumption in* CO_2 *compressors and therefore a highest improvement of the COP* [28]
Table 22.1 shows some representative parameters of the cycle. The average COP and compressor work are presented, along with the maximum gas cooler pressure and cycle's temperature. They allow the reader to have a taste of the system's performances along the year.

23 Vapour-compression CO_2 cycle with ejector, part-load operation

The ejector cycle was already introduced in chapter 9. The analyses on this cycle, under design conditions operation, showed the highest performances improvement, respect the 1-stage reference cycle and all the other improved cycles (see chapter 10 for their comparison). The COP increase of 26% respect the reference cycle was the most promising result found during the investigations conducted during the first part of the project. In this chapter, the ejector cycle's performances are tested also during part-load operation of the system, along a real-year of operation.
The components that the ejector cycle has in common with the previously described cycles, were modelled in the same way already explained. See chapter 18 for the common modelling part of all the cycles. Concerning the ejector, no particular expedients were required to model it. The isentropic efficiencies of the ejector were considered constant for all the different operating conditions, similarly to what it is done for the compressor. During the computational simulations regarding the part-load operation, the separator pressure was assumed controlled in order to follow whatever pressure recovery the ejector was able to perform. Especially for low ambient temperature, very small pressure differences across the expansion valve were faced. Anyway, for this preliminary investigation, it was assumed that whatever pressure difference across the valve is acceptable.
The part-load analysis of the system was performed assuming the cycle always operating at optimal conditions. During transcritical operation, the gas cooler pressure was optimized, along with the ejector's mixing pressure. When the system was running subcritical, only the mixing pressure was optimized for each different ambient condition.

23.1 Part-load behaviour of the system

As for the previous cycles, the compressor work profiles are here reported, figure 23.1. One representative day from each season was chosen. Moreover, as a reference, the July's daily compressor work profile for the 1-stage reference cycle is reported. The system's consumptions are higher in the hottest months, as expected. Anyway the energy input required is considerably

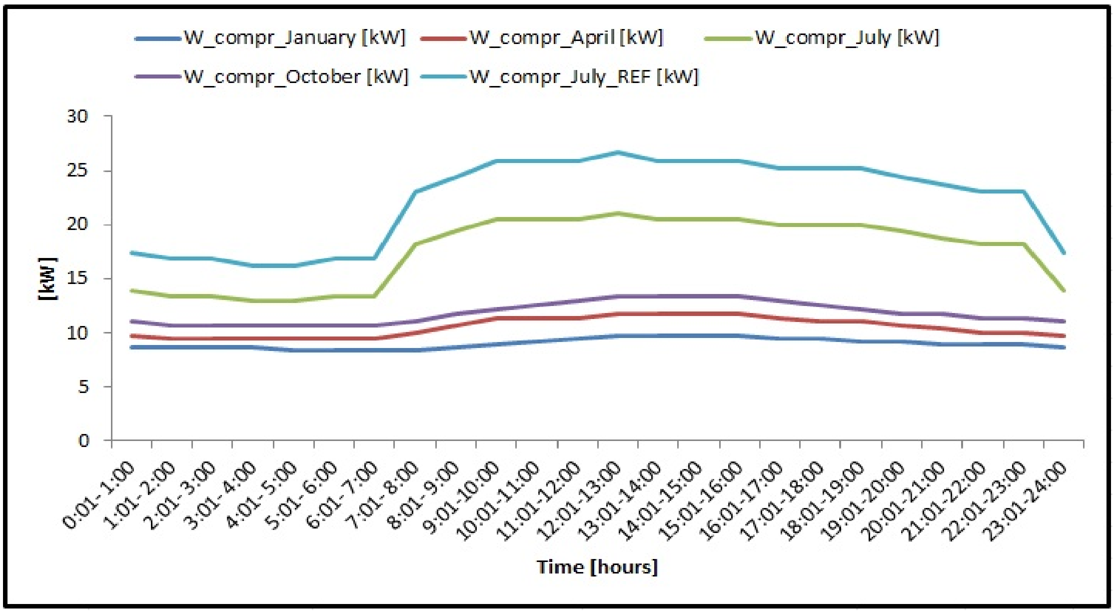

Figure 23.1 – Comparison of the daily compressor work profiles for a representative day from each season, ejector cycle

COP_{av}	$p_{gc,opt,max}$	T_{max}	$W_{compr,av}$
[-]	[bar]	[°C]	[kW]
5.24	95.62	81.57	12.33

Table 23.1 – Average and maximum system performances for part-load operation of CO_2 ejector-expansion cycle

reduced respect the reference cycle, as visible from the figure.
Table 23.1 shows some representative parameters of the cycle. The average COP and compressor work are presented, along with the maximum gas cooler pressure and cycle's temperature. They allow the reader to have a taste of the system's performances along the year.

It is worthy to say that the assumption about the constant isentropic efficiencies of the ejector should be revised. As reported by Kornhauser, *"in general, ejectors perform poorly away from their design points"* [35]. Anyway, it is difficult to establish a pattern for the ejector's isentropic efficiencies during part-load operation. So the constant efficiencies assumption used to simplify a bit the investigation. It must be kept in mind that the results here obtained might be "too good to be true".

24 Comparison of the studied cycles, part-load operation

Part III of this report deals with the description of the part-load operation of the studied cycle. The systems under consideration are meant to bring improvements to the CO_2 vapour-compression cycles, especially when operating in hot climates.
All these solutions (mechanical subcooling, PCE and the use of an ejector for expansion power recovery) were proved to have higher performances than the 1-stage vapour-compression reference cycle, when operating at design conditions (see chapter 10 for the design operation performances of the systems). As the reader surely remembers (chapter 4), those systems were tested under extremely hot conditions. Thus, it appears clear that the considered cycles are promising solutions to be exploited in warm locations, like South Europe.
The third part of this project tries to answer the following questions: are these cycles providing performances' improvements all along the year, respect the 1-stage cycle? Or are they better only for the design conditions assumed in the first part of the project? The part-load analysis allows for a simulation of the systems' behaviour when operating away from their design conditions. In this way, it is possible to perform an investigation about the systems' performances during a realistic year of operation. This is exactly what was performed in this third part of the project. The chosen location for the systems' operation was Rome, Italy.
The way all the systems were modelled to allow for the part-load analysis was already explained. This chapter is about a comparison of their performances along the operation year. Figure 24.1 presents the maximum cycle's temperature and the maximum optimal gas cooler pressure faced by the cycles during their year of operation. As expected, the highest optimal gas cooler pressure is faced by the 1-stage reference cycle. All the improved systems allow for a reduction of the maximum (optimal) gas cooler pressure, the greatest reduction is provided by the mechanical subcooling cycle. Anyway, the reduction is quite small and all the cycles offers very similar behaviour from this point of view. A similar observation can be made for the maximum temperature in the cycle. Again, the highest temperature observed along the year is faced by the reference cycle. All the remaining cycles allows for a similar reduction of it, the greatest reduction is provided by the ejector cycle. These results reflect what it was found in chapter 10 for the design operation of the cycle and this makes perfectly sense. The highest temperatures and gas cooler pressures are faced in the summer months, when the ambient

temperature are at their maximum. These are exactly the conditions studied during the design operation, in fact high ambient temperatures were assumed.
Figure 24.2 and 24.3 present the average values of compressor power and COP faced by the systems along the year. The computational simulations allowed for an hourly investigation of the system operation. The yearly averages of these values are shown in the two cited figures. Average values are quite interesting because they allow the reader to have a taste of the systems' performances in part-load operation. In fact, if one cycle has higher average COP than another one, this represents a big clue that the first cycle is more efficient than the second, when operating along a real year. As visible from figure 24.2, the average compressor power is quite similar for the considered cycles. Anyway, the 1-stage reference cycle presents the higher values, while the lowest is the one of the ejector cycle. Figure 24.3 is probably the most interesting because summarises the systems' performances with one number: the COP. Actually, the figure presents the yearly-averaged COP. The ejector cycle allows, on average, for a performances improvements of 27% respect the 1-stage cycle. Really good performances are offered also by the part-load operation of the PCE systems. On average, the improvements it provides are around 21%. The cycles with the lowest performances, among the improved systems, are the two mechanical subcooling cycle. Anyway, even these systems allow for a considerable improvement of the COP, on average, about 17-18% respect the reference cycle.
Based on the value of the compressor power for each hour of operation along the year, the yearly energy consumptions of each system were calculated. Figure 24.4 presents graphically the results. Of course the highest energy consumptions are faced by the 1-stage reference cycle (the purpose of this project is precisely to improve the performances of this cycle). All the improved cycles offers a similar reduction of the energy consumption. The lowest reduction is provided by the two mechanical subcooling cycles. Once again, they have similar performances and they ensure a reduction of the consumptions around 17% respect the reference cycle. The ejector cycle offers a reduction of the consumption of 21%, the highest energy saving among the considered cycles. The PCE system, as already proved along this report, is the cycle with closest performances to the ejector one. 19% of energy saving is allowed by the PCE system.
Figure 24.5 presents the daily compressor power profiles for the different cycles. One representative day for each season was considered. Having a first general look to the graphs, it is clear that the scale is very different among them. Going towards the summer months the consumptions are higher. Regardless which cycle is considered, high ambient temperature brings higher energy input requirements. The consumptions are lowered down again during autumn, when the ambient is colder. No matter which season it is chosen, the ejector cycle presents the lowest compressor power consumptions. The energy savings allowed by this system during cold and medium-temperature months are considerably higher than the rest of the cycles considered. It is visible in figure 24.5 the gap between the ejector profiles and the other profiles for January, April and October. During the hot months, all the improved systems performs similarly and this gap is almost impossible to notice in the figure. Anyway, as already said, the ejector cycle offers slighlty higher energy savings.
The different behaviour of the systems depending from the ambient temperature is quite

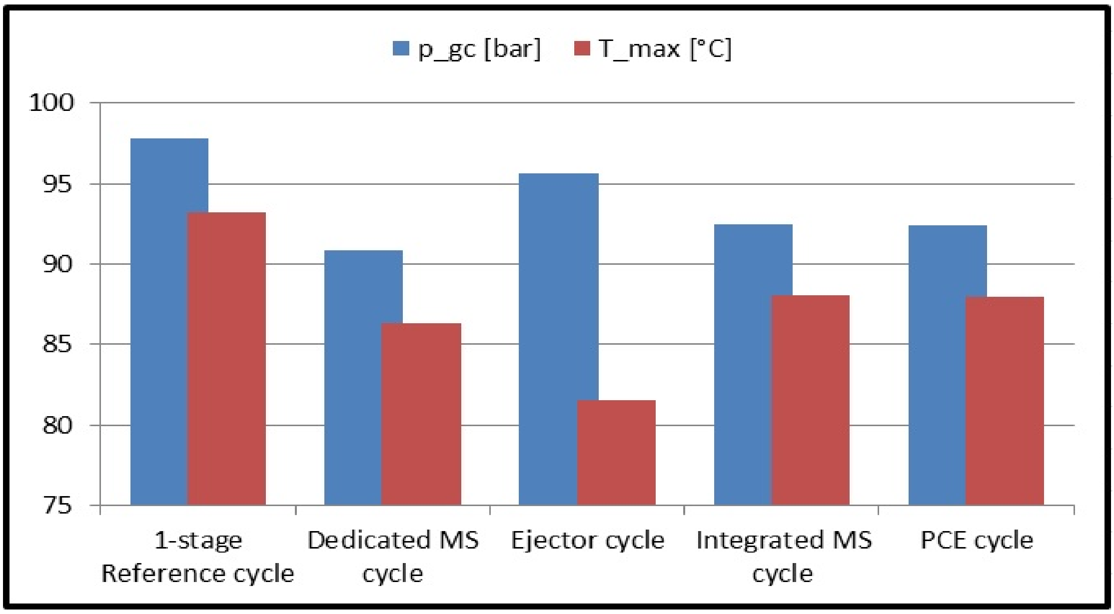

Figure 24.1 – Maximum cycles' temperature and optimal gas cooler pressure for the different cycles

interesting and it is further developed with the help of figure 24.6. In this figure, the energy consumptions for the different cycles, in January and August, are compared. These two months were chosen because they include the lowest and highest ambient temperatures along the year. Once again, the figure shows that the energy required is higher when the ambient temperature is higher. Anyway, this figure is more interesting from the savings point of view. All the cycles provides a greater reduction of the energy consumptions in August. In other words, when the system is strongly negatively affected by the high ambient temperature, the improved cycles perform even better, ensuring reduction of the consumption around 20%. Once again, the ejector cycle is the one that provides the greatest energy saving but the difference respect the other cycles is really small. If looking at the January's consumptions, the savings provided by the improved cycles are much lower, except for the ejector cycle. Even if the ejector works better with high ambient temperatures, the reduction in energy consumptions provided in January is only slightly lower than the one provided in August. This is for sure a strong point for the ejector cycle: it ensures a performances boost all along the year. On the other hand, the other improved cycles present a considerable difference in the energy input reduction during the operation year.

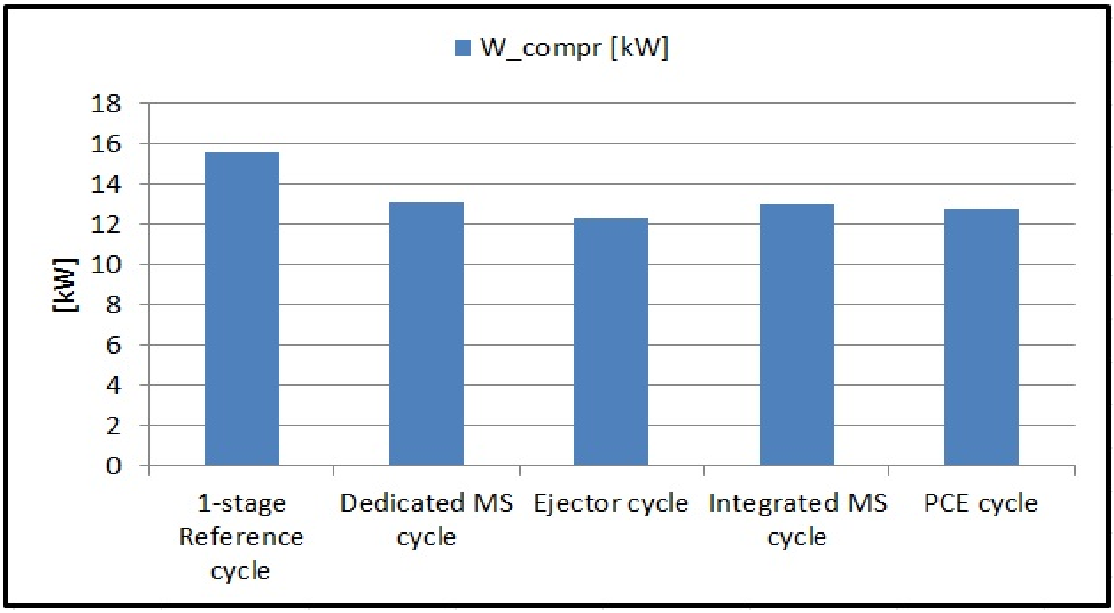

Figure 24.2 – Yearly-averaged value of the compressor power for the different cycles

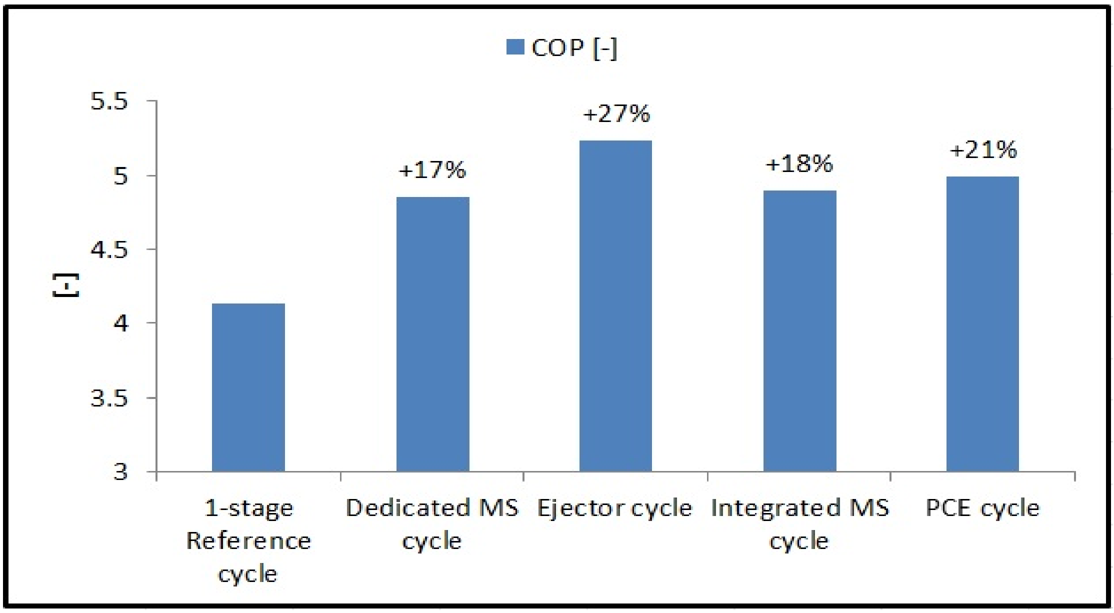

Figure 24.3 – Yearly-averaged value of the COP for the different cycles

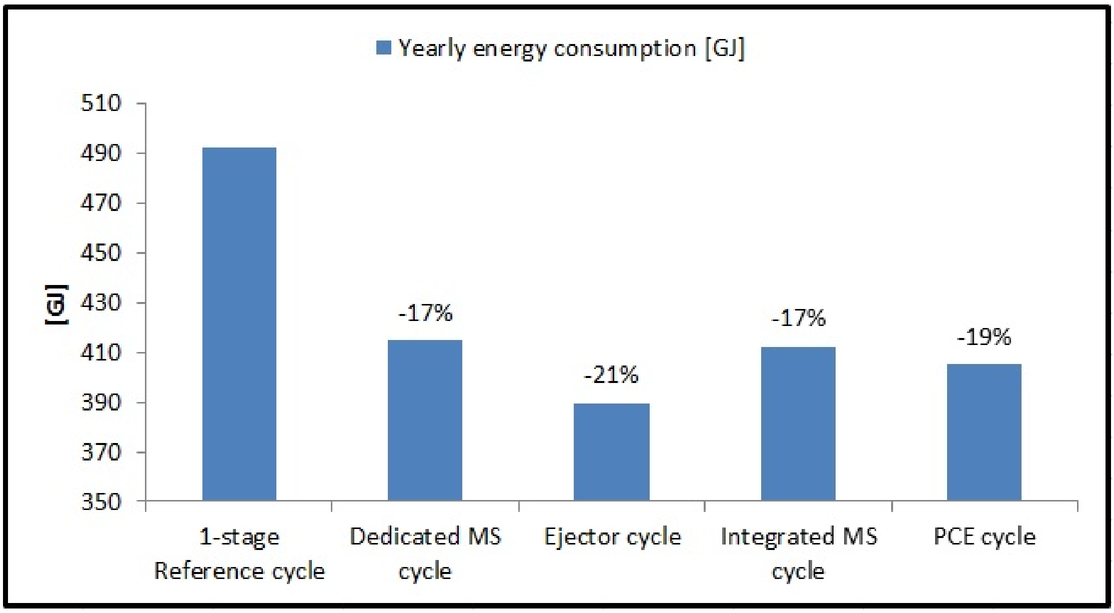

Figure 24.4 – Yearly energy consumptions for the different cycles

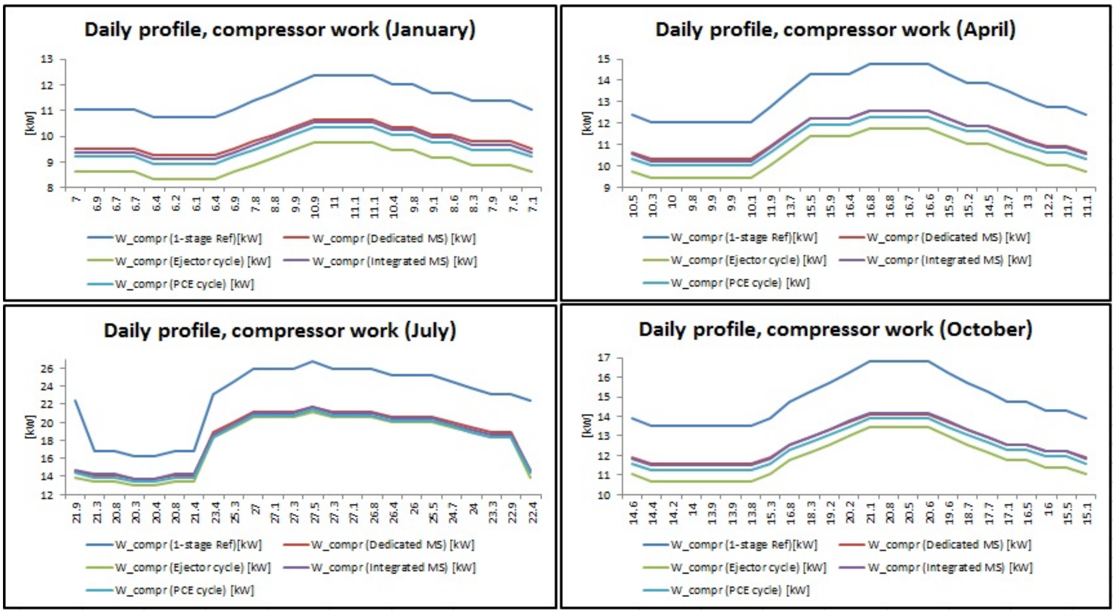

Figure 24.5 – Comparison of the daily compressor power profiles for the different cycles, one representative day for each season was considered

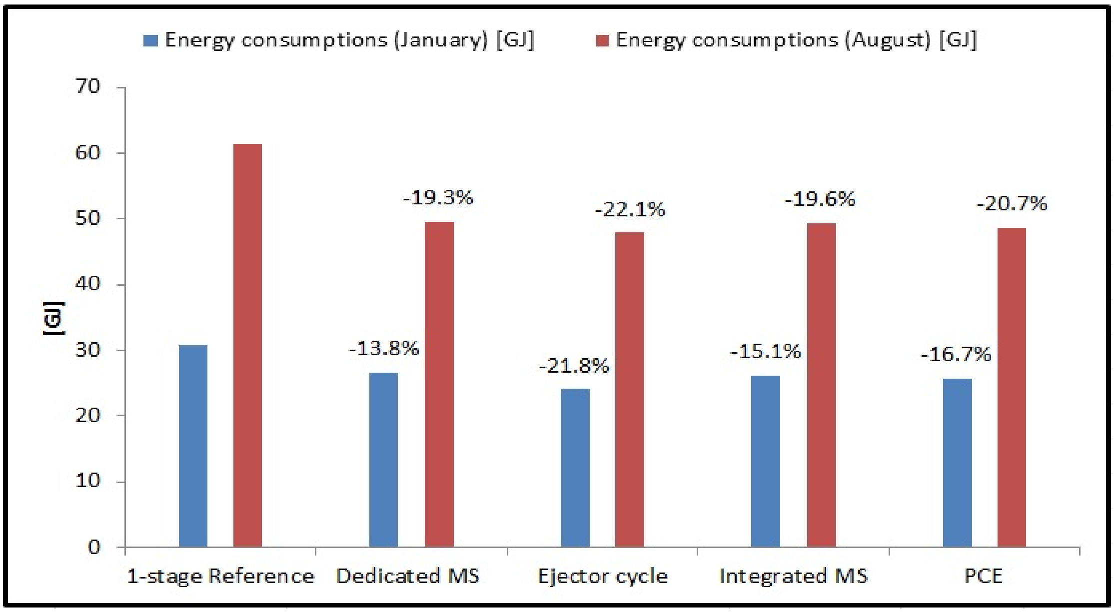

Figure 24.6 – Comparison of the monthly energy consumptions of January and August for the different cycles

25 Part-load operation using real compressors and real cooling capacity profile

The results presented in chapter 24 allow for a comparison of the part-load behaviour of the improved cycles. Even if the models are based on quite strong assumptions, like constant isentropic efficiency of the compressors, the trends and the observations proposed represent a very good base to understand the real-life cycles' behaviours.
In this chapter, a bit more realistic models of the cycles are proposed. The cooling capacity is assumed as changing during the day and the compressors are characterized by varying isentropic efficiencies. The behaviour of the systems is not different from what it was shown in chapter 24, most of the graphs and trends proposed are the same, just the number are different (hopefully more close to the one of real cycles). For this reason, many figure proposed in the previous chapter are not repeated here.
As already said, the main changes in the cycle's models concern the cooling capacity and the η_{is} of the compressors. During all the investigations performed and presented so far, the cooling capacity of the system was fixed to 60kW. In the analysis here proposed, this value is kept for the day operation of the system. The cycles are assumed to be exploited for supermarket applications. The shops are assumed to be open from 7am to 9pm, during these hours the required cooling capacity is assumed to be 60kW. During the hours the shops are closed, a reduced value of the cooling capacity was assumed. In fact, during night, the refrigerators and freezers are not open and closed continuously. Consequently the load of the system is lower and the required output from the refrigeration system is lower. This is the same approach used by Minetto et al. in their yearly consumptions investigation of CO_2 supermarket refrigeration systems [3].
If the change regarding the cooling capacity is quite a small one, the big improvement here proposed is the use of real compressors in the systems. To collect the required informations about real compressors, the web-database made available by the German company Bitzer was used [52]. In their on-line software, they propose a great variety of compressors for different application, each one with detailed information about the component's behaviour. The type of compressor chosen is semi-hermetic reciprocating one. This choice was forced by the lack of alternatives for the considered application and refrigerant. The chosen compressor model is 4MTC-10K. It allows both subcritical and transcritical operation, moreover the great variety

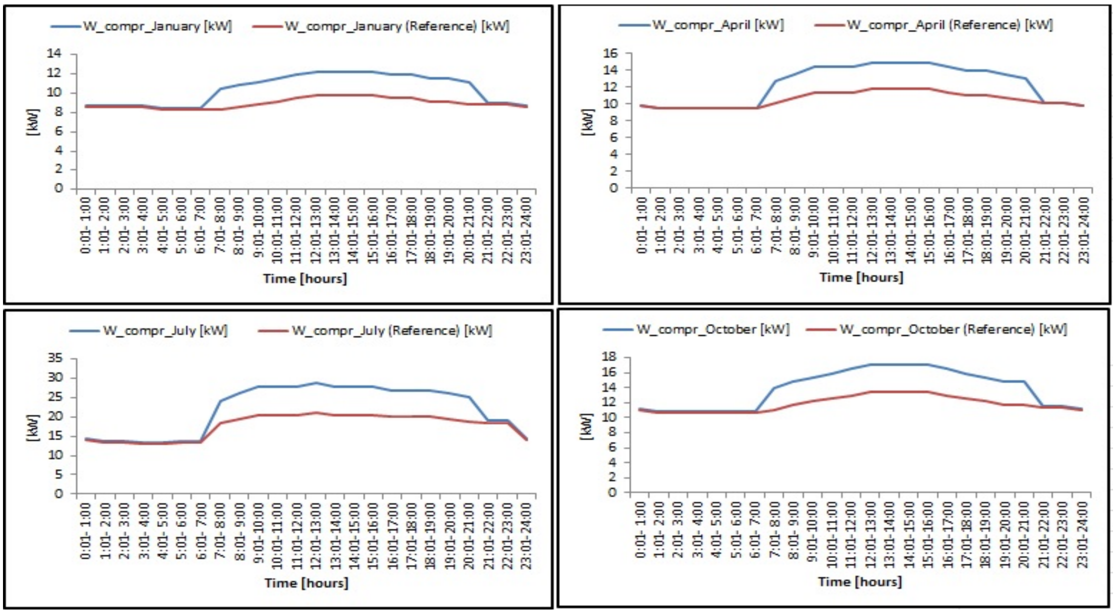

Figure 25.1 – Daily compressor power profiles for the ejector cycle

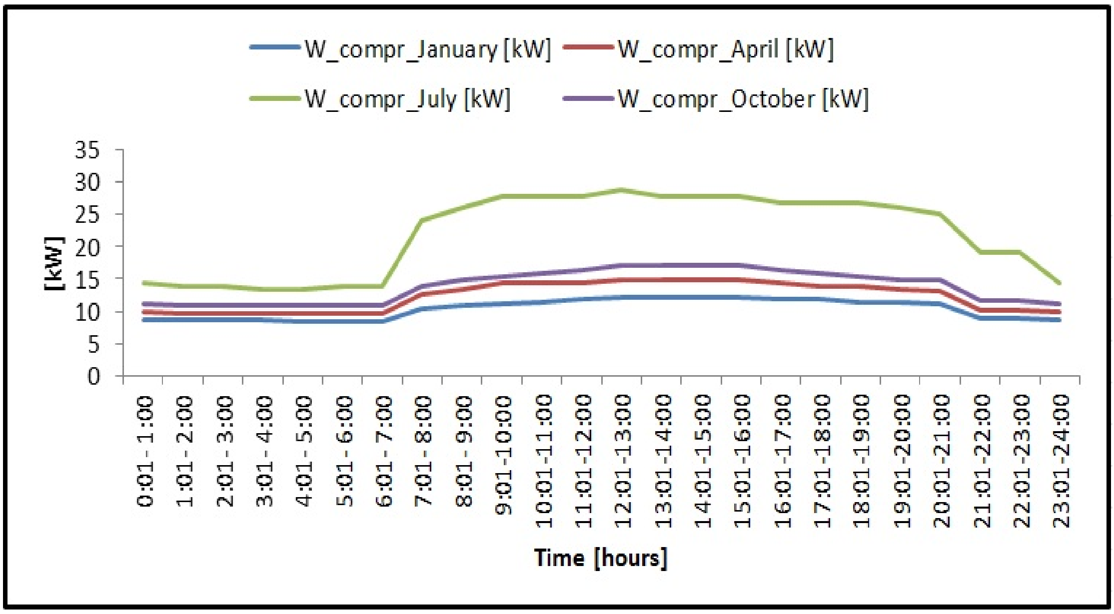

Figure 25.2 – Comparison of the daily compressor power profiles for different months, ejector cycle

of pressure ratio possible inside its working envelope pushed the author towards the choice of this compressor. Two small database were built in order to represent in the models the behaviour of this compressor: one for subcritical operation and one for transcritical one. For the subcritical operation, the isentropic efficiency of the compressor for different evaporation and condensation pressure was stored. In this way, it was possible to built correlations between isentropic efficiency and condenser pressure, for different constant evaporation pressures. In the same way, it was dealt with the transcritical operation. It was possible to built correlations between isentropic efficiency and gas cooler pressure, for many constant evaporation pressure. In the cycles' models, on the base of the lower compressor pressure (the evaporation pressure set in the Bitzer software) and the operating conditions (subcritical or transcritical), the right correlation to represent the isentropic efficiency was chosen. It was proved with some simulations that this approximation allows for a very good representation of the real compressor for different pressure levels and pressure ratios.
The models adjustments used in this more accurate investigation of the systems' behaviour are only the ones just explained. The remaining parts of the cycles were modelled as already explained in chapter 18.
The daily compressor power profiles are not here presented again for each cycle. The shape of these profile is similar to the ones already seen in the previous chapters, only the absolute value of the compressor power changes. As a proof of this, the compressor power profiles for the ejector cycle (and not for the other systems!) are here reported. Figure 25.1 presents these daily profiles for one representative day from each season. Along with them, the profiles coming from the previous part-load investigation are reported to provide a benchmark for some observations. It is visible how the shape of the two profiles presented in each graph is really similar. The only difference are the higher consumptions during the working hours of the shops. This is due to the fact that, for the same requirements in cooling capacity, the systems has much lower compressors' isentropic efficiencies. The previously assumed η_{is} was 0.9, while the real values of this parameter ranges between 0.6 and 0.7 for the chosen compressor. If the consumptions are now much higher during the working hours, if compared with the constant compressor's isentropic efficiency analysis, the same cannot be said for the night hours. For those hours the consumption are quite similar, regardless the month considered. This is due to the fact that the lower cooling capacity (required to the system during night hours) compensates for the lower isentropic efficiencies of the compressor.
Figure 25.2 shows the just observed ejector cycle's profile sorted all in the same graph. In this way it is possible to see that the behaviour of vapour-compression cycles does not change even if real components models are introduced in the system. Still, the months with the higher ambient temperature are the ones with the highest energy consumptions.

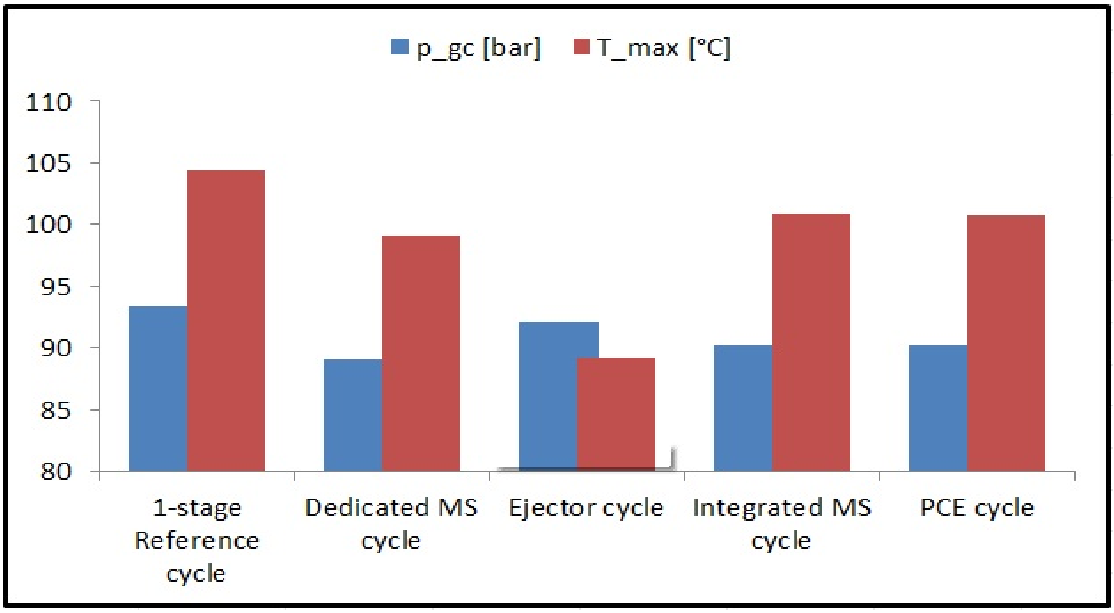

Figure 25.3 – Comparison of the maximum cycle's temperature and optimal gas cooler pressure for the different cycles

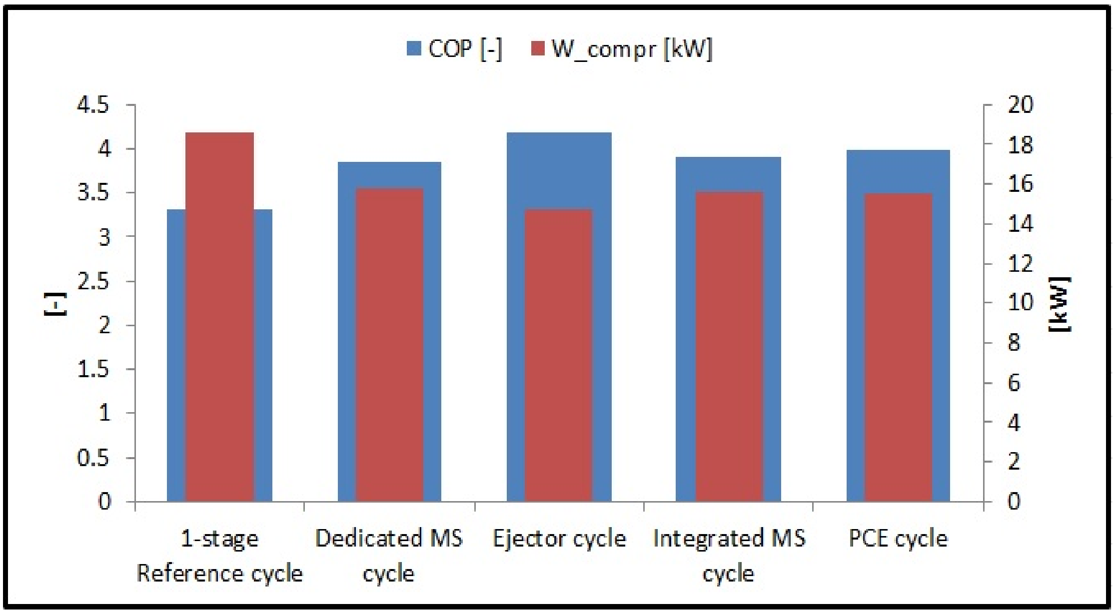

Figure 25.4 – Comparison of COP and compressor power for the different cycles

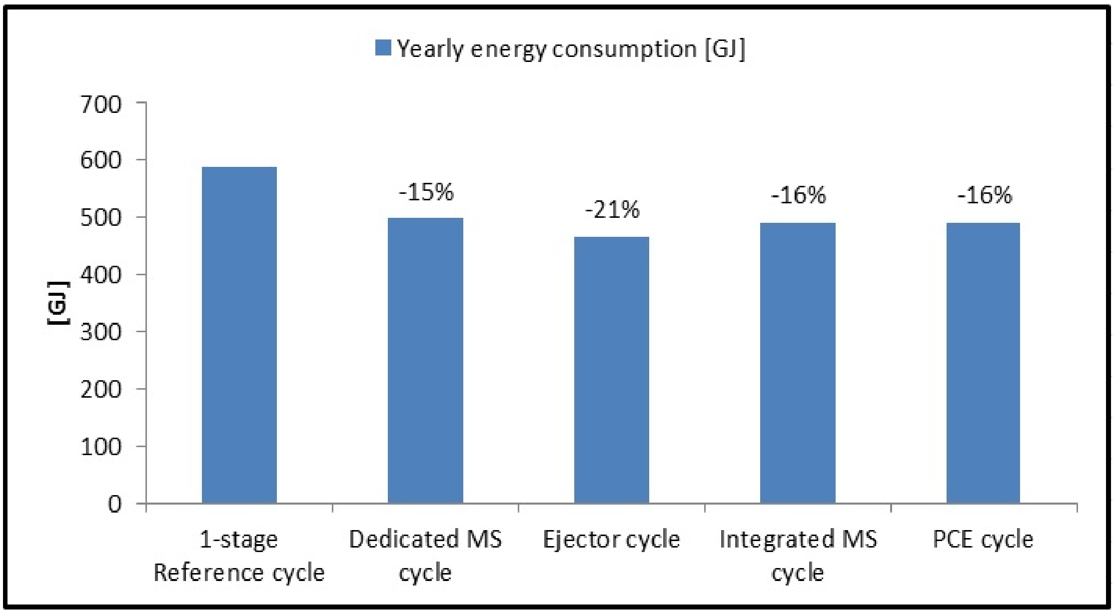

Figure 25.5 – Comparison of yearly energy consumptions for the different cycles

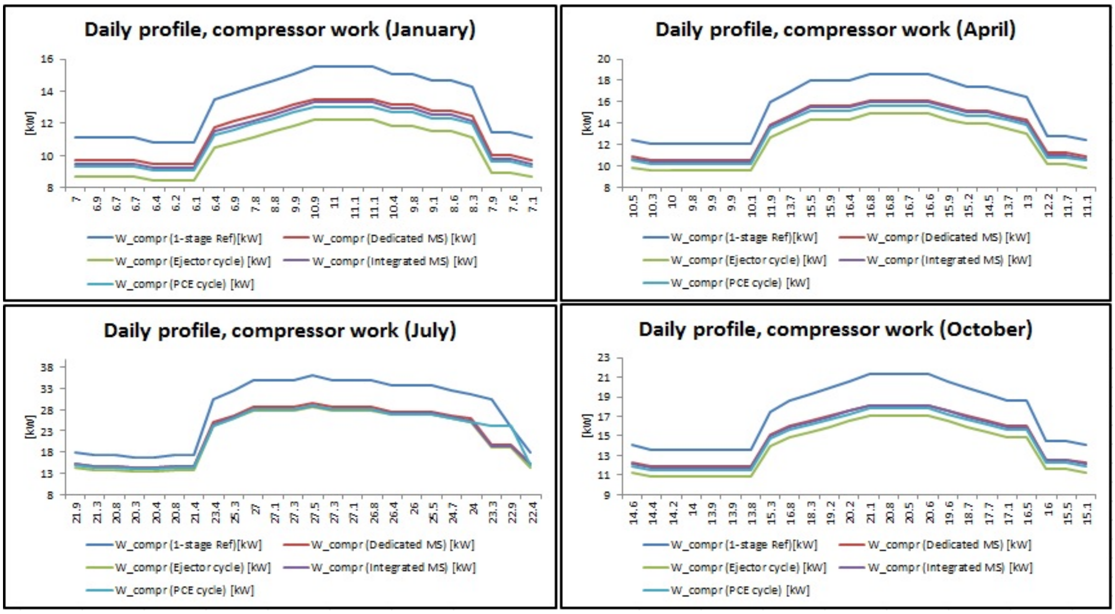

Figure 25.6 – Comparison of daily compressor power profiles for the different cycles, considering one representative day for each season

25.1 Comparison of the cycles behaviour

This section provides a comparison of the improved CO_2 cycles, under the considered (more accurate) assumptions. Figure 25.3 is the first here proposed. It represents the maximum value for the cycle's temperature and optimal gas cooler pressure faced by the systems during a typical year of operation in Rome, Italy. As visible, the dedicated mechanical subcooling cycle provides the lowest between the maximum optimal gas cooler pressures, while the 1-stage reference cycle presents the highest. Anyway, this difference is very small, no significant improvement is faced from this point of view. The maximum cycle's temperature is considerably reduced by mean of the ejector cycle. From this point of view, the ejector cycle offers quite an improvement if compared with the other cycles. The remaining systems present almost the same value for the maximum temperature faced in the cycle.
Figure 25.4 depicts the cycles' COP and compressor power. As expected the two parameters have opposite trends, if the compressor power is high for one cycle, the corresponding COP is low. The system with the worst performances is the 1-stage reference cycle, while the one with the best ones is the ejector cycle, with a COP boost around 26%. A great improvement is offered also by the PCE cycle, with a COP improvement of 20%. The mechanical subcooling cycles offer the smallest increase of performances, around 17% each.
The results are completely aligned with the ones from chapter 24. Due to the realistic behaviour of compressors and cooling load, all the cycles lose around 1% of their COP improvement respect the 1-stage reference system. Anyway the difference is almost negligible. The "cycles' ranking" defined in chapter 24 (from the best to the worst cycle in terms of COP) still holds up even adding more accurate assumptions.
From the yearly energy consumptions point of view, some differences can be observed respect what found in chapter 24. As visible in figure 25.5, the two mechanical subcooling cycles and the PCE one present almost the same reduction of the consumptions. They all ensure a decrease around 16%. This is quite interesting because the PCE cycle seemed to be a higher performances system from the investigations made so far in this project. Anyway, the reason for this change of behaviour lays in the use of a real compressor. The same compressor was used in all the cycles, the different performances of this component for different pressure ratio and pressure level is probably the reason of the increased consumptions in the PCE cycle. Using a more proper compressor for the application, the PCE cycle would still show highest performances respect the two subcooling cycles. The ejector cycle is not considerably affected by the use of a real compressor. The yearly energy consumption reductions it provides is around 21%. It is still the cycle with the best performances and this reduction is the same found in chapter 24.
Figure 25.6 presents the daily compressor power profiles for the considered cycles, one representative day for each season is reported. If compared with the one from chapter 24, the figure shows highest consumptions for the considered months. As already said, the real compressors have isentropic efficiencies much smaller than the 0.9 assumed in the previous chapters. This represents the main reason for the higher consumptions. Regardless the month considered, the 1-stage reference cycle is the one with the higher consumptions, while the ejector one has

the lowest. As already observed in chapter 24, the ejector cycle performs much better than the others for all the cold and temperate months, while the other improved cycles offers a performances boost smaller during the cold months. On the other hand, all the improved cycles offer almost the same improvement when considering the hottest months of the year. This is visible from figure 25.6, in July the consumptions of the boosted cycles are almost the same, the different profiles are overlapping and difficult to distinguish.

This chapter offers an interesting investigation of mechanical subcooling, PCE and ejector cycles under more realistic assumptions. Some differences respect the findings of chapter 24 where observed during this analysis. The main one observed regards the yearly energy consumptions of the PCE cycle, now identical to the ones of the two mechanical subcooling cycles. Anyway, these observed differences are not huge ones and they are all connected to the assumptions made. The use of the same compressor for all the cycle is not a fair assumption. Different applications require specific compressors that fit with the system's requirements. As already said, if each cycle is equipped with a proper compressor, able to maximize the performances of the cycle for the specific application, the "cycles' ranking" of chapter 24 would be found.
When looking at the results from chapter 24, the only thing to keep in mind is that the consumptions of a real cycle are higher, due to the use of real compressors, with losses and inefficiencies. The effect of even more complicated cooling load profiles can be studied, anyway, for weather conditions similar to the Rome's ones, the ejector cycle is surely the most promising one.

26 Real cycles implementation

The part-load operation implemented in the previous chapters gives a perfect overview of the systems' performances under a real year of operation. Interesting findings were extracted from this analysis. The considered cycles (mechanical subcooling , PCE and ejector cycle) offer considerable improvements of the performances respect the 1-stage cycle, if used under hot weather conditions like the Rome's ones. The ejector cycle resulted to be the one with the highest reduction in the yearly energy consumptions, under the considered working conditions. Anyway, the considered analysis cannot considered completely realistic, specifically due to one strong assumptions made for the EERC.

When investigating the part-load operation of the ejector, whatever separator pressure was assumed as allowed. Anyway, this is not a realistic assumption. In particular for low ambient temperatures, the pressure recovery performed by the ejector is very small and the pressure at the diffuser outlet is only slightly higher than the evaporation pressure. Consequently, the pressure difference across the expansion valve (separator pressure minus evaporation pressure) is not high enough to ensure a proper functioning of this component. Interesting was a conversation that the author had with representatives of the famous refrigeration systems' firm DANFOSS. This company offered a fundamental partnership for the realization of this project. Speaking about ejector cycles, they explained that they usually control the separator pressure in order to follow the pressure recovery performed by the ejector. Anyway, they do not allow the separator pressure to be less than 4 bar higher than the evaporation pressure. This regards the already introduced minimum pressure difference across the expansion valve that allows a proper functioning of the component.

This control on the separator pressure was not performed during the part-load simulations presented in the previous chapters. The purpose of this final chapter is to present even more realistic results, introducing this control in the ejector cycle. This means that the ejector cycle cannot be used below certain ambient temperatures because the pressure recovery is insufficient. In this cases, the ejector cannot longer be used and it is considered a switch from the ejector cycle to PCE or mechanical subcooling one. It is assumed that the system is provided of a valves set able to perform this switch: the ejector is by-passed and the secondary loop (PCE or mechanical subcooling one) is used.

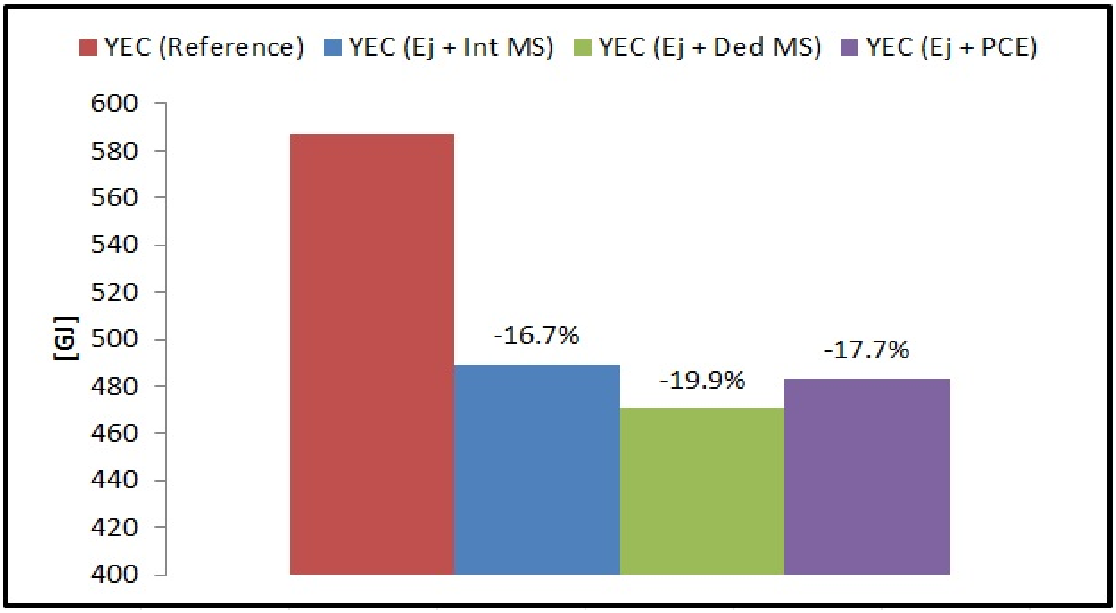

Figure 26.1 – Yearly energy consumptions for the considered cycles

For this analysis, the models from chapter 25 were used. Due to the fact that a real ejector cycle's control is here introduced, the choice was to use also the more accurate systems results coming from the use of real compressors. No more simulations were required to perform this final investigation. Simply some spread sheets were used to combine together the results obtained for the ejector cycle and the others. The switch from the ejector cycle to the "back-up" one was assumed to happen when the separator pressure falls below $p_{evap} + 4\ bar$.

26.1 Comparison of the real systems behaviours

Three cases are here considered. The common point of this systems is the use of an ejector cycle for expansion power recovery at high ambient temperature. When the pressure recovery is not high enough for a proper operation of the expansion valve, a "back-up cycle" is used:

- ejector cycle + integrated mechanical subcooling cycle: the back-up cycle is the integrated mechanical subcooling one. When the ambient temperature are too low for the ejector cycle, the ejector is by-passed and an integrated mechanical subcooling cycle is use instead. The behaviour of the other systems here considered is exactly the same, only the used back-up cycle changes.

- ejector cycle + dedicated mechanical subcooling cycle

- ejector cycle + PCE cycle

Figure 26.1 presents the yearly energy consumptions for the considered cycles. As visible, all the solutions here adopted to improve the 1-stage cycle actually perform their job. The yearly energy consumptions are reduced, regardless which one of the improved cycle is considered. Moreover, the consumptions reduction they realize is almost the same, they deviate each other of 3% at most. Between this cycle, the one with the highest performances seems to be the ejector cycle using a dedicated mechanical subcooling cycle as "back-up" cycle. Anyway, it will not be put to much stress on it. The slightly better performances of this cycle are connected to the compressor's choice. In fact, the chosen one fits better with the requirements of this system and a bit worse with the ones of the others. The choice of a proper compressor could be beneficial for all the cycles, especially the PCE one. In this situation, the ejector cycle using PCE as back-up, could be the one with highest performances (as the reader would expect due to the considerations from the previous parts of this report).

27 Conclusion

The refrigerants the cold industry have relied on for the last fifty years are arising an increasing concern about their environmental impact. CFCs and HCFCs are already abandoned solutions due to their worrying ODP. Even the HFCs, introduced as a replacement of the former, are now object of international political actions to reduce their utilization. The HFCs actually reduce their impact on the ozone layer but they raise another current (and great) issue: the global warming. During the last decades, the refrigeration industry was forced to face the increasing concern and to embrace the changes required. The "natural refrigerants" were rediscovered due to their low environmental impact, especially CO_2 that seems the most suitable among them.

This report is focused on CO_2 applications in the refrigeration industry. As many of the other "natural refrigerants", CO_2 has negligible ODP and GWP, moreover presents good refrigerant's characteristics. The main problem connected with carbon dioxide is the low critical point. For application in cold/temperate climates this does not represent a problem. Due to the low ambient temperature, the system runs mainly subcritical with competitive performances respect the "traditional refrigerants". $CO_2 - only$ cycles arise issue when applied in hot climates, due to the fact that the cycle is forced to release heat over the refrigerant's critical point. Transcritical CO_2 cycles are strongly penalized from the performances point of view. The expansion process in particular, due to the great pressure difference throttled, is cause of considerable irreversibility that tears down the system's COP.

Three main solutions were here proposed in order to improve the performances of CO_2 transcritical cycles: mechanical subcooling, PCE and ejector-expansion device. These expedients were accurately investigated to estimate their actual impact on the system's COP.

In the first part of the project, the boosted systems were compared under the same fixed design conditions. The cycles were considered as installed in a very hot location, in fact the assumed gas cooler outlet temperature was set to 40°C. It was found that all the studied cycles offers considerable COP improvements. Mechanical subcooling and PCE increase the systems COP around 22% respect the 1-stage reference cycle. The ejector cycle was discovered to be the one with the higher performances boost for the given working conditions, the COP increases around 26%.

In the second part of the project more complex systems improvements were studied, always in order to improve the performances of carbon dioxide systems. Many of the solution adopted were built combining together the devices studied in the first part of the report. Anyway, also alternative cycle's improvements were studied, as the use of an expander instead the traditional expansion valve or the use of a heat-driven dedicated subcooling loop. The assumed working conditions were still fixed and identical to the one used in the first part of the project. The PCE cycle with additional dedicated MS loop was discovered to give a considerable boost to the systems: the COP raises around 30% respect the 1-stage reference cycle and 8% respect the "traditional" PCE system.
It was discovered that the most efficient subcooling is performed by the EPR dedicated MS cycle. The systems faces a COP improvements around 47% respect the 1-stage cycle and around 25% respect the "classic" dedicated MS cycle. Of course, the increased investment cost faced when introducing an expander in the cycle must be taken into account.
Very interesting results were obtained also from the ejector cycle with additional MS loop. The COP, respect the reference vapour-compression 1-stage cycle, raise around 37%.
The first two parts of the report offer a great overview of the possible solution adoptable to improve the performances of CO_2 transcritical systems, especially when applied in hot climates. Anyway, rarely a refrigeration system works under the same ambient conditions all along the year. Considering a typical air-cooled system, the ambient air represents the heat sink of the cycle. In this way, the ambient conditions affect considerably the system operation. This is particularly important in CO_2 cycles because high ambient temperatures force the system to work many hours as transcritical, lowering down its performances.
The last part of the project is precisely a part-load investigation of the improved cycles: PCE, mechanical subcooling and ejector cycle. The cycles were studied under the real weather conditions of Rome, Italy. This location is particularly suitable for the study here conducted because of the high ambient temperatures reached for many months of the year.
Even when considering the off-design operation, the ejector cycle is still the most promising solution. The COP boost that it provides is around 27%, respect the 1-stage cycle (value based on the yearly-averaged COP). Differently from what it was seen in the design investigation of the systems, the PCE cycle presents an yearly average COP higher than the ones of the two mechanical subcooling cycle. The improvement provided by the PCE system is around 21% (in terms of COP), against the 17% offered by the MS cycles. Using the ejector cycle, the yearly energy consumptions can be reduced of 21% respect the reference case. Interesting results are obtained also from the other systems: the energy required is lowered down around 19% by the PCE cycle and 17% by the MS systems.
The part-load analysis completes the study started in the first two parts of the project. The ejector cycle, the most promising at the specific design conditions assumed initially, is still a very interesting solution when analysed for an entire year of operation in a warm climate. The PCE cycle, surprisingly, performs even better than expected when considering its off-design operation. This solution seems to be slightly better that the two MS cycles for a real-year of operation.
The report is concluded with a more realistic study of the considered cycle, introducing real

compressors and real cooling load profiles. The results reflect what it was found during the first part-load analysis, only the absolute value of the consumptions are a bit higher for all the cycles (due to the lower isentropic efficiency of the compressors).

The project offers an important review of the CO_2 cycles improvements. The design conditions analysis and the part-load study clearly state the effectiveness of the studied solution when used to boost hot-climate CO_2 refrigeration systems. A considerable number of studies are already available on the topic, the first test application already started in many countries of the South Europe. The way is already paved for a massive introduction on the market of these solutions, to finally reduce the human footprint on our precious Earth.

28 Further work

This report represents a good overview of the most promising solutions to boost CO_2 transcritical cycles. These tools are applied to a generic vapour-compression refrigeration cycle, in order to evaluate their potential without complicating to much the investigation. The results are promising, either if considering mechanical subcooling, parallel compression or ejector cycle.

To complete the work here started, some other investigation are required. Each one of the considered cycle's improvements should be tested for specific applications, from reefer containers to supermarket refrigeration. The results presented in this report suggest the effectiveness of the considered boosts but still they must be tested for different operating conditions and cycles' configurations. Some, more detailed, simulations should be performed, including all the details here neglected. Heat losses and/or pressure drop can affect considerably the system performances. The impact of the fan power on the yearly consumptions can be analysed and the control strategy can be optimized to increase the savings.

Some experimental results are needed to support the findings presented in this report. The part-load analysis offers an interesting overview of the yearly performances of the cycle but surely not compared to what would be real-cycles' data.

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A Appendix A

In this project, the chosen ejector model is the one proposed by Kornhauser in 1990 [35]. In his paper, Kornhauser gives an accurate description about how to model a constant-pressure mixing ejector. See chapter 9 for the model description.

This model was chosen mainly for its simplicity. The behaviour of the ejector cycle is well approximated by the Kornhauser's model, even if the coding required is quite easy. The model proposed by Kornhauser is one of the most spread in the ejector cycle's literature, mainly because well-performing and simple.

Going through the literature, only another model is as widely used as the Kornhauser's one. This model represents a constant-area mixing ejector and it is well explained by Cen et al. in their work from 2012 [48]. The reader is recommended to see their paper for more information about the coding proposed by the authors. Also this model offers reliable results and quite simple coding, respect other models proposed in the literature. For this reason, it is significantly employed for ejector cycles' studies.

Anyway, when comparing the two models, Cen's and Kornhauser's one, the former adds considerable more complexity when setting it up in the software EES. This additional complexity creates converging difficulties when performing the already complicated (and thus delicate) part-load analysis. On the other hand, the simplicity of the Kornhauser's model allows for a more fluent investigation of the off-design operation of the ejector cycle. The choice made during this project was to award this peculiarity of the Kornhauser's model.

The selection of the proper ejector model was made on the base of the level of complexity, but it was not the only requirement. Many models were found in the literature, a lot of them were characterized by a high level of complexity that was considered too detailed for the aim of this project. Excluding these models, Cen's and Kornhauser's one remained (actually they are the most spread in the literature, papers that propose different models are really few). An investigation on this two models was performed and it helped during the decision process. It was discovered that the ejector proposed by Kornhauser is not only more simple than the one proposed by Cen, it also describes almost the same ejector's behaviour as the other one. Here, some results from the investigation performed on the two cycles are presented.

Figure A.1 presents a comparison, for different ambient temperatures, of COP resulting from

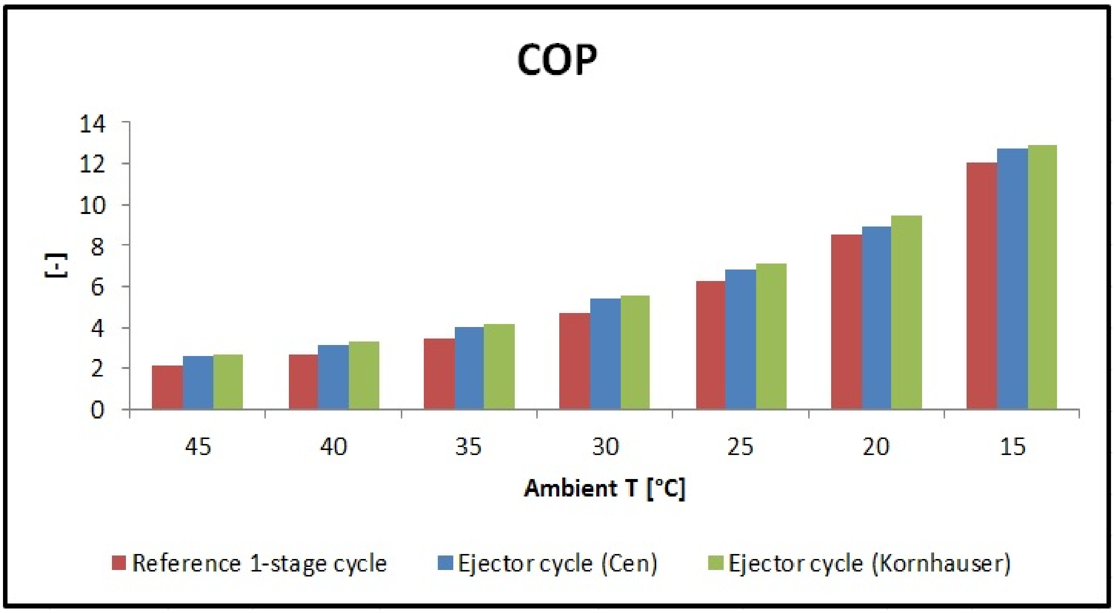

Figure A.1 – COP trends for different ambient temperatures: 1-stage reference cycle, ejector cycle using Cen model and ejector cycle using Kornhauser model

1-stage reference cycle and two ejector cycles, one using Cen's ejector model and the other using Kornhouser's one (of course all the cycles were simulated using CO_2 as refrigerant). As visible, the two ejector cycles offer higher performances respect the 1-stage cycle at all the considered ambient temperatures. The only difference between the two models seem to be the slightly lower COP offered by the use of the Cen model.
Figure A.2 and A.3 present a comparison of the trends for compressor work and separator pressure for the considered cycles. Again, it is visible that the two ejector's models represent almost in the same way the behaviour of this component and consequently of the overall cycle. The differences between the compressor power and separator pressure are negligible for all the considered ambient temperatures. Moreover, the observed trend is the same regardless which model is considered.

Due to the similar behaviour and performances of the two models, the Kornhauser's one was chosen because of its lower level of complexity.

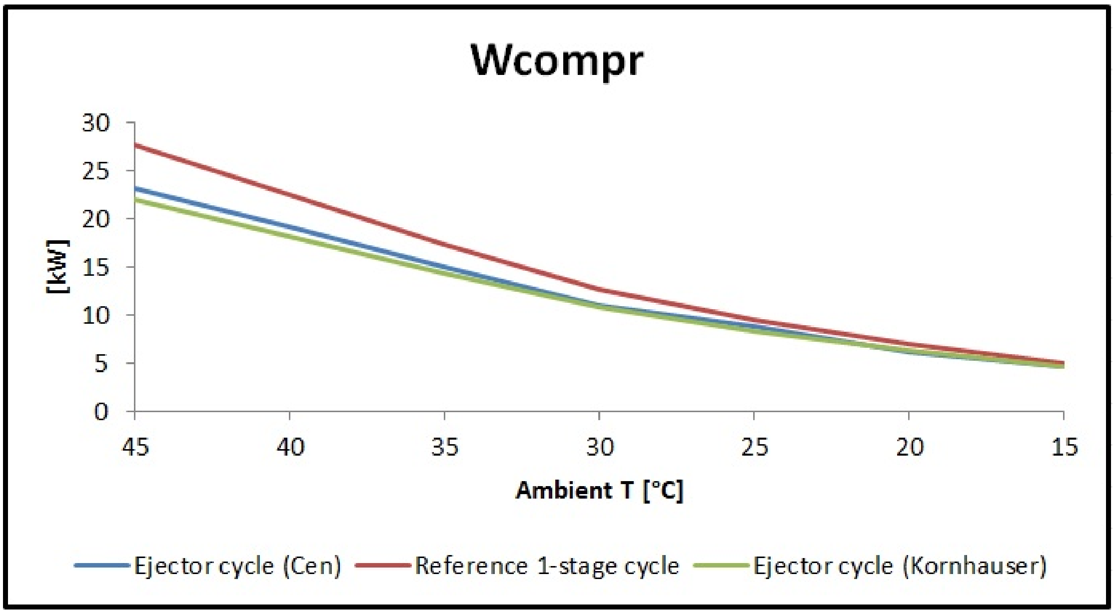

Figure A.2 – Compressor power trends for different ambient temperatures: 1-stage reference cycle, ejector cycle using Cen model and ejector cycle using Kornhauser model

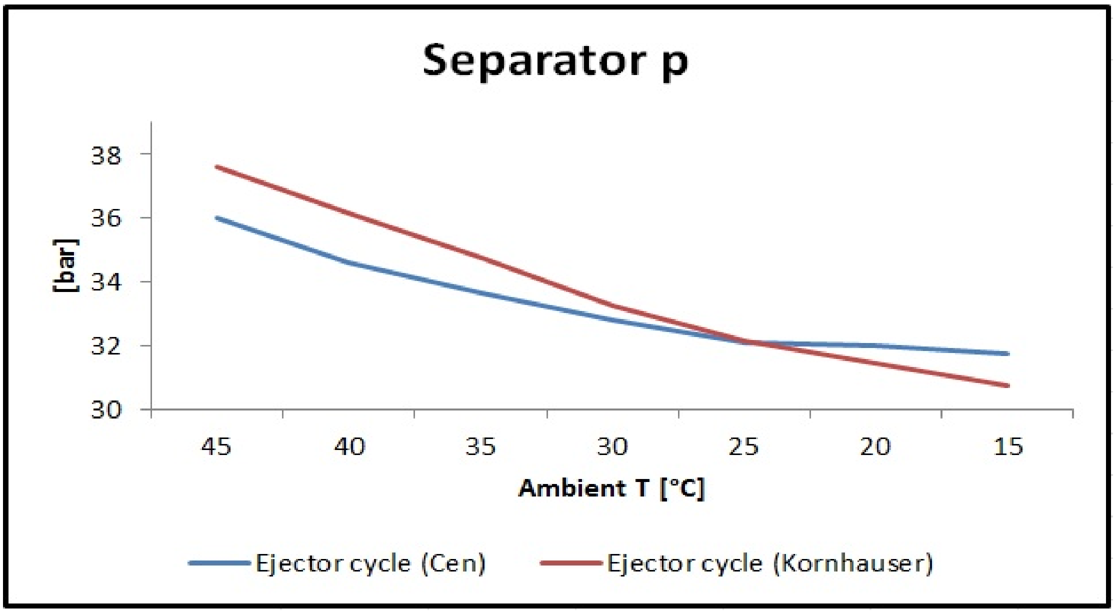

Figure A.3 – Separator pressure trends for different ambient temperatures: ejector cycle using Cen model and ejector cycle using Kornhauser model

B Appendix B

The part-load operation of the two mechanical subcooling cycles offered room for discussion about the modelling of the subcooler. The choice made was to model it considering fixed the temperature difference at one side of it.
Another possibility was to design this component in the same way as condenser, evaporator and gas cooler. A fixed UA-value for the design conditions of the subcooler has to be defined. After that, the part-load operation is determined using a simple correlation between off-design UA-value and off-design mass flow rate. See chapter 18 for a detailed explanation of this method, applied to gas cooler, evaporator and condenser.
An investigation about the two modelling possibilities was performed to help in the decision process. It was discovered that the two models give almost the same results. Figure B.1 and B.2 present a comparison of this two subcooler models applied for a yearly part-load investigation of the dedicated and integrated mechanical subcooling cycle. As visible, the differences in the considered parameters are almost impossible to observe in many cases!
As for the ejector model, the choice was to award the most simple model. The model using a design UA-value is much more complicated and generated convergence problems when studying the part-load operation of the cycle. On the other hand, the model using a fixed ΔT at one side of the gas cooler allows for a more fluent analysis of the off-design operation of the two mechanical subcooling cycles.

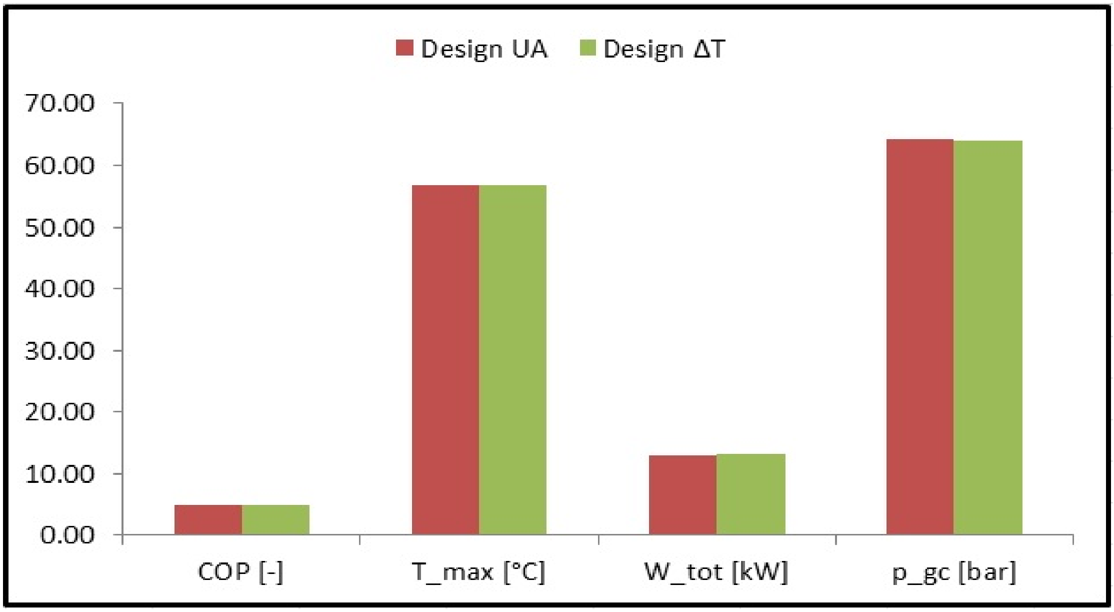

Figure B.1 – Comparison of the two subcooler models applied to the dedicated mechanical subcooling cycle, yearly part-load operation of the cycle

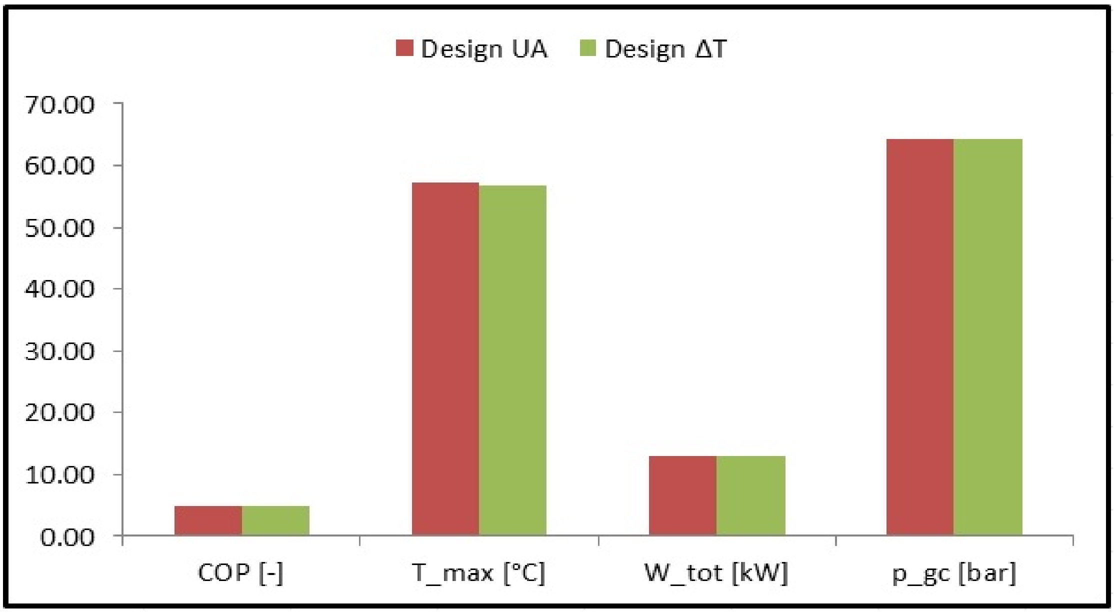

Figure B.2 – Comparison of the two subcooler models applied to the integrated mechanical subcooling cycle, yearly part-load operation of the cycle

C Appendix C

The dedicated MS cycle was introduced in chapter 8. Together with that, it was also introduced the possibility to use different refrigerants (than CO_2) in the subcooler loop. This project is about carbon dioxide systems, for this reason the focus was kept on this refrigerant. Anyway, for sake of knowledge, the impact of alternative substances in the subcooler loop was also studied.
The refrigerants taken into account are other environmentally friendly substances, characterized by low GWP levels. The results are presented in figure C.1 and, as visible, they are quite interesting. When CO_2 is used in the secondary loop, the COP gains are considerable, but still not as good as for the other refrigerants tested. All the alternative substances allow for a further improvement of the performances, around 16-17% respect the $CO_2 - only$ MS cycle. The $CO_2 - only$ MS cycle is not the proper solution to mitigate the carbon dioxide's drawbacks. From the practical point of view, alternative (low-impact) refrigerants must be used in the subcooler cycle, only in this way the system performances can be maximized.

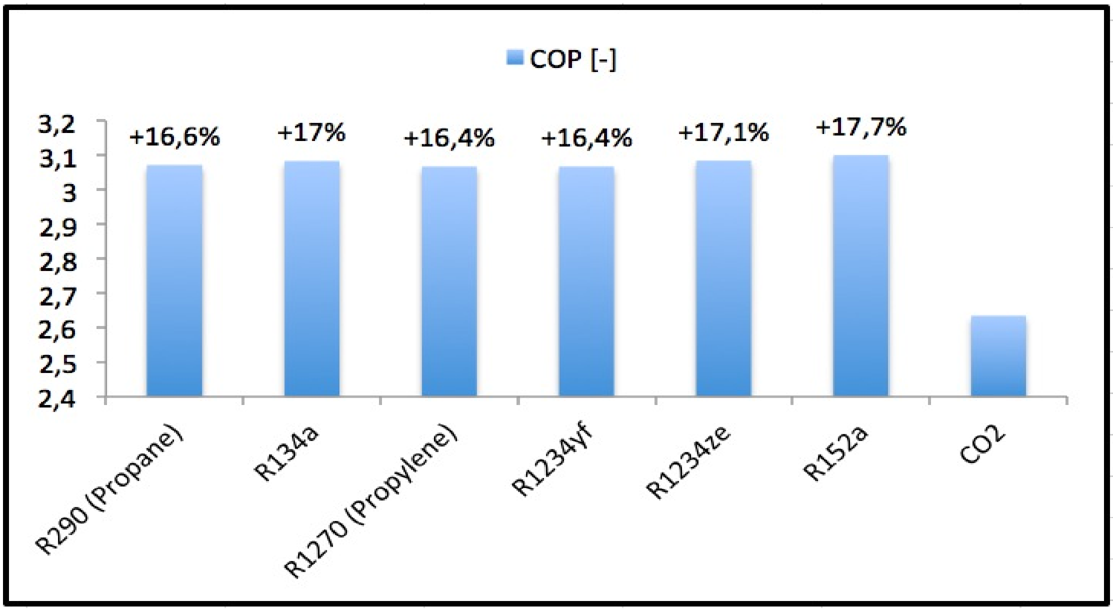

Figure C.1 – Impact on the MS cycle's performances when using alternative refrigerants in the subcooler loop

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