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E DELL'INFORMAZIONE

Operating strategies to tackle the effect of the increase of ambient temperature on the performance of dry-cooled coal power plants based on supercritical CO2 power cycles

Tesi di Laurea Magistrale in
Energy Engineering - Ingegneria Energetica

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Abstract

Nowadays one of the biggest challenges is the battle against climate change. The mean global temperature is increasing and the catastrophic weather events are becoming more frequent with the years passing by. This situation is mainly caused by the greenhouse gases emissions from the energy sector and is therefore pressing to shift towards more sustainable energy sources. However the transitioning towards renewable energy sources cannot be immediate. Their intermittent nature may cause problems to the grid operation. Conventional plants could foster the integration of renewable energy sources (such as wind and solar) by off-setting their intermittent nature providing fluctuating back-up power, however this plants are currently not able to work in this operation and is therefore introduced the sCO2-Flex project. The project aims to design a power plant capable of satisfying this operation in term of power and flexibility, and it plans to do that with a supercritical CO2 brayton cycle. This cycles however present the major problem of the reduction of the maximum power producible with the increase of the ambient temperature. In this thesis work different solutions in order to tackle this problem have been investigated. After an introduction of the plant, the ambient temperature effect will be analyzed and also how the problem is dealt by now. Afterwards different alternative strategies are presented. From the results is assessed that strategies with no variation of the main compressor inlet pressure showed no significant advantage. Instead, the strategies that adopt a regulation of the compressor inlet with a pressure variation present huge improvements and could be considered a viable solution in order to tackle the ambient temperature effect problem for the sCO2 cycles.

Keywords: Climate Change, Supercritical CO2, Coal power plant, Ambient temperature, Off-design.

Abstract in lingua italiana

Al giorno d'oggi una delle più grandi sfide è la battaglia contro il cambiamento climatico. La temperatura media globale sta aumentando e gli eventi climatici catastrofici stanno diventando sempre più frequenti con il passare degli anni. Questa situazione è causata principalmente dalle emissioni di gas serra provenienti dal settore energetico ed è quindi urgente il passaggio verso fonti energetiche più sostenibili. Tuttavia la transizione verso le energie rinnovabili non può essere immediata. La loro natura intermittente potrebbe causare problemi al funzionamento della rete elettrica. Impianti convenzionali possono incoraggiare l'integrazione delle fornti di energia rinnovabile (come eolico e solare) compensando la loro natura intermittente e fornendo potenza di back-up, tuttavia questo tipo di impianti non è al momento in grado di funzionare in questa operatione ed è quindi introdotto il progetto sCO2-Flex. Questo progetto ha come obbiettivo di progettare un impianto in grado di soddisfare questa operatione in termini di potenza e flessibilità, e pianifica di farlo utilizzando un cyclo Brayton a CO2 supercritica. Questi cicli però presentano il grande problema della riduzione della massima potenza producibile con l'aumento della temperatura ambiente. In questo lavoro di tesi diverse soluzioni mirate a risolvere questo problema sono analizzate. Dopo un'introduzione dell'impianto, l'effetto della temperatura ambiente verrà analizzato e verrà presentato anche come è trattato ora il problema. Successivamente diverse strategie operative saranno presentate. Dai risultati si è definito che strategie che non adottano una variazione della pressione di ingresso di compressore primario non presentano vantaggi signiticativi. Invece, le strategie che adottano una regolazione delle condizioni di ingresso compressore con una variazione di pressione, dimostrano grandi miglioramenti e possono essere considerati una soluzione fattibile per affrontare il problema dell'effetto della temperatura ambiente sui cicli ad anidride carbonica supercritica.

Parole chiave: Cambiamento Climatico, CO2 supercritica, Centrale a carbone, Temperatura ambiente, Off-design.

Contents

Introduction

Climate Change

Nowadays one of the world biggest challenges is the battle against climate change. The mean global temperature is increasing with every year passing by, and in Figure 1 is possible to notice the progression of this effect.

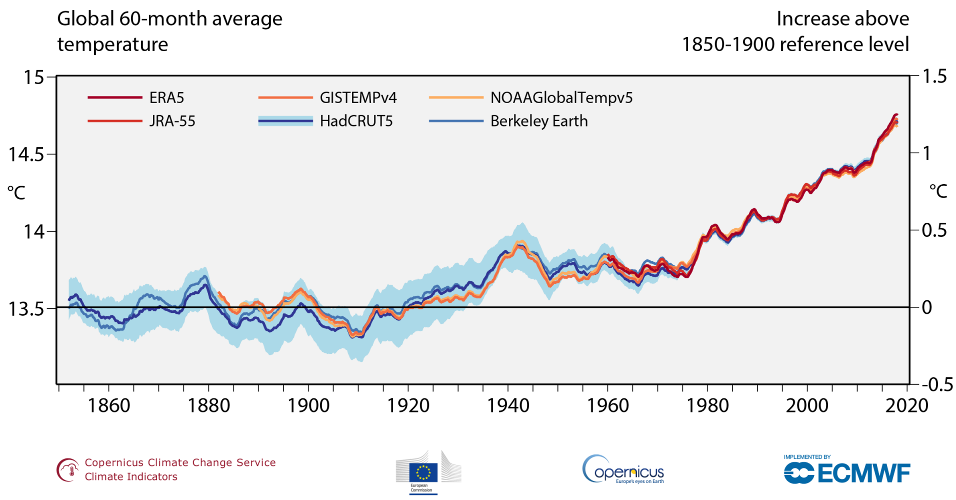


Figure 1: Global average near-surface temperature for centred running 60-month periods (left-hand axis) and increase above the 1850–1900 level (right-hand axis) according to different datasets[11].

But why is so important to act against climate change?
Global warming and the impact on the Earth weather conditions are seriously threatening the liveability of this planet. Different regions around the globe are experiencing extreme weather events and rainfalls, while others heat waves and droughts, and this effects are expected to intensify in the following years[10]. The polar ice caps are melting and and heavy rains are becoming more and more frequent, this is causing a downward spiral where the sea levels continues resulting in floods and erosion of coastal and low lying areas. This will affect the climate that will favor even further rains. This floods other

than dangerous will lead to a decrease of the water quality and the decrease of available water resources. Keeping the picture at a global level make it a bit difficult to understand, to make it more clear is possible to focus on the effects on the European climate. With the years passing by Europeans have seen the progressive changes in the climate conditions within their territories. The Mediterranean area with the southern and central part of the continent are becoming dryer, heat waves are more and more frequent making this zones vulnerable to wildfires and droughts. The norther area instead is getting significantly wetter with winter floods that are becoming more common. Despite Europe has many of the richest countries in the world, urban areas are ill-equipped and not resilient against climate change, hence 4 out of 5 Europeans are exposed to all the effects mentioned above[10]. Swiss Re Institute's stress-test analysis reveals that World economy is set to lose up to 18% GDP from climate change if no action is taken[19].

Causes of Climate change, the greenhouse gases effect

Now that is clear that the problem exist and has to be solved, before jumping in search of a solution is important to define its causes. The main driver of climate change is the greenhouse effect. Humans are increasingly influencing the climate and the earth's temperature by burning fossil fuels, cutting down forests and farming livestock. This adds enormous amounts of greenhouse gases to those naturally occurring in the atmosphere, increasing the greenhouse effect and global warming. This gases act similarly as the glass covering a greenhouse, the sun's heat is trapped within the atmosphere and are stopped to leak back into space[9]. As mentioned before different human actions contribute to the emissions of greenhouse gases into the atmosphere, in the Figure 2 is possible to see the percentage of each sector contribution.

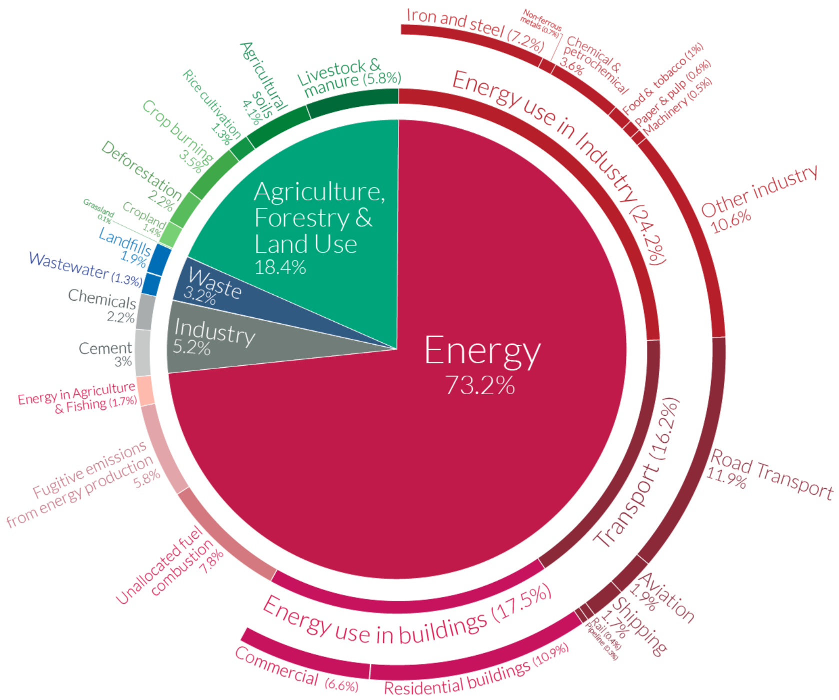


Figure 2: Global GHG emissions by sector, year 2016, "Our World in Data"[13]

Is clear that the energy sector is the main player for GHG emissions, different actions can be made in order to reduce the emissions from this sector, like promote an efficient use of energy and also leave the conventional energy plants shifting towards renewable and sustainable energy production methods.

Renewable energy

Renewable energies are in fact the key player in the majority of the discussions about climate change, but before proceeding further in the discussion is necessary to assess what are those, and why is important to proceed with this energy sources. Renewable energy is generated from sources that naturally replenish themselves and never run out, the most common sources that nowadays are discussed are solar, wind, hydro, geothermal and biomass. However this are not the only renewable sources of energy that can be found around the globe, day by day new ways of exploiting natural phenomena such as tidal waves are studied, leading to a future with several possible renewable solutions. The advantage of this energy sources such as solar and wind is that they don't directly emit carbon dioxide or other greenhouse gases that, as mentioned before, contribute to the greenhouse effect and thus to global warming.
The only emissions that they produce are indirect, meaning those that result from man-

ufacturing parts, installation, operation and maintenance, but even those are kept at minimal values. Moreover renewable energies are a reliable source of power, in opposition of conventional sources of power such as oil and natural gas this sources will never run out. This sources such as the kinetic energy of the wind and the energy from the sun are often free, the major cost component of the energy produced will thus be the plant construction. However, renewable energies do not come without downsides, it is in fact difficult to generate power in the same scale of fossil fuels. Furthermore sources of energy like wind and solar are intermittent and not programmable, they generate power only while the sun is shining or the wind is blowing[15]. Batteries could help in off setting this nature but are nowadays still expensive and not always able to effectively solve the problem[1]. While renewable energies present some challenges they also offer an environmentally friendly alternative to tackle the greenhouse gas emissions and pollution generated by fossil fuels. For these reasons the International Renewable Energy Agency (IRENA), stated that the shift towards renewable sources needs to happen fast declaring that with new policies by 2050 four-fifths of the world's electricity could be produced from renewable energy sources.

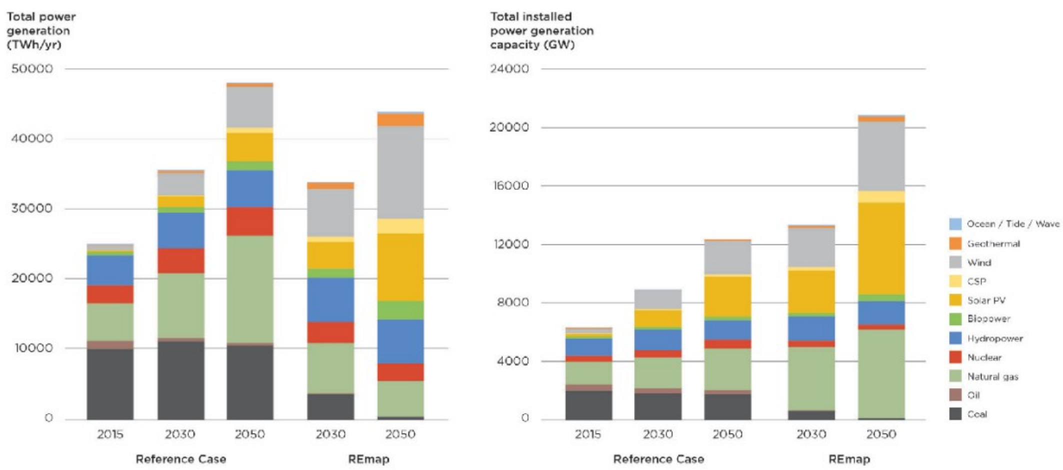


Figure 3: Power generation capacity and total electricity generation by technology, 2015-2050, based on today's policies and with a more renewable oriented policy (REmap)[2]

In the REmap scenario is possible to notice that the power sector will present a high share of renewables, providing more than 80% of the electricity by 2050, virtually eliminating the coal and oil power sources[2]. This would be an exciting result, however it will not come without major difficulties. In terms of energy production the transitioning between conventional and renewable power sources cannot be immediate[14]. The latter due to their intermittent nature may cause problems in the distribution grid such as power shortages and grid instability. Conventional plants could foster the integration of renewable energy sources (such as wind and solar) by off-setting their intermittent nature providing fluctuating back-up power and thus helping stabilize the grid. However, these plants are currently not designed to work with high power fluctuations and here is where it comes the introduction of supercritical CO2 cycles.

sCO2 flex project

EU objectives for highly flexible and efficient power plants could be met with sCO2 technologies. This thesis will develop around the H2020 sCO2-Flex project that aims to design a power plant capable of satisfying the EU objectives in term of power and flexibility, while also reducing greenhouse gas emissions, residue disposal and, above all, water consumption. This project aims at building expertise around sCO2 for electricity generation, within it the main advantage/disadvantage and possible future application of this technology are presented. General characteristics of this project will be presented in the first chapter but the main objective of this Thesis is to find viable solutions for the common problem of sCO2 cycles with the increase of the ambient temperature.

This problem is critical for the sCO2 cycles application an its related to the difficulties of keeping the operational points of the plant in the supercritical fluid area. An ambient temperature increase above the design conditions of 20°C will, move the inlet compressor operating point away from its design condition losing the advantages of compressing a supercritical fluid and greatly affecting the electric power producibility of the plant. The second chapter of the thesis will present an overview of off-design conditions such as partial load operation, the effect of the ambient temperature on the power plant and how the problem is dealt by now.

In the subsequent chapters different operating strategies of solving this problem will be presented, and the advantageous or disadvantageous effects of each one will be highlighted.

1 | sCO2-Flex project

“On the 18th-19th of January 2018 in Paris, the EU funded project “Supercritical CO2 Cycle For Flexible and Sustainable Support to the Electricity System” (sCO2-FLEX) has been kicked off by all partners"[8].
This project, led by Électricité de France (EDF), brought together an interdisciplinary group of academics, partners, technology provider and industry experts, that together aim to adapt conventional power plants to the future energy system requirements. In this future renewable energy sources will need a backup energy source that is capable to counteract their intermittent nature. This new power plants will need to provide fluctuating back-up power and grid stabilization, and conventional power plants are not currently capable to work, or work efficiently with huge power fluctuation. Since this problem is going to get more and more relevant with the years passing by, with the continuous increase of the share of renewables the need of this new kind of power plant will also became more significant in order to stabilize the grid. This project directly addresses this challenge by developing and validating a design of a 25Mwe Brayton cycle using supercritical CO2 as working fluid, this will enable an increase in the operational flexibility and in the efficiency of existing and future coal and lignite power plants, thus reducing also the environmental impact that this kind of plants have. Moreover even though theoretically the supercritical CO2 Brayton cycle is capable of reaching the objective defined by the project, a plant like this is practice was never developed. This add another challenge to this project, and also it make it useful to build expertise on sCO2 for electricity generation and its stated that Électricité de France (EDF), is also interested in investigating possible application to other renewable thermal sources such as biomass and Concentrated Solar Power (CSP[8]).

1.1. Supercritical CO2 power cycles

Before proceeding further is important to define the key player of this project, the supercritical CO2, because sCO2 cycles for power generation are gaining a large interest from industry, institutions and academia as demonstrated by the large amount of investments, funded projects and research papers.

The potential of this technology are hinted by exploring the characteristics of supercritical fluids and in particular supercritical CO2.

1.1.1. Supercritical fluid

Supercritical fluids differ quite significantly compared to the properties of real fluids. They cannot be defined neither as liquid nor as a gas but as a substance in a state, defined *"supercritical state"*, above its critical temperature (T_C) and critical pressure (P_C). In Figure 1.1 is shown a phase diagram of a single pure substance where is possible to identify four phases (Solid, liquid, gaseous and supercritical), are also shown the boundary lines between those phases where multiple phases exist in equilibrium[18].

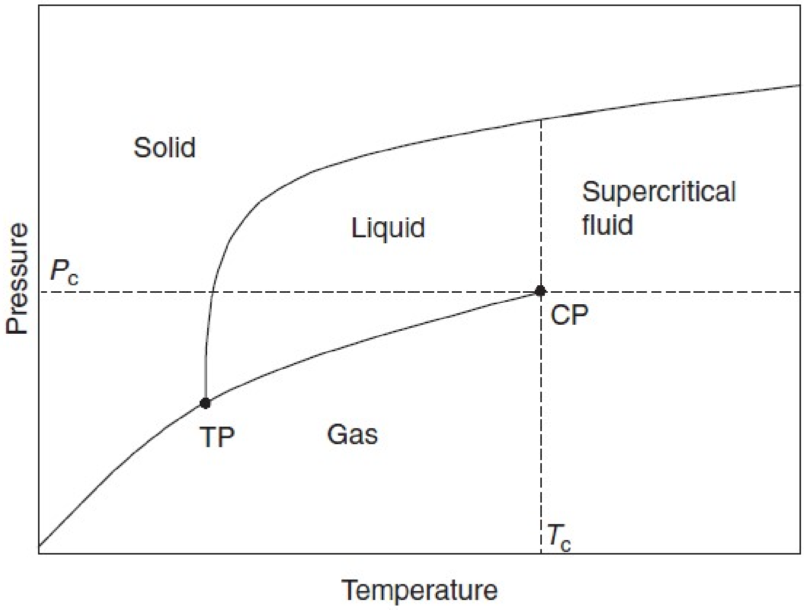


Figure 1.1: Phase diagram of a pure substance[18].

A significative point in this regard is the triple point (T_P) where liquid, solid and gas phases coexist. Following the vapor pressure curve, that defines the equilibrium points between gas and liquid phase, its possible to reach the critical point (C_P). However when P_C and T_C are simultaneously attained both liquid and gaseous phases cease to exist and merge in order to produce a supercritical fluid.

Being a coalescence of a liquid and a gas phase supercritical fluids possess both gas and liquid like physical properties. In particular the trend is to possess viscosities more similar to gases while retaining densities more similar to liquids. In Table 1.1 the typical range of values is shown.

Table 1.1: Typical range of values of several physical properties of gases, liquids, and supercritical fluids[18].

	Density **(kg m^{-3})**	**Viscosity** **(g s cm^{-}1)**
Gas ***P=1 bar; T=303 K***	(0.6-2)	10^{-4}
Supercritical-fluid ***P=P_c bar; T=T_c***	200-1100	10^{-4}-10^{-3}
Liquid ***P=1 bar; T=288-303K***	500-1500	10^{-2}

1.1.2. Advantages of supercritical CO2 power cycles

Carbon dioxide is the most widely used supercritical fluid in analytical and industrial applications since it provides various advantages, the main one being easy to manage from an engineering perspective due to low T_c (304.1 K) and low P_c (7.377 MPa). Moreover it is nonflammable, nontoxic, and it can be obtained from existing industrial processes without further contribution to the greenhouse effect.
The thermodynamic behavior of CO2 in the vicinity of critical point, and the motivation for operating the compression process close to the vicinity of the critical point is explored. In particular in Figure 1.2 is possible to observe the specific work for an isentropic compression process with a compression ratio of two.

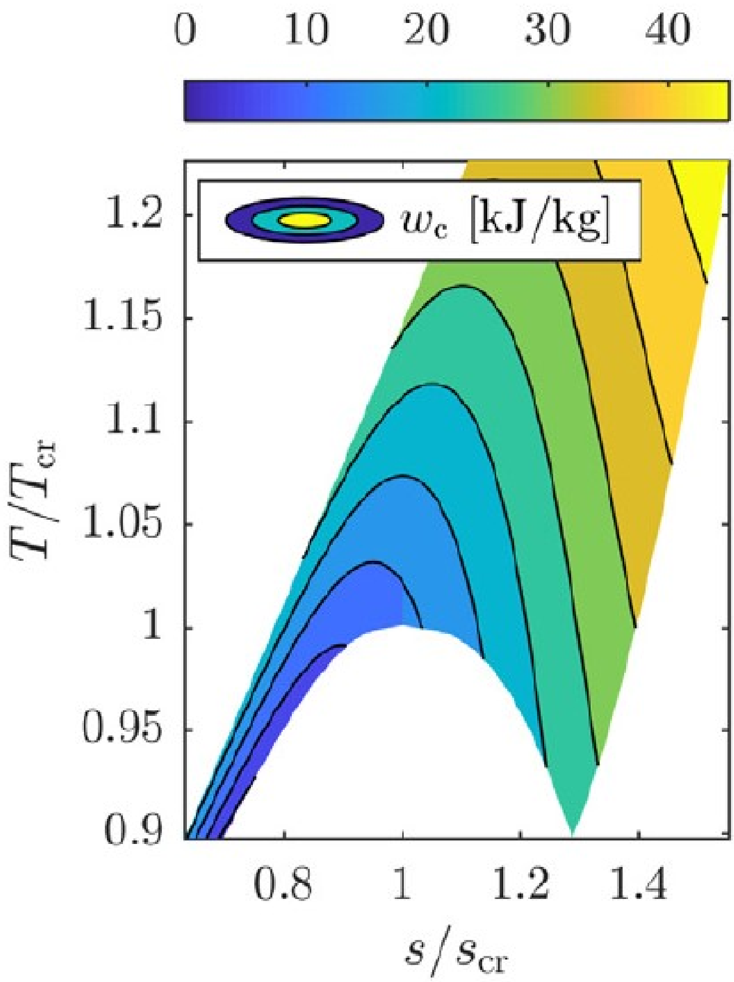


Figure 1.2: Specific work for an isentropic compression for a compression ratio of 2[25].

Noting that a lower compressor work increase the cycle efficiency the direct advantage of supercritical CO2 become clear.

Comparison with technology competitors

The Ultra Super Critical (USC) steam cycle technology is defined as the possible competitor of supercritical CO2 cycles in the future energy market. Hence is necessary to assess how this new kind of power plant can do better in respect to USC[12].
The main advantages are of sCO2 power plants are:

- **Potential higher efficiency.** As mentioned before compressing around the critical point greatly reduce the compressor work of the turbomachinery;
- **Compactness of the turbomachinery.** The high power density of CO2 leads to

small dimensions and high revolution speeds of the turbomachines.

- **No need of water treatment.** Working always in pressurized condition there is not risk of air infiltrations and components like deaerator and vacuum pump are not needed.
- **High performances at part-load.** Good plant efficiency maintained for an electrical load ranging from 100% to 20%.
- **Fast transients.** The lower fluid mass moved in the cycle grants the flexibility that USC, heavily bound by thermal inertia, cannot provide.

Disdvantages of supercritical CO2 power cycles

However not only advantages comes from transitioning towards supercritical CO2 cycles[12]. Operating under supercritical pressures, however, has implications on both cycle operation and component design. Firstly, operation pressures that are within the range of 50 to 250 bar require all parts to be designed to safely operate under both high pressures and high pressure differentials. The high pressures also lead to high densities, and consequently low volumetric-flow rates through the system.
There are also challenges related to starting the compression process close to the critical point. Specifically, there are significant variations in the thermodynamic properties of CO2 in the vicinity of the critical point.
Finally, the property variations also influence heat-exchanger design and operation. This is both from a more fundamental design perspective point, where the sudden change in specific-heat capacity at constant pressure influences the effectiveness of internal heat-exchange processes, to more practical aspects relating to define off-design performance[12].

1.2. sCO2-flex project objectives

The final objective of this project is well defined but as mentioned before this kind of plant have never been implemented before, hence it comes with various additional challenges that needs to be overcome. This challenges can be divided in three major categories, each category contains a multitude of different objectives defined fundamental to develop the sCO2 Brayton cycle desired that will be able to support the future energy system[20].

1.2.1. Technical objectives

As mentioned before this kind of power plant has never been developed before, hence there are several technical challenges that need do be overcome. The general aim is to increase coal and lignite power plants flexibility, that means to:

- **Define and simulate future coal power plant's work flows and scenarios**, for advanced and complex flexible demand operation;
- **Enable entire load optimization**, with fast load changes, fast start-ups and shut downs;
- **Develop advanced equipment solutions**, contributing to increase modularity and reduce costs through the reduction of turbomachinery size;
- **Analyze and define new boiler models**, with advanced thermos-hydraulics and mechanical characteristics to work with sCO2;
- **Design and validate a new turbomachinery system**, that will be able to work in harsh conditions. In particular is important the design of the compressor near critical point operation and modelling the turbine to be able to work with sCO2 expansion;
- **Design and test new compact heat exchanger** able to work with high efficiency, speed and reliability under high CO2 pressures and low pinch temperature;
- **Define and test state of the art materials** used for power generation at high temperature applications. This material, like stainless steel, needs to be suitable to sCO2 environment;
- **Develop Instrumentation & Control (I&C)** solutions for fast load change transients;

All of this objectives are necessary steps required to design a working cycle ready to be implemented[23].

1.2.2. Economic objectives

The technical pillar of this project alone is not enough to justify the investment; its undoubtedly important to develop a working sCO2 power plant to built expertise in this technology but is important as well that this kind of plant will be able to do well in the market.
This project states its economic goal as **enabling market uptake and industrial**

deployment of sCO2 Brayton cycle from 2025.
The necessary steps to reach this goal are divided in three economic challenges:

- **Establish sCO2 Brayton cycle as optimal coal and lignite power plants solution**, in terms costs through optimized design and components integration;
- **Optimize sCO2 cycle operation and maintenance costs** through selection of resilient materials and components through dynamic I&C solutions, this will help also increase by 5% the capacity factor of the coal plant;
- **Lower the Levelized Cost of Electricity (LCOE)** of the sCO2 Brayton cycle for coal and lignite power plant to reach a value 6 to 16% lower than the LCOE of future supercritical coal plant with CCS;

This three steps, and in particular the last one, necessary needs to be tackled in order to have this plant running in the future, because if its not advantageous in terms of economic benefit they may be able to be developed but they may never run or the market share will be minimal[21].

1.2.3. Environmental objectives

Nowadays the environmental impact of a project cannot be neglected, the continuous threat of climate change and global warming make the environmental objectives of the project a key factor in determine its overall performances. The sCO2-flex projects starts a little behind since the designed heat source is coal and lignite, surely not the cleanest ones, but its undeniable that this kind of power plants are required nowadays. Hence this project knowing that is not possible to directly exclude this plants from the energy market aims to **mitigate coal and lignite plants environmental impact**. Doing that it will also **foster sCO2 technology's acceptance**.
The path to reach this goals is not easy and the environmental objectives in details are:

- **Reducing water consumption by 100%**; high water consumption is usually a critical aspect in coal power plants. In the sCO2 project the aim is to avoid the use of water in the thermodynamic cycle and heat rejection unit (HRU), this also enabling geographical flexibility;
- **Reducing CO2 emissions and fossil fuel consumption for an equivalent power output**;
- **Enabling potential addition of a CO2 capture solution** by a post combustion capture system;

- **Access benefits of the solution** through a comprehensive Life-Cycle Assessment (LCA);
- Developing a methodology **fostering sCO2 technology** overall acceptance;
- **Reducing overall environmental footprint of the plant by at least 25%** and metal alloys needs for the turbomachinery through size reduction of power block's components; that translates also in a power density increase.

The objective of making coal and lignite power plants more environmentally friendly is definitely ambitious, is possible to notice how the main problems that usually defines this kind of plants are immediately addressed and planned to be tackled. This problems are the CO2 emissions and the excessive water consumption of this kind of plants. Moreover thanks to the use of sCO2 technology is possible to address the environmental impact of the power block components, that will undergo a significative size reduction translating in a reduction of the environmental footprint of the plant[22].

1.3. Cycle Layout

Several different sCO2 cycle layouts have been proposed in literature for different applications, 21 of them have been chosen within the sCO2-Flex project ad promising for this specific application[12]. The selection of the architecture was made considering: cycle efficiency, turbomachinery performance, boiler integration and system simplicity.
The cycle configuration selected for the final sCO2-Flex plant design is the recompressed cycle with HTR bypass. This cycle configuration (plant layout in Figure 1.3, T-s diagram in Figure 1.4, thermodynamic data in Table 1.2) has been selected by the sCO2-Flex consortium as the most promising solution for small modular coal-fired applications thanks to its high thermodynamic efficiency and the possibility to design the coal boiler with similar features to steam USC power plants with an adequate cooling of the flue gases[4].

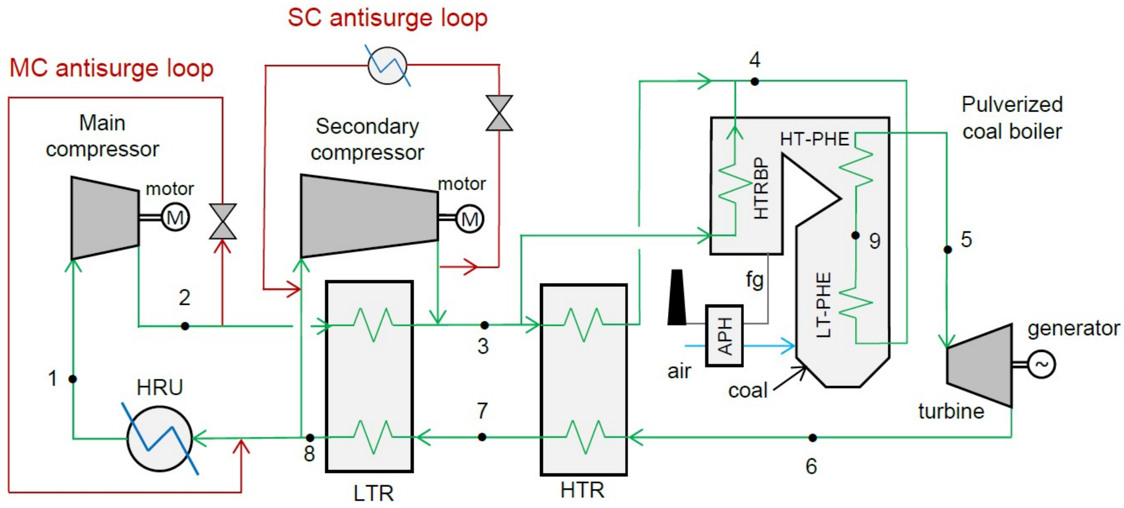


Figure 1.3: Plant layout: Recompressed cycle with HTR bypass configuration[4].

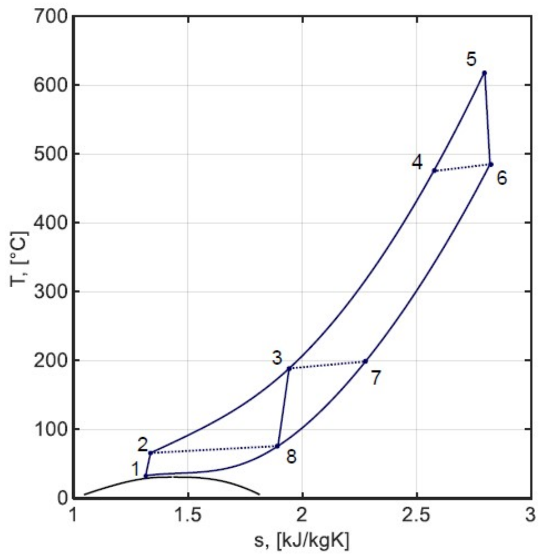


Figure 1.4: T-s diagram of the recompressed cycle with HTR bypass in nominal condition (dotted line represents the internal recuperation processes, black line represents the saturation curve)[4].

Table 1.2: Thermodynamic conditions of sCO2 in each relevant thermodynamic point[4].

stream	$\dot{m}$ [kg/s]	T [°C]	p [bar]	ρ [kg/m^3]
1	172.6	33.0	79.8	609.4
2	172.6	69.7	250.0	738.3
3	268.2	195.2	250.0	332.7
4	268.2	478.1	249.9	170.2
5	268.2	620	237.1	133.9
6	268.2	488.1	81.0	55.9
7	268.2	205.2	80.6	95.5
8	268.2	79.9	80.2	161.5
9	268.2	571.4	248.5	148.6

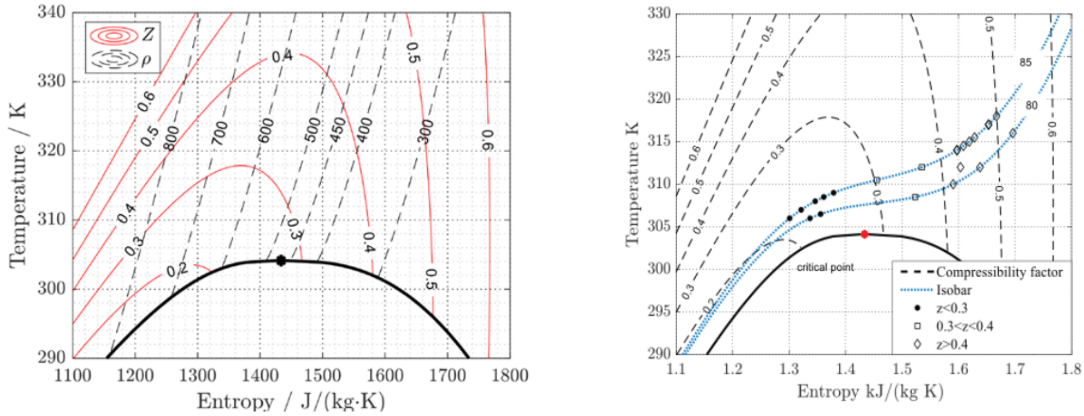
Form the plant layout is possible to notice that the supercritical CO2 cycle consists of four main components.Namely, the turbomachinery, the boiler, the recuperators and the heat rejection unit. While for steam cycles this components are well established, for sCO2 cycle they are still in the development phase due to the fluid properties and cycle conditions that pose several challenges to their design and operation[24].

1.3.1. Turbomachinery

Turbomachinery are one of the key components for the sCO2 cycle. They are the components assigned to power extraction and pressurization of the cycle, furthermore in the sCO2-flex cycle they need to grant high flexibility, hence they will require to be able to work with high fluctuations of mass flow rate. The particular working operation of the turbomachinery in addition with the peculiar properties of the CO2, especially in supercritical operation, requires an in depth study for the component design.

Compressors

The sCO2 cycle requires two compressors to operate and hence to be designed. However, while the secondary compressor is a more conventional component due to inlet operating condition around 80°C, the main compressor works slightly above the critical point with the temperature and pressure respectively around 33°C and 80 bar. This region is tricky in terms of operating properties, a minor variation in terms of temperature or pressure condition may result in a strong variation of thermodynamic properties as shown in Figure 1.5a. Moreover in Figure 1.5a is also possible to notice that this area shows a substantial deviation from ideal gas condition with a compressibility factor (Z) that undergo huge variations.



(a) Properties variation of CO2 in the supercritical state

(b) Selected compressor inlet conditions

Figure 1.5: T-s graphs of CO2 in supercritical state[24].

Hence, when the CO2 flow in a real gas behaviour approach the critical point may result in phase change phenomena, this may cause cavitation phenomena making necessary a proper design of the impeller to mitigate the effects of phase change and also maintaining the performance target of the turbomachine.

Another not negligible challenge for the compressors operation is the flexibility required by the selected cycle. The plant is supposed to operate from 100% to 20% of load. To achieve this high level of flexibility Inlet Guide Vanes (IGV) and variable-speed drivers have been implemented in both compressors design. This means that the two centrifugal compressors need to operate on two completely different shafts coupled each with its own variable speed electric motor[4].

Turbine

The operating area of the turbine is far from the supercritical area, hence the CO2 in an ideal gas behaviour permits an easier design. In this situation the turbine is composed by 5 expansion stages working at 50% reaction degree. For partial load operation the limited number of stages makes the partial arc admission a not recommended strategy, since it would penalize the efficiency at full load operation, and eith the state of the art technology due to the high pressure and temperature, 250 bar & 620°C, at which the turbine operates a variable geometry at the inlet is also a strategy that cannot be adopted. Moreover adjustable guide vanes are not suitable for this application due to their limited range of control, making for them impossible to reach the 20% load target. The turbine at partial load will hence operate in sliding pressure as the only possibility to maintain good efficiency. The turbine will have a trip and control valve at its inlet with the characteristics shown in Table 1.3, and in order to simplify the control strategy and operability a single shaft configuration has been selected[4].

Table 1.3: Turbine valves main modelling characteristics.

Component	Loss factor of the valve fully open	Reference section diameter, mm
Trip valve	0.5	160
Control valve	0.5	140

Summary of the turbomachinery characteristics

The set of turbomachinery designed for the cycle layout are therefore the following:

- A main compressor with 2 centrifugal stages, equipped with inlet guide vanes (IGV) and variable speed electrical motor. With nominal polytropic efficiency of 82.4%.
- A secondary compressor with 3 centrifugal stages, equipped with inlet guide vanes (IGV) and variable speed electrical motor.With nominal polytropic efficiency of 81.8%.
- A 5 reaction stages axial turbine characterized by a fixed rotational speed. With nominal isentropic efficiency of 88.4%.

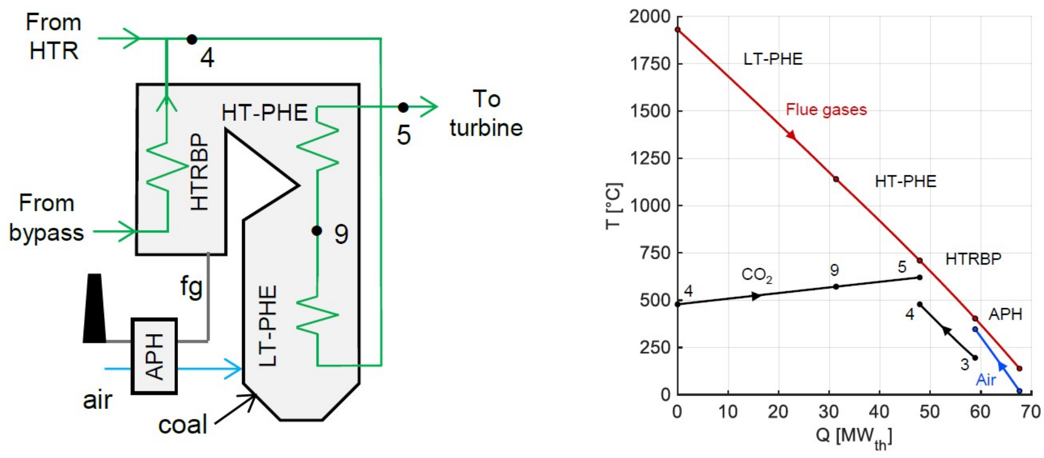
1.3.2. Pulverized coal Boiler

This section is dedicated to the analysis of the pulverized coal boiler design. The general requirements for this component where high efficiency, high flexibility, low pressure losses and a unit of power output of 25 MWe. Considering the cycle architecture this translate in a thermal load required by the boiler of 59 MWth. In order to minimize the NOx emissions, the boiler is equipped with low-NOx swirling burners and a NOx reduction system based on selective catalityc reduction (SCR).

The boiler design can be divided in four major heat exchangers that will provide heat to the CO2 to reach the desired value of 620°C at Turbine Inlet Temperature (TIT). These heat exchangers are the following:

- **Air Preheater (APH)**: This component is the first heat exchanger encountered by the inlet air and the last by the flue gases. It's a Ljungström rotative heat exchanger that helps increase the temperature of the air til the limit of 350°C favoring the combustion process, moreover it sets the stack temperature equal to 138°C. Value chosen in order to maximize boiler efficiency while avoiding acid condenses.
- **Low Temperature Primary Heat Exchanger (LT-PHE)**: This zone include the section of the boiler fed by CO2 after the mixing with the outlet of the HTR. It is designed as a membrane wall heat exchanger and is the area where the combustion process happens.
- **High Temperature Primary Heat Exchanger (HT-PHE)**: This zone is a radiative convective zone with high temperature. Designed according to cocurrent arrangement is the last heat exchanger encountered by the CO2 and the one that brings the temperature up the the 620°C value.
- **High Temperature Recuperator Bypass (HTRBP)**: In this area a counterflow tubular heat exchanger is present. The scope of this component is exploit the high temperatures in the final part of the boiler, an improve the boiler efficiency. A portion of the CO2 mass flow of the cycle is in fact extracted right before the inlet of the cold side of the HTR, heated up in this heat exchanger and mixed back at the outlet of the HTR cold side. Granting a higher temperature value at the inlet of the LT-PHE.

These considerations are confirmed by the T-Q diagram shown in Figure 1.6b, while in Figure 1.6a is possible to see the boiler design[16].



(a) (b)

Figure 1.6: (a) scheme of the heat transfer sections in the pulverized coal boiler for the recompressed cycle with HTR bypass; (b) T-Q diagram of the boiler and air preheater[4].

In conclusion in the Table 1.4 is possible to summarize the main characteristics presented by the boiler:

Table 1.4: Boiler design results[4].

	Boiler Result
Coal thermal power [MWth]	63.76
Thermal load [MWth]	58.83
Boiler efficiency [%]	92.33
Coal consumption [kg/s]	3.77
Inlet Air Temperature [°C]	20

1.3.3. Recuperators

The plant layout chosen present two internal recuperative heat exchangers: a low temperature recuperator (LTR) and a high temperature recuperator. This components needs to recover the highest amount of heat in order to achieve the highest cycle effectiveness. Operating conditions are a parameter to consider carefully, this components must withstand high temperature and pressures and also high pressure differences between the hot and cold exchange fluids. Therefore is chosen for both recuperators to opt for Printed Circuit Heat Exchangers (PCHE)[24].

1.3.4. HRU

The last component required from the plant is the Heat Rejection Unit (HRU), this heat exchanger is used to release the excessive heat from the cycle to the environment.
This is constituted by several dry cooled heat exchangers arranged in parallel, each unit is made up of batteries of small copper tubes and alluminium fins and fans provided by variable speed electric motor.
Due to the similar temperature and pressures of operation the technology transfer from CO2 refrigeration cycles can be made, is then not required to design a component from scratch. As shown in Figure 1.7 the cold-end temperature is equal to 13°C, with an air inlet temperature of 20°C and CO2 outlet temperature of 33°C, despite that due to real gas behaviour of the CO2 the pinch point temperature is around 9°C.

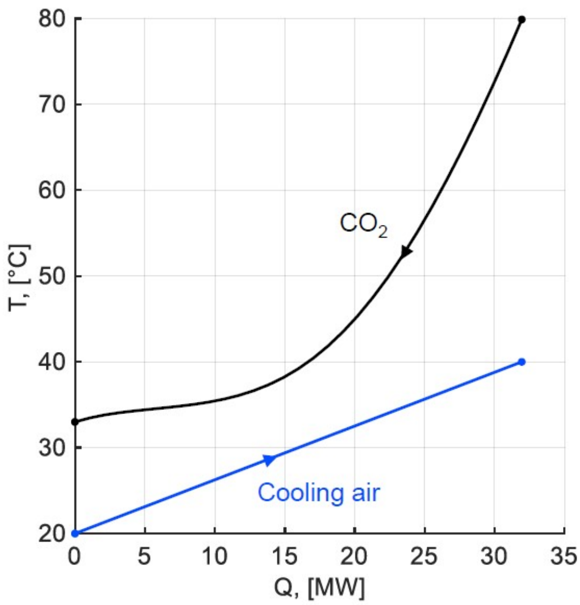


Figure 1.7: TQ diagram of the HRU system[7].

Acting on the rotational speed of the fans is possible to change the CO2 set point temperature, there is an upper limit to this intervention set at 125% of the design rotational speed[4].

1.4. Design operating conditions

The cycle layout and the main components have been defined in the previous chapters, is now possible to evaluate the overall performances of the plant in nominal operating conditions.

In the Table 1.5 are reported the main results of the simulation.

Table 1.5: System performances, power balances and main results of the 25 MWel sCO2-Flex plant[4].

System performance	
Cycle thermodynamic efficiency [%]	42.47
Boiler efficiency [%]	92.33
Plant gross overall efficiency [%]	39.21
Net plant efficiency [%]	37.01
Power balance	
Input thermal power [MWth]	58.87
HRU thermal power [MWth]	31.94
Main turbine electric power [MWel]	38.78
Main compressor electric power [MWel]	5.30
Secondary compressor electric power [MWel]	8.20
HRU auxiliaries consumption [MWel]	0.28
Gross plant electric power [MWel]	25
Total auxiliaries consumption [MWel]	1.40
Net plant electric power [MWel]	23.60
Mass flow rates	
CO2 mass flow at turbine inlet [kg/s]	268.15
CO2 mass flow at HRU [kg/s]	172.63
CO2 mass flow at HTR bypass [kg/s]	29.95

Combining the cycle efficiency of 42.47% with the boiler efficiency of 92.33% a plant gross overall efficiency of 39.21% is achieved. From a preliminary estimation is possible to obtain the power consumption from auxiliaries that is not included in the computation of the gross overall efficiency. This value estimated around 1.4 MWel permits to compute the net plant efficiency of 37.01% that can be used as a comparative value with current technologies. The comparison puts the sCO2-flex power cycle below modern large (650 MW) supercritical steam coal fired plants with an efficiency around 39.5%, but the result is satisfactory compared to with similar size steam plants[4].

2 | Off-Design performances

2.1. Part Load Operation

The role of conventional power plants in the future energy scenario as mentioned in the introductory chapter need to change. This kind of plant need to evolve from a base-load only operation to provide fluctuating back-up power to meet the unpredictable and short noticed load variation caused by the renewable energies that will dominate the energy market. The sCO2-flex project aims to design this kind of new generation power plant, capable of granting flexibility and high performances at part load.
For this reason is important to properly model and analyze the plant behaviour in part-load operation, in particular this analysis focus on the optimal part load cycle operation at design constant ambient temperature. Given the air-cooled cycle this condition is usually not valid since the temperature highly varies in a daily and seasonal base. The constant ambient temperature is thus a simplification required to reduce the complexity of the analysis and helping in the identification of the most promising operation strategies[7]

2.1.1. Part Load Operating strategies

Several options are available for the part load operating strategy of the cycle, different possibilities have been investigated in order to select the best strategies that maximizes the plant performance for the whole operability demanded by the plant (100% down to 20% of the nominal load), taking into account the turbomachinery performance maps in part load and the boiler constrains. The sCO2-flex cycle components can provide a form of plant regulation; the turbomachines adopts a sliding pressure expander and the compressors are able to vary their rotational speed and are also equipped with Variable Inlet Guide Vanes. The heat rejection unit is equipped with variable speed fans that will help in the regulation of the compressor inlet temperature and is also possible to exploit the fact that is a closed cycle and act on the cycle working fluid inventory varying the set cycle minimum pressure.
In conclusion to operate and part load it is required to start reducing the pulverized coal

mass flow rate, doing that is than necessary to to correct the excess air and the APH rotational speed in order to have a good operation of the boiler. Turbine inlet temperature is set equal to the nominal one by changing cycle maximum pressure according to the turbine operation curve and CO2 mass flow rate by acting on compressor operating point. The main compressor temperature and pressure inlet conditions are kept constant acting on the HRU fan rotational speed and on the working fluid inventory respectively. Concluding the efficiency is maximized acting on the IGV and RPM for the compressors and on valves in order to keep ΔTmix=0 in the two mixing points of the cycle (high pressure exit of both LTR and HTR).

With this strategy is possible to regulate the inlet compressor conditions for the part load operation, but is fair to ask what are the optimal inlet conditions of the compressor that maximize the plant efficiency.
Two possible operational strategies have been investigated in this regards:

- **CASE 1:** The turbine operates in sliding pressure and fixed minimum cycle pressure equal to the design value. The thermodynamic conditions at the inlet of the main compressor does not change in part-load operation;
- **CASE 2:** The turbine operates in sliding pressure but this time the cycle minimum pressure may vary in order to optimize the plant efficiency. The thermodynamic conditions at the inlet of the main compressor varies in order to choose the optimal operating conditions for the plant overall performance.

This two cases have been evaluated in steady state conditions for the whole electrical load range required, from 100% to 20%. This range of operation is again the expected requirement that the future small sized conventional power plants will need to have and sCO2-flex aims to be one of them[7][3].

2.1.2. Part load operation of the components

Before proceeding further and evaluating the performance of the cycle in part-load operation, is important to assess how each component work in off-design operation. While in Chapter 1 is possible to have a look at the design operation condition of the components, in this paragraph the off-design assumptions and modelling for the performance assessment will be briefly summarized.

Recuperators off-design modelling

The part load simulation of the recuperators has been tacked modelling the component following a discretized approach, the recuperator is divided in a specified number of sections in order to ensure an accurate description of the thermodynamic properties throughout the stream.
The thermodynamic conditions of the stream will then be defined for each off-design simulation, this conditions needs to satisfy the constrains defined by the cycle operation and the component. In particular the mass and energy balance of the recuperator needs to be satisfied as well as the characteristic equation of the heat exchanger:

$$\dot{Q}_{HX} = UA_{HX}\Delta T_{ml} \tag{2.1}$$

Where $\dot{Q}_{HX}$ is the thermal power exchanged in the heat exchanger, U is the global heat transfer coefficient referred to A_{HX}, that is the heat exchange area, and ΔT_{ml} is the logarithmic mean temperature across the component[3].

Boiler off-design modelling

The boiler part-load operation is obtained reducing the mass flow rate of the coal and subsequently evaluate the excess air ε_{air} coefficient of the combustion process in order to obtain an effective combustion. The correction of the excess air is made according to the Equation 2.2 provided by ÚJV Řež (UJV), research institute that provided the detailed design of the boiler.

$$\begin{cases} if \quad \dot{Q}_{LHV} \geq 0.7 \cdot \dot{Q}_{LHV,nom}; \quad \varepsilon_{air} = \varepsilon_{air_nom} \\ if \quad \dot{Q}_{LHV} < 0.7 \cdot \dot{Q}_{LHV,nom}; \quad \varepsilon_{air} = \varepsilon_{air_nom} + 0.7 - \dfrac{\dot{Q}_{LHV}}{\dot{Q}_{LHV,nom}} \end{cases} \tag{2.2}$$

In the equation $\dot{Q}_{LHV}$ and $\dot{Q}_{LHV,nom}$ are the thermal power that the coal combustion will generate respectively in part-load end design operation. While in parallel the value ε_{air} correspond to the excess air value required by the boiler in part-load operation, in contrast with the $\varepsilon_{air,nom}$ that is the design value of excess air correspondent to nominal operation.
From the equation 2.2 is possible to notice that until the boiler reaches the 70% load threshold the ε_{air} coefficient will remain constant to the nominal value. Afterwards this value will be not enough to guarantee an efficient combustion and it will increase linearly with the load decrease ($\frac{\dot{Q}_{LHV}}{\dot{Q}_{LHV,nom}}$ always < 0.7 in this operation). Is important to remember that ε_{air} is a relative value and keeping this value constant means that a decrease of

the coal mass flow rate will translate also in a decrease of the absolute mass flow of the inlet air necessary for the combustion. Hence $\varepsilon_{air} = \varepsilon_{air,nom}$ doesn't translate into $\dot{m}_{air,combustion}$=$\dot{m}_{air,combustion,nom}$[5].

Compressors off-design modelling

Compressor off-design optimal operation is evaluated thanks to the the non-dimensional performance maps that are shown in Figure 2.1. The maps are evaluated as function of the volumetric flow rate ratio $\frac{\dot{V}_{in,comp}}{\dot{V}_{in,comp,nom}}$ and the enthalpy rise ratio $\frac{\Delta h_{comp}}{\Delta h_{comp,nom}}$. Thanks to the map is possible to see that is convenient to operate near the surge line of the compressors where the polytropic efficiency is higher, this translate in preferring a control regulation via variation of the shaft rotational speed adopting IGV control only at very low or high volumetric flow rates.

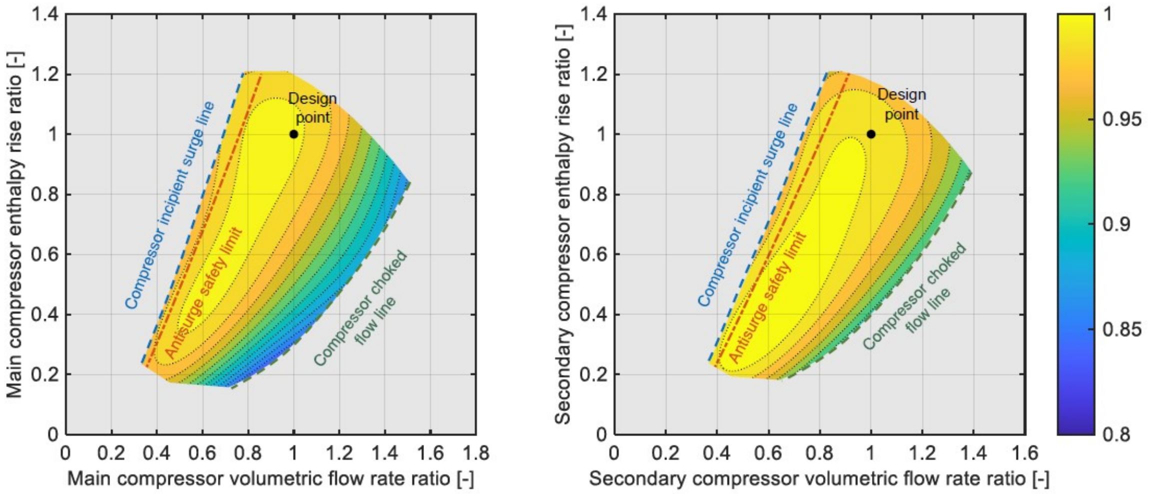


Figure 2.1: Dimensionless operational map of the main (left) and secondary (right) compressors[4].

In order to preserve compressors operability and lifetime it is important to avoid the surge phenomenon. For this reason, as is possible to notice in the Figure 1.3 an anti-surge loop is installed in the system for both compressors. When the volume flow rate circulating in a compressor is lower than the safety limit defined as 1.1 times its corresponding surge limit value, the antisurge valves are opened ensuring stable compressor operation. The drawback of this procedure is the increase of the compressor consumption and thus penalize the overall plant efficiency.

The upper limit of the compressor maps is also an interesting condition, this limit shows the maximum volumetric flow rate that the compressor is able to process due to the reach of the maximum value of RMP and IGV regulation. This situation contrary to the surge operation is not easily solved and if neglected could lead to the plant forced shutdown[4].

Turbine off-design modelling

The lacks of control systems of the turbine makes the evaluation of the part load operation easier than for the compressors. The results shows that the Turbine efficiency remains very close to its nominal values until a strong decrease of the pressure ratio moving from chocked to the non-chocked conditions described by Stodola law. Φ is the flow coefficient of the turbine that is used to evaluate the efficiency of the component at allowing fluid flow.

$$\Phi = \frac{\dot{m}_5 \sqrt{T_5}}{p_5 A_{in,turb}} \tag{2.3}$$

Where $\dot{m}_5$ stands for the CO2 mass flow rate in [kg/s] that enters the turbine, T_5 is the turbine inlet temperature (TIT) measured in [K], p_5 is the turbine inlet pressure [bar] and $A_{in,turb}$ is the area at the inlet of the turbine [m^2].
The maximum CO2 temperature of the cycle ($\dot{m}_5$) is kept constant in order to avoid overheating in the furnace by changing the cycle maximum pressure (p_5) according to the turbine curve and CO2 mass flow rate acting on compressor operating point[4].

Air cooled heat rejection unit off-design modeling

In off-design conditions the HRU operation is fundamental for the control of the CO2 set point temperature at the inlet of the main compressor.
As mentioned in the paragraph 1 the heat rejection unit can be controlled by acting on the rotational speed of the electronically commutated fan, which can be increased up to the limit of 125% of its operation. CO2 heat transfer coefficients have been computed in part load operation similarly to the heat exchanger design using a slightly modified Gnielinski correlation for supercritical CO2[5].

2.1.3. Performance in part load operation

In the paragraph 2.1.1 the operating strategies adopted in case of part-load operation have been showed, in particular the distinction between two possible operating solution has been proposed.

- **CASE 1:** Turbine in sliding pressure and fixed minimum cycle pressure
- **CASE 2:** Turbine in sliding pressure and optimized minimum cycle pressure

The main difference between the two cases is in the variation of the minimum operating pressure of the components. In CASE 1 the pressure ratio decrease due to the sliding pressure operation of the turbine with consequent reduction of the maximum pressure. This will induce a non optimal operation of the turbine as shown in Figure 2.2, in CASE 2 this effect is partially compensated by the controlled reduction of the minimum pressure of the cycle that goes from the nominal value of 79.78 bar to around 77 at minimum load.

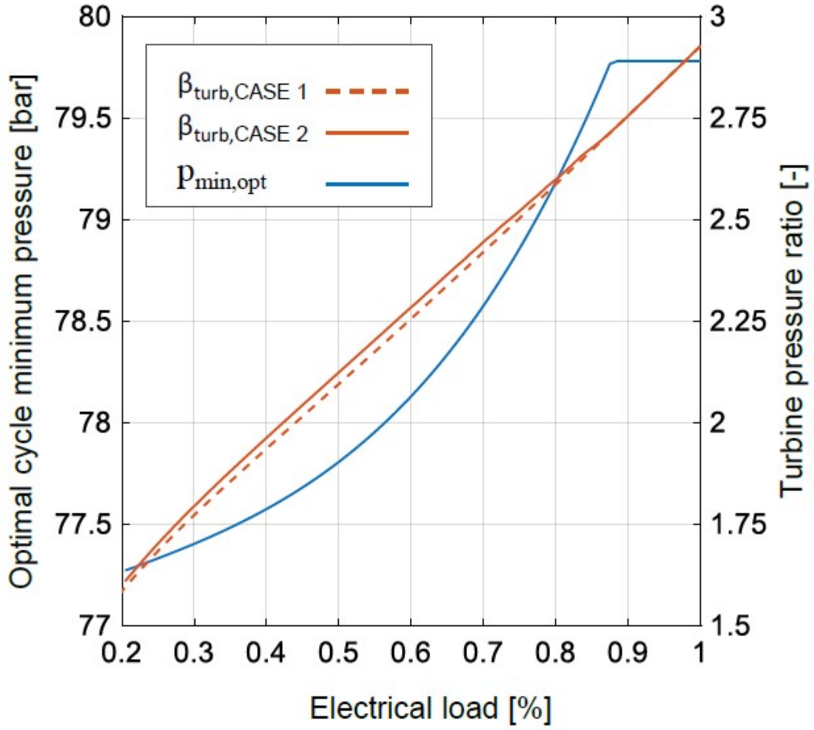


Figure 2.2: Trend of the optimal minimum pressure of CASE 2 together with the values of pressure ratios for CASE 1 and CASE 2 as function of the electrical load of the plant[5].

The effect of the two investigated part load strategies on the overall plant performance is shown in Figure 2.3.

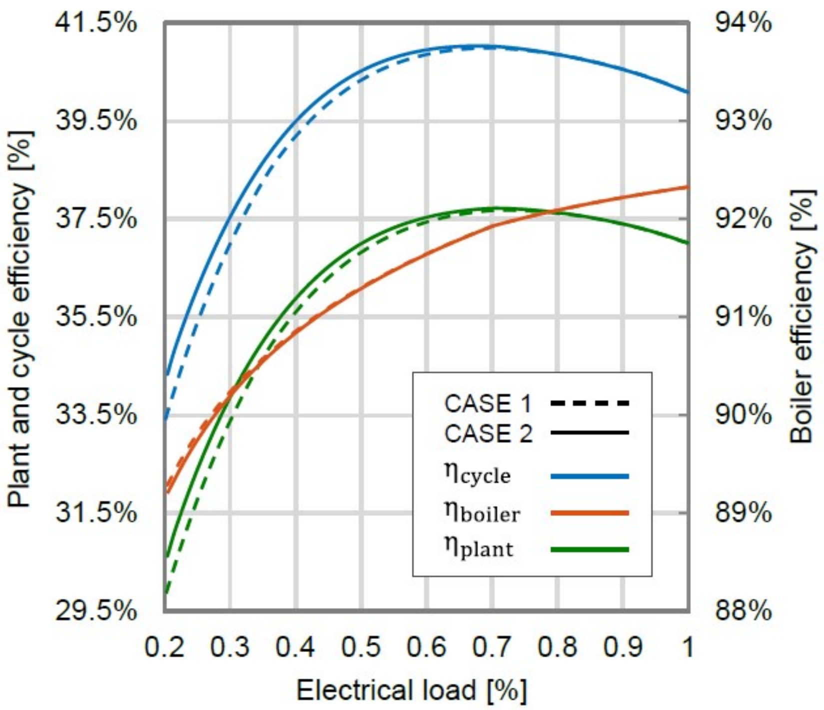


Figure 2.3: Trend of cycle efficiency, boiler efficiency and the plant efficiency for the two investigated part-load operational strategies as function of the electrical load of the plant[5].

For both cases the efficiency of the cycle tends to remain almost constant for a wide range of electrical loads as the negative effect related to the decrease of the pressure ratio of the cycle is compensated by the increase of the effectiveness of the heat exchangers. This is due to the heat exchangers being oversized for part load operation, hence they can perform more effectively working with smaller temperature differences. For electric loads below 70% of the nominal value, it is possible to notice a drastic change in boiler efficiency, the excess combustion air at the boiler increases as assessed in the Equation 2.2 causing a significant increase of the stack losses.
Furthermore when the plant reaches the value of around 60% of load cycle the cycle efficiency starts to drop considerably due to an excessive decrease of the pressure ratio that will considerably step away from its optimal values causing also a decrease in the turbine isentropic efficiency.
In regard of the main and secondary compressor operation as function of the electrical load is possible to observe the performance maps showed in Figure 2.4 and in Figure 2.5.

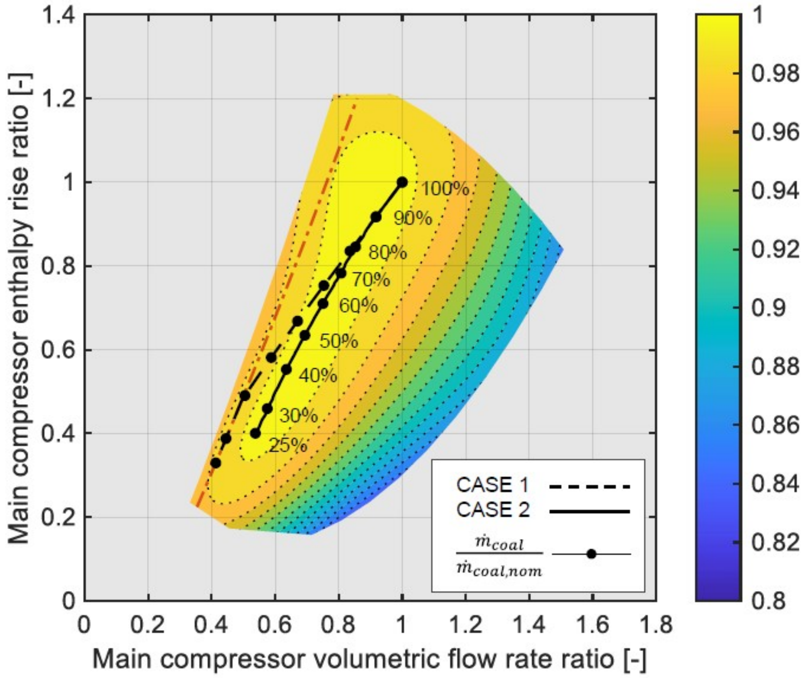


Figure 2.4: Main compressor operating points for the fixed minimum cycle pressure equal to the nominal value (CASE 1) and for optimal minimum pressure case (CASE 2) as function of the coal mass flow rate fed to the boiler[4].

It is possible to notice in Figure 2.4 that adopting the **CASE 1** operating strategy, for the main compressor the curve of operation tends to shift towards the compressor surge line in a greater measure than for **CASE 2**. In this case at 50% of electrical load the antisurge loop activates that translates in a decrease of the overall efficiency due to the increase flow processed by the turbomachinery.

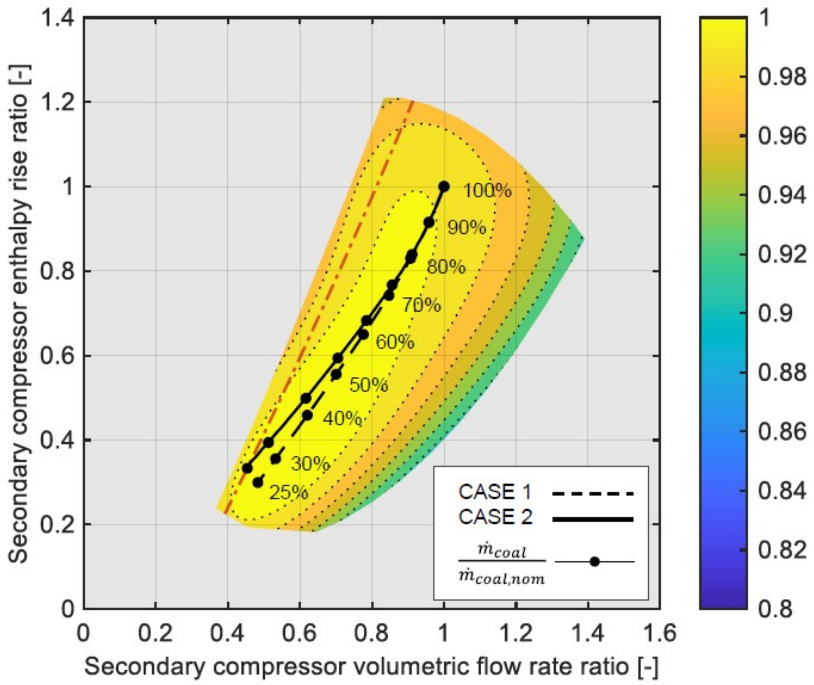


Figure 2.5: Secondary compressor operating points for the fixed minimum cycle pressure equal to the nominal value (CASE 1) and for optimal minimum pressure case (CASE 2) as function of the coal mass flow rate fed to the boiler[4].

For **CASE 2** the reduced minimum pressure allows a greater compressibility factor than for **CASE 1**, keeping the operating condition for the main compressor always in the high efficiency region of the map. The secondary compressor as reported in Figure 2.5 do not show criticalities unless for **CASE 2** operation at very low electrical loads (25%) where the antisurge loop activates.

2.1.4. Conclusions on part load operation

In conclusion is possible to point out the main results of the part load operation analysis that have been previously presented.

- Is introduced the possibility to vary the cycle inventory to reduce and optimize the minimum pressure of the cycle. This will bring positive effects related to a higher pressure ratio and higher secondary compressor efficiency.
- Is noticed that the cycle efficiency reaches its maximum value around 70% of the electrical load thanks to the increase of the recuperators effectiveness at part-load.
- Finally in regard to the operability range of the plant it has been shown that is

possible to run the cycle down to the minimum load required of 20%. In this case both compressors are able to run and the cycle efficiency shows very little penalization demonstrating the very good part-load performances of sCO2 power plants.

2.2. Operation at variable ambient temperature

Another operating condition that is interesting to explore is how the plant is affected by the variations of the ambient temperature. This condition, not usually critical in standard Brayton cycles, it's important to be assessed in the sCO2-flex cycle due to its peculiar characteristics.

2.2.1. The problem of the ambient temperature increase

The critical component that act as interface between the cycle and the environment is the HRU. As shown in the plant layout an throughout Chapter 1 this component adopts multiple air-cooled units with small cold end temperature difference (13°C) and high working fluid temperature variation(35°C).

This peculiarity allows to meet the main environmental objective of the sCO2-flex project, a significant reduction of water consumption, namely 100%. Furthermore, it permits an easier installation for such systems, free from geographical constrains such as the availability of a river or see in the proximity of the site location. However, differently from water-cooled HRU that can benefit from a relatively stable minimum temperature of the cooling medium, for dry-cooled units the ambient temperature variation on daily and seasonal base can affect the cycle minimum temperature with a consequent impact on sCO2 power plant performance and operability.

In order to define the gravity of this problem is important to assess in what margin the HRU can mitigate the ambient temperature variation on its own. The component have a way of improving its heat rejection acting on the rotational speed of the electronically commutated fans, this action will lead to a decrease of the pinch point temperature difference of the HRU, as well as a lower temperature increase across the air section of the component. However the electric motors that operates this fans have a maximum operation limit of +25% of their nominal value.

This solution however is far from enough, the increase of the rotational speed of the fans permits an operation at design conditions only for a little increase of the ambient temperature from the design conditions. The simulations assessed that this range of operation is only from the design ambient temperature of 20°C to the maximum temperature of 22°C,

above that temperature the HRU will not be able to dissipate enough thermal power in order to keep the main compressor inlet temperature constant, that thus will increase.
In Figure 2.6 is possible to see the meteorological data of two site locations studied to assess the plant performances.

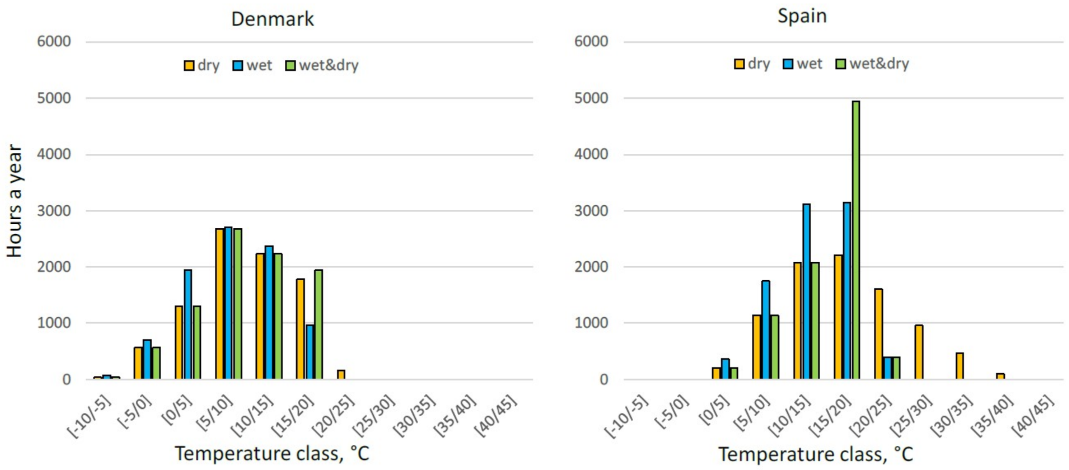


Figure 2.6: Statistical ambient temperature data for Denmark and Spain in 2019[7].

For Denmark (Copenaghen) average annual dry bulb temperature is low (9.6°C), and is above 20°C only few hours a year with a maximum value of 22 °C. Spain (Seville), on the contrary, shows a completely different annual temperature distribution: the average dry bulb temperature is higher (17.66°C) with a maximum value of 40°C, that is largely above the nominal temperature at design condition of 20°C.
From this data is possible to notice that while in Denmark the increase on the ambient temperature may not be an issue, in Spain this situation is strongly present and is then important to assess what an increase of the main compressor inlet temperature leads in term of plant performance and what strategies are possible to adopt in order to mitigate this effect[7].

2.2.2. Effects of the main compressor inlet temperature increase

The last paragraph concluded assessing that if no further action is adopted the increase of the ambient temperature causes an inevitable increase of the main compressor inlet temperature.
This means also a progressive departure from the critical point region, resulting in a marked drop of the working fluid density.
Leaving the proximity of the critical point translate also in a loss of real gas behaviour at the compressor inlet, this leads to a higher specific consumption for the compressor (at

given pressure ratio) and a rapid increase of the volumetric flow rate of the CO2. While the first effect penalizes the efficiency of the cycle, the second may be a significant limiting factor for the operability of the system.

The limiting component will be the main compressor, as shown in its performance map, that can be seen in Figure 2.4 and also Figure 2.1, it is possible to notice that just an increase of 20% of the volumetric flow rate from the nominal condition is able to push the operating point at the map limit. A similar but less strict limitation exist also for the secondary compressor

In order to assess when this limit will be reached in Figure 2.7(left) is depicted the ratio between the density at main compressor inlet in design conditions varying the compressor inlet temperature and pressure conditions. While in Figure 2.7(right) is displayed the same graph but for the secondary compressor.

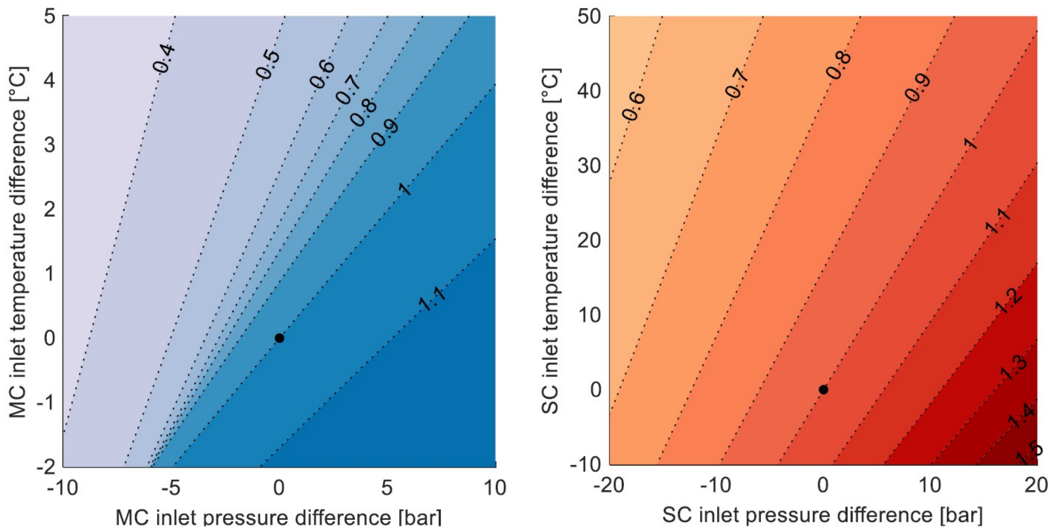


Figure 2.7: Ratio between the local density of CO2 and the density at main compressor (left) and secondary compressor (right) inlet point in nominal conditions[6].

It is possible to notice that operability limit of the main compressor (-20% density) can be reached just with a compressor inlet temperature increase of 1°C while keeping the cycle minimum pressure unchanged. While for the secondary compressor an increase of around 30°C at constant inlet temperature is required to exceed the map operational limits.

Is also important to notice that this effect can be mitigated with the increase of compressor inlet pressure, allowing to keep the density ratio equal to 1 and thus the volumetric inlet flow rate equal to the nominal one. However, from the graph is possible to notice that an inlet temperature increase of only 2°C will need a corresponding compressor inlet pressure increase of 5.2 bar to keep the nominal volumetric inlet flow rate, thus causing a reduction of the plant pressure ratio and consequent efficiency loss.

Therefore this method is put aside but definitely not discarded, the benefits of adopting a variable pressure inlet for the main compressor in a variable ambient temperature operation are exploited in Chapter 4.
As said before in paragraph 2.2.1 due to the adoption of a dry air cooled heat rejection unit the departure from the nominal operating temperature it can easily happen, translating in a important issue for closed loop Brayton cycles that adopts supercritical CO2 with compressor conditions close to the critical point (sCO2-flex project situation). Therefore is important to implement a strategy that preserves the operability of the plant also for higher ambient temperature contitions.

2.2.3. Variable ambient temperature: operating strategy

In this paragraph is presented the strategy that is currently adopted in order to tackle the effect of the increased ambient temperature, that however present numerous limitations at high ambient temperatures.
The strategy can be divided in four different operative regions each defined by its ambient temperature limits. For each region a strategy to tackle the effect of the ambient temperature is suggested[7].

1. **$T_{amb} \leq 20°C$**
   This operating condition is considered not significative in term of ambient temperature variation on plant operation. A decrease of the ambient temperature would make possible the reduction of the compressor inlet temperature, however this operation of the turbomachinery is not recommended. Operating closer to the critical point due to real gas behaviour may lead to cavitation issues at the impeller inlet as the formation of vapour bubbles could occur[17].
   Hence when the ambient temperature decrease the strategy adopted is to keep the main compressor inlet temperature as constant leading back to design operating conditions of the cycle. This procedure is made by reducing the HRU fan rotational speed.

2. **$20°C < T_{amb} \leq 22°C$**
   This operative region is the first characterized by an increase of the ambient temperature, in this condition the temperature increase is still minimal, hence like before the operating strategy adopted is to keep the main compressor inlet temperature constant. This is made acting the rotational speed of the electronically commutated fans of the HRU. This will lead to a decrease of the pinch point temperature difference of the heat exchanger as well as a lower temperature increase across the

air section. However, this strategy quickly reaches its limits due to the maximum variation of the fan rotational speed, this limit is set by electric motors at +25% of the nominal value.

The advantage of this strategy is the possibility to work in design operation, since the main compressor inlet conditions remains unchanged, all other thermodynamics points will remain the same as shown in Chapter 1. As result the specific power of the cycle will remain almost constant while the plant overall efficiency and the net power output will slightly decrease due to the increase of the power consumption of the HRU.

3. **22°C<T_{amb} ≤23.6°C**

   When the ambient temperature increase again and it reaches 22.1°C the maximum rotational speed of the fans its reached, and the compressor inlet temperature will inevitably start to rise. However, it is still possible to fuel the plant with the nominal coal mass flow rate, however as mentioned in the paragraph 2.2.2 the compressors operating points will change. The nearly constant pressure ratio and the increase of the main compressor inlet temperature will translate in an increase of the main compressor volumetric flow rate as well as an increase of the compressor enthalpy head. The main compressor operative point is pushed towards the upper limit of the performance map as shown in Figure 2.9a. While for the secondary compressor the operating point slightly changes because the higher temperature inlet in off-design conditions (+6.8°C), as shown in Figure 2.7(right), is still not enough to significantly affect the fluid properties.

   The larger main compressor specific consumption will lead to an overall decrease of the plant efficiency.

4. **T_{amb}>23.6°C**

   When the ambient temperature reaches the value of 23.6°C the compressor inlet temperature will be of 33.9°C, in this situation the upper limit of the main compressor operative map is reached, making not possible the operation with the previous strategy with higher ambient temperatures.

   In this condition the only possibility is to reduce the amount of fuel fed to the boiler. The reduction of the heat input in the cycle leads to a decrease of the CO2 mass flow rate allowing to maintain the operation of the compressor within its volumetric flow limits. The main compressor operative point in this strategy will all be placed onto the upper limit of the performance map, as shown in Figure 2.9a, and the regulation of the fuel input will be made accordingly.

   On the contrary as seen in Figure 2.9b the operative point of the secondary compressor is progressively pushed towards the surge safety limit, that will inevitably reach at high enough temperatures activating the antisurge loop.

This strategy can also be summarized in the scheme below.

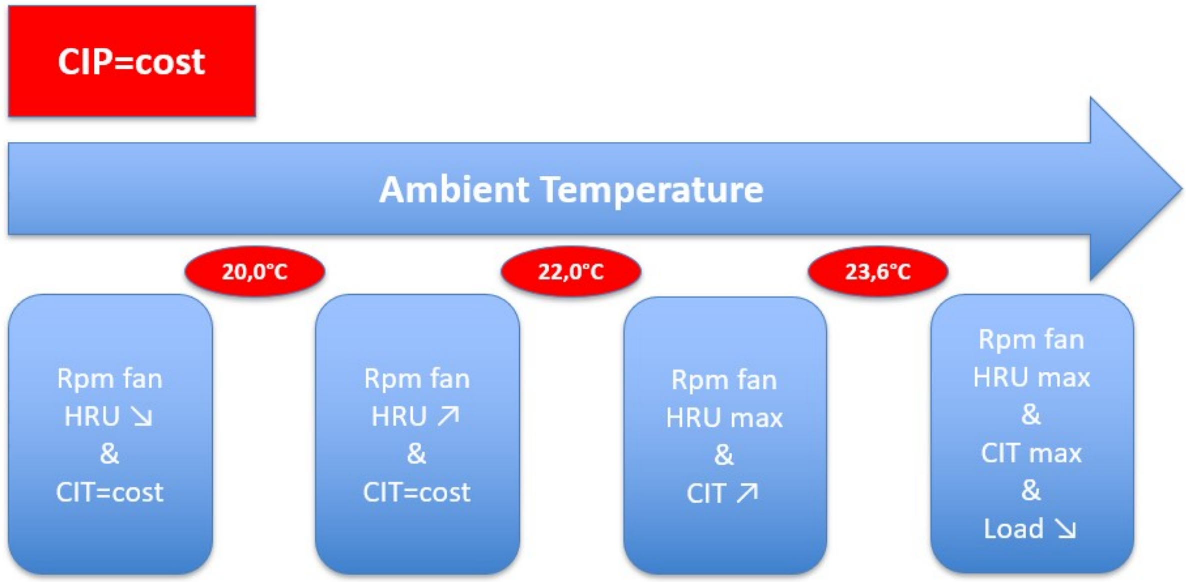


Figure 2.8: Standard operating strategy to mitigate the ambient temperature effect

Where *"CIP"* stands for the main Compressor Inlet Pressure that is always kept constant, *"CIT"* is the main Compressor Inlet Temperature, *"rpm fan HRU"* is the rotational speed of the HRU fans and *"Load"* corresponds to the fuel input. It is important to remember that the nominal condition chosen for the operation is T_{amb}=20°C, therefore is the only operating condition where no action on the plant components have to be made

2.2.4. Plant performance at variable ambient temperature

Is now important to assess the the performance operations of the cycle in this particular off-design condition.

Compressors

The compressors operation is surely the most interesting to assess the cycle performance. The Figure 2.9 shows how the operating points in both compressors move with the increase of the ambient temperature from the design conditions. As already explained in the paragraph 2.2.3 that describes the operating strategy the main compressor will operate in design conditions until the temperature of 22.1°C is reached, when that happens the compressor inlet temperature will start to rise until it reaches the map limit at 23.6°C of ambient temperature, at that point with greater ambient temperatures the compressor operation will move along the limit of the map thanks to a fuel input decrease. For the secondary compressor after a not significant increase of the enthalpy ratio in the temperature range from 22.1 and 23.6, the secondary compressor operating point moves towards the surge line activating the antisurge loop.

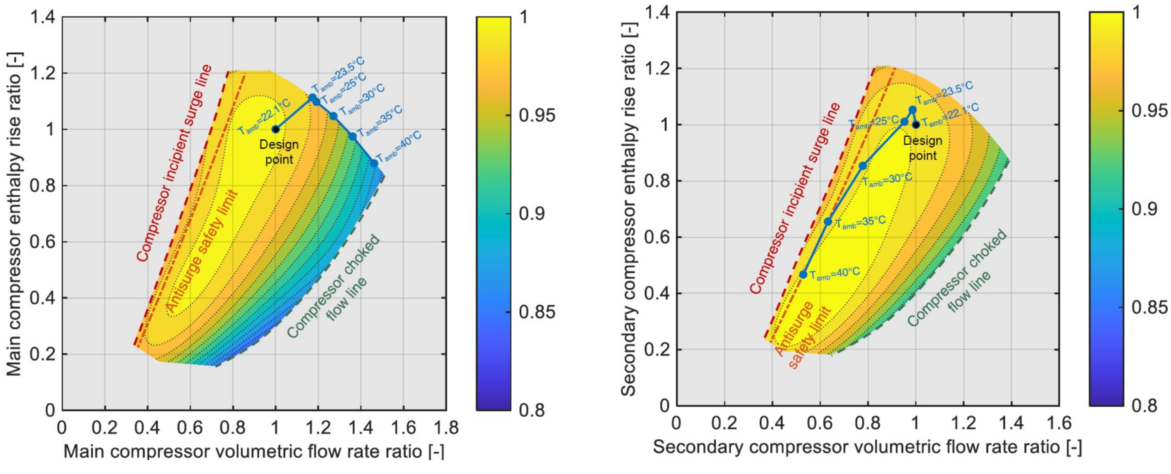


Figure 2.9: Main (a) and secondary (b) compressor operating points as a function of the ambient temperature[7].

Plant performance

Regarding plant performance the major issue is regarding the significant limitation of the maximum power output that this strategy involve. This effect is showed clearly in Figure 2.10a where is possible to notice an abrupt decrease of the maximum power producible by the plant after the 23°C threshold of ambient temperature.

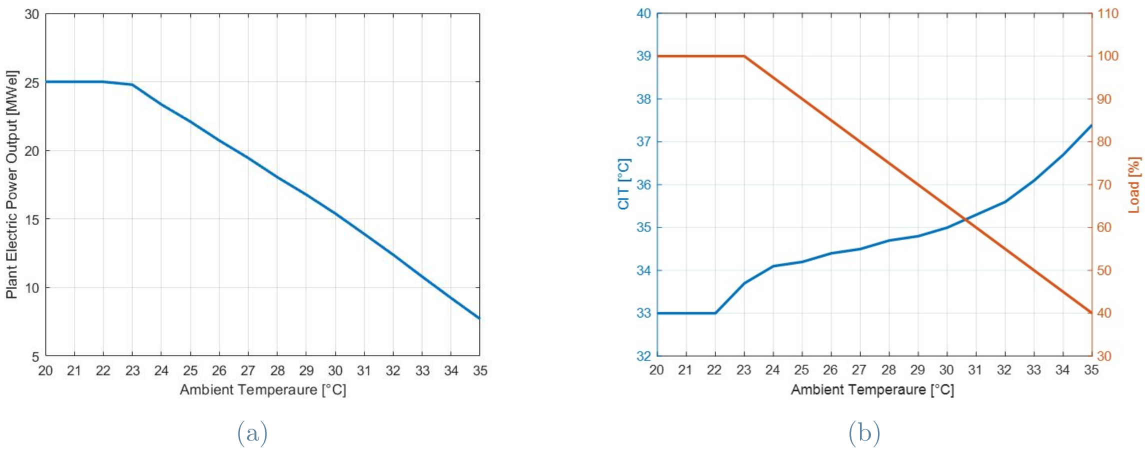


Figure 2.10: Maximum plant electric power output (a) and regulation values (b) with the increase of the ambient temperature[6].

In Figure 2.10b are showed the the trend of the adjustment input values necessary to run the plant as intended by the chosen operating strategy in paragraph 2.2.3. This result will be also presented in the Table 2.1 where the specific numbers that form this graphs can be seen, thus permitting a better understanding of the performances..

Table 2.1: Plant performance with current operating strategy at variable ambient temperature

T_{amb} [°C]	CIT [°C]	Load [%]	Wel_{max} [MWel]	Wel_{max}/W_{nom} [%]
20	33.0	100	25.0	100
21	33.0	100	24.9	99.7
22	33.0	100	24.8	99.2
23	33.9	100	24.5	98.3
24	34.1	95	23.2	92.9
25	34.2	90	22.0	87.8
26	34.4	85	20.6	82.5
27	34.5	80	19.3	77.3
28	34.7	75	18.0	72.0
29	34.8	70	16.7	66.8
30	35.0	65	15.3	61.4
31	35.3	60	13.9	55.6
32	35.6	55	12.4	49.7
33	36.1	50	10.9	43.5
34	36.7	45	9.4	37.5
35	37.4	40	7.9	31.6

Is the last column is showed the percentage of electric power that the plant is able to supply to the grid compared to the nominal one of 25 MWel. From this value is possible to understand how critical the increase of ambient temperature is for the sCO2-flex cycle, as reported in the table for a temperature of 35°C the maximum power output is around 30 % of the nominal one, a huge downgrade.
This is particularly significant also for due to the Peak Load operation that this plant is meant to do, due to the importance of this operation the plant cannot afford this strong reliance with the ambient temperature that will strongly limit its introduction in the energy market.

2.2.5. Conclusion for variable ambient temperature operation

In conclusion is possible to point out the main results of the off-design operation described above. The main advantage is that the flow rate operability limit of the main

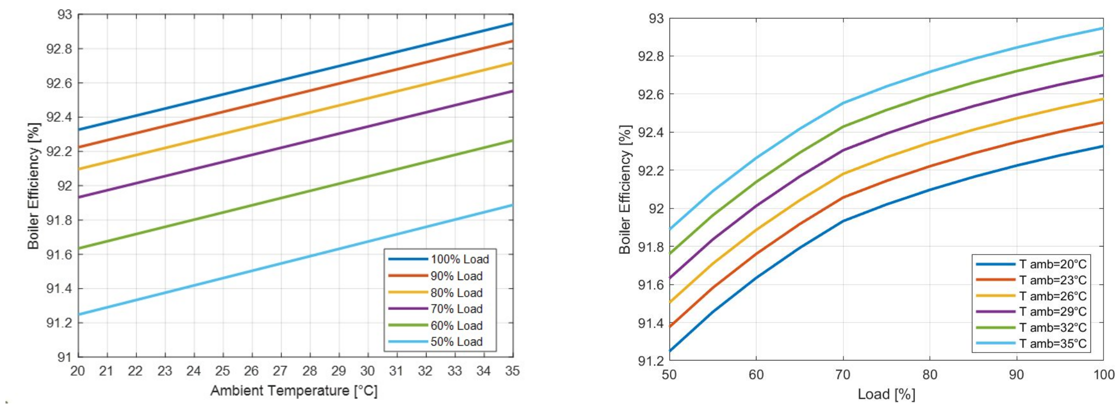
compressor is overcame, granting the possibility to operate the plant at higher ambient temperatures that the ones previously thought of 23.6°C (limit of the main compressor map reached). However this is still not a solution of the problem due to the important limitation of this strategy in term of plant performances. Simulations have been carried out considering steps of 0.1°C for the CIT and 5% for the Load, that showed, at slightly higher temperatures than the nominal one, that the maximum power output of the plant is already strongly limited. Additional increases of the ambient temperature accentuates even further this limitation, reaching a value of around 8 MWel as maximum power output at 35°C (Table 2.1), in contrast with the nominal operation that is supposed to grant 25 MWel, thus making the plant unreliable for peak load operation.
This strong limitations require the necessity to investigate different alternative solutions that will be exploited in the subsequent chapters.

3 | Alternative operating strategy at constant cycle minimum pressure

In the Chapter 2 is shown how the sCO2-flex cycle behaves in off-design conditions; in particular, despite the good performance in term of flexibility assessed in the the part-load analysis in paragraph 2.1, in the paragraph 2.2 is introduced the strong limitation that the ambient temperature increase pose to the plant performance.
This limitation has been partially overcame with an operation strategy that makes possible to operate after the limit of the main compressor map is reached, however this comes at not negligible cost in term of plant performance. The maximum electric power that the plant is able to produce at slightly higher temperatures than the nominal one will be already strongly limited, and at even higher temperature value the situation gets worse, reaching a maximum electric power production of around 8 MWel at 35°C in opposition to the 25 MWel at design conditions as shown in Table 2.1.
This is a strong limitation for the plant that will not be able to effectively work in Peak Load operation, where is designed to operate, if the ambient temperature is too high. Hence different methods will be evaluated in order to mitigate the ambient temperature effect and still be able to obtain a satisfying power produced by the plant in any condition.
Remembering the considerations made in paragraph 2.2.2 regarding the compressor inlet conditions, increasing the minimum operating pressure of the cycle may allow to keep a constant volumetric flow rate but it will also cause a reduction of the plant pressure ratio and consequent efficiency loss. Hence in this chapter the operating strategies proposed will focus on keeping the minimum pressure of the cycle as constant.
However operating strategies with variable minimum pressure will still be exploited in Chapter 4.

3.1. Assumption on the Boiler operation

Before the considering new operating strategies is important to assess how the boiler operates in this particular off-design conditions. In Chapter 2 it has been presented how the boiler operates in Part-Load conditions, in the the paragraph 2.1.2 is defined how the component is modelled for part load operation and the trend of its efficiency as a function of the load of the plant is showed in Figure 2.3. Still, the boiler operation will be also influenced by the ambient temperature, different values of the inlet air temperature could lead to changes in the boiler performances in term of combustion and heat exchange. An higher value of air temperature could possibly increase the temperature at the outlet of the air preheater improving the combustion efficiency, however as mentioned in Chapter 1 the materials of the APH are designed for a limited temperature at the air hot side of 350°C. Hence, it will not be possible to surpass that temperature limit and the rotational speed of the Ljungström will need to be reduced in order to limit the heat exchange. This will still translate in an increase of the boiler efficiency as shown in Figure 3.1 due to the increase of the effectiveness and a reduction of the energy consumption of the APH.



(a) (b)

Figure 3.1: Boiler efficiency at variable ambient temperatures and operating loads

Despite granting an increase of the boiler and consequently plant performance this benefit is negligible in the overall efficiency computation, the boiler efficiency will increase by less than 0.6% of its own value only when the ambient temperature reaches a value of 35°C . Figure 3.1 shows also that the temperature affects the efficiency of the boiler independently by its Load of operation. From the trend of both graphs is possible to assume that the Load and ambient temperature affect the boiler independently through each other.
This considerations will be useful for further analysis, in fact in the investigation of the following operation strategies will be made with the assumption that the boiler works with

air in design temperature condition. This will result to be helpful in order to simplify their computational weight and still obtaining valid results. The strategies that will be explored in this chapter will consider acting on different control variables of the cycle, in order to find a beneficial effect, that will be able to overcome the compressor volumetric flow rate limitation set by the ambient temperature increase.

3.2. Operation at variable TIT

The first strategy suggested consisted in operating at different Turbine Inlet Temperatures (TIT variation). Regulation is possible acting on the IGV and RPM of the main compressor in order to vary the CO2 mass flow rate value. The variation of the mass flow rate will then translate into a TIT variation thanks to the turbine performance curve. Thanks to the assumption pose in paragraph 3.1 is thus possible to analyze the plant performance separately from the ambient temperature and observe if the TIT variation is beneficial in term of reducing the main compressor volumetric flow rate.

3.2.1. Effects on the plant performance

In Figure 3.2 is shown the T-s diagram with different TIT operations at constant inlet compressor values, is possible to notice the variation of the operating thermodynamic points. Is important to highlight that only operations with a lower TIT value are analyzed due to the limits on the materials of the components that are not able to stand higher temperatures.

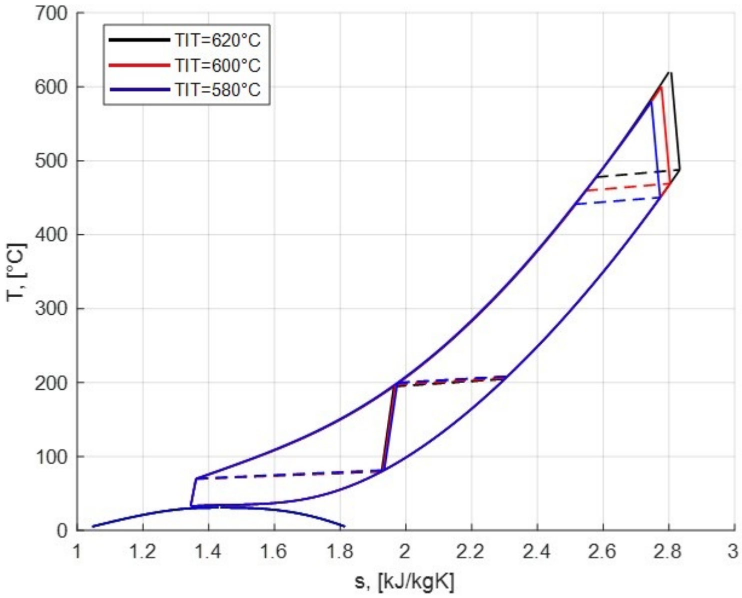


Figure 3.2: T-s diagram, operation at different TIT

From the graph above is possible to notice that the cycle operates at very different conditions according to the TIT, as is also possible to see from the main thermal and electric powers exchanges shown in Table 3.1.

Table 3.1: Main thermal and Electric Powers values of the cycles according to different TIT

TIT [°C]	Q_{IN} [MWth]	Q_{HTR} [MWth]	$Q_{HTR,Bypass}$ [MWth]	Q_{LTR} [MWth]	Q_{HRU} [MWth]	$W_{el,gross}$ [MWel]
620	58.8	87.1	11.0	41.2	31.9	25.8
610	58.8	84.7	10.8	41.6	32.1	25.0
600	58.8	82.2	10.6	42.1	32.4	24.7
590	58.8	79.6	10.4	42.6	32.8	24.4
580	58.8	77.0	10.3	43.1	33.1	24.1
570	58.8	74.3	10.1	43.6	33.4	23.7

From this table is possible to notice that decreasing the TIT the plant will still absorb the same thermal power from the boiler but the recuperative effect is much lower as well as the electric production due to the operation at lower temperature conditions.

This factor leads also to a plant with a lower value of cycle efficiency, thus assessing that in term of plant performances is not beneficial to lower the TIT.

3.2.2. Key values to define variable ambient temperature operation

Is now important to evaluate if despite being not beneficial in terms of plant performances a TIT decrease could still lead to better values for the compressor operation, key component for the ambient temperature limitations.
Unfortunately this is not the case, in the Table 3.2 are showed the variation on the mass flow rates according to the TIT variation and is possible to notice that the total mass flow rate as well as the mass flow rate that flows in the main compressor both increases.

Table 3.2: Mass flow rates at different TIT

TIT	**620°C**	**610°C**	**600°C**	**590°C**	**580°C**	**570°C**
$\dot{m}_{CO2}$ **[kg/s]**	267.83	270.20	272.75	275.35	278.00	280.71
$\dot{m}_{CO2,MC}$ **[kg/s]**	172.43	173.15	174.04	174.98	175.99	177.06
$\dot{m}_{CO2,SC}$ **[kg/s]**	95.40	97.06	98.71	100.36	102.00	103.65

This is detrimental for the main compressor operation, higher values of mass flow rate at the same inlet conditions leads to higher volumetric flow rates that wants to be avoided for the operation at higher ambient temperatures.
Regarding compressor operations both appear to operate at higher enthalpy and volumetric flow ratios as showed in Figure 3.3.

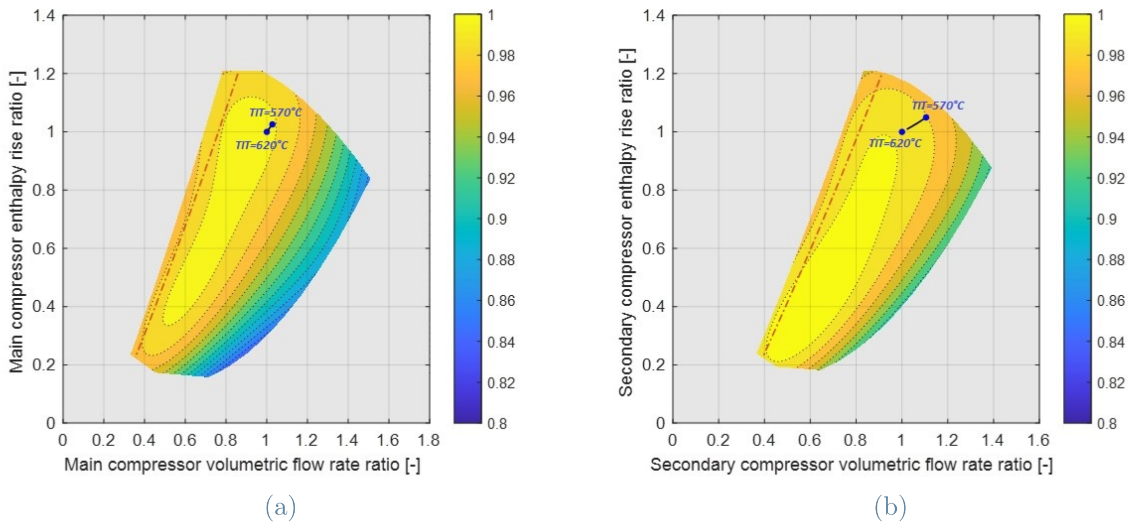


Figure 3.3: Main (a) and Secondary (b) Compressor maps at variable TIT operation

In conclusion, the operation at variable TIT thus showed that a reduction of this value will not bring any benefit in term of plant performance at design or with the variation of the ambient temperature, and even if an increase of this value in theory could bring beneficial results this operation is not possible due to the components constrain.

3.3. Operation at variable Split Ratio

Throughout the paragraph 2.2.3 is defined how the increase of the ambient temperature is a problem for the plant operation due to the increase of the volumetric flow rate at the main compressor inlet. In this paragraph is decided to intervene on this problem taking advantage of the cycle layout that presents two separate compressors that suffer to opposite problems with the increase of the ambient temperature as shown in paragraph 2.2.4. The secondary compressor the volumetric flow rate goes through a decrease, while the main compressor volumetric flow rate increase to a point that is not possible to operate without reducing the load.

Thanks to that peculiar condition, in previous works, when the operation at different ambient temperatures is discussed, is suggested to exploit that in order to make the problems cancel each other.

This operating strategy consists in the variation of the split ratio as the ambient temperature increase. This could make possible to redirect the excessive flow rate of the main compressor to the secondary, preserving the system operability at higher temperatures without limiting the fuel mass flow rate.

3.3.1. Split Ratio definition

Before the investigation of this solution is important to determine what is the Split Ratio, the key parameter on which this operating strategy will base on.
In Figure 3.4 is showed a portion of the plant layout, in particular is a focus on the secondary compressor with its close connections.

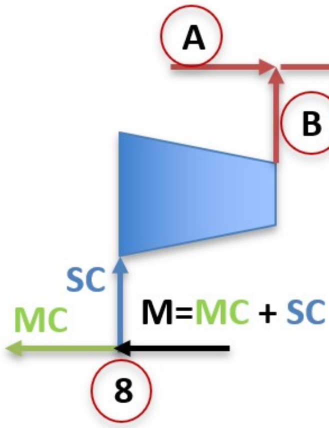


Figure 3.4: Simplified plant layout: Focus secondary compressor

Where is possible to notice the already known thermodynamic points **3** and **8** that are respectively the inlet of the cold side of the HTR and the outlet of the hot side. In this scheme is also shown the value **M** that corresponds to the mass flow rate that comes from the HTR, the value **SC** that is the mass flow rate that goes into the Secondary compressor and lastly **MC** that corresponds to the mass flow rate that goes into the HRU and subsequently into the main compressor. Thanks to this values is possible to define the split ratio (**SR**) as:

$$SR = \frac{MC}{M} \tag{3.1}$$

This value thus define how much mass flow rate with respect to the total each compressor process. An increase of the SR with constant total mass flow of CO2 translates into higher flow rate towards the main compressor and lower towards the secondary and vice versa.

3.3.2. Variation of the split ratio

From the previous paragraph the definition of split ratio is obtained, an is suggested how this value is important to define how the two compressors divide the mass flow rate that needs to be processed.
Thanks to a decrease of the value of the split ratio it would therefore be possible to shift

the excess of mass flow rate processed by the main compressor towards the secondary, however this is not as simple as it seems to be. The split ratio is useful to assess how the mass flow rates are divided but is not a value that is possible to be directly controlled. Hence for the simulations another input variable that will indirectly changes the SR needs to be found

Connection between $\Delta T_{mix,LTR}$ and Split Ratio

The input variable able to affect the split ratio is the $\Delta T_{mix,LTR}$, this value is defined as the temperature difference of the two streams at the outlet of the LTR cold side. In particular thanks to Figure 3.4 this value can be defined as:

$$\Delta T_{mix,LTR} = T_A - T_B \tag{3.2}$$

With T_A and T_B being the temperatures of the streams A and B respectively. Where **A** defines the outlet LTR cold side stream and and **B** the secondary compressor outlet stream.
This value is usually set to 0 in order to avoid mixing losses, however increasing this value leads to an higher value of T_A that can be reached with the LTR only if the mass flow rate processed by the cold side decreases. Hence, the split ration with the mass flow rate directed towards the main compressor decreases.
In Figure 3.5 is shown the variation of the split ratio as respect to the $\Delta T_{mix,LTR}$.

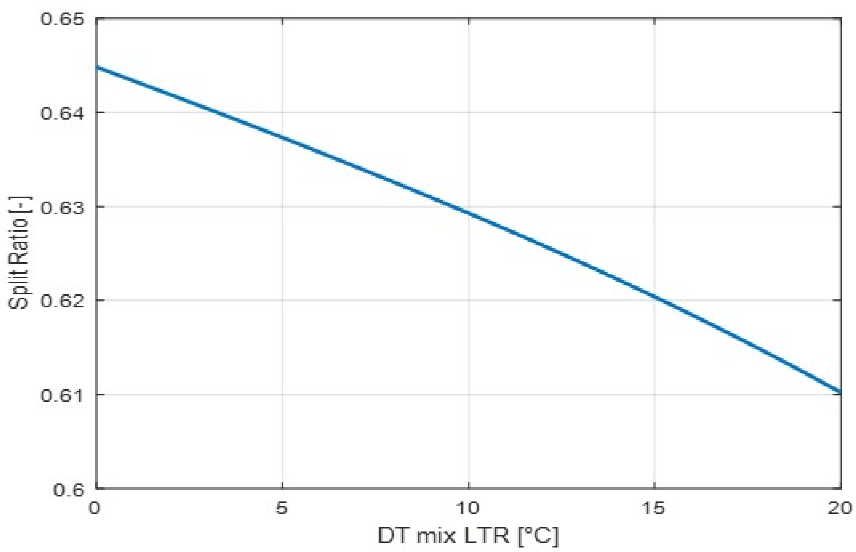


Figure 3.5: Effect of the $\Delta T_{mix,LTR}$ on the SR

From the graph is possible to notice that the range of regulation is very limited, the value of the SR from the design value of 0.6438 gradually decreases with the increase of the $\Delta T_{mix,LTR}$ but even after reaching a temperature difference of 20°C the SR will only be 0.6102.
Moreover the variation of the split ratio above certain limits presents also another problem for the operation of the secondary compressor. Due to this limit at nominal load is not possible to operate with $\Delta T_{mix,LTR}$ greater than 20°C. This limitation it is better described in the paragraph 3.3.4.

3.3.3. Effects of the variation of the $\Delta T_{mix,LTR}$

Now that the connection between the SR and the $\Delta T_{mix,LTR}$ it has been properly described is possible to reconsider the suggested operating solution at paragraph 3.3 as respect to this latter value instead of the SR. The decision to operate with the $\Delta T_{mix,LTR}$ is made because it is simpler to impose from a numerical point of view
Therefore, in this paragraph is analyzed how the variation of the $\Delta T_{mix,LTR}$ affects the cycle and plant performance.

T-s Diagrams

Changing the $\Delta T_{mix,LTR}$ leads to a clear variation of the thermodynamic points of the cycle as shown in Figure 3.6.

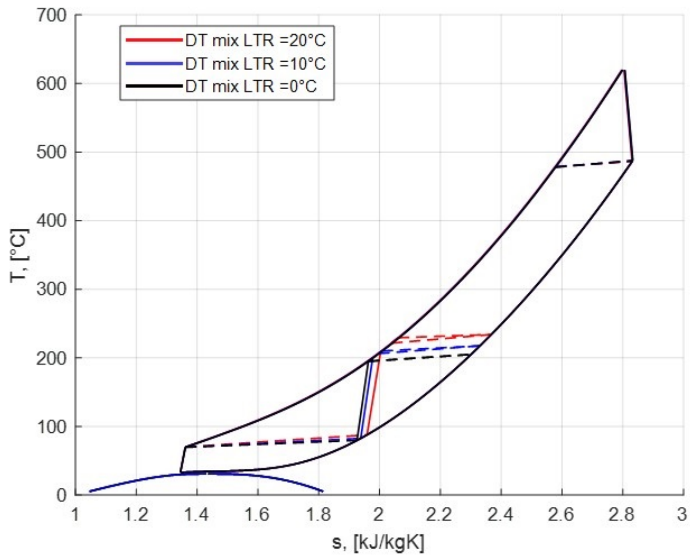


Figure 3.6: T-s diagram, operation at different $\Delta T_{mix,LTR}$

From the T-s diagram is possible to notice that the increase of $\Delta T_{mix,LTR}$ leads to different operating points in particular in the LTR area of operation, where the hot-end temperatures significantly increases as well as the outlet temperature of the hot side. However the thermodynamic points related to the turbine, main compressor as well as the PHE remains almost unchanged.

Mixing outlet cold side LTR

A particular point that is interesting to evaluate is the mixing outlet cold side of the LTR. Is possible to notice in Figure 3.7 that an increase of the $\Delta T_{mix,LTR}$ leads to an increase of not only the outlet temperature of the cold side of the LTR($T_{LTR,cold,out}$), but also the outlet temperature of the secondary compressor ($T_{SC,out}$) and consequently the inlet temperature of the cold side of the HTR($T_{HTR,cold,in}$) will increase.

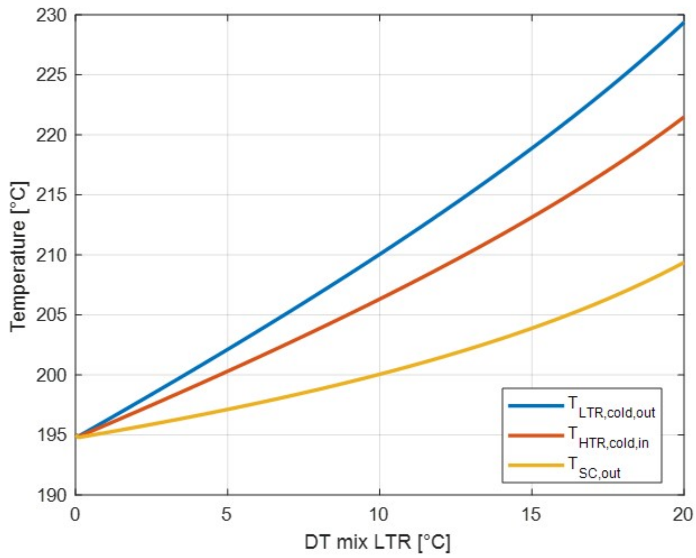


Figure 3.7: Mixing temperatures at LTR cold side outlet at different $\Delta T_{mix,LTR}$

This increase is caused by the $T_{LTR,cold,out}$ of increasing at a higher rate in comparison to the $\Delta T_{mix,LTR}$ due to the LTR characteristics, hence $T_{SC,out}$ needs also to increase to preserve the balance. From the T-s diagram in Figure 3.6 is showed that in order to correctly operate the secondary compressor the temperature at the inlet of the component will slightly increase, granting also an higher temperature difference in the HRU.

3.3.4. Plant performance at different values of $\Delta T_{mix,LTR}$

This new operation of the will lead to a different performance values of the plant. Also in this situation the boiler fueled with the same coal flow rate provides the cycle with a constant heat power throughout all the range of $\Delta T_{mix,LTR}$ examined. However the plant gross electric power produced as well as the cycle efficiency will vary as shown is Figure 3.8, highlighting a detrimental effect of the $\Delta T_{mix,LTR}$ in term of plant performances.

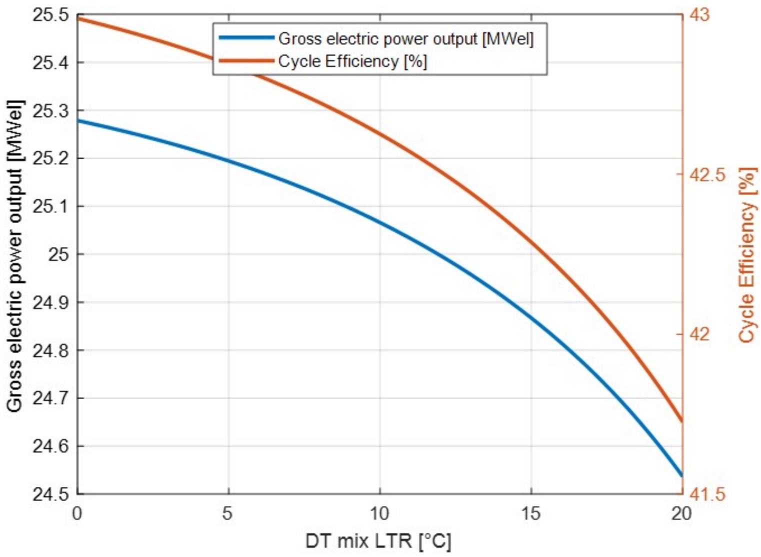


Figure 3.8: Gross electric power production and cycle efficiency at different $\Delta T_{mix,LTR}$

The plant performance can also be further evaluated thanks to the Table 3.3 where the main thermal and electric powers exchanged throughout the cycle are showed for three different $\Delta T_{mix,LTR}$.

Table 3.3: Main electric and thermal powers at different $\Delta T_{mix,LTR}$ operation

	Q_{IN} [MWth]	Q_{HRU} [MWth]	Q_{HTR} [MWth]	Q_{LTR} [MWth]	Q_{REC} [MWth]	$W_{el,gross}$ [MWel]
$\Delta T_{mix,LTR}$=0°C	58.80	31.88	87.07	41.23	128.30	25.28
$\Delta T_{mix,LTR}$=10°C	58.80	32.07	83.51	44.17	127.68	25.07
$\Delta T_{mix,LTR}$=20°C	58.80	32.56	78.82	47.55	126.37	24.54

Is possible to notice how the increase of $\Delta T_{mix,LTR}$ affects the thermal power exchanges in the cycle. Despite the constant value of inlet thermal power as shown in the column Q_{IN}

the gross electric power generates by the plant decreases and is confirmed the assumption made before that the blame can be put on the recuperators performance. From the column Q_{REC} that represent the thermal power exchanged in both LTR and HTR, is possible to notice that the energy recovered decreases, thus causing an higher value of thermal power that needs to be disperse in the environment as noticeable in the column related to the Q_{HRU} and also a lower cycle efficiency. In particular the LTR increases its thermal recovery due to the increase of the its operation temperatures as well as the temperature variation across the cold side. However, this cuts out a large operating area for the HTR that operating in inefficient conditions is not able to recover as in design condition and the effect on the latter showed to be more prominent than the benefit on the LTR.

Effects on the mass flow rates

In order to evaluate if changes of $\Delta T_{mix,LTR}$ could be beneficial in conditions with variable ambient temperatures is necessary to assess how the mass flow rates changes throughout the cycle. This evaluation is important because the reduction of the split ratio being a relative value doesn't directly translates into a decrease of the mass flow rate towards the main compressor but only that its percentage on the total mass flow rate decreases.
The evolution of the mass flows is thus showed in Table 3.4.

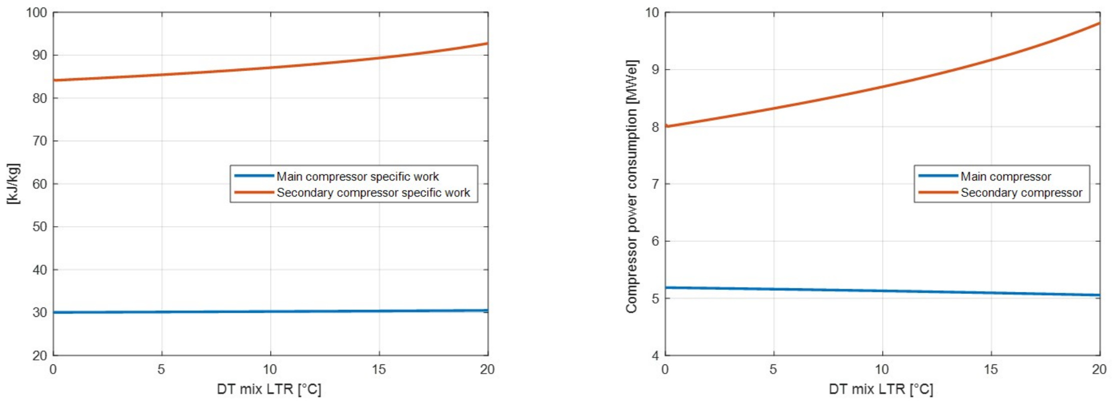
Table 3.4: Mass flow rates at different $\Delta T_{mix,LTR}$ operation

$\Delta T_{mix,LTR}$	**0 [°C]**	**5 [°C]**	**10 [°C]**	**15 [°C]**	**20 [°C]**
M [kg/s]	267.83	268.51	269.37	270.31	271.41
MC [kg/s]	172.43	171.13	169.51	167.70	165.62
SC [kg/s]	95.40	97.38	99.86	102.61	105.97

From the Table 3.4 is noticed that the new thermodynamic conditions of the cycle leads to and increase of the total CO_2 mass flow rate that may lead to a constant(or higher) mass flow towards the main compressor even with a decreasing SR. Fortunately this is not the case, in Table is showed that even if the total mass increases part of the mass flow rate from the main compressor shift towards the secondary compressor. However, both this effects combined leads to a significant increase of the mass flow rate processed by the secondary compressor causing operational problems.

Compressors operation with the $\Delta T_{mix,LTR}$ variation

In the previous chapter has been assessed how the shift of the mass flow rate could be beneficial for the compressor performances. Therefore, is important to perform a thorough evaluation of how the operation conditions changes for the compressors.



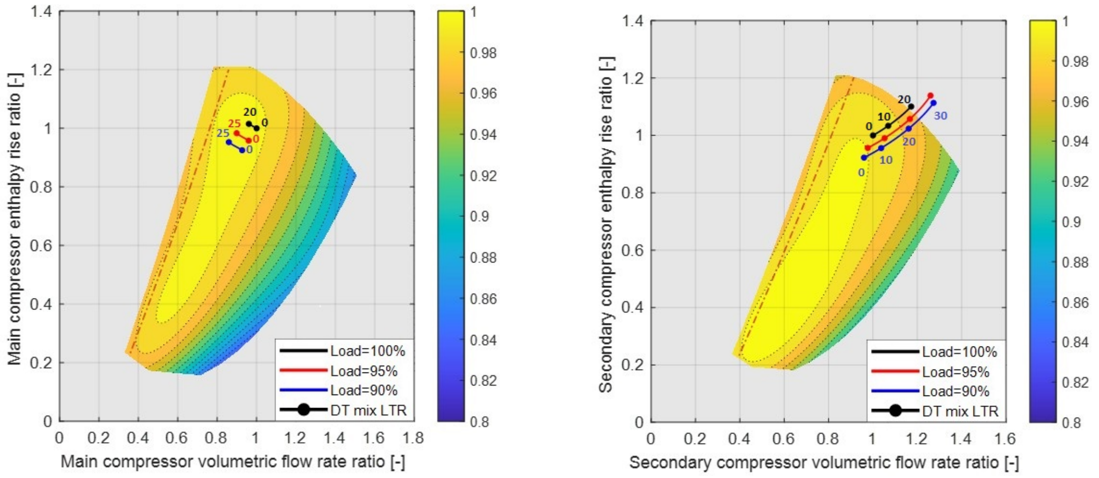
(a) (b)

Figure 3.9: Specific work (a) and electric consumption (b) of Main and secondary compressor at variable $\Delta T_{mix,LTR}$ operation

In particular regarding the main compressor from Figure 3.9a is noticeable that the specific work remains almost constant due to the unchanged thermodynamic operating conditions, however the impact of the mass flow rate decrease can be showed in Figure 3.9b where the consumption of the main compressor decreases.
However, for the secondary compressor the situation is different, from Figure 3.9a is possible to notice that the turbomachine operates in different thermodynamic conditions that requires a bigger specific work to operate, and additionally the increase of the processed mass flow rate showed an huge impact on the compressor consumption as showed in Figure 3.9b.

Compressor maps and operation limits

The performance of the compressors can be further evaluated with the aid of the compressor maps showed in Figure 3.10.



(a) (b)

Figure 3.10: Main (a) and secondary (b) compressor performance maps at variable Load and $\Delta T_{mix,LTR}$ operation

In this maps is showed again that the $\Delta T_{mix,LTR}$ variation moves the main compressor towards and operating condition at lower volume flow rate and this effect is similar for the operation at different loads. However, the operation of the secondary compressor showed far more interesting results. From Table 3.4 it has been noticed that this operation leads to a significant increase of the mass flow rate towards than the secondary compressor, thus leading to operation conditions outside its performance map. Therefore the secondary compressor will limit the range of operation of the $\Delta T_{mix,LTR}$ greatly affecting the possibilities of this operation strategy. From the map at Figure 3.10b is noticeable that the $\Delta T_{mix,LTR}$ limit, being directly related to the volumetric flow rate, increases with the operation at lower loads. However the variation is small, moving from 19.1°C at design condition to 23.0°C at 90% load.

3.3.5. Operating strategy considering $\Delta T_{mix,LTR}$ variation

In this paragraph a set of control strategies that exploit the possibility to operate at different $\Delta T_{mix,LTR}$ are presented. The strategy is strongly based on the already adopted one presented in paragraph 2.2.3, hence for ambient temperatures greater than the design condition it can be divided in three different regions defined by its ambient temperature.

1. **20°C$<$T$_{amb}$ $\leq$22°C**
   In this operative region the operation strategy is the same as the standard one. Given that the ambient temperature increase is still minimal, in this area of op-

eration the aim is to keep the main compressor inlet condition constant. This is made again increasing the rotational speed of the electronically commuted fans of the HRU, granting the same power dispersed in the environment even with the increase of the air temperature. However, at 22°C the fans reaches their maximum rotational speed, due to the +25% operational limit of the electric motors. Hence, it will not be possible to keep the main compressor inlet conditions fixed.

2. **22°C<$\mathbf{T}_{amb}$ ≤23.6°C**
   As well as before also the operation strategy in this region is the same adopted in Chapter 2. The HRU fan reaches its limit and its not possible to keep the design condition at the main compressor inlet, hence the compressor inlet temperature starts to rise. This area is characterized by changes in the main compressor operation that is represented in its performance map by an increase of both the enthalpy head and the volumetric flow rate, pushing the operating point towards the upper limit of the performance map. This changes are caused by the different thermodynamic condition at the inlet of the compressor like the density coefficient that causes higher volumetric flow rate at same mass flow rates.
   When at 23.6°C the operation point reaches the limit of the map, it is necessary to change temperature region, hence operating strategy.

3. **$\mathbf{T}_{amb}$ > 23.6°C**
   When the ambient temperature reaches the value of 23.6°C the compressor operation is on the limit of the performance map and is necessary to operate differently. Is thus decided to combine the standard operation strategy with the results obtained by the analysis of the $\Delta T_{mix,LTR}$ variation on the compressor performances. In this operative region it will be possible to follow three different routes with different $\Delta T_{mix,LTR}$, $\Delta T_{mix,LTR}$=5°C, $\Delta T_{mix,LTR}$=10°C and $\Delta T_{mix,LTR}$=15°C. This three value have been chosen in order to have multiple possible routes to analyze and considering also the limits on the secondary compressor operation with the increase of the $\Delta T_{mix,LTR}$.
   For each route is then reduced the heat input of the cycle in order to operate the main compressor on the upper limit of its performance map.

This strategy can also be summarized in the scheme below.

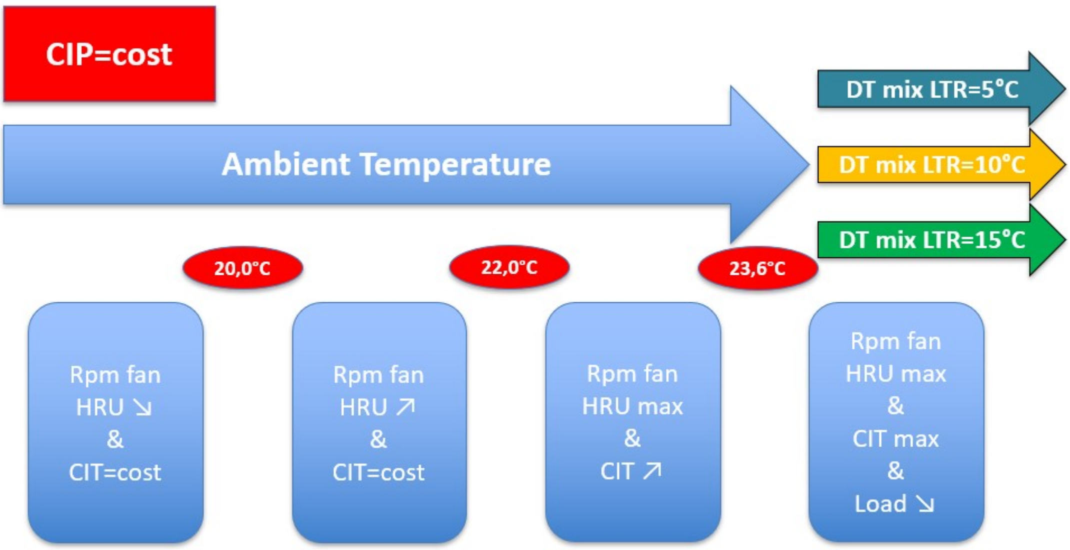


Figure 3.11: Operating strategy to mitigate the ambient temperature effect with $\Delta T_{mix,LTR}$ variation

Where "CIP" stands for the main compressor inlet pressure that is always kept constant, "CIT" is the main compressor inlet temperature, "rpm fan HRU" is the rotational speed of the HRU fans and "Load" corresponds to the fuel input. The three separate arrows of corresponds to the three different possible routes that can be traveled as respect to the three different $\Delta T_{mix,LTR}$.

3.3.6. Plant Performance

Is now possible to assess the performance of the plant with this new operating strategy. The major issue regarding the standard operation was the significant limitation of the maximum power output caused by the abrupt decrease of the Load after the 23°C threshold. This new operating strategy with the aid of the $\Delta T_{mix,LTR}$ aimed to mitigate this limitation thanks to the redistribution of the mass flow rate between the two compressors. Unfortunately, as showed in Figure 3.12, even in this condition the maximum power generated by the plant is very limited with the increase of the ambient temperature, showing values similar to the standard case.

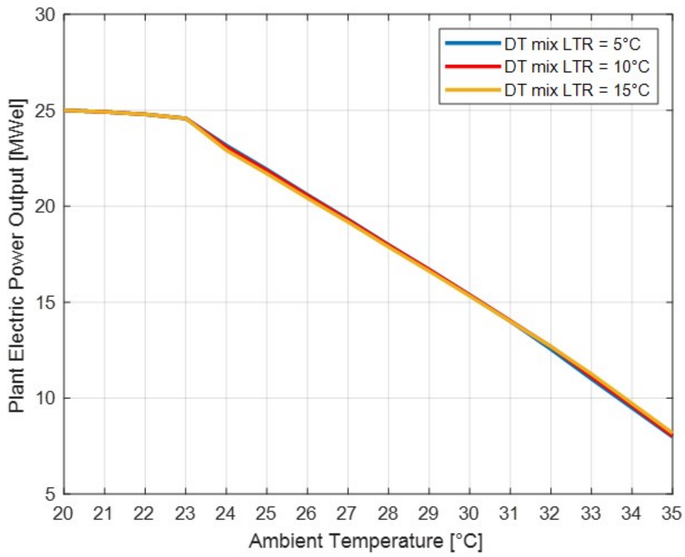


Figure 3.12: Maximum electric power output at variable ambient temperature with different $\Delta T_{mix,LTR}$.

The results showed are consistent with what was showed in the analysis of the plant performance with the $\Delta T_{mix,LTR}$ in paragraph 3.3.4. The shift of the mass flow rate between the main and secondary compressor was supposed to allow a greater range of operation for the plan at higher loads. However this effect resulted to be minimal, due to a little difference between the standard and the new volumetric flow rate of the main compressor. Furthermore, the efficiency drop due to the inefficient operation on the recuperator showed to have a much greater impact, resulting in a total power generated usually lower than the standard case.

3.3.7. Comparison with the standard operating strategy

From the graph in Figure 3.12 is possible to say that the variation of the $\Delta T_{mix,LTR}$ showed no noticeable variation in term of plant performance as respect to the standard operating strategy. However in order to assess that more detailed analysis is required, in this paragraph it will be presented how the main characteristic values of the plant changes in the different operation strategy.
At first in Table 3.5 is possible to notice that despite being very similar the value of maximum power produced slightly changes between the 4 strategies of operation. In particular is showed that a greater $\Delta T_{mix,LTR}$ at temperatures directly above the 23°C threshold have a negative effect in term of power production being lower than the standard one. However the situation is the opposite when the ambient temperature goes above 30°C.

Table 3.5: Maximum electric power output values at variable ambient temperature with different $\Delta T_{mix,LTR}$.

$\mathbf{T}_{amb}$ [°C]	$\mathbf{Wel}_{\Delta T_{mix,LTR}=0}$ [MWel]	$\mathbf{Wel}_{\Delta T_{mix,LTR}=5}$ [MWel]	$\mathbf{Wel}_{\Delta T_{mix,LTR}=10}$ [MWel]	$\mathbf{Wel}_{\Delta T_{mix,LTR}=15}$ [MWel]
20	25.00	25.00	25.00	25.00
21	24.92	24.92	24.92	24.92
22	24.79	24.79	24.79	24.79
23	24.58	24.58	24.58	24.58
24	23.22	23.17	23.08	22.92
25	21.96	21.92	21.85	21.70
26	20.62	20.60	20.55	20.42
27	19.34	19.32	19.28	19.18
28	18.00	17.99	17.97	17.88
29	16.70	16.70	16.70	16.62
30	15.35	15.37	15.37	15.32
31	13.91	14.01	14.01	14.00
32	12.42	12.55	12.55	12.69
33	10.89	11.00	11.13	11.26
34	9.38	9.49	9.61	9.72
35	7.90	7.98	8.04	8.18

In order to better understand what changes in the plant operation two significant ambient temperatures are chosen and an overview of the plant parameters with the $\Delta T_{mix,LTR}$ variation is showed. The temperatures chosen for the analysis are at 24°C where the $\Delta T_{mix,LTR}$ showed negative overall effect, and at 32 where the standard operation is the one with the lowest maximum electric power output.

Table 3.6: Overview plant operation at T$_{amb}$=24°C and T$_{amb}$=32°C at diferent $\Delta T_{mix,LTR}$.

T$_{amb}$=24°C	$\Delta T_{mix,LTR}$=**0°C**	$\Delta T_{mix,LTR}$=**5°C**	$\Delta T_{mix,LTR}$=**10°C**	$\Delta T_{mix,LTR}$=**15°C**
Wel [MW]	23.22	23.17	23.08	22.92
Q$_{in}$ [MW]	55.83	55.83	55.83	55.83
η_{cycle} **[%]**	42.32	42.21	42.04	41.77
$\Delta \dot{m}_{MC}$ **[%]**	0	-0.799	-1.683	-2.692
ΔW_{comp} **[%]**	0	1.849	4.013	6.707
T$_{amb}$=32°C	$\Delta T_{mix,LTR}$=**0°C**	$\Delta T_{mix,LTR}$=**5°C**	$\Delta T_{mix,LTR}$=**10°C**	$\Delta T_{mix,LTR}$=**15°C**
Wel [MW]	12.42	12.55	12.68	12.69
Q$_{in}$ [MW]	32.04	32.04	32.04	32.04
η_{cycle} **[%]**	39.54	39.92	40.31	40.36
$\Delta \dot{m}_{MC}$ **[%]**	0	-0.876	-1.888	-3.211
ΔW_{comp} **[%]**	0	-0.093	-0.239	-0.302

Where $\Delta \dot{m}_{MC}=\frac{\dot{m}_{MC,\Delta Tmix=i}}{\dot{m}_{MC,\Delta Tmix=0}}$, and $\Delta W_{comp}=\frac{W_{comp,\Delta Tmix=i}}{W_{comp,\Delta Tmix=0}}$ with i=5°C,10°C,15°C.
Thanks to the data showed in Table 3.6 is possible to determine that despite higher $\Delta T_{mix,LTR}$ could lead to operations at higher load percentage the thermal input is showed to be the same in both operations. Therefore, the variation of the electric power produced is caused by more or less efficient operation of the cycle as highlighted by the η_{cycle}.
The answer to this different operation can be found in the variation of the impact that the $\Delta T_{mix,LTR}$ on the redistribution of the mass flow rates. From the Table 3.6 looking at the $\Delta \dot{m}_{MC}$ value is possible to notice that at higher temperatures the same $\Delta T_{mix,LTR}$ variation cause higher decreases of the mass flow rate processed by the main compressor. This resulted beneficial in term of compressor performances, while for the T$_{amb}$=24°C the power consumed by the compressors increases, for the T$_{amb}$=32°C case the reduction of the mass flow rate towards the main compressor is enough to overcome the increase of the secondary compressor consumption thus leading to an overall consumption of the compressors lower than the $\Delta T_{mix,LTR}$=0°C case.

3.3.8. Conclusions on the operating strategy considering $\Delta T_{mix,LTR}$ variation

In conclusion is possible to point out the main results of the off-design operating strategy described above. The analysis of the split ratio at first showed promising results, the possibility to redirect the mass flow rate from the main compressor towards the secondary thus tackling the problem of excessive volume flow rate in the main compressor that reduces the system operability with the increase of the ambient temperature. This was suggested also due to the reduction of the volumetric flow rate that the increased ambient temperature causes to the secondary compressor operation.

Unfortunately this method presented the huge limitation of requiring high $\Delta T_{mix,LTR}$ variations in order to have minimal changes of the SR and also a reduced range of operation due to the excessive increase of the volumetric flow rate of the secondary compressor. In fact at 100% load the $\Delta T_{mix,LTR}$ range of operation is limited from 0°C to 19.1°C that translates into a SR reduction of less than 5% (From 0.6438 to 0.6122).

Taking into account this limitations, in order to tackle the ambient temperature effect on the cycle, after the limit of the main compressor map is reached, is added to the standard operation the possibility to operate at three different $\Delta T_{mix,LTR}$ values, 5°C, 10°C and 15°C. However this three possibilities showed to produce no clear improvement in term maximum electric power produced despite the the possibility to operate at slightly greater loads granted by the decrease of the main compressor volumetric flow rate. This is because the inefficient operation of the recuperators is showed to decrease significantly the cycle efficiency, thus having a much greater impact on the maximum electric power produced.

This solutions proved to have only a slight advantage at high ambient temperatures where the $\Delta T_{mix,LTR}$ showed to have a greater impact on the decrease of the main compressor mass flow rate, thus reducing the power consumption of the compressors.

However, this cannot be considered enough in order to consider this operation strategy a viable solution for the problem of the ambient temperature increase. The maximum power output of the plant is similar to the case with standard operation and thus strongly limited.

Again, this strong limitations require the necessity to investigate different alternative solutions. Since the alternative operation strategies at constant minimum pressure that have been presented in this chapter have shown no benefits, in the following chapter operative strategies at variable minimum pressure will be exploited.

4 | Alternative operating strategy at variable cycle minimum pressure

The purpose of this chapter is to evaluate the benefits of adopting variable minimum pressure for the cycle operation. In chapter 2 it has been presented how operate ad different minimum pressure in part-load operation could improve the performances of the plant. Decreasing the minimum pressure accordingly to the decrease of the load showed to improve the efficiency of the cycle due to the increase of the pressure ratio.
However, when the problem of the increase of the ambient temperature needs to be tackled a reduction of the minimum pressure will result to be detrimental. In Figure 4.1 is in fact possible to notice that a decrease of the main compressor inlet pressure leads to areas of operation where the ratio between the local density and the one in nominal conditions is lower than 1.

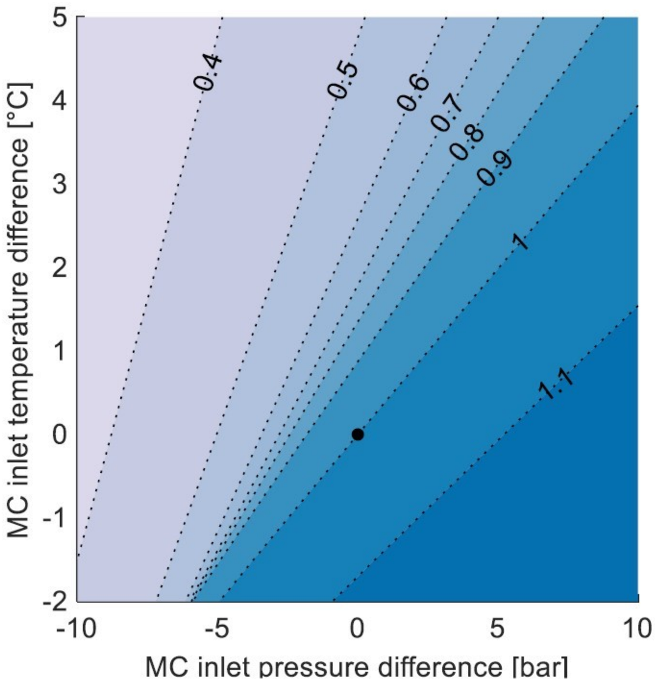


Figure 4.1: Ratio between the local density of CO2 and the density at main compressor inlet point in nominal conditions[6].

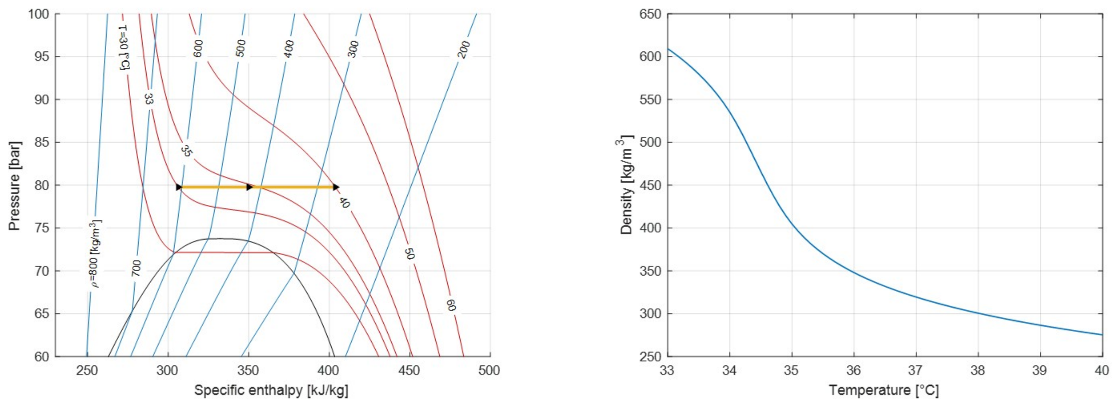
The lower density presented by the new thermodynamic conditions of the main compressor inlet translates into a greater volumetric flow rate with the same mass flow processed by the turbomachine. In case of increased ambient temperature operation, this will prematurely lead the main compressor to reach the limit of its performance map.
However, also the opposite regulatory procedure is possible, from the map in Figure 4.1 is shown that increasing the pressure at the inlet of the compressor leads to areas with densities higher than the nominal one.
In chapter 2, the operation strategy that focus on increasing the minimum pressure of the cycle was discarded. This was due to the lower cycle efficiencies caused by the reduction of the pressure ratio of the cycle.
However, a regulated increase of the compressor inlet pressure could effectively tackle the abrupt increase of the volumetric flow rate that limits the compressor operation at high ambient temperature operations. This possibility, with the strong limits presented by the previously exploited solution strategies, showed that despite leading to possibly lower cycle efficiencies strategies that includes the possibility to increase the minimum pressure are worthy to be investigated.

4.1. Variation of Compressor inlet temperature and pressure

Before proceeding into the actual operation strategies suggested is important to define what does actually mean to change the main compressor inlet conditions. In particular the compressor inlet temperature (CIT) and compressor inlet pressure (CIP) will be evaluated.

4.1.1. Compressor inlet temperature

The increase of the main compressor inlet temperature is a critical issue in term of plant operation. From the pressure enthalpy diagram in Figure 4.2a is possible to notice that at constant pressure the thermodynamic conditions significantly vary even with slight increases of temperatures. The operating condition gradually moves away from the supercritical area, thus losing real gas behaviour.



(a) (b)

Figure 4.2: Pressure enthalpy diagram (a) and density at different CIT(b)

This causes a huge variation in the density at the inlet compressor. From the graph in Figure 4.2b] the density is shown to abruptly decrease after a little increase in temperature condition, dropping by a third after only 2°C (from 33 to 35°C). After that it continues to decrease reaching half of its nominal value at 38°C. This will translates into a direct increase of the volumetric flow rate by a value reciprocal of the density decrease.

How is possible to control the CIT

In order to use the CIT as a control variable a method to vary the temperature at the inlet compressor is necessary. Therefore, the possibility of controlling the rotational speed

of the electronically commutated fans of the HRU will again prove to be useful.
Different rotational speeds will decrease or increase the heat power released into the environment by the HRU.

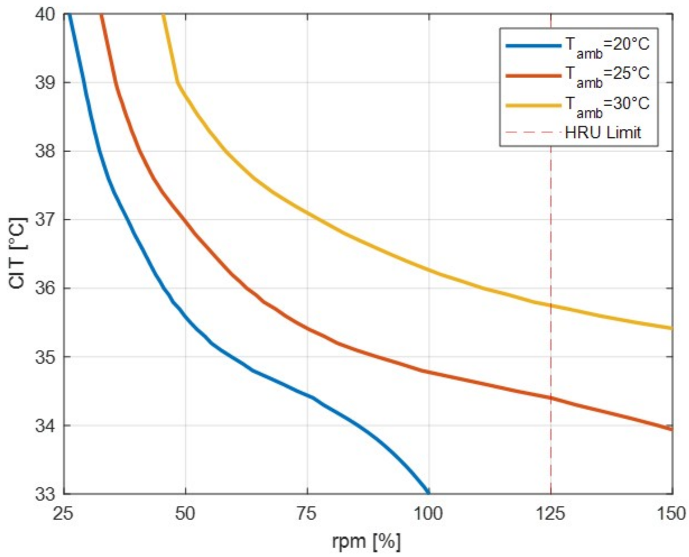


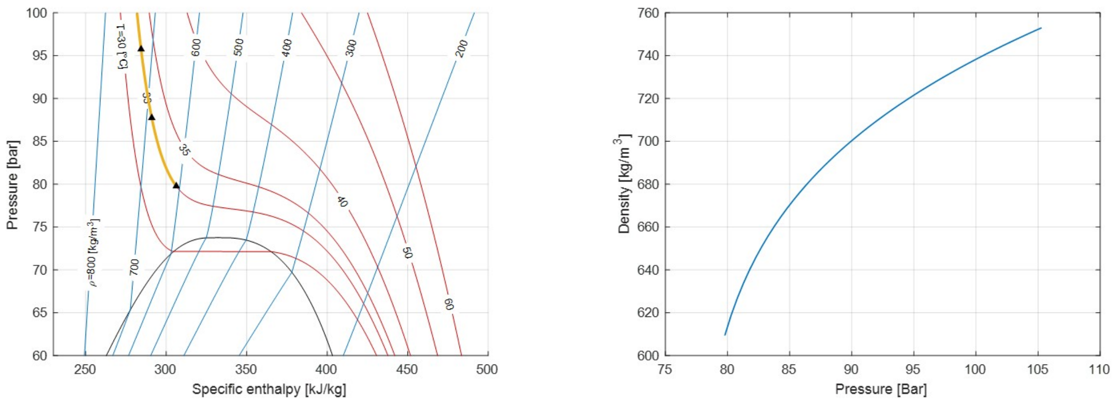
Figure 4.3: Compressor inlet temperature variation as respect to HRU fans rotational speed percentage at different T_{amb}.

From Figure 4.3 in particular is possible to see the graph that shows the percentage of rotational speed that the HRU fans needs to run to achieve a determined value of CIT with different ambient temperatures. Is possible to point out that while for ambient temperature design conditions (T_{amb}=20°C) the range of operation in order to increase the CIT til 40°C is from 100% to around 25%, when the ambient temperature increases it will became slightly easier to reach higher temperatures as the curves shift upwards. However, the main issue is that it will become impossible to keep the CIT low through this method. As mentioned in the HRU characteristics the fan rotational speed has a limit of +25% and form the graph is clear that the limit prevents from reaching CIT=34,5°C when T_{amb}=25°C and CIT=35,7°C when T_{amb}=30°C.
However, is important to point out that the evaluation is made considering only the variation of the rotational speed of the HRU fans and the T_{amb} as variables. Other operating conditions may shift the results towards higher or lower CIT at the same rotational speed.

4.1.2. Compressor inlet pressure

The variation of the compressor inlet pressure is important to evaluate in order to effectively define the new operating strategy. It has already been introduced in Figure 4.1

the possibility to keep the density ratio constant with the increase of the CIT. From the pressure enthalpy diagram in Figure 4.2a is possible to notice that at constant temperature the operating condition moves along the isothermal curve towards lower enthalpy and higher density conditions.



(a) (b)

Figure 4.4: Pressure enthalpy diagram (a) and density at different CIP(b)

The density variation caused by this operation is showed in Figure 4.4a, is possible to notice that the value increases along with the pressure but the rate of increase is shown to be much lower than the increase caused by the CIT.

The pressure inlet needs to increase by 10 bar in order to generate an increase of the density value of 15%, while for an increase of 20 bar the increase is still only around 22% if its nominal value. However, despite having a less sensitive regulation of the density value than the CIT regulation, the increase of the pressure inlet of the compressor could help since increasing the density value the volumetric flow rate will decrease accordingly.

How is possible to control the CIP

In order to use the CIP as a control variable however a method to vary the pressure at the inlet compressor is necessary. Therefore, the possibility to act on the plant inventory is introduced.

The sCO2-Flex cycle is in fact a closed loop cycle, therefore injecting additional fluid mass into the plant will change the operating conditions and due to the increase of the mass in the same volume the pressure increases.

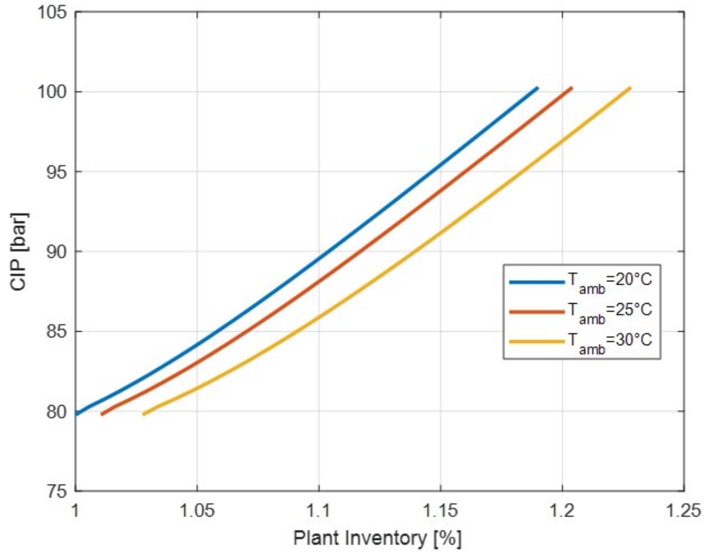
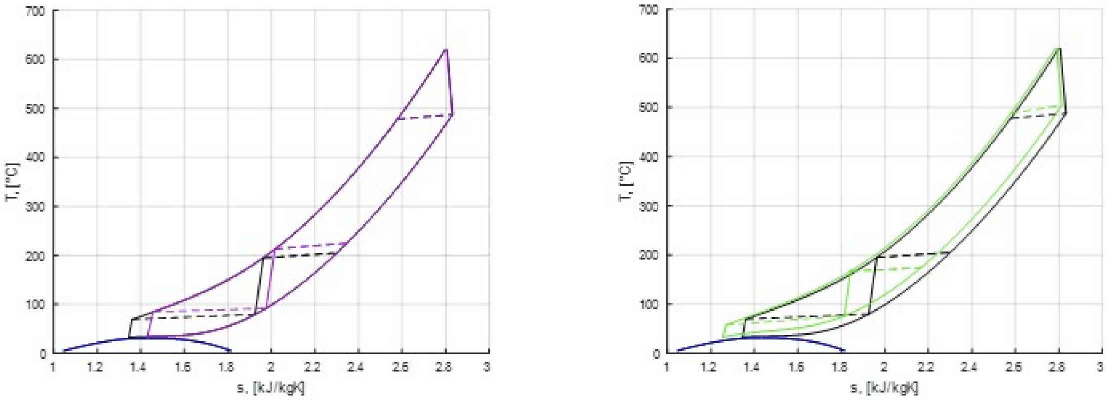


Figure 4.5: Compressor inlet pressure variation as respect to plant inventory percentage variation

From Figure 4.5 in particular is possible to see the graph that shows the percentage of plant inventory increase needed to reach determined values of CIP with different ambient temperatures. Is possible to point out that for the ambient temperature design conditions(T_{amb}=20°C) in order to reach the compressor inlet pressure value of 100 bar is needed to inject almost additional 20% of the inventory into the plant. At higher ambient temperatures is shown that the situation is very similar with a slight shift towards the right part of the graph. This translates into an increase of the necessary inventory injected into the cycle in order to reach the same CIP.

4.1.3. Effects on the thermodynamic points (T-s graphs)

In the previous paragraph an analysis of the effect of the CIP and CIT variation on the thermodynamic conditions at the compressor inlet has been presented.
However, varying this values will not only change the main compressor operation but it will affect the cycle thermodynamic conditions. In this paragraph are presented the new T-s graphs of the cycle at limit conditions.



(a) (b)

Figure 4.6: T-s diagrams at design condition compared to (CIT=34.3°C; CIP$_{design}$=79.78 bar) (a) and (CIT$_{design}$=33°C; CIP=100.28 bar) (b)

In Figure 4.13 are shown the T-s diagrams of two limit cases in comparison with the design one (in black), the limit conditions that have been defined for the simulation are 100,28 bar for the CIP value and 40°C for the CIT value. However, the limit for the CIT value at design pressure condition is at 34.3°C, above that value the simulation will not be able to define an operating condition. Hence, for the T-s diagram that value has been chosen.

From the graphs is possible to notice that with higher CIT the T-s diagram slightly varies and becomes shorter. The situation has already been defined in chapter 2 while talking about the increase of the CIT in the ambient temperature increase standard strategy. The variation affects the operation of some components like the compressors and the LTR where the inlet/outlet temperature conditions increases as well as the temperatures at the cold side of the HTR.

With higher CIP however the situation is different, the variation of the minimum operating pressure affects the entirety of the cycle. The major considerations that are noticeable are the increase of the operating area of the HTR and a slight increase of the maximum pressure of the cycle.

4.2. Operating strategy for variable cycle minimum pressure strategies

The previous paragraphs presented a general overview of the compressor inlet temperature and compressor inlet pressure values. This two values showed sometimes to have

opposite impact on the cycle performance, in particular regarding the density value and the volumetric flow rate of the main compressor.
In previous chapters this showed to be the main issue in term of operation with the ambient temperature increase, therefore the CIP value can be considered an interesting control variable in order to define a new operating strategy.
The definition of the strategy however is not immediate, from previous analysis is showed that the higher ambient temperature leads to a higher operation of the HRU with the fans, after that the compressor inlet temperature starts to increase in order to preserve the heat dissipation into the environment, and this leads to higher volumetric flow rates in the main compressor until it reaches its limit. It has been showed that the volumetric flow rate increase can be mitigated through the variation of the compressor inlet pressure, however the best value is difficult to define in each condition. Due to the particular operation of the fluid in the supercritical region the best operation of the plan is not always the of the is not always the optimal one.
Therefore, the proposed operating strategy will not define a specific action to be made in different ambient temperature areas. For each ambient temperature it will be needed to operate on both the commutated fan of the HRU and on the inventory in order to reach the best compressor inlet conditions. For each temperature condition thank to a simulation it will be possible to obtain the best pair of values that grants the maximum power produced by the plant.

4.2.1. Definition of the simulation

The operation strategy is thus based on the simulation of the plant performance in different operating conditions. However before proceeding in the description of the simulation is important to define the input variables, each of this values will generate a specific plant operation that will be subsequently analyzed. It is also important to assess the range and the step of operation of this variable in order to have a clearer overview of the results.
The first input variable necessary for the simulation is the **load** defined as the percentage of coal input as respect to the total, this value is set at the 100% in nominal conditions and is decreased with a step of 1% when necessary.
Another input variable is the **CIT**, this value is previously mentioned to be the a main player in the problems caused by the increase of the ambient temperature. In the simulation is set at 33°C at design conditions and increased by a step of 0.1°C until it reaches 40°C.
In order to counteract the problems generated by the increase of the CIT, before in this chapter has been assessed how the possibility to control the compressor inlet pressure

could result useful. Therefore in this simulation the **CIP** is one of the input variables that is set at 79.78 bar at design conditions and reach 100.25 bar with a step of 0.5 bar. The last input variable is the key factor for this operation and is the $\mathbf{T}_{amb}$, this value is important to determine the net cycle performances and how the cycle operation changes with the increase of the ambient temperature. This value is set at 20°C at design condition and with a step of 1°C until it reaches 32°C.
In order to better understand the following statements on the simulation, however is important to say that this input variables don't have the same weight. The simulation is in fact based on matrices that generates different operating conditions of the cycle in each square as showed in Figure 4.7.

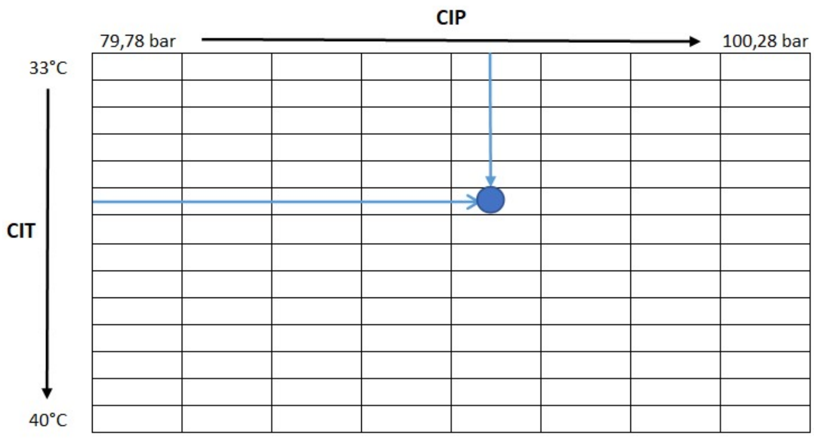


Figure 4.7: Explanatory compressor inlet temperature and pressure matrix in the simulation

Is possible to notice that the matrix is defined by all the possible couple of compressor inlet operating conditions, on the vertical axis is in fact possible to see the CIT that varies from 33°C to 40°C from top to bottom and an the horizontal the CIP that varies from 79.78 bar to 100.28 bar from left to right.
This kind of matrix is generated for each load of operation that will thus have a higher weight as a variable in the simulation. The $\mathbf{T}_{amb}$ value will be used afterwords when the operation of the HRU is assessed. In the following paragraphs more details of the simulation procedure are exploited.

4.2.2. Cycle operation limits

For each load it will thus generated a matrix of cycles operations for each CIP and CIT. However is important to consider that not every couple of compressor inlet temperature and pressure produce a feasible cycle operation. In Figure 4.8 is possible to see the area that will be generated by the unfeasible cycle operations.

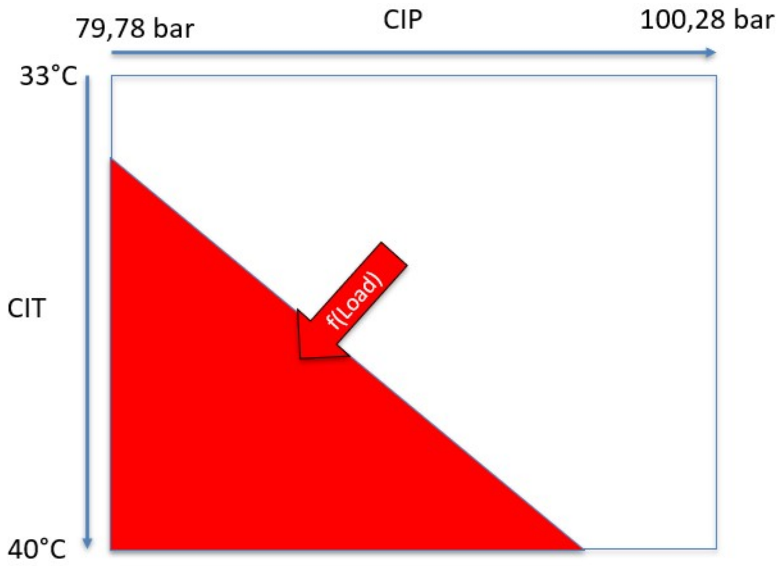


Figure 4.8: Compressors operability limit

This operation limits that are excluded correspond to the exceeds of the compressor operation limits, therefore the operation points in this area cannot be considered useful. Another interesting characteristics of this area is that is a function of the load percentage, the area will in fact decrease in the direction showed in the figure along with the load decrease.

4.2.3. HRU operation limits

Afterwards is important to consider also the HRU operation and the ambient temperature. In the simulation is in fact determined the gross operation of the cycle without taking into account the external conditions, and after that the limits on the HRU are checked. As mentioned many times in previous chapters the HRU presents a limit in its operation set as +25% of the commutated fans rotational speed. With the increase of the ambient temperature the cycle will not always be able to respect this limit, hence some points will be considered unfeasible for the operation. The point excluded by this limitation are showed in the Figure 4.10.

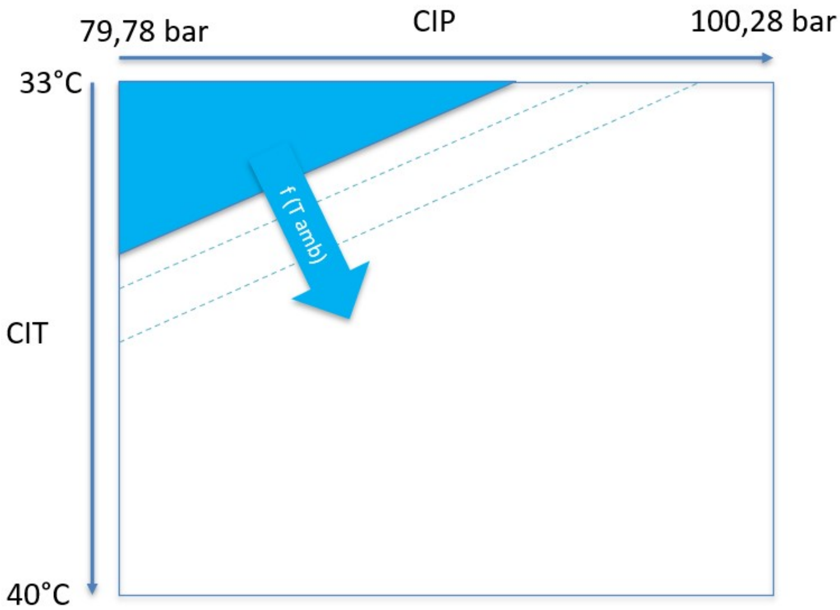


Figure 4.9: Heat rejection unit operation limit

Is important to notice that this area is clearly function of the ambient temperature, the area will in fact increase in the direction showed in the figure along with the ambient temperature increase.

4.3. Solution A) Maximum power generated

The main problem that the increase of the ambient temperature caused to the plant operation was the abrupt decrease of the maximum power producible by the plant. Therefore, the first solution exploited will focus entirely in reaching the operating points able to produce the maximum electric power.

4.3.1. Operating strategy at variable minimum pressure and maximum power generated

In this paragraph is presented how the curve at maximum power generated is obtained. The strategy is based on the information given in paragraph 4.2 and in the description of the simulation.
Afterwards, more strategies will be presented that follows a similar approach, to differentiate them this strategy will be defined as ***"Maximum power strategy"***. For each ambient temperature greater than the design one the map of the feasible cycle operating conditions will be generated. The unfeasible operating points are excluded via the areas

explained before. From the feasible points it will then chosen the one with the maximum electric power production. The strategy can also be summarized in the scheme below:

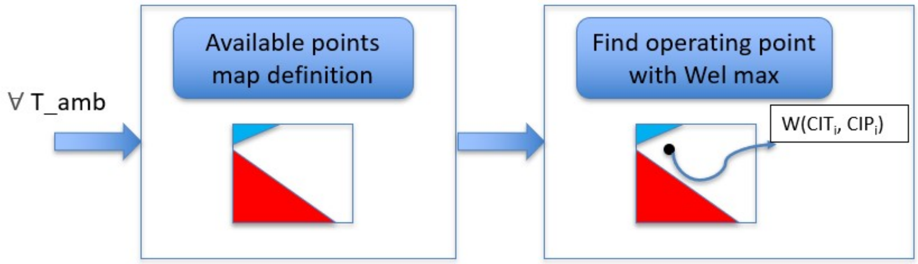


Figure 4.10: Operating strategy to mitigate the ambient temperature effect with CIT, CIP variation and $W_{el,max}$

Below is also showed one of the real maps used to assess the operation curve.

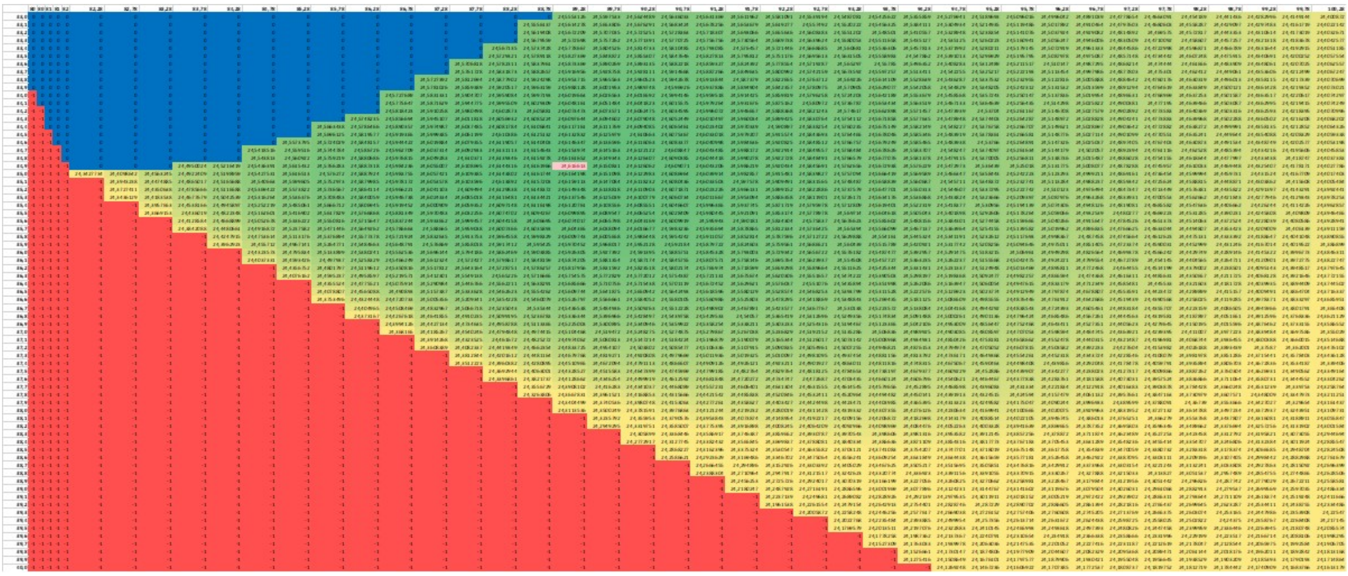


Figure 4.11: Map of operation at Load=100% and T_{amb}=25°C

In the figure is possible to notice the unfeasible points cancelled by the the areas, for the remaining feasible points a color scale is used in order to assess the area of feasible points where the power production is maximal. Moreover in light red is detected the operating condition with the maximum power and therefore the one chosen for the curve.

4.3.2. Plant performance

Is now possible to assess the performance of the plan in this new operating strategy. This new operating strategy with the aid of a controlled regulation of the compressor inlet pressure and compressor inlet pressure aims to reach a curve of operation able to maintain high power production even at high ambient temperatures.
The graph in Figure 4.12 showed that this solution demonstrated to be successful.

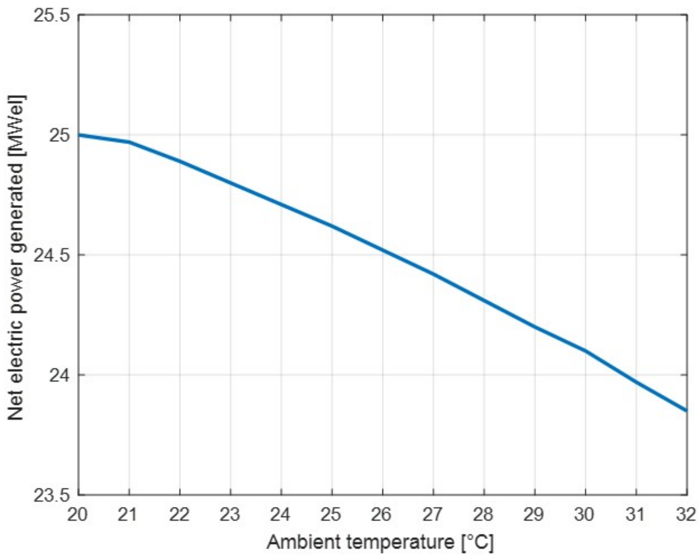


Figure 4.12: Maximum electric power output at variable ambient temperature with compressor inlet conditions regulation

The drop in power production is in fact minimal showing a net electric power generated of almost 24 MWel with an ambient temperature of 32°C. In Table 4.1 the curve of operation given by the operating strategy for every ambient temperature is showed.

Table 4.1: Curve of operation at variable ambient temperature with regulation of compressor inlet conditions

T_{amb} [°C]	Wel_{max} [MWel]	CIT [°Cl]	CIP [bar]	Wel_{max}/Wel_{nom} [%]
20	25.00	33.0	79.78	100
21	24.97	33.0	82.78	99.9
22	24.89	33.0	84.28	99.6
23	24.80	33.6	86.28	99.2
24	24.72	34.3	87.78	98.8
25	24.62	34.9	89.28	98.5
26	24.51	35.6	90.78	98.1
27	24.42	36.3	92.28	97.7
28	24.31	37.0	93.78	97.2
29	24.20	37.7	95.78	96.8
30	24.01	38.5	97.28	96.4
31	23.97	39.2	98.78	95.9
32	23.85	39.9	100.28	95.4

Is possible to notice from the $\mathbf{Wel}_{max}/\mathbf{Wel}_{nom}$ column that the maximum power produced drops only by less that 5% at 32°C as respect to the nominal power. This is a huge step forward compared to the standard operation where at the same temperature the loss in maximum power production at the same temperature was around 50%.
This advantage is reached thanks to the regulation of the compressor inlet conditions as showed in Table 4.1 and more in detail in Figure 4.13a. In the Figure is possible to notice that the regulation can be splitted in two different sections before and after 22°C. Before the 22°C threshold is possible to notice that the best operating conditions with a slight increase of ambient temperature are not anymore the design conditions. While the CIT is kept constant the CIP increases with steps of around 2 bar for each degree of increase of the ambient temperature.
However, after the 22°C threshold the situation changes, the limit on the HRU kicks in and it is not possible to keep the CIT constant. Therefore, the regulation strategy changes and is possible to notice that the best operating conditions follows a linear trend. This result could prove to be useful in further studies regarding this operating strategy.

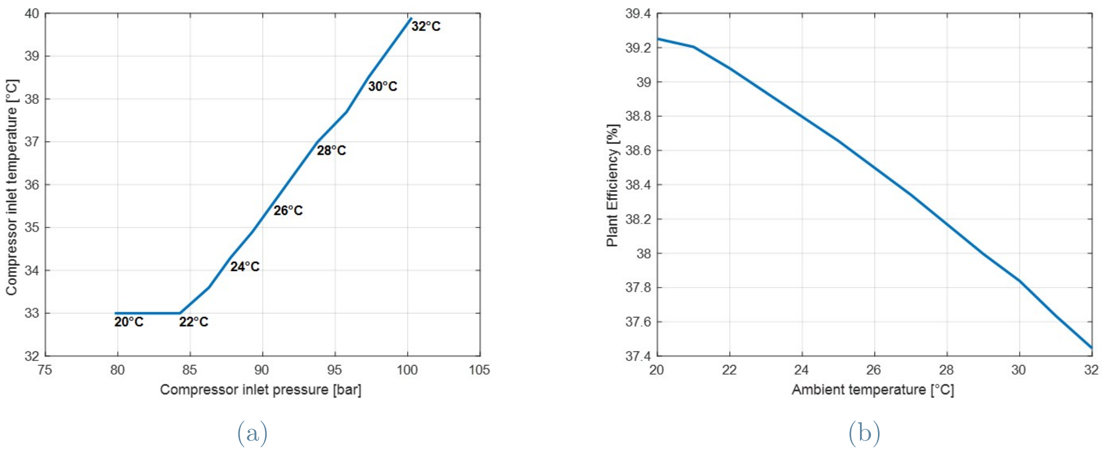


Figure 4.13: Inlet compressor operating conditions (a) and plant efficiency (b) at different ambient temperature conditions

The important consideration to make is that with this operation is always possible to keep the plant running with 100%, the decrease in the maximum power produced that is showed in Figure 4.12 is all due to the inefficient operation of the plant. In Figure 4.13b is in fact possible to notice that this new regulation has a significant impact on the net plat efficiency, that is showed to loose almost two percentage points from design to 32°C of ambient temperature.

4.3.3. Comparison with previous solutions

From the analysis of the plant performance, showed in the previous paragraph, it results that a controlled regulation of the compressor inlet conditions proved to be a valid option in order to tackle the ambient temperature problem.
Is now important to compare this new strategy with the standard operation, starting with the main issue, the maximum power produced.

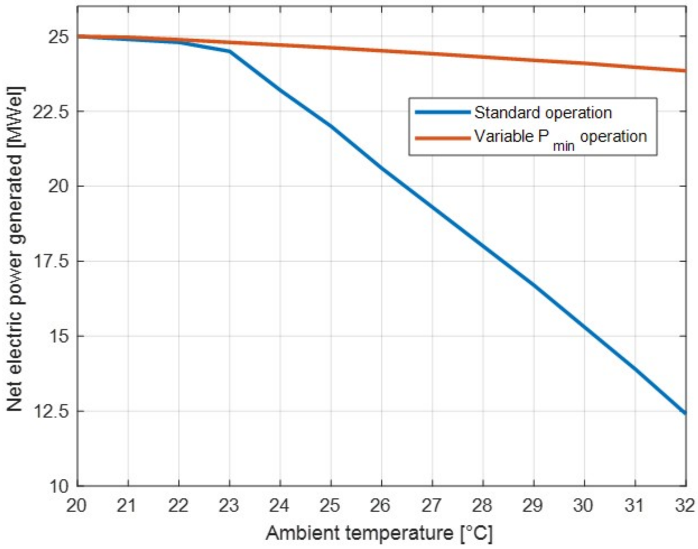


Figure 4.14: Maximum electric power output at variable ambient temperature with different operation strategies

From Figure 4.14 is possible to notice that the difference in performance is abysmal. The new operation strategy performs a lot better than the standard one, showing a new maximum power production slightly limited in comparison.
However, this new operation is not without flaws. The main one is considered to be the progress of the maximum cycle pressure with the increase of the ambient temperature.

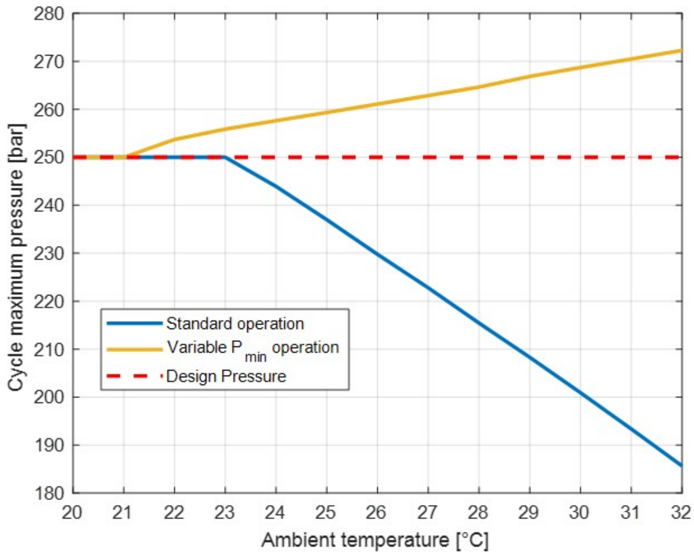


Figure 4.15: Maximum cycle pressure at variable ambient temperature with different operation strategies

The two operating strategies showed to perform really differently in this regard, for the standard operation, due to the decrease of the load, the maximum pressure decrease significantly, from 250 bar to almost 180 bar. Instead, the new operation strategy pays the higher performance in term of power producibility with an increase of the maximum pressure of the cycle.
This increase of maximum cycle pressure is not negligible reaching a value of 270 bar. This results in a problematic operation of the components that may not endure the operation at this high pressure thus crashing. Is therefore important to find a new operating strategy that could overcome this limit granting a safe operation of the cycle.

4.4. Solution B) Pressure constraint

The chapter showed that a controlled regulation of the compressor inlet conditions proved to generate promising results in term of tackling the problem of the increase of the ambient temperature. Therefore, it is possible to leave behind the standard operation and consider this controlled regulation the new foundations that will lead to the optimal strategy.
The strategy presented in 4.3 is far from optimal, the huge increase of maximum pressure is not negligible and leads to a non safe operation of the plant. Therefore, another solution is exploited where the maximum pressure of the cycle will be limited. A new limit for the operation is thus introduced.

4.4.1. Maximum pressure operation limit

In order to define a new curve of operating points that does not perform at pressures higher than the nominal one a new limit area is introduced. This area will define unfeasible all the operating points where the maximum pressure of the cycle is above the nominal one of 250 bar. In Figure is possible to see the shape that the area will have in a general operation matrix.

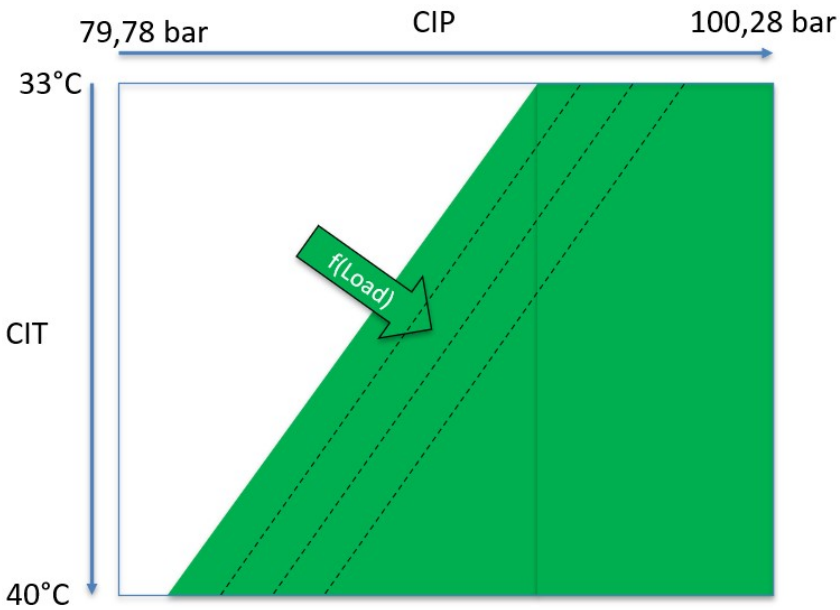


Figure 4.16: Maximum cycle pressure operation limit

Interesting characteristic of this area is that is a function of the load percentage, the area will in fact decrease in the direction showed in the figure along with the load decrease.

4.4.2. Operating strategy at variable minimum pressure and maximum pressure constraint

Operating strategy at variable minimum pressure and maximum power generated In this paragraph is presented how the curve of the new strategy defined ***"Pressure Constraint strategy"*** is computed. The strategy is based on the information given in paragraph 4.2 and in the description of the simulation like in the ***"Maximum power strategy"***. It will also take into account the new *maximum pressure operation limit* previously defined. For each ambient temperature greater than the design one, a map of feasible operating conditions will be generated. The unfeasible operating points are excluded with the limit areas explained before. From the feasible points it will then chosen the one with the maximum electric power production, that in this case will also respect the maximum pressure limit. However, due to the new limit added in the strategy it is possible that the matrix of the operating points will present no feasible solution. Therefore, in that situation the characteristic of the *maximum pressure operation limit* area and the *Cycle operation limit* area is exploited. The two areas are in fact function of the load percentage and shrinks with the decrease of it. Hence, when no feasible points are found, the load is decreased until the shift of the two areas will free some operating conditions and the one

with the greater power production will be chosen.
The strategy can also be summarized in the scheme below:

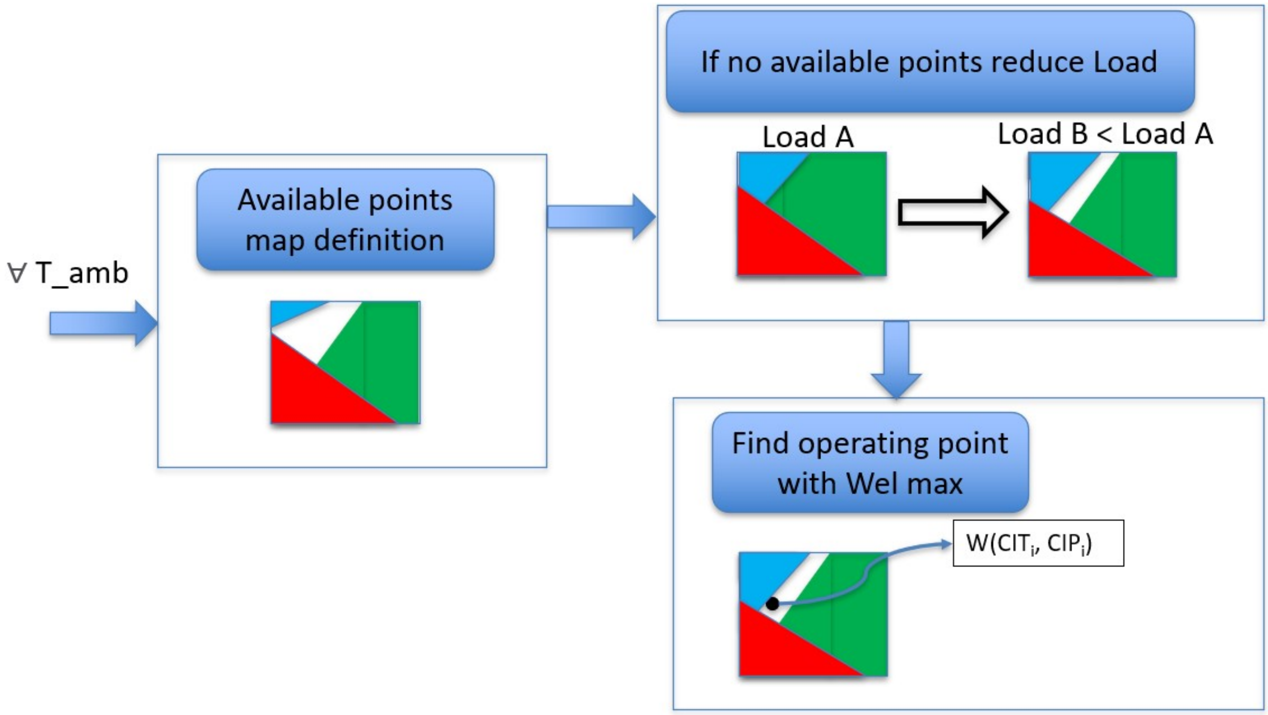


Figure 4.17: Simplified scheme for *"Pressure Constrain strategy"*

4.4.3. Plant performance

Is now possible to assess the performance of the plant in this new operating strategy. The new operating strategy is again based on the controlled regulation of the compressor inlet conditions and despite the new limitations added, the curve of operation generated proved again to be able to maintain high power production even with great ambient temperature variations.
In the graph in Figure 4.18 is showed the net electric power generated with the increase of the ambient temperature.

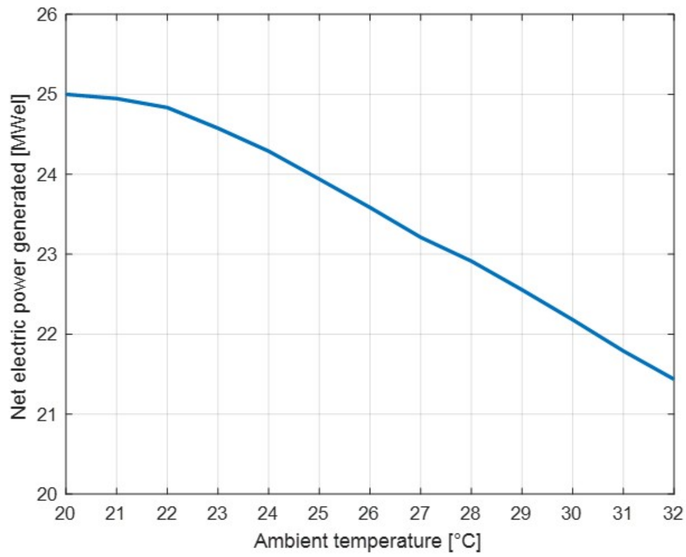


Figure 4.18: Maximum electric power output at variable ambient temperature with *"Pressure Constrain strategy"*

The drop in maximum power production showed again to be minimal compared to the design condition with almost 21.5 MWel with an ambient temperature of 32°C. However, despite being minimal this reduction of the maximum power produced is not negligible being 3.5 MWel less than nominal conditions.
In Table 4.2 the curve of operation given by the operating strategy for every ambient temperature is showed.

Table 4.2: Curve of operation at variable ambient temperature with *"Pressure Constrain strategy"*

$\mathbf{T}_{amb}$ [°C]	$\mathbf{Wel}_{max}$ [MWel]	CIT [°Cl]	CIP [bar]	Load [%]	$\mathbf{Wel}_{max}/\mathbf{Wel}_{nom}$ [%]
20	25.00	33.0	79.78	100	100
21	24.95	33.0	80.78	100	99.8
22	24.83	33.0	80.78	100	99.3
23	24.58	33.7	79.78	100	98.3
24	24.29	34.2	81.28	99	97.2
25	23.94	34.8	82.28	98	95.8
26	23.58	35.4	83.28	97	94.3
27	23.21	36.1	84.28	96	92.8
28	22.91	36.7	85.78	95	91.7
29	22.55	37.3	86.78	94	90.2
30	22.18	38.0	87.78	93	88.7
31	21.79	38.7	91.28	91	87.2
32	21.44	39.4	92.28	90	85.7

From the data showed in the table is possible to notice a few of interesting results. First of all from the column $\mathbf{Wel}_{max}/\mathbf{Wel}_{nom}$ is noticeable that the maximum power produced have a particular trend as showed in Figure 4.19.

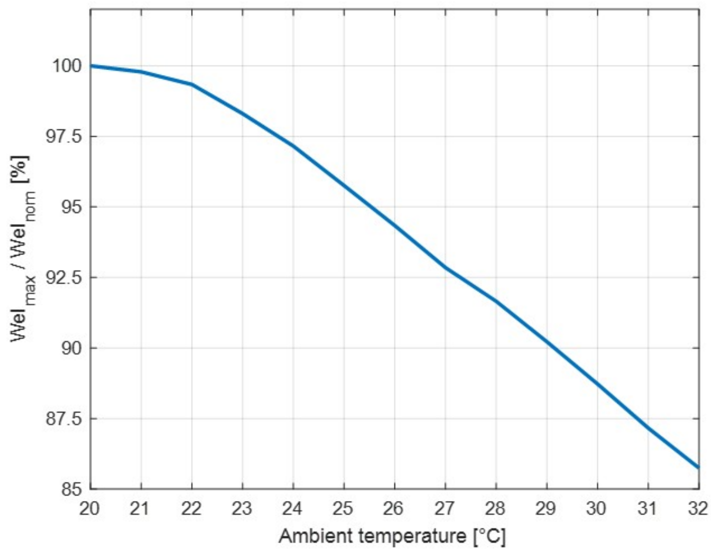


Figure 4.19: Ratio of maximum electric power output at variable ambient temperature with *"Pressure Constrain strategy"*

At ambient temperatures slightly above the design condition the shift from the nominal power value is minimal, however the situation will not be the same afterwards. When the ambient temperature reaches the value of 23°C, the decrease started to become more significant and after 24°C this effect is even more accentuated. In the graph is possible to notice three different slopes:

- **From 20°C to 22°C**: The area is characterized by a minimal variation of the maximum power produced loosing around 0.5% of the nominal value per degree of ambient temperature increase. In this area the regulation is made thanks to the increase of the compressor inlet pressure that allows to keep constant the values of load and compressor inlet temperature;
- **From 22°C to 23°C**: This little area is characterized by a higher impact of the ambient temperature increase on the maximum power produced. The 1°C of increase translates into a loss of maximum power produced of around 1% of the nominal value. This area is also particular due to the shift in the trend of regulation variables, due to the limit on the maximum pressure to reach the best operating condition is reached bringing back the pressure to its nominal value and increasing the CIT. Operation that previously result to be less efficient in terms of power production;
- **From 23°C onward**: In the last area of operation the impact of the ambient temperature increase is the highest. In this condition each degree of increase of

the ambient temperature generates a loss of power produced of around 1.5% of the nominal value. This result can be explained by the regulation method adopted. After 23°C of ambient temperature the cycle is not able to work at 100% load anymore, hence it will be necessary to reduce the fuel input in order to obtain a feasible operating point. This with a regulation that still needs to rely on both CIT and CIP increase translates into a greater loss in maximum power produced compared to the previous areas.

4.4.4. Comparison with previous solutions

The analysis of the plant performance showed interesting results from the *"Pressure Constrain strategy"*, showing that proceeding towards variable minimum pressure strategies is the best solution in order to tackle the effect of the increase of the ambient temperature. In the *"Maximum power strategy"* comparison with the standard operation is showed that the first one is far better than the latter despite the limits on the high pressure operation. Therefore it is interesting to assess how the newly assessed *"Pressure Constrain strategy"* performs compared to the standard and *"Maximum power strategy"*. In Figure 4.20 is showed the comparison between the strategies analyzed until now.

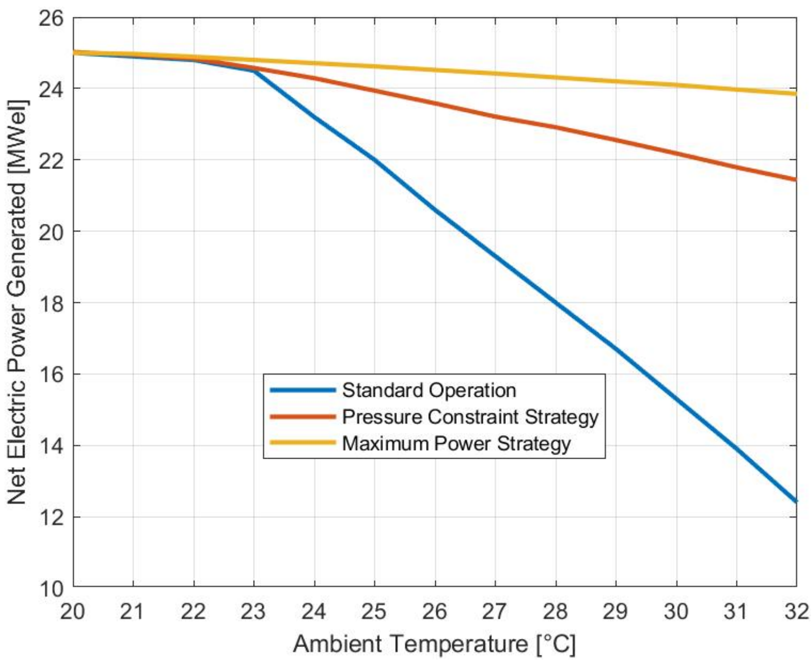


Figure 4.20: Maximum electric power output at variable ambient temperature with standard, *"Maximum power strategy"* and *"Pressure Constrain strategy"*

Is possible to notice that also in this case the difference between the standard operation

and a strategy that adopts controlled compressor inlet conditions is huge. Despite proceeding in a similar way from 20°C to 23°C after that the two curves diverge and the gap between the maximum power produced of the two operations continues to increase. At 32°C there is a difference of about 9 MWel, that's one third of the power produced at design conditions. Moreover the *"Pressure Constrain strategy"* does not suffer the same problem of the *"Maximum power strategy"*, since the maximum pressure of the cycle is limited and it's showed to be always around the design value of 250 bar.
However *there ain't no such thing as a free lunch*. The difference between the *"Pressure Constrain strategy"* and the *"Maximum power strategy"* curves even in this case is significant, the operation within the pressure limits comes with the cost of a lower value of electric production from the plant. Despite proceeding similarly from the ambient temperature values of 20°C to 22°C, afterwards the load decrease due to the limit on the maximum pressure of the cycle starts to have a significant impact. The two curves begin to diverge, leading to a difference of around 2.5 MWel at the ambient temperature of 32°C. That despite being far less than the 9 MWel difference from the standard operation is still almost 10% of the power produced at design conditions. Therefore is possible to assess that in term of power production the new operating strategies places between the standard and the *"Maximum power strategy"*. However, is important to point out that the *"Pressure Constrain strategy"* does not present any significant technical limit or warning that may result in a problematic operation of the plant, making the *"Pressure Constrain strategy"* a more than valid option due to safety issues and a moderate power losses.

4.5. Solution C) No inventory variation

The *"Pressure Constrain strategy"* showed that is possible to obtain promising results with the regulation of the compressor inlet conditions while keeping a maximum pressure of the cycle around the nominal value of 250 bar.
At the beginning of the chapter is presented how the regulation of the compressor inlet pressure is made, exploiting the characteristic of being a closed cycle the inventory of the cycle is changes via extraction or injection of fluid. This causes a variation of the pressure at the compressor inlet.
In this paragraph it will be evaluated if is possible to operate at variable ambient temperature even without the extraction or injection of new fluid into the cycle, therefore avoid to changes to the plant inventory. In order to do that a new graphic component needs to be included.

4.5.1. No Inventory variation limit

In order to define the new curve of operating points that performs with no inventory variation some considerations have to be made. Is important to state that despite the inventory variation was the regulating method of the compressor inlet pressure, operating without changing the inventory does not translate into a constant pressure operation. This however means that the pressure is not anymore an independent variable, for each load the operating points in the map will not be anymore all the possible interconnections between the CIT and CIP values. They will be on a line where the CIP is a function of the CIT as shown in Figure 4.21.

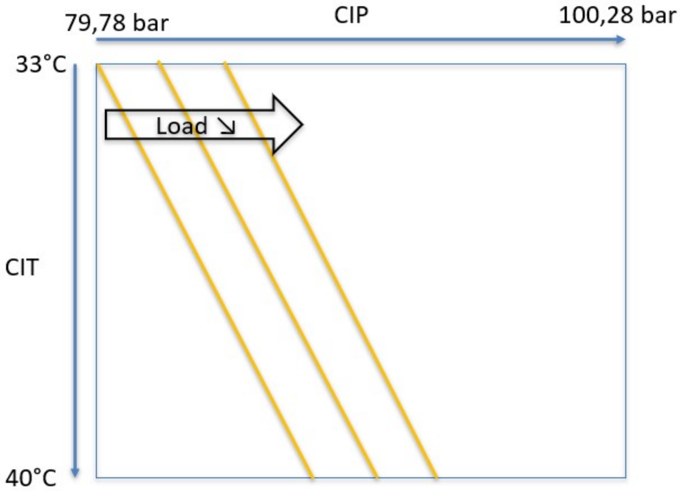


Figure 4.21: No inventory variation limit

Each line correspond to a different load. Decreasing the load, the line will move as showed in the figure. Interesting result is that decreasing the load at the same CIT the compressor inlet pressure will be higher. Is also important to remind that a line with 0 inventory variation is very difficult to obtain due to the discretization of the simulation, therefore is considered a *No inventory variation* operation when the variation is under $1^{o}/_{oo}$ of its nominal value.

4.5.2. Operating strategy at variable minimum pressure and no inventory variation

In this paragraph is presented how the curve of the new strategy defined ***No Inventory Strategy*** is computed. The strategy, like the ones exploited before in this chapter, is based on the information given in the paragraph 4.2 and taking into account all the limits

presented until now: *Cycle operation limits*, *HRU operation limits*, *maximum pressure operation limits* and *No Inventory variation limit*.
For each ambient temperature greater than the design one, a map of feasible operating conditions will be generated. The unfeasible operating points are excluded with the limit areas explained before. From the feasible points it will then chosen the one with the maximum power production, that respects the limit of pressure and no inventory variation. However, like for the *Pressure Constrain Strategy*, is possible that due to the new limit added no feasible solution is found. Therefore, in that situation the characteristic of the *Cycle operation limits*, *maximum pressure operation limits* and *No Inventory variation limit* is exploited. This limits changes as a function of the load percentage, in particular the limits on the cycle operation and maximum pressure shrinks with the load decrease. The no inventory variation curve does not change its size and instead shifts towards the right side of the map. Hence, when no feasible points are found, the load is decreased until the shift of the limits will free some operating conditions and the one with the greater power production will be chosen.
The strategy can also be summarized in the scheme below:

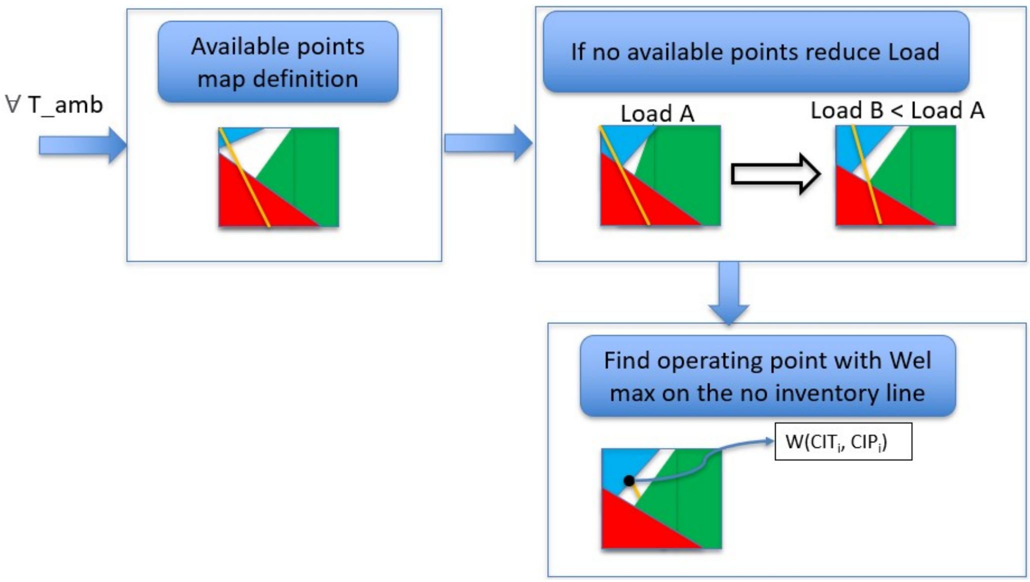


Figure 4.22: Simplified scheme for *No Inventory Strategy*

Is important to reiterate that all the limits have to be respected in order to find the operating point of the curve.

4.5.3. Plant performance

Is now possible to assess the performance of the plant in this new operating strategy. The new operating strategy is based on the regulation of the load and compressor inlet temperature, and also the compressor inlet pressure will vary accordingly to this two values.
Even in this situation, without a direct control of the pressure, the curve of operation generated proved to be able to maintain a fairly high power producibility even with great ambient temperature variations.
In the graph in Figure is showed the net electric power generated with the increase of the ambient temperature.

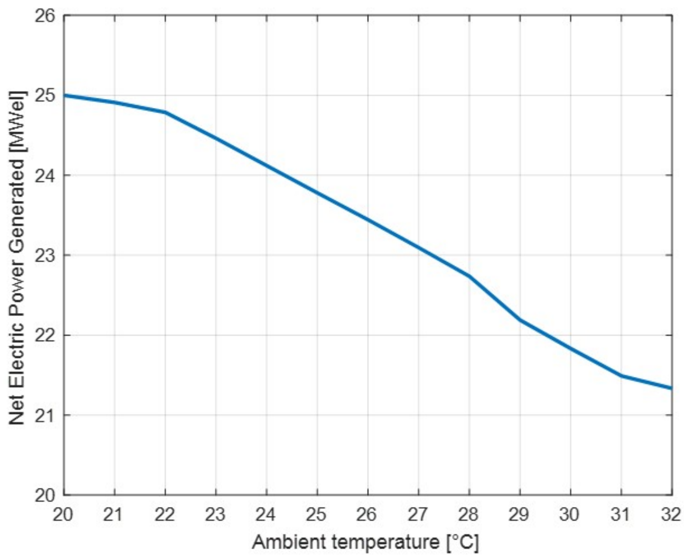


Figure 4.23: Maximum electric power output at variable ambient temperature with *No Inventory Strategy*

Despite operating with a limited regulation even in this case the drop in maximum power production showed to be minimal with around 21 MWel at an ambient temperature of 32°C. However like for the *"Pressure Constrain strategy"* this reduction is not negligible since it is still around 3.7 MWel less than the nominal operation. The curve doesn't result as smooth as the other previously showed, this is due to the peculiar characteristic of the operation without inventory variation that for each load have only a line of feasible values on the map. Due to the discretization of the simulation is not always possible to get the exact value and the curve will result rough.
In Table 4.3 is showed the curve of operation given by the operating strategy for every ambient temperature.

Table 4.3: Curve of operation at variable ambient temperature with *No Inventory Strategy*

T_{amb} [°C]	Wel_{max} [MWel]	CIT [°Cl]	CIP [bar]	Load [%]	Wel_{max}/Wel_{nom} [%]
20	25.00	33.0	79.78	100	100
21	24.91	33.3	80.28	100	99.6
22	24.78	33.1	79.78	100	99.1
23	24.46	33.9	81.28	99	97.8
24	24.12	34.5	82.28	98	96.5
25	23.78	35.4	84.28	97	95.1
26	23.44	35.9	84.78	96	93.8
27	23.01	36.5	85.78	95	92.4
28	22.74	36.8	86.28	94	90.9
29	22.19	37.9	88.28	92	88.8
30	21.83	38.8	89.78	91	87.3
31	21.79	39.1	90.28	90	86.0
32	21.33	39.8	91.28	90	85.3

From the data showed in the table is possible to notice that the situation is very similar to the *Pressure Constrain Strategy*. This will be exploited in the following paragraph where the main operating strategies are compared.

4.5.4. Comparison with previous solutions

From the analysis of the plant performance, showed in the previous paragraph, it results that a limited controlled regulation of the compressor inlet condition it is still a valid option to tackle the problem of the increase of the ambient temperature.
It is now important to compare this strategy with the strategies exploited in this chapter. In Figure 4.24 is shown the maximum electric power production of the three operating curves.

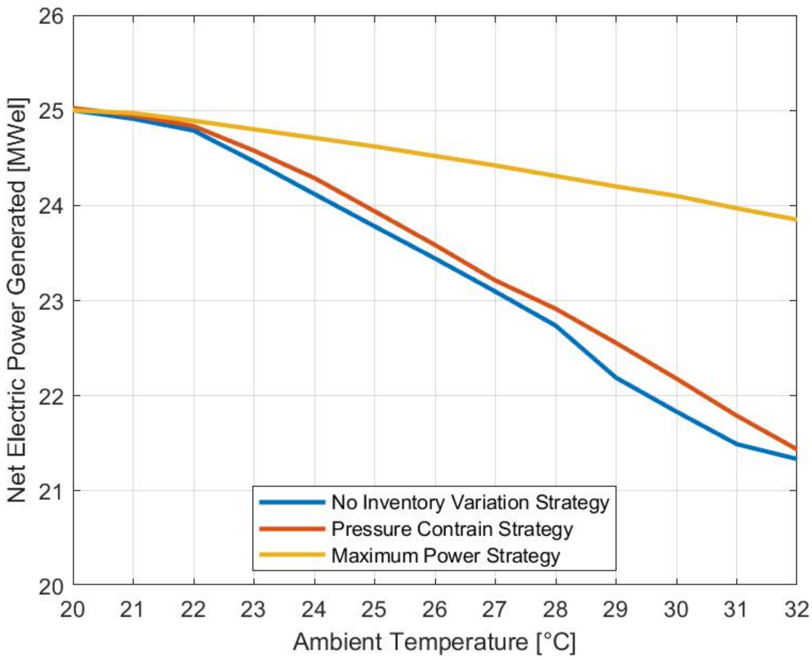


Figure 4.24: Maximum electric power output at variable ambient temperature with different strategies

Is possible to confirm the statement made in the previous paragraph that says that this new *No Inventory Strategy* is really similar to the *Pressure Constrain strategy*. In the figure is in fact shown that the two curves are really close to each other. Therefore, for the comparison with the *Maximum Power Strategy* please refer to *Pressure Constrain strategy* comparison since it will reach the same results.
The comparison between the two other strategies instead could do further exploited, to do that in the following paragraph an analysis of the inventory is made.

4.6. Inventory analysis

In this paragraph it will be analyzed the inventory of the plant, being the inventory the distribution of the mass flows in the cycle during its operation.
At design conditions the overall mass of CO2 that circulates inside the cycle will have a value of around 8175 kg. While in Figure 4.25 is showed a pie chart where the distribution of the mass inside the cycle is exploited.

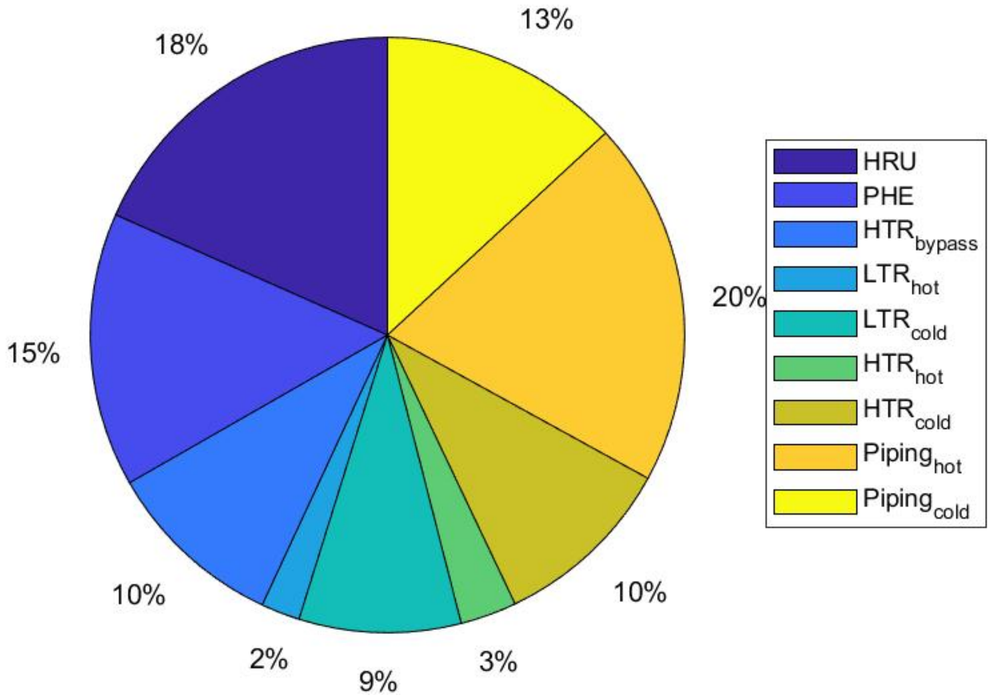


Figure 4.25: Distribution of the inventory at design conditions

From the graph is possible to notice that the distribution of the fluid mass is clearly not constant throughout the components. In the chart is visible that just the hot and cold piping holds a third of the total mass, the second component with the greater inventory is the heat rejection unit with 18% of the total inventory and the third is the primary heat exchanger with a value of 15%. Afterwards the remaining mass will be distributed inside the recuperators and the bypass. Now that the design conditions are assessed is showed how this values changes in the various curves defined in the *Pressure Constrain strategy* and *No Inventory Strategy*.

4.6.1. Breakdown inventory *Pressure Constrain strategy*

The first operating strategy that will be analyzed is the *Pressure Constrain strategy*. In Figure 4.26 is showed the overall value and the distribution of the inventory throughout the curve obtained by the strategy.

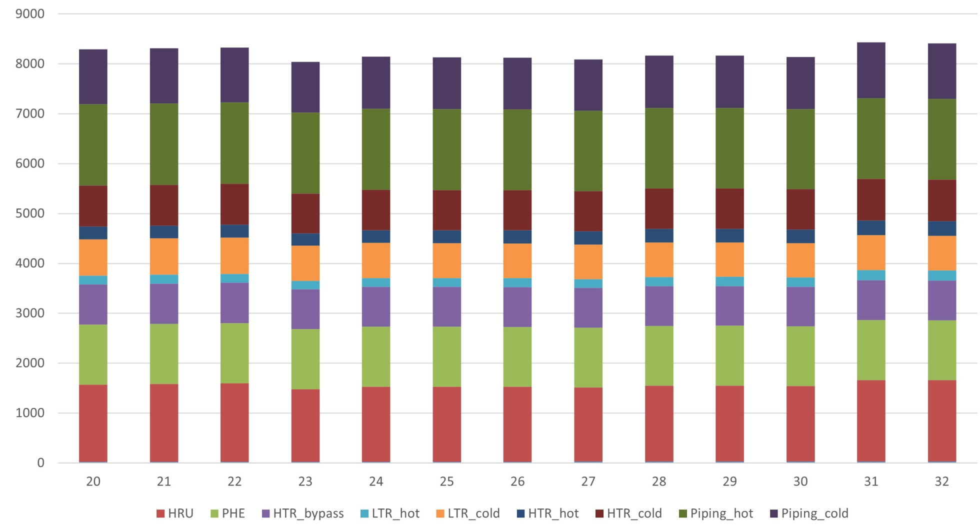


Figure 4.26: Breakdown inventory at different ambient temperature with *Pressure Constrain strategy*

Is possible to notice that there isn't any prominent difference at the different ambient temperature operations, the value of the total inventory seems to don't vary much that could mean that there isn't a lot of mass of CO2 injected or extracted from the cycle. Moreover another advantage is that also the distribution of the mass doesn't seem to change significantly.
In order to define how much the inventory changes, an analysis on the deviation of the values from the design conditions is showed in Figure 4.27.

Delta plant %	0,000%	0,184%	0,381%	-3,087%	-1,851%	-1,992%	-2,114%	-2,570%	-1,649%	-1,696%	-2,032%	1,499%	1,287%
T_amb	20	21	22	23	24	25	26	27	28	29	30	31	32
HRU	0,0%	1,0%	2,0%	-5,9%	-2,9%	-3,0%	-3,1%	-4,0%	-1,8%	-1,7%	-2,4%	5,6%	5,2%
PHE	0,0%	0,0%	0,0%	0,1%	0,0%	-0,1%	-0,2%	-0,2%	-0,3%	-0,4%	-0,4%	-0,6%	-0,6%
HTR_bypass	0,0%	0,0%	0,0%	-1,5%	-1,0%	-1,2%	-1,4%	-1,7%	-1,5%	-1,6%	-1,9%	-0,7%	-0,9%
LTR_hot	0,0%	0,0%	0,0%	-3,6%	-1,1%	-0,1%	1,0%	1,8%	4,1%	5,3%	6,1%	13,2%	14,3%
LTR_cold	0,0%	0,0%	0,0%	-3,4%	-2,8%	-3,5%	-4,1%	-5,1%	-4,8%	-5,3%	-6,1%	-3,6%	-4,2%
HTR_hot	0,0%	0,0%	0,0%	-3,0%	-0,5%	0,7%	1,9%	3,0%	5,3%	6,6%	7,7%	14,1%	15,3%
HTR_cold	0,0%	0,0%	0,0%	-2,6%	-1,6%	-1,6%	-1,6%	-1,8%	-1,1%	-1,1%	-1,3%	1,5%	1,4%
Piping_hot	0,0%	0,0%	0,0%	-0,7%	-0,6%	-0,7%	-0,9%	-1,1%	-1,0%	-1,2%	-1,4%	-0,8%	-0,9%
Piping_cold	0,0%	0,0%	0,0%	-7,4%	-5,0%	-5,4%	-5,7%	-6,7%	-4,9%	-5,0%	-5,7%	1,5%	1,1%

Figure 4.27: Deviation of the inventory within the components from the design conditions at different ambient temperatures with *Pressure Constrain strategy*

A few things can be noticed from the table and the main one is that is confirmed that despite not being a focus of the strategy the deviation of the total mass into the plant from the design conditions is minimal. In the Delta plant row is in fact showed that there is a range of around +/-3% deviation that can be considered a fairly low value.
In the same figure is showed also the percentage deviation from its design condition of each

component. The one that shows the greater variation are the the LTR and HTR hot sides, this component in fact present an increase of around 15% from its nominal conditions. However this variation could be not as huge at it seems, in Figure 4.28 is showed the distribution of the inventory in the cycle for each value of ambient temperature.

T_amb	20	21	22	23	24	25	26	27	28	29	30	31	32
HRU	18,7%	18,8%	19,0%	18,2%	18,5%	18,5%	18,5%	18,4%	18,7%	18,7%	18,6%	19,4%	19,4%
PHE	14,6%	14,5%	14,5%	15,0%	14,8%	14,9%	14,9%	14,9%	14,8%	14,8%	14,8%	14,3%	14,3%
HTR_bypass	9,8%	9,7%	9,7%	9,9%	9,8%	9,8%	9,8%	9,8%	9,8%	9,8%	9,8%	9,6%	9,5%
LTR_hot	2,1%	2,1%	2,1%	2,1%	2,1%	2,2%	2,2%	2,2%	2,3%	2,3%	2,3%	2,4%	2,4%
LTR_cold	8,8%	8,8%	8,8%	8,8%	8,7%	8,7%	8,6%	8,6%	8,5%	8,5%	8,4%	8,4%	8,3%
HTR_hot	3,1%	3,1%	3,1%	3,1%	3,1%	3,2%	3,2%	3,3%	3,3%	3,3%	3,4%	3,5%	3,5%
HTR_cold	9,9%	9,9%	9,9%	10,0%	10,0%	10,0%	10,0%	10,0%	10,0%	10,0%	10,0%	9,9%	9,9%
Piping_hot	19,7%	19,7%	19,6%	20,2%	20,0%	20,0%	20,0%	20,0%	19,9%	19,8%	19,9%	19,3%	19,3%
Piping_cold	13,3%	13,3%	13,3%	12,7%	12,9%	12,8%	12,8%	12,7%	12,9%	12,9%	12,8%	13,3%	13,3%

Figure 4.28: Distribution of the inventory within the components at different ambient temperature with *Pressure Constrain strategy*

The figure showed that despite having a 15% increase from their design conditions the impact of this components into the overall cycle is minimal. It is in fact possible to see that the distribution remains roughly constant throughout the curve of operation.

4.6.2. Breakdown inventory *No Inventory Strategy*

The second operating strategy that will be analyzed is the *No Inventory strategy*. In Figure 4.27 is showed the overall value and the distribution of the inventory throughout the curve obtained by the strategy.

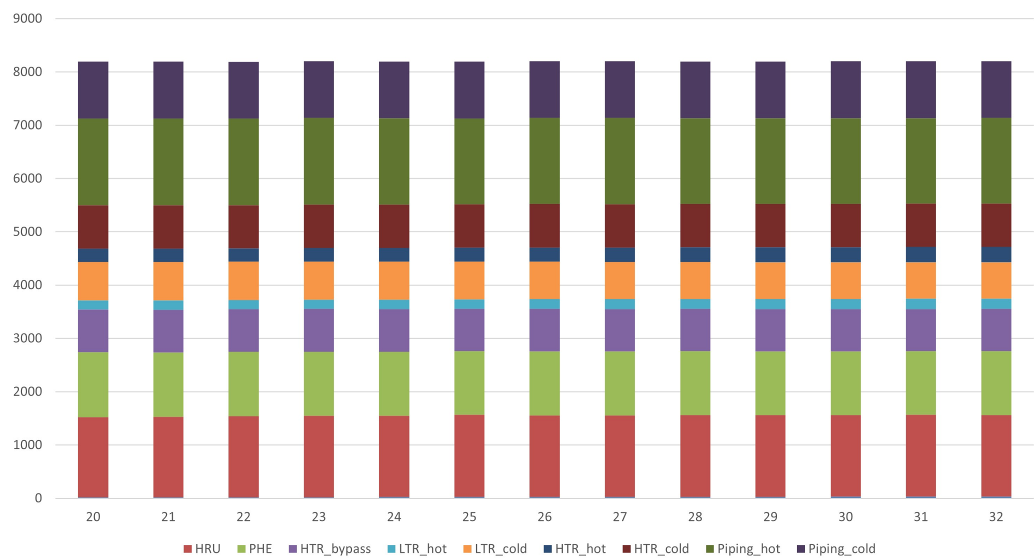


Figure 4.29: Breakdown inventory at different ambient temperature with *No Inventory strategy*

The value of total inventory is confirmed constant as stated during the construction of the operating curve. Is also possible to notice that there isn't also any prominent difference also in the distribution of the mass throughout the components. An analysis on the deviation of the values from the design conditions is made again and therefore showed in Figure 4.30.

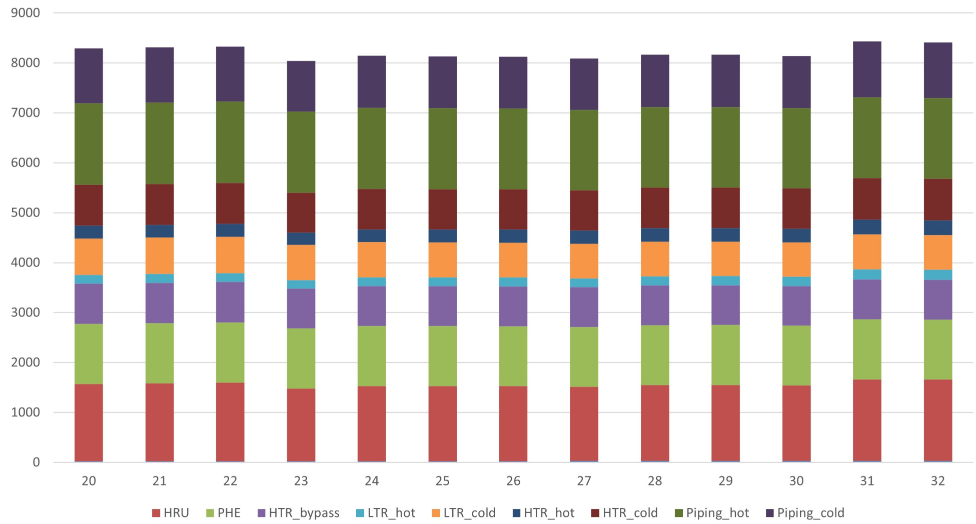


Figure 4.30: Deviation of the inventory within the components from the design conditions at different ambient temperatures with *No Inventory strategy*

A few things can be noticed from the table such as the deviation of the total mass into the plant from the design condition that is confirmed to be kept under the $1^{o}/_{oo}$ value. Like for the *Pressure Constrain strategy* is showed the percentage deviation from its design conditions for each component, and the main elements that present the greater variation are again the hot sides of the LTR and HTR. This components present an increase of around 14% from its nominal values. However is needed again to confirm how much of this variation is reflected on the overall cycle, therefore in Figure 4.31 is showed the distribution of the inventory in the cycle for each value of ambient temperature.

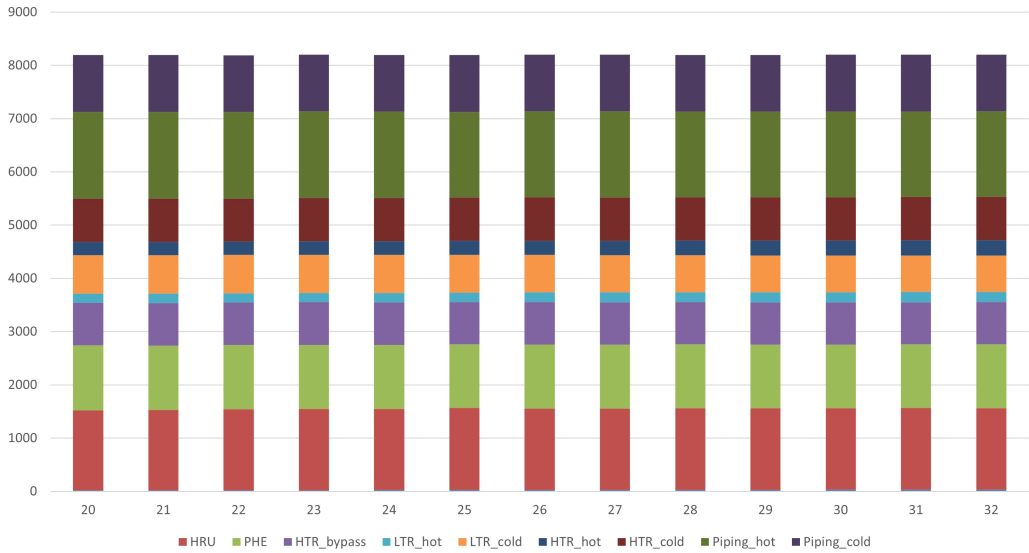


Figure 4.31: Distribution of the inventory within the components at different ambient temperature with *No Inventory strategy*

Also in this condition the Figure showed that the impact of the 14% increase in both HTR hot and LTR hot is negligible. The distribution of the CO2 mass throughout the cycle will indeed remain roughly constant in all ambient temperatures operation.

4.6.3. Inventory analysis conclusion

The result of the inventory analysis showed that the two operating strategies are again very similar. Despite not being the main focus of the strategy the *Pressure Constrain strategy* showed to not depart very much from a no inventory variation condition, the plant inventory varies in a range from -3% to +1.4% of its design conditions and more importantly in both cases there is no significant variation of the mass distribution throughout the components of the cycle. Thanks to this, is possible to assess that in the operation, the mass will remain roughly in the same position and there will not be large mass flows between the components that could cause malfunctions.

4.7. Conclusion of variable minimum pressure operating strategies

In conclusion is possible to point out the main results of the off-design operating strategies described in this chapter. Unlocking the value of the compressor inlet pressure opened new great possibilities in the operating strategies that aims to tackle the problem of the ambient temperature variation. If no actions are made the increase of the ambient temperature is showed to lead to an increase of the compressor inlet temperature. This variation of thermodynamic conditions pushes the operating point of the inlet compressor away from the supercritical area, and in particular to operations at lower density that translates into the increase of the volumetric flow rates, that lead to main compressor operations above the limit of its performance maps. In this chapter this problem is meant to be tackled thanks to the a controlled regulation of the CIT and CIP values, made possible via the variation of respectively the HRU fan rotational speed and the plant inventory. Is in fact showed that increasing the CIT, leads to thermodynamic conditions with higher density, thus counteracting the increase of density caused by the CIT and limiting the volumetric flow rate in the main compressor.

The operating strategies will be carried out thanks to a simulation that generates operating matrix where each operating condition is a function of CIT, CIP and Load, and limits on the compressor operation and the HRU will be transformed in areas that will be overlapped onto the maps and erase the unfeasible operations. Three main strategies have been defined. The standard operating condition with variable ambient temperature present the huge limitation of the maximum power produced, therefore the first operating strategy aims to create the operation curve with the highest maximum power produced. This strategy showed to be successful. The increase of the ambient temperature still cause a decrease in the maximum power production of the plant. However, thanks to the regulation of the compressor inlet conditions is possible to operate till 32°C at 100% with a loss of maximum power produced of just 1.2 MWel, compared to the 12.5 MWel of the standard operation. Unfortunately this operation presents the disadvantage of having a huge increase of the maximum pressure of the cycle that makes the operating points at high ambient temperatures not realistically feasible. Therefore another solution is found and thanks to the new limit on the maximum pressure a new curve of operation is generated. This power produced in this operation is lower than the one without the limitation on the pressure component and at 32°C the loss in power will be of around 3.5 MWel, almost three times the one assessed by the previous strategy but still much lower than the standard one. Moreover it does not present the problem on the maximum cycle pressure. The last operation strategy assessed is constructed removing the direct regulation of the minimum pressure via injection and extraction in the cycle of new mass of CO_2, therefore keeping the inventory constant. In this case is showed that the operating curve generated is really similar to the previously defined with the pressure limit. This result could be interesting if the plant works in an on off operation throughout the year, suggesting the removal of the inventory tank. The strategy with no inventory variation showed in fact to be able to work properly without the injecting or extracting mass from the design conditions until the last temperature considered of 32°C. Is important to highlight that removing the inventory variation, the plant will not be able to work anymore at part load operation, thus loosing part of the benefits of the sCO2-Flex cycle that is designed to have good performances till 20% of load.

5 | Conclusions and future developments

This thesis work goal is to find feasible operating strategies able to tackle the effect of the ambient temperature increase on the performance of dry-cooled coal power plants based on supercritical CO2 power cycles.
The analysis is made focusing around the sCO2-Flex project that aims to create an innovative sCO2 brayton cycle able of flexible and efficient operation at various loads. However the ambient temperature increase could obstruct the achievement of that goal. At higher temperature in Section2.2 is in fact showed that the operating conditions at the main compressor inlet are pushed away from the supercritical area thus loosing the benefits of real gas behaviour. Moreover the new operating condition, due to the increase of the compressor inlet temperature, presents a lower density value that generates a significant increase in the volumetric flow rate. Due to this increase the main compressor reaches the upper limit of its performance map and in order to continue to operate the plant, the thermal input into the cycle is rapidly decreased. This showed to be a critical aspect of the cycle since an increase of ambient temperature of around 12°C (T_{amb}=32°C), the maximum power that the plant can produce is just around 50% of it design value.
Two major kind of strategies are therefore studied in order to tackle this issue. The first type is defined as *Constant minimum pressure*, the decision of keeping the minimum pressure constant is based on the consideration that decreasing the pressure ratio the plant will work in a less efficient operation thus producing less power. At first it is examined the possibility to change the TIT, but decreasing that value showed to have a detrimental effect on the plant operation with the variation of the ambient temperature. The increase of the TIT value therefore could result beneficial in this situation, however it is not a possible regulation due to the limits on the materials of the components. Therefore is defined a strategy that exploit the peculiar cycle layout of being a recompressed cycle. From the analysis of the compressors is in fact showed that while the main compressor suffers an increase of the volumetric flow rate, the secondary is in the opposite situation. In this operation is introduced the concept of Split Ratio that defines the percentage of

the total mass flow that is processed by the main compressor. Therefore, decreasing the Split Ratio could redirect the excess of mass flow rate toward the secondary, and not being possible to directly changing the Split Ration a way of doing it indirectly is found acting on the mixing temperature difference at the outlet of the LTR cold side. However this solution is not efficient, in fact is showed that despite the huge variations of temperatures the split ratio changes only slightly. Moreover due to the different thermodynamic conditions at the inlet of the compressors, the redirection of the mass flow have a much greater impact on the secondary compressor than for the main one, greatly limiting the range of regulation. At 100% load is in fact showed that with a variation of Δ $T_{mix,LTR}$ of 19.1°C the secondary compressor also reaches the upper limit of its performance map. Due to this regulation limits three curves at different values of Δ $T_{mix,LTR}$ (5°C,10°C and 15°C) have been computed. However neither of them showed to significantly improve the operation at variable ambient temperature, and in some conditions it aggravates the situation due to the loss in efficiency.

Constant minimum pressure strategies are therefore set aside and *Variable minimum pressure* strategies have been analyzed. This strategies aimed to tackle the abrupt increase of the volumetric flow rate in the main compressor, with a controlled regulation of the main compressor inlet condition. Is in fact showed that the decrease of density caused by the increase of the CIT can be limited by an increase of CIP that will push the thermodynamic point towards an area with higher density. This set of strategies showed to perform much better than the one at constant minimum pressure, the curve of the maximum power generated is in fact much higher than the standard operation one. There is still a reduction of the maximum power produced, however now at T_{amb}=32°C the percentage reduction of power produced will be only less than 5% contrary to the 50% of standard operation. Unfortunately this strategy showed to have a disadvantage in term of safety of operation, it is showed that in order to reach values similar to the design condition the maximum pressure of the cycle increase significantly. This strategy is therefore considered not optimal due to the safety issues that much higher pressures of operation could bring to the plant. It is therefore assessed a new operation strategy that follows the same logic of the previous one but with a limit on the maximum pressure value. Despite performing slightly worse than the previous one this strategy still showed to perform much better than the standard operation. The percentage reduction of the power produced at T_{amb}=32°C will be around 14%, more than the previous strategy but still far better than the nominal one.

Lastly it has been analyzed the the no inventory operation, is in fact considered interesting to assess if its possible to operate without the inventory tank. In this condition the CIT is the only controlled variable at the compressor inlet while the CIP will change accordingly.

From the results is showed that the operating curve in this condition it is not far from the previous one. This outcome could lead to new interesting cycle layouts, is in fact showed that in an on/off operation the inventory variation is not necessary in order to reach almost optimal values of power production, and the inventory tank will not be necessary even with the increase of the ambient temperature. However it is important to highlight that removing the inventory tank the plant will not be able to work anymore at part load operation, thus loosing part of the benefits of the sCO2-Flex cycle that is designed to be flexible and have good performances till 20% of load. Therefore, the best strategy that is proposed to adopt is the variable minimum pressure strategy that also takes into account the limit on the maximum pressure.

5.1. Future developments

This thesis work could be developed further in other studies. It is in fact possible to remind that the whole analysis of the alternative operation strategies has been made considering a constant ambient temperature at the boiler input. This assumption, despite having been justified, generate a non realistic plant performance. Therefore it will be interesting to evaluate how the operating curves assessed in the thesis work will change including the effect on the boiler performance.

Future works could also focus on reducing the discretization and range of the input variables used in the simulation of the performance curves. This will expand and improve the analysis given that it will give the chance of extract the very best operation for each ambient temperature condition. Moreover expanding the range of values will make possible to evaluate the plant performance at higher temperature conditions than the one assessed in this thesis work.

Finally is important to highlight that one of the objective of the sCO2-Flex project was to build expertise around sCO2 for electricity generation. This solution could be in fact used as a starting point in order to study other applications of supercritical CO2 Brayton cycles. One of them being the CSP, that could benefit from the adoption of this cycle but also present the need of a good operation strategy that will be able to tackle the effect of the ambient temperature.

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GHG	*Greenhouse Gases*
EDF	*Électricité de France*
$\mathbf{C}_P$	*Critical Point*
$\mathbf{T}_C$	*Critical Temperature*
$\mathbf{P}_C$	*Critical Pressure*
HRU	*Heat Rejection Unit*
HTR	*High Temperature Recuperator*
LTR	*Low Temperature Recuperator*
PHE	*Primary Heat Exchanger*
APH	*Air Preheater*
IGV	*Inlet Guide Vanes*
TIT	*Turbine Inlet Temperature*
CIT	*Compressor Inlet Temperature*
CIP	*Compressor Inlet Pressure*
SR	*Split Ratio*
ε_{air}	*Excess air coefficient*
ρ	*Density*