sCO2flex

Technology Brochure

Making fossil fuel-based electricity production more flexible using supercritical CO_2 technology

This project has received funding from the European Union's Horizon 2020 research and innovation programme under grant agreement N° 764690.

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List of abbreviations

BOP Balance Of Plant
CAPEX CAPital EXpenditure
CFD Computational Fluid Dynamics
CW Civil Works
DGS Dry Gas Seal
IGV Inlet Guide VaneHTR – High-temperature Recuperator (plate heat exchanger used to preheat the CO_2 before it enters the boiler)
IPCC Intergovernmental Panel on Climate Change
LHV Lower Heating Value (amount of heat released on combustion of a fuel, assuming the steam released by the combustion does not condense)
LTR Low-Temperature Recuperator (plate heat exchanger used to preheat the CO_2 before it enters the boiler) PCHE - Printed Circuit Heat Exchanger
PFHE Plate-Fin Heat Exchanger
PI Proportional-Integral controller
PID Proportional-Integral-Derivative controller
sCO_2 Supercritical Carbon Dioxide
SH Super Heater

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1. Introduction

1.1. Context

The EU has set itself a long-term goal of reducing greenhouse gas emissions by 80-95%, when compared to 1990 levels, by 2050. The EU 2050 Energy Strategy includes therefore new challenges and opportunities. Renewable energy (such as wind and solar) is gaining momentum and now raises the question of grid stability in the event of large power output fluctuations. In this context, enhancing the flexibility and the performance of conventional power plants is seen as a good opportunity to both secure the energy grid while reducing their environmental impact. Therefore, it is necessary to develop innovative and cost-effective ways of enabling existing and future fossil fuelled power plants to be flexible enough to deal with load fluctuations and to reduce emissions.

1.2. Key Results

Supercritical carbon dioxide (aka sCO_2) is a fluid state of carbon dioxide where it is held at or above its critical temperature and critical pressure. The fluid presents interesting properties that promise substantial improvements in conventional power plant system efficiency. Due to its high fluid density, sCO_2 enables extremely compact and highly efficient turbomachinery.

The sCO_2-flex consortium therefore sought to develop a scalable and modular design of a 25 MWe plant using supercritical CO_2 to enable an increase in the operational flexibility (fast load changes, fast start-ups and shutdowns) and efficiency of existing and future solid-fossil power plants, thus reducing their environmental impacts, in line with EU targets. To reach these goals, the project secured a €5 million grant from the European Commission's Horizon2020 Programme.

The results currently available indicate that a cycle net efficiency of 42.5% can be reached, with a boiler efficiency of 92.3% (based on the fuel's lower heating value). The resulting plant net efficiency of 37% (including boiler and flue gas treatment auxiliaries) allows a reduction by 8% of the greenhouse gases emissions in operation compared to a water/steam plant of the same power output, and can be operated flexibly between 20% and 100% of its nominal load. Those improvements in performance come at an investment cost comparable to that of a similar water/steam plant, making sCO_2 cycles a competitive option at a 25 MWe scale.

1.3. The main technological challenges of sCO_2

While state-of-the-art power cycles involve water/steam cycling between around 40°C / a few millibars and 620°C / 300 bars, the cycle designed in sCO_2-flex uses CO_2 between 33°C / 81 bars and 620°C / 250 bars. This change in both working fluid and operating conditions implies several challenges in the process of designing individual equipment as well as a whole plant. The figure below (figure 1) is a simplified flow diagram of the cycle developed in the project. While some components such as the secondary compressor and the cold source benefit, to a limited extent, from industrial experience in other fields (refrigeration, oil and gas), most components are faced with challenges relative to corrosion, high flowrates and structural reliability.

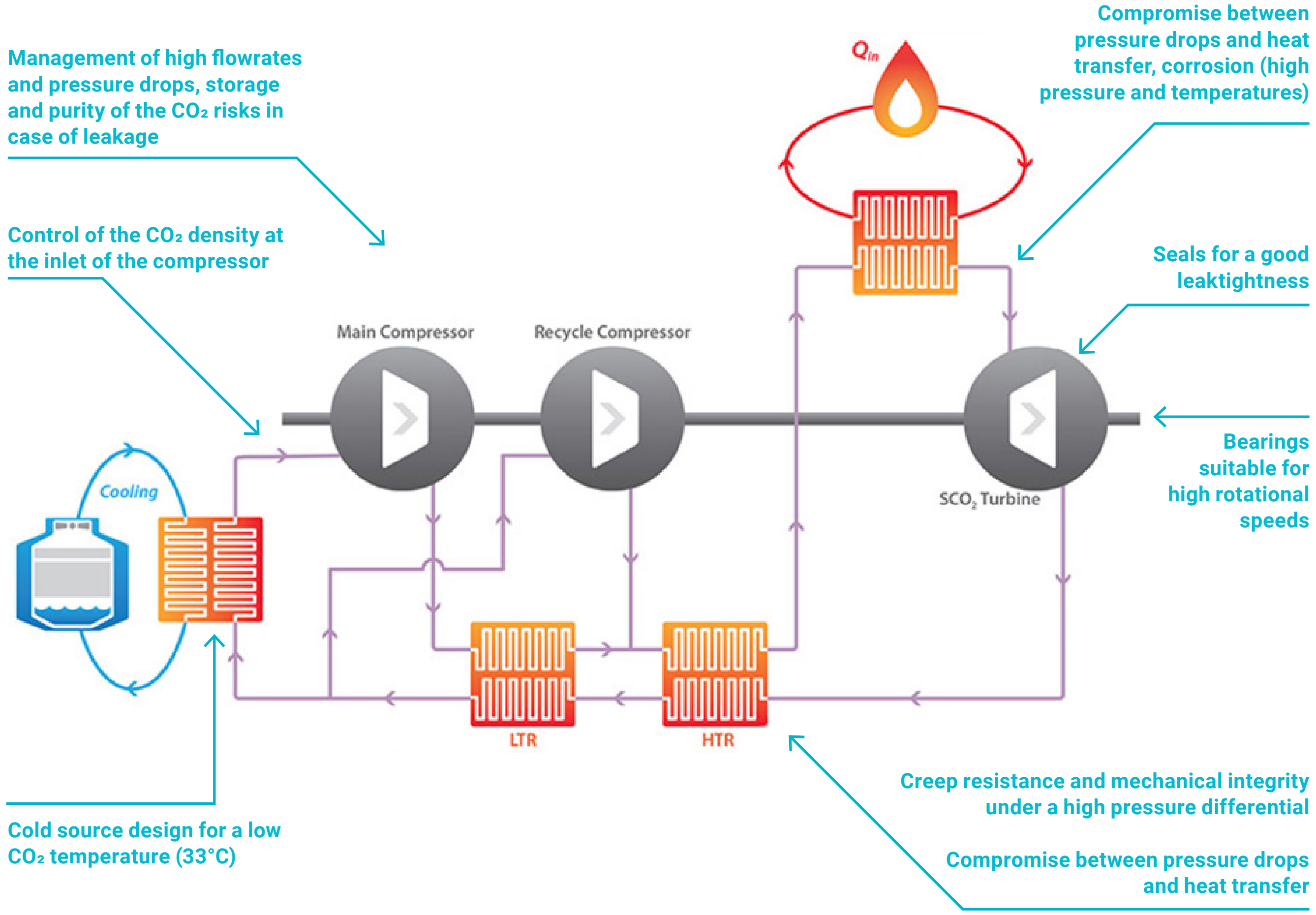

Figure 1: Main technical hurdles prior to the commercial deployment of sCO_2 cycles

1.4. Selecting the right architecture

While the state-of-the-art architecture of water/steam cycles results from a roughly 250-year-old experience on thousands of different cycles, the interest in sCO_2 is much more recent. Tens of different cycle architectures have been suggested in scientific literature, serving different purposes and/or leading to different benefits. An assessment of around 40 different architectures was performed, considering the preliminary requirements of each individual equipment. The selected architecture shown below is a compromise between cycle efficiency, turbomachinery efficiency and heat recovery in the boiler. Compared to the simple recuperated cycle, it involves two improvements: recompression allows a much better heat recovery in the recuperators, substantially increasing the efficiency, while the high-temperature recuperator bypass improves the boiler's efficiency.

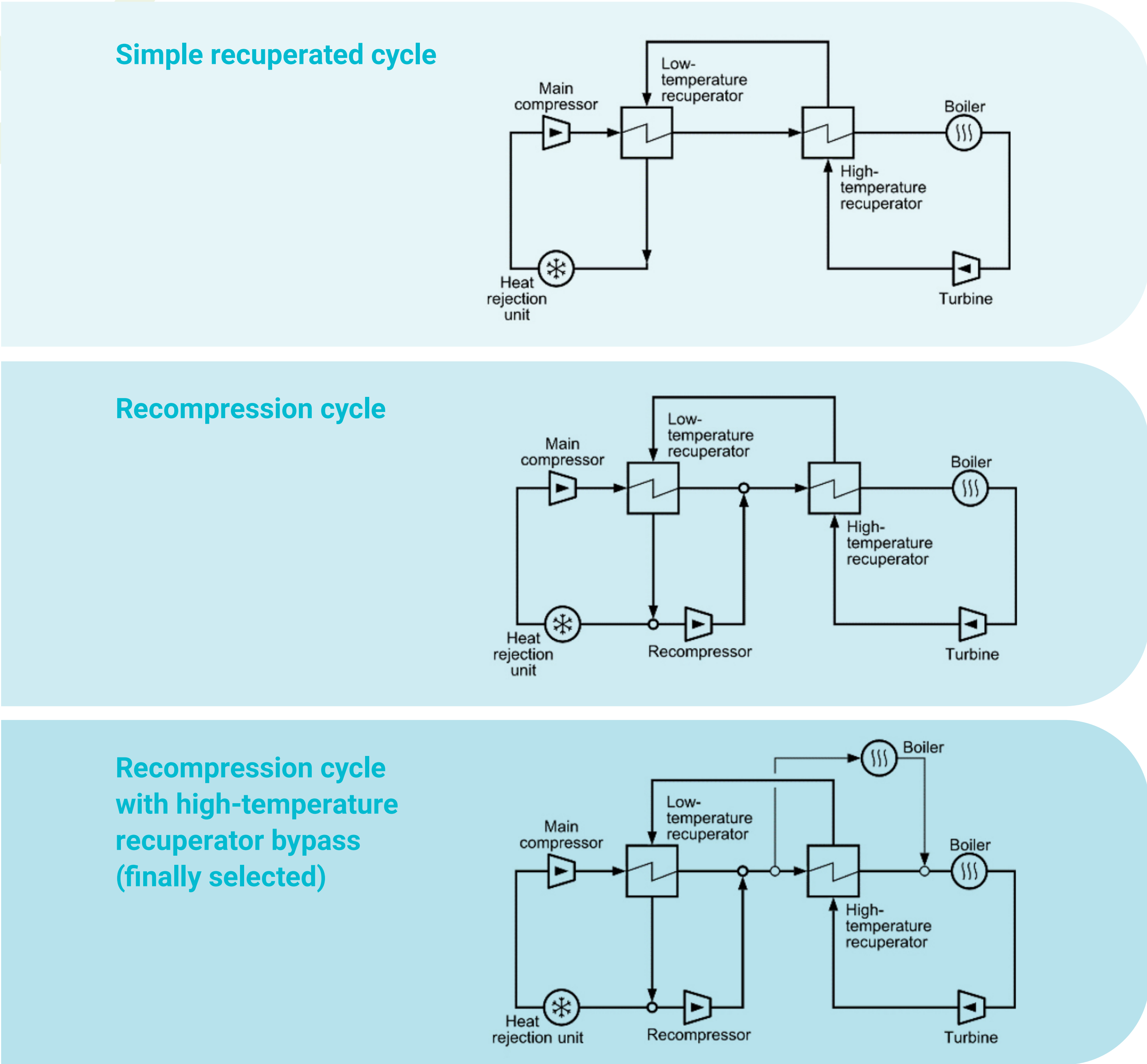

Figure 2: Improvements in the sCO_2 cycle's architecture leading to the final structure of the cycle designed in sCO_2-flex.

2. Plant design

The sCO_2 cycle consists of three main components. Namely, the turbomachinery, the boiler and the recuperator. While the components of conventional power cycles as the steam Rankine cycle are well established, sCO_2 cycle components are still in the development phase. sCO_2 properties and cycle conditions pose several challenges to their design and operation. For instance, the corrosive concerns or the much higher pressure of the cycle at the low-pressure side. Therefore, industrial and research institutions are cooperating to bring this technology to its next technological readiness level (TRL).

2.1. Materials

Corrosion and erosion (due to high flowrates), as well as the vulnerability of welds linked to the presence of CO_2 are not so well known as for components exposed to water and steam. The stakes are high: while noble, high-alloyed metals can ensure a long service, with a good and lasting mechanical resistance under severe conditions, their use is detrimental to the plant's environmental impact and economics. To address these challenges, several materials (mostly designed for supercritical steam applications) were selected for each major component and have been tested in the project after hundreds of hours spent in industrial- or research-grade CO_2, under severe conditions. These investigations provide more accurate guidelines for material selection in the design of sCO_2 components.

2.2. Boiler design

This section is an overview of the coal-fired boiler designed for re-compression supercritical CO_2 Brayton cycle. The general requirements for the boiler defined by the project are high efficiency, high flexibility, low pressure losses and a unit power output of 25 MWe (with possible scale-up to 100 MWe). The boiler design is based on the cycle architecture and its required thermal load of 59 MWth.

The Boiler was designed for brown coal from the Czech mining location Bilina with 29.7 % of water content, 9.2 % of ash content and a Lower Heating Value (LHV) of 16.9 MJ/kg. But another coal may be used with different operation characteristics, such as fuel consumption, flue gas temperatures, ash production and so on.

Thermal design

There are two possible types of combustion — fluidized bed combustion and pulverized combustion. For this application, a pulverized coal-fired boiler was selected as more advantageous than a fluidized bed boiler, due to its higher flexibility and its ability to work at smaller loads. The duct system and auxiliary components of the designed boiler is arranged as two parallel channels with the combustion chamber in the first pass and convective heat transfer surfaces in the second. This type of boiler is called two-pass or TT-shape. Its advantages are a simple construction and an easy cleaning of heat transfer surfaces. In the combustion chamber, there are six Low-NOX burners in two rows placed in the front wall. This arrangement is suitable for both 25 MWe and 100 MWe boilers. Additionally, to meet emission limits, a Selective Catalytic Reduction (SCR) should be taken into an account.

BOILER PARAMETERS

- **Heated CO_2 temperature: 620°C**
- **Thermal load: 59 MW**
- **Fuel consumption: 13.53 t/h**
- **Efficiency: 92.5 %**

The overall height of the boiler is 19.5 m. There are five heat exchangers in order to meet the required outlet CO_2 temperature, which is 620 °C. Their connection is depicted in figure 3. The heat exchangers are marked as SH00-SH04. The heat exchanger SH04 is a bypass of a high temperature recuperator (fraction of CO_2 main stream flows there). After heating the bypass to the required temperature, it is mixed with the main stream and subsequently the mixed streams enter the boiler, respectively the furnace enclosure (SH00). The furnace superheater consists of a membrane wall and it is the first part of a high-pressure system with a full flow of CO_2, where the main part of heat is transferred. The main stream is lead into the boiler in the furnace due to the relatively low temperature (478 °C). The furnace membrane wall must be very well cooled down and the mean wall temperature should be low to minimize material demands due to the extremely high heat flux in this part of the boiler. The CO_2 stream passes from the membrane wall to the platen heat exchanger (SH01) placed in the upper part of the furnace and then continues to the vertical convective pass where a convective superheater SH02 is placed. The final superheater SH03 is placed in the crossover pass where there are the appropriate flue gas temperatures to reach the desired outlet temperature of CO_2 stream (620 °C).

BOILER GEOMETRY

- **Boiler type: supercritical once through PC boiler**
- **Dimensions of furnace: 4.60 x 6.90 x 13.33 m**
- **Total boiler house height from ± 0,00: 19.5 m**
- **Total mass ~ 220 t**

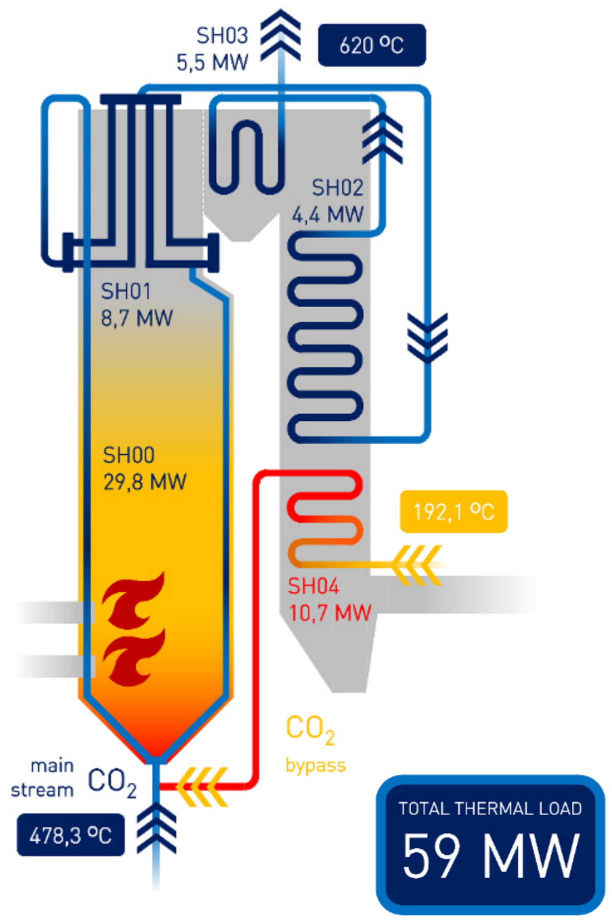

Figure 3: CO_2 path in the boiler

Impacts of CO_2 as Working Fluid

From the boiler construction point of view, the essential difference compared to conventional steam boilers is the CO_2 temperature at the boiler inlet, which can reach extreme values depending on the CO_2 cycle configuration. The higher temperature of working fluid at the inlet to the boiler leads to a higher thermal stress on the furnace enclosure and more difficult extraction of residual flue gas heat. To achieve a reasonable boiler efficiency, the flue gas must be cooled to a low temperature, which can be done by preheating the combustion air to very high temperatures. However, such a measure causes a further increase of the temperature in the combustion chamber and hence thermal stress of the membrane wall material. Protective linings could be used here, but this does not correspond to the requirement for the boiler flexibility. As a solution, there is the high temperature recuperator bypass used for the last heat transfer surface of the pressure system (SH04), which respectively corresponds to the economizer in conventional steam boilers. Advantage of the bypass is relatively low CO_2 inlet temperature (188.7 °C) which allows to cool down the flu gas before entering the air preheater from 637 °C to 373 °C. By preheating combustion air up to 350 °C in the air preheater, the flu gas is then cooled to 138.5 °C, which provides boiler efficiency 92.5 %.

Main challenges:

- **Extraction of residual flue gas heat**
- **Reduction of pressure losses**

Another significant impact of CO_2 is its high mass (and volume) flow rate causing high pressure loss. The pressure loss can be reduced by using more tubes of the heat exchanger parts and by increasing their inner diameter. However, increasing the diameter brings a disadvantage - greater wall thickness of the tubes - and thus higher investment costs and lower heat transfer due to greater conductivity thermal resistance. To obtain low pressure losses and a high heat transfer coefficient on CO_2 side (which are contradictory requirements), tubes were proposed with the inner diameter in the range from 22.22 mm to 23.88 mm in the case of the CO_2 main stream and 19.84 mm in the case of the bypass (SH04). A moderate overall pressure loss of 491 kPa was obtained.

2.3. Turbomachinery

Turbomachines play a key role in the sCO_2 cycles. They represent the main components for pressurization and power extraction from the cycle. Furthermore, as the main objective of the cycle is to provide high flexibility, the compressors and turbine are required to handle high variations of mass flow rate. This, in addition to the peculiar properties of sCO_2, necessitates an in-depth study on the design, control strategy and material selection for reliable and efficient operation of the turbomachinery components.

Modeling challenges for the design of sCO_2 compressors

Two sCO_2 compressors are required for the selected cycle layout. While the secondary compressor has an inlet condition at around 80°C and thus represents to some extent a conventional component, the main compressor inlet condition is near the critical point with a pressure and temperature of around 80 bar and 33°C, respectively. In this region, the thermo-physical properties of sCO_2 are characterized by a high gradient with respect to pressure and temperature. For instance, a variation of one Celsius degree in temperature can lead to a density ρ variation by as much as a factor of three. Moreover, sCO_2 shows a substantial deviation from the ideal gas law with compressibility factor Z ranging between 0.2-0.6. It is therefore essential to understand the effect of sCO_2 properties on the compressor design and performance. At the University of Duisburg-Essen (UDE), a series of investigations has been conducted for this purpose

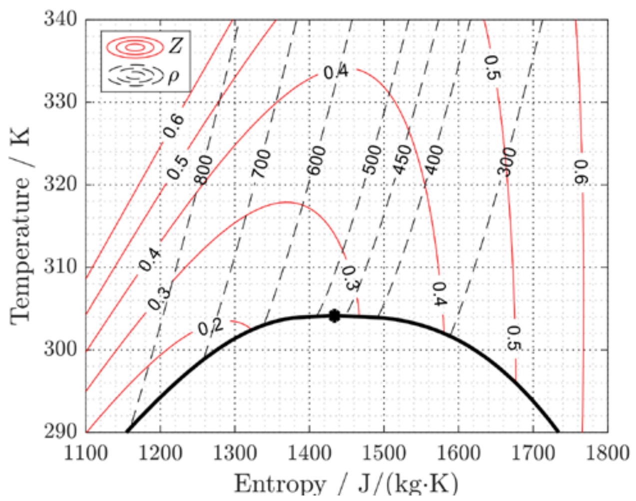

Figure 4: Properties variation of CO2 in the supercritical state

Boundary Layer

The main role of the compressor is to raise the fluid pressure from the low-pressure side (around 80 bar) to the high-pressure side (around 250 bar at the design condition) by transferring energy to the fluid. In turbocompressors, this is realized by turning the flow with the blades. Boundary layer stability plays, therefore, a decisive role in the reliability and operating range of the compressor. It will also directly affect skin friction losses. To acquire insights into sCO_2 flows, a fundamental study on the boundary layer over a NACA airfoil is conducted by UDE. Air is selected for comparison as a typical fluid that obeys the ideal gas law. Simulations are conducted under Reynolds number and Mach number similarities. These similarities are assigned based on the fluid properties at the inlet of the numerical domain. Although sCO_2 experience much higher properties variation along the blade compared to air, results show a negligible effect on the lift coefficient and the drag coefficient that shows excellent agreement between sCO_2 and air results. This suggests that sCO_2 properties do not affect the compressor operating range and established loss correlation can be applied to sCO_2 compressors.

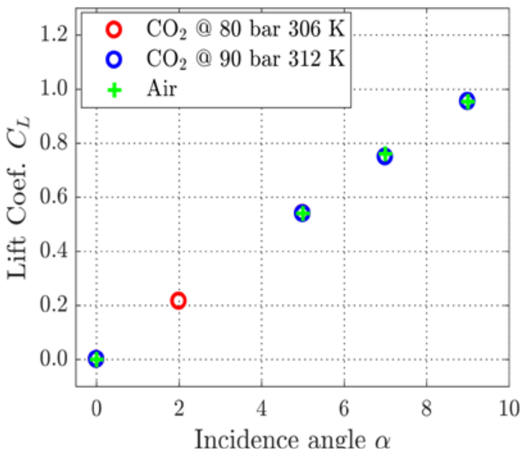

Figure 5: Variation of the lift coefficient in terms of the incidence angle

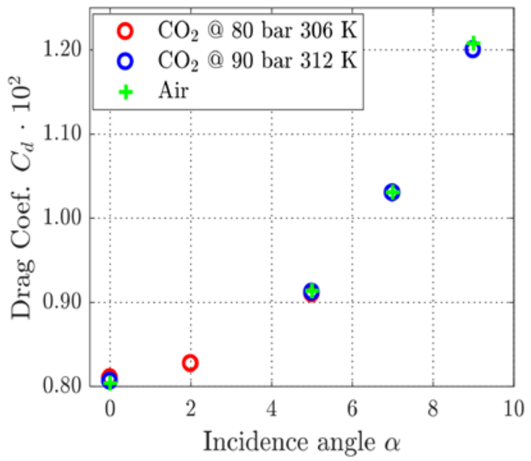

Figure 6: Variation of the drag coefficient in terms of the incidence angle

Aerodynamic design

Due to the peculiarity of the sCO_2 properties, questions arise regarding the applicability of conventional design methods on sCO_2 compressors and the impact of different inlet conditions on the compressor. The boundary layer study of sCO_2 suggested the applicability of established loss correlations on sCO_2 compressors. Therefore, a design tool is developed based on a 1-D performance model derived from the experience with ideal gases. First, the model plausibility is checked against experimental pressure ratio measurement taken by UDE during the sCO_2-HeRo project. Thereafter, an extensive set of inlet conditions is selected and for each, a unique compressor design is generated. With the help of Computational Fluid Dynamics (CFD), the validity of the developed design approach is verified. Moreover, CFD results show that the departure from the ideal gas law has no appreciable impact on the compressors' design practice. Instead, the performance of sCO_2 compressors correlates with the specific speed, which is a machine parameter with a known effect on the performance.

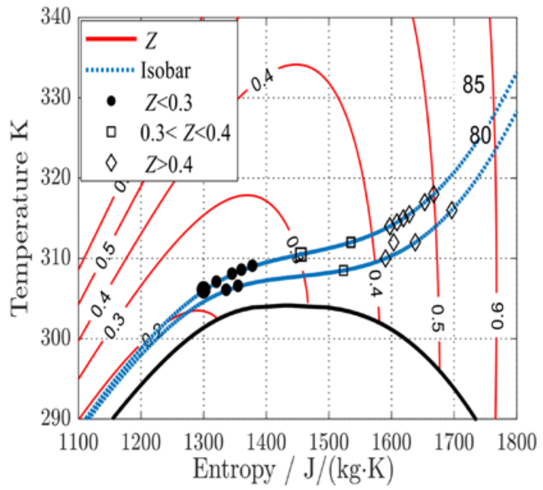

Figure 7: Selected compressor inlet conditions

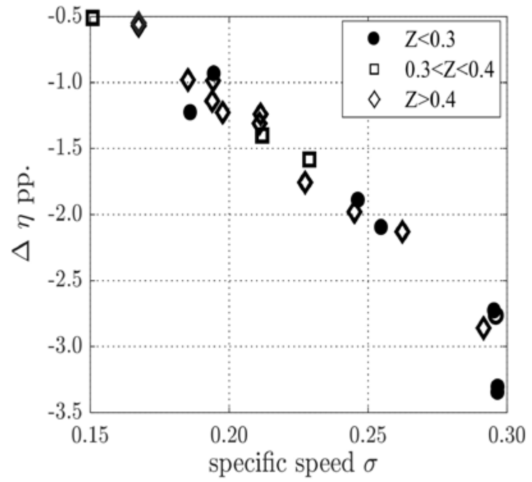

Figure 8: Deviation of 1-D efficiency from CFD result in terms of the specific speed

Performance map

The simulation and control of the cycle require performance maps for the turbomachinery. A dimensionless map is used for this purpose to allow the consideration of the high property variations with respect to the pressure and temperature at the compressor inlet. Established map formulations neglect the impact of the isentropic exponent variation, which is limited to a range between 1 and 2 from a fluid to another that obeys the ideal gas law. sCO_2 can, however, exhibit much higher variation from 1.5 to more than 10. A study was conducted by UDE to investigate the impact of this variation on the compressor performance by considering three different isentropic exponent values at the compressor inlet. The results show a reduction in efficiency as the isentropic exponent increases. Based on that, another map formulation was proposed that uses the compressor exit flow coefficient instead of the reduced mass flow rate at the inlet. In this case, results show the ability of this formulation to consider the isentropic exponent variation at the compressor inlet.

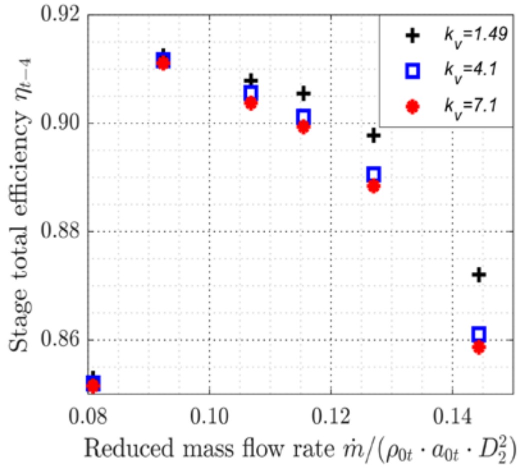

Figure 9: Effect of the isentropic exponent on the compressor efficiency

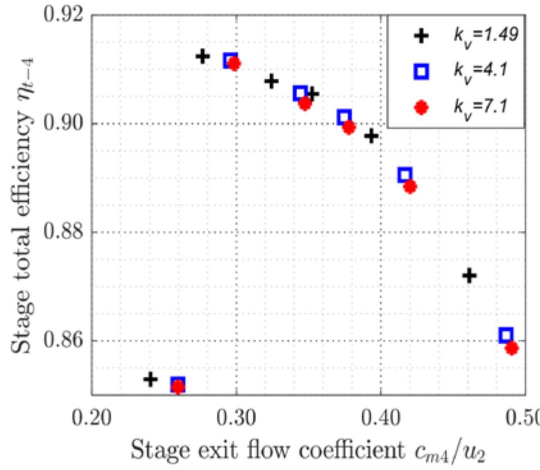

Figure 10: Correlating the compressor efficiency with the exit flow coefficient

Compressor impeller and Dry Gas Seal system

Beyond the challenges they raise modeling, the large variation of CO_2 properties in the supercritical region may have technological implications on the design of turbomachinery, where the flow field combined with departures from the ideal-gas behavior approaching the critical point, could promote phase-change phenomena resulting in non-conventional flow configuration, with coexistence of cavitation phenomena and compressibility effects. In the Brayton cycle selected the compressors are working in parallel evolving respectively around 2/3 and 1/3 of the total plant mass flow rate. The main compressor suction conditions are very close to CO_2 critical point. In this condition the intake flow approaches the region of large gradients where relative Mach at impeller shroud increases, reaching the sonic limit with consequent blockage and direct implications on machine performance and operability. For the first impeller, a proper design was necessary to mitigate effects of phase change keeping at the same time the performance target in terms of pressure ratio and efficiency. The optimization of blade shape and loading has been carried out by numerical simulations. As outlined in a paper written by Baker Hughes in collaboration with Politecnico di Milano* also the CFD methodology used to verify the impeller's detailed design has to be carefully considered because the fluid properties are far removed from ideal conditions and only a proper thermodynamic gas modeling is able to capture the impact of phase change.

The design of the Dry Gas Seal (DGS) system was influenced by the special operating conditions and sCO_2 properties. Special care has been given to the seal gas flow heating to avoid liquid formation in the seals during normal operation. The methodology developed by Baker Hughes to evaluate the system behavior in case of dry gas seal failure, in accordance with API692, was then applied to the sCO_2-Flex centrifugal compressor. The analysis highlighted the risk of formation of solid CO_2 inside the seal gas vents due to the low temperature of the CO_2 processed. The addition of Pressure Safety Valves with dedicated dimensioning and set points was validated to ensure pressures above the triple point pressure inside the seal gas vent lines and avoid possible blockage of these lines. The Valves' selection was made to avoid possible ice formation inside the valves' bodies. The discharge lines' size and routing were also designed to ensure that solid phase transition, if any, would occur in atmosphere without blocking the lines.

This solution is recommended for all centrifugal compressors operating with CO_2 in supercritical conditions and at low temperatures.

Another main challenge of the selected cycle is the flexibility in terms of net power. For this reason, turbomachinery are required to operate from 100% (nominal operating point) to 20% load. To achieve this level of flexibility, design choices such as compressor Inlet Guide Vanes (IGV) and variable speed drivers have been implemented.

* Giacomo Persico, Paolo Gaetani, Alessandro Romei, Lorenzo Toni, Ernani Fulvio Bellobuono, Roberto Valente, Implications of Phase Change on the Aerodynamics of Centrifugal Compressors for Supercritical Carbon Dioxide Applications, 2021, https://doi.org/10.1115/1.4049924

Compressor prototype

The design concepts introduced in the main compressor has been finalized in a 5.4MW compressor prototype produced and tested, reaching the supercritical conditions at the inlet of the compressor, in Baker Hughes Florence site.

Dealing with sCO_2 has been the main challenge of the design of the prototype and of the test facility. The CO_2 thermodynamic properties (e.g. compressibility, Mach number, etc.) near critical point shows large gradients. Therefore, a dedicated detailed aerodynamic analysis of movable IGV and 1st impeller has been carried out to understand the impact on overall compressor performance and to ensure no local conditions with liquid CO_2 occurred at design condition. Compressor inlet temperature control is an important parameter that has been taken in account carefully during the design phase to meet compressor requirements.

Cooler control needs to guarantee that, in all operating conditions, steady state and transient, inlet temperature range (min/max oscillation) allows to avoid CO_2 liquid formation on the first impeller and to operate the compressors without any limitations.

In addition, sCO_2 density is comparable with the CO_2 liquid one. The extreme power density (power to inertia ratio) for this application required special attention on mechanical verification of the main compressor components, on the compressor rotor dynamic behaviour and on the evaluation of the thrust load acting on compressor thrust bearing. On the other hand, the extreme power density, allow the selection of a smaller size compressor, providing benefits in terms of footprint and equipment cost. Nevertheless, this aspect brings challenges to the design and manufacturing of the compressor, being much smaller than typical process centrifugal compressors.

Figure 11: Compressor prototype built and tested in Florence by Baker Hughes

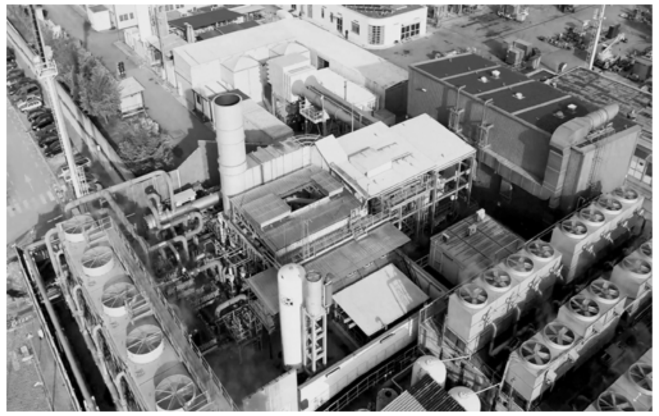

Figure 12: Baker Hughes' testing facility in Florence

Turbine

The flow-path of the turbine is composed by 5 stages working at 50% reaction degree. The number of stages has been selected after a thorough optimization turned to balance conflicting requirements like efficiency, rotor-dynamic behavior and limitation of the thermal stresses in order to maximize the flexibility of the operation: improvements of the efficiency and of thermal stresses during transient operation are typically achievable by reducing the diameter of the rotor (the efficiency takes advantage by the possibility to increase the aspect ratio and the radius ratio of the flow path) but this is conflicting with the necessity to have a first natural mode of the rotor-bearings system with an adequate logarithmic decrement to ensure the stability. The stability of the rotor-bearings system has been achieved by adopting conventional tilting pads journal bearings.

Due to the high pressure at the exhaust of the expander and due to the fact that this pressure is the lowest of the cycle, hence any leakages through the shaft end seals cannot be recovered in a turbomachine operating at lower pressure, the expander has been equipped with dry gas seals (DGS), in order to minimize the leakages and not to impact the efficiency of the overall cycle. The current technology of the DGS does not allow to adopt them at the temperature at which the expander operates. For this reason, a cooling system has been designed with the aim to limit the operating temperatures in the areas of the DGS within the allowable limits. The design of the cooling system has been optimized by analysing the flow field both with a mono-dimensional flow solver and with 3D CFD, realizing finally a conjugate convective heat transfer model.

Conjugate heat transfer models have been used also to calculate the heat transfer coefficients (HTC) used in the thermo-mechanical analysis of the flow-path components. The thermodynamic properties of supercritical CO_2, in fact, results in huge values for the HTC and this is extremely challenging for the thermo-mechanical design of the main items, even if the dimensions of the expander are relatively small, compared to the power size, due to the very high density of the working fluid.

Specific design choices for partial load operation were validated for the turbine as well. Due to the limited number of stages driven by the thermodynamic cycle it has been decided not to employ the partial arc admission, since this choice would have penalized the efficiency at 100% load, privileging an almost flat behavior at partial load. It is remarked that, with the state of the art of the technology, a variable geometry at the inlet (adjustable nozzle guide vanes) is not a viable option for the combination of pressure and temperature peculiar for this application (250 bar and 620°C): the containment of the fluid leakages through the stem of the NGVs, at this level of pressure, is extremely challenging. Moreover, with adjustable guide vanes it would be possible to control the flow only in a limited range, not suitable to reach a partial load of ≈20%, which is the target. Summarizing, the operation at partial load in sliding pressure, avoiding the losses associated to a pure throttling at the inlet, is the only possibility to maintain a good efficiency.

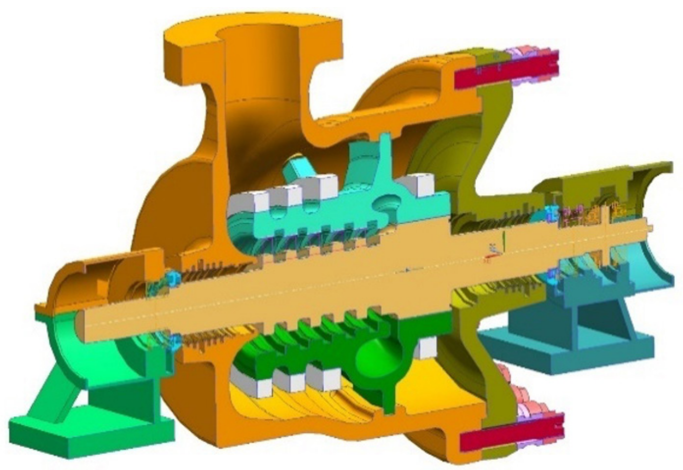

Figure 13: Longitudinal section of the sCO_2 turbine

2.4. Recuperators

A specificity of sCO_2 cycles, as compared to water/steam cycles, is that no phase change is occurring in the heat exchangers used for heat recovery. Thus, the absence of a pinch in the temperature profile of the fluids circulating enhance heat transfer performance.
In parallel there is a clear need of highly efficient technology of the heat exchanger to capitalize on this observation. This is mainly dictated by the need of recovering the largest amount of heat in both recuperators (Low Temperature and High Temperature) with the aim of achieving the highest cycle effectiveness.

Operating conditions are significant parameters of sCO_2 power systems. Recuperators must withstand high temperatures and pressures and at the same time guarantee safe operation taking into account the large pressure difference between hot and cold exchange fluids. It is challenging to achieve a solution that satisfies mechanical, thermomechanical and thermohydraulic performances.

Obviously having a high pressure on one side of the cycle helps to reduce the size of the principle components. Considering heat exchangers, high pressure permits to reach great heat exchange characteristics and to reduce pressure drop to raise the system efficiency at the end. Nevertheless, increasing pressure induces the need to have greater material thicknesses for piping and heat exchangers. A balance must be found between the optimized technical requirements and the economic cost. A key point is to consider a convenient heat exchanger technology respecting compactness and robustness at a reasonable cost.

Another aspect to consider is the highest design temperature in the heat exchanger. The use of expensive high content nickel alloys should be avoided has this raises costs considerably. As heat exchangers constitute a large part of component cost, it is preferable to mitigate this item of expense. Thus, the temperature setting should be compatible with the use of stainless steels. This seems to be the most promising choice, especially for recuperators. The selected grade should be however in accordance with corrosion constraints.

A close temperature approach also leads to a better cycle performance. This requires more heat exchange surface anyway. Again, the plant efficiency has to be considered versus the Capital Expenditure (CAPEX). A rational analysis was carried out within the project to define the best solution between the size and cost of the components by comparison with the advantages for cycle performance.

Compact heat exchangers appear to be the most suitable under these conditions. These are known to exhibit high thermal efficiency and accept high thermomechanical strain tolerance. The reduction of the overall footprint and the limitation of the fluid inventory are other great advantages.

Different compact heat exchangers are promising candidates to provide an adequate solution. For example, Printed Circuit Heat Exchangers (PCHE) are known and employed in sCO_2 cycles. A limitation is the capital cost which is increasing compared to conventional Shell and Tube Heat Exchangers. The Plate-Fin Heat Exchanger (PFHE) is a good challenger. This technology brings gains in terms of flexibility. This particular heat exchanger is not only composed of plates. It then offers a lighter weight and therefore minimizes the metal mass. This characteristic is favourable during transient operating conditions as it allows faster dynamic responses. This type of brazed plate heat exchanger can provide very good thermohydraulic performance at moderate capital costs.

The ability of PFHE to withstand high pressure at the demanded sCO_2 cycle temperature requires a specific development. The choice of materials, manufacturing process and assembly are essential.

The selection of a convenient base material is the first step. An austenitic stainless steel is the preferred option, presenting high mechanical characteristics at 500°C and compatible with brazing technique in a vacuum furnace. It should also be easily formed to produce fins with hydraulic presses. The corrosion aspect was considered too.

Specific geometric parameters of the fins have been defined to offer channels adapted to mechanical resistance and good thermohydraulic performance.

The burst tests results show that Plate and Fin Heat Exchanger can be a convenient solution for heat recuperation in the sCO_2 system. Indeed, a burst

Figure 14: Plate and fin heat exchangers can withstand pressures up to 1200 bar before bursting.

pressure of almost 1200 bar was obtained. This qualifies the use of this type of technology at a pressure of 204 bar at 500°C, taking into account restrictive regulatory rules.

Further upgrades are in progress to increase even more the mechanical resistance of the structure by changing the base material from 316Ti to another austenitic stainless with higher mechanical characteristics. An intensive job is realised also on the brazing issue to control the metallurgical microstructure of the brazed joint by reducing the appearance of a brittle phase limiting the strength of the assembly. All these joined efforts must lead to a heat exchanger solution combining the best technical and economic accommodation for operation in flexible sCO_2 power plants.

The manufactured PFHE prototypes were tested in a pilot to evaluate hydraulic and thermal performance. Consolidated heat transfer and pressure drop data were generated that enable the design of future heat exchangers.

2.5. Engineering design

The individual design of every major component led to a preliminary engineering design of a complete plant. One of the main outputs of this design is a 3D model of the plant, shown in figure 15. This includes the sizing of the main pipes, plant auxiliaries and inventory management systems, but excludes flue gas treatment. However, the 3D plan of the plant includes enough space around the boiler and the stack for such processes to be added.

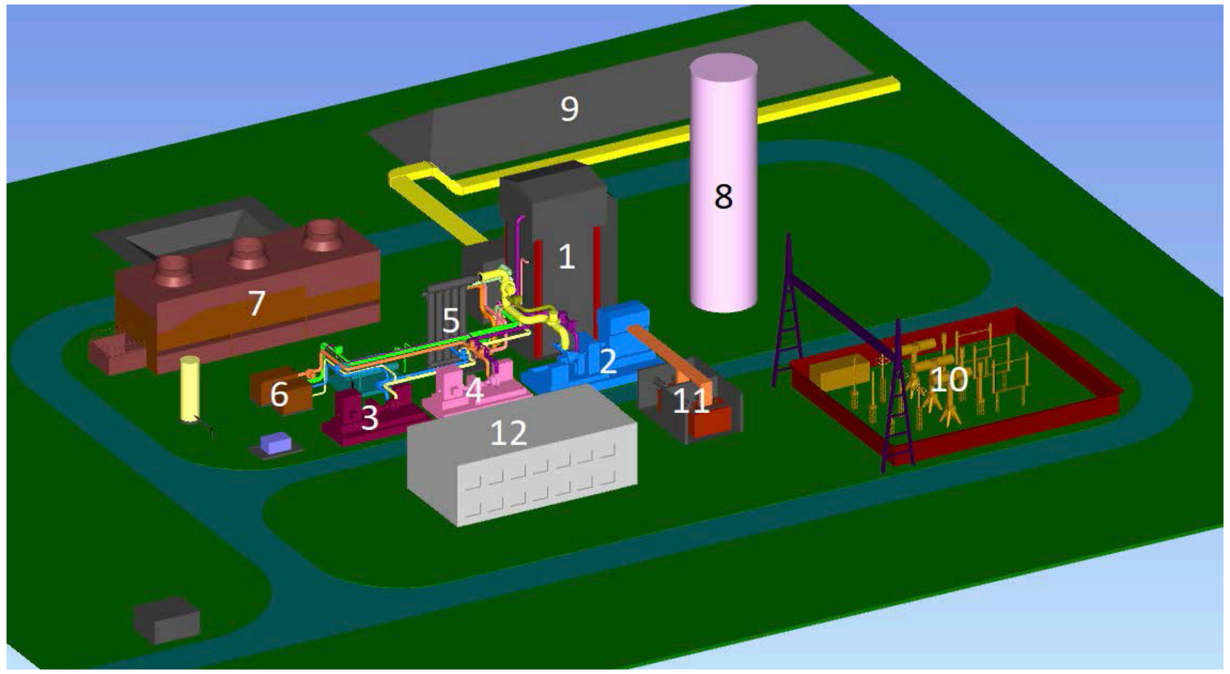

Figure 15: Overview of the 3D model of the whole plant
(1: boiler house, 2: turbine, 3: main compressor, 4: secondary compressor, 5: low- and high-temperature recuperators, 6: CO_2 inventory management tanks, 7: heat sink, 8: stack, 9: fuel storage area, 10: electrical substation, 11: transformers, 12: electrical & administration building)

3. Plant Operation

The role of fossil fuelled power plants in a future energy scenario dominated by variable renewable energies (VRE) will have to evolve from base-load operation to provide fluctuating back-up power to meet unpredictable and short-noticed load variations. The novel sCO_2-based coal power plants developed within the sCO_2-Flex project will thus have to guarantee flexibility and high performance at part load. For this reason, it is fundamental to properly model the plant behaviour and performances during transients and in part load-operation. This task has been assigned to Politecnico di Milano (Polimi), which led the activities related to the overall system modelling and analysis in design, off-design and transient conditions.

3.1. Off-design performance

The design of the selected recompressed sCO_2 cycle with a High Temperature Recuperator (HTR) bypass, its optimization and the preliminary sizing of the different plant components was performed by Polimi as first modelling step. Then, different part-load operating strategies have been investigated down to 20% of the nominal load in order to select the one that maximizes the plant performance.

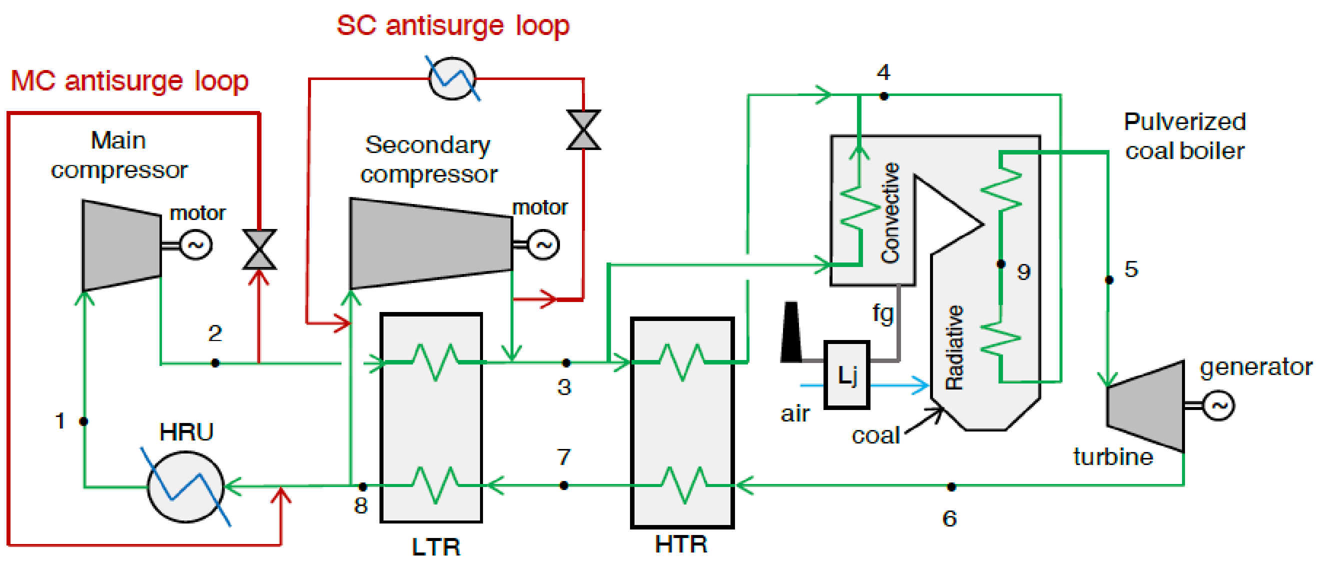

Figure 16: Schematic of the recompressed sCO_2 cycle with HTR bypass selected within sCO_2-flex

The load reduction in a Joule-Brayton cycle using an ideal gas as working fluid (e.g. nitrogen, helium) can be easily obtained by varying the cycle pressurization and proportionally of the working fluid mass flow rate: the cycle simply "shifts horizontally" in the temperature-entropy diagram as shown in figure 17, left, with a constant compression ratio and thus with an efficiency that can even be higher than the nominal one thanks to the improved heat exchangers' effectiveness. The load and flow coefficients of the turbine and compressor are kept constant in off-design conditions, guaranteeing no performance decay.

The selection of sCO_2 as working fluid allows the design of the main compressor near the CO_2 critical point (31°C, 73.7 bar), giving large benefits in terms of cycle performance due to the decreased compressor consumption, but leads to significant challenges in part-load operation: a proportional reduction of the cycle pressurization at constant compression ratio may not be the best choice for sCO_2 cycles, as the reduction of the minimum cycle pressure may imply to step away from the critical point with a significant and uncontrolled variation of the volumetric flow rate at the compressor inlet (see figure 17, right).

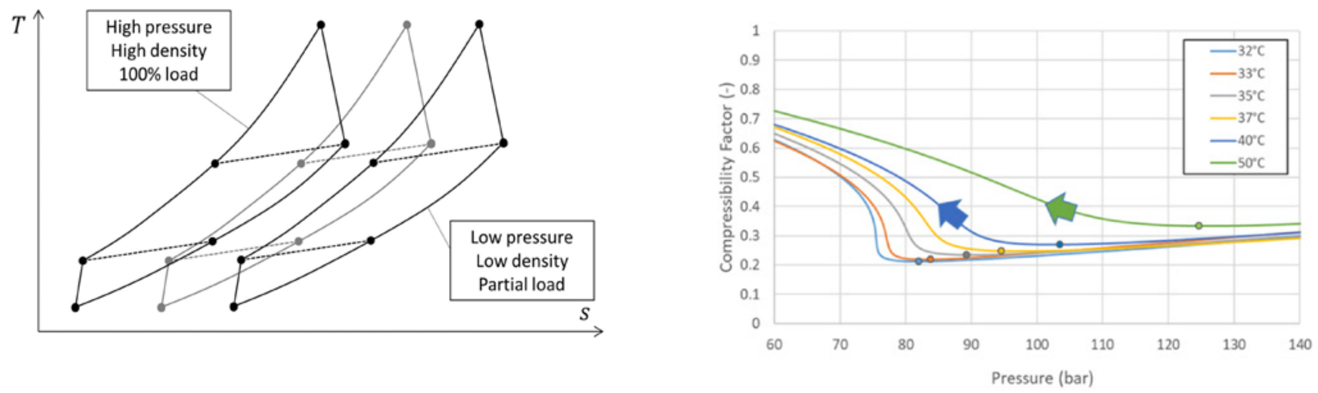

Figure 17: (left) Translation of a Joule-Brayton cycle using an ideal gas in the temperature-entropy (T-s) diagram as function of the load and (right) variation of the compressibility factor at the compressor inlet as function of temperature and pressure.

In the selected recompressed cycle with HTR-bypass, different operating strategies for load reduction are investigated acting on the different operational parameters of the system. As the amount of fuel burnt in the boiler is decreased, the best operating strategy is the one that maximizes the plant efficiency, taking into account turbomachinery part load performance maps and boiler constraints. The sCO_2 cycle adopts an expander operating in sliding pressure and two compressors able to vary their rotational speed and equipped with Variable Inlet Guide Vanes (VIGV). Moreover, the cycle heat rejection unit (HRU) is equipped with variable speed fans and the cycle working fluid inventory can be varied to set the cycle minimum pressure. The chain of operations performed to regulate the system is reported in figure 18.

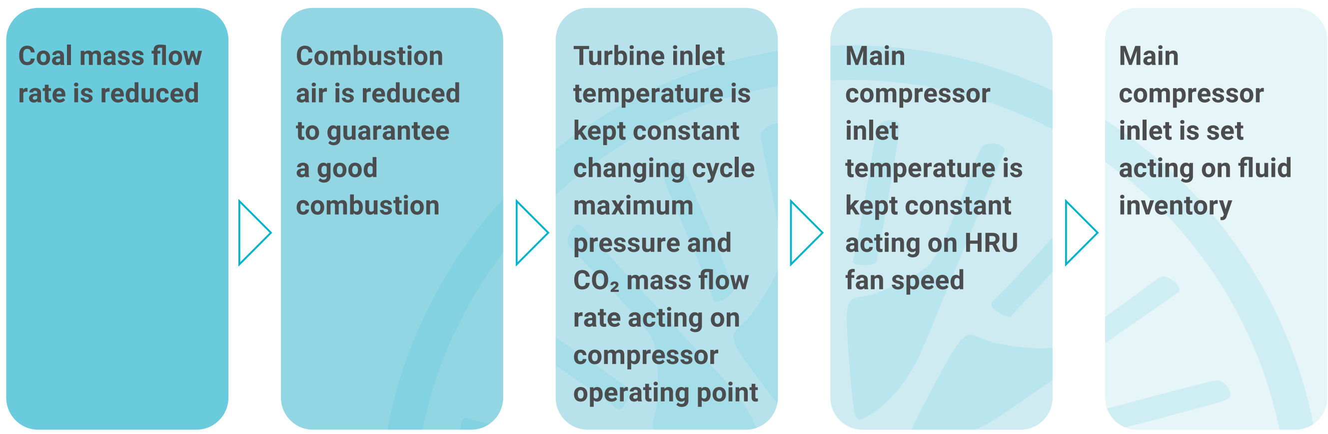

Figure 18: The set of operation performed on the boiler, turbomachinery and fluid inventory to reduce the plant load

How to select the best compressor inlet conditions?

The answer to this question is not trivial and different options have been compared with the simulation tools developed by Polimi.

CASE 0 – Clinging to the critical point: as the design point of the compressor is very close to the critical point, a good option may be to keep fixed and equal to the design value the compressor inlet pressure and temperature with the variation of the load in order to exploit the working fluid real gas effects. The performed calculations showed that this solution is not convenient from the plant efficiency point of view as the reduction of the maximum cycle pressure with the load (as the turbine operates in sliding pressure) at constant minimum pressure strongly reduces the turbine expansion ratio and thus its specific work (kJ/kg). From the compressor point of view, the reduction of the volumetric flow at the compressor inlet with the load would move the compressor operating point towards the surge conditions (see figure 19). The use of an anti-surge system or a reduction of the HRU fan speed may overcome the problem, but with further penalization on the cycle efficiency.

CASE 1 – Do what would be done with an ideal gas Joule-Brayton cycle: a reduction of the minimum cycle pressure proportional with the maximum cycle pressure is the solution that would be applied in an ideal Joule-Brayton cycle and it is feasible from the turbomachinery point of view also with CO_2, but with a certain efficiency decay. Is there a better solution?

CASE 2 – Maximizing the efficiency: a controlled reduction of the minimum cycle pressure can further improve the efficiency as the compressor operational point is selected with an eye on the overall system efficiency. The result is shown in figure 19, right against CASE 1.

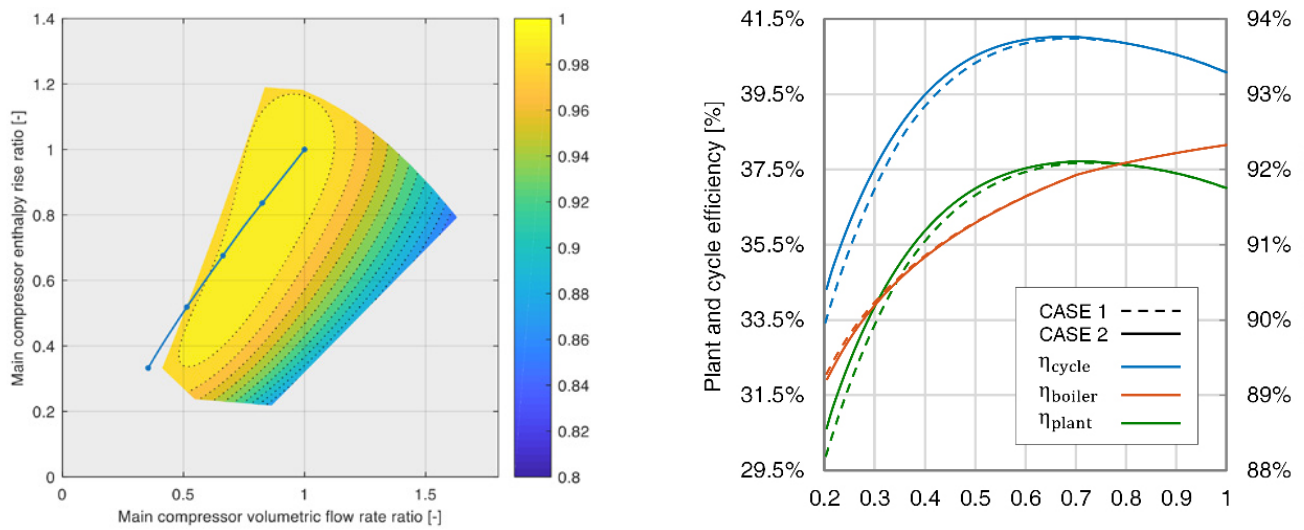

Figure 19: (left) Main compressor operational points with respect to their operative maps. Markers are representative of system power output 100%, 80%, 60%, 40% and 20%

The selection of the best minimum cycle pressure is obtained through CO_2 inventory variation and the adoption of an external tank in which part of the CO_2 can be stored leading to a flexible cycle that can operate down to 20% of the electric load with limited efficiency decay.

3.2. Control and transient behavior

Truly flexible operation of the plant in a context of increasing penetration of intermittent renewable sources requires the ability to undergo fast load ramping rates, stable operation at reduced electric load levels as low as 20%, and the ability to quickly provide extra power output for primary frequency control. The behaviour of the plant from this point of view was analysed by means of first-principles dynamic modelling and simulation, with a very interesting and positive outcome.

The first result is that it is possible to change the power output of the turbine quickly, without the need of expensive partial arc turbine admission or of inefficient turbine throttling, contrary to traditional supercritical steam power plants. This is due to the inherent structure of the sCO_2 cycle, whereby the CO_2 being heated in the recuperators and boiler is a relatively lightweight fluid behaving almost as an ideal gas, contrary to the case of supercritical steam power plants (and even more of drum-boiler based plants), where the fluid has a very high liquid-like density in a significant part of such circuit. Consequently, the mass and energy storage in that part of the circuit, which determines the inertia of the dynamic response of boiler pressure, is much lower than in steam power plants. An increase of the compressors flow rate results in an increase of the boiler pressure, and consequently of the turbine power output, in a matter of a few seconds. All other critical process variables, in particular the turbine inlet temperature and the pressure in the low-pressure part of the circuit, react much more slowly, so they can safely be kept close to their set points by the control system even under such a fast transient without too much difficulty.

The second result is that the control problem turns out to be a relatively easy one; the dynamic analysis shows that it is possible to effectively control the plant with a fairly straightforward PID-based (Proportional-Integral-Derivative) control strategy, and to do so effectively all the way down to 20% power output. The plant-wide control system structure is shown in figure 20; PI (Proportional-Integral) controllers in green require gain scheduling as a function of the load levels, and pre-computed feed-forward action based on the load set point must also be provided.

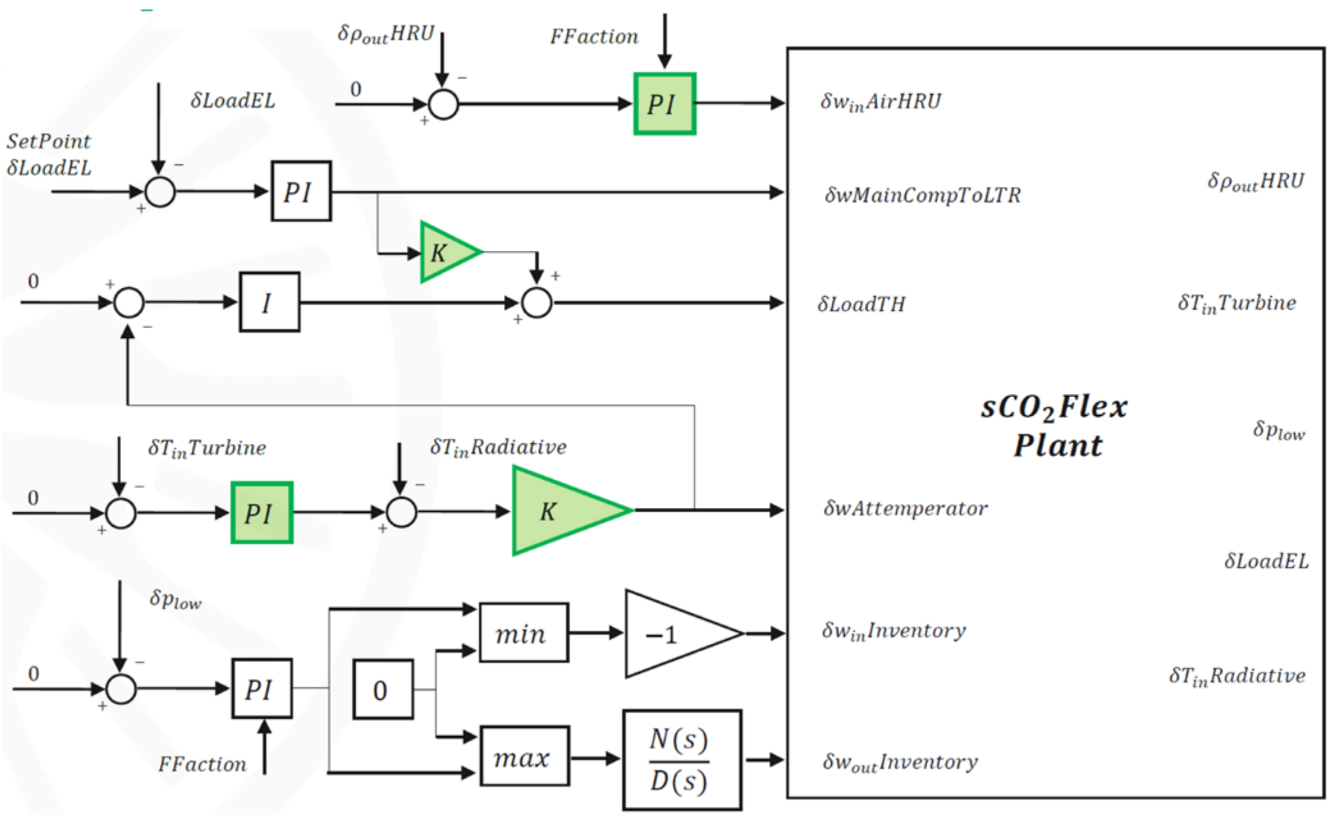

Figure 20: Plant-wide control architecture

Of course, this result does not preclude the possibility of using more sophisticated control algorithms, such as, e.g., model-predictive control; however, the fact that the plant can be controlled effectively with such a simple strategy makes the commissioning and early operation of the first plant prototype less critical, and also provides a fall-back option for more sophisticated solutions.

Figure 21 shows the result of a 5%/min load ramping rate transient, obtained with this control system in place: the set point is followed perfectly, and the two main control variables, namely the main compressor flow rate and the coal flow rate, undergo smooth changes with no overshoots. Similar results are obtained with ramp load increase transients.

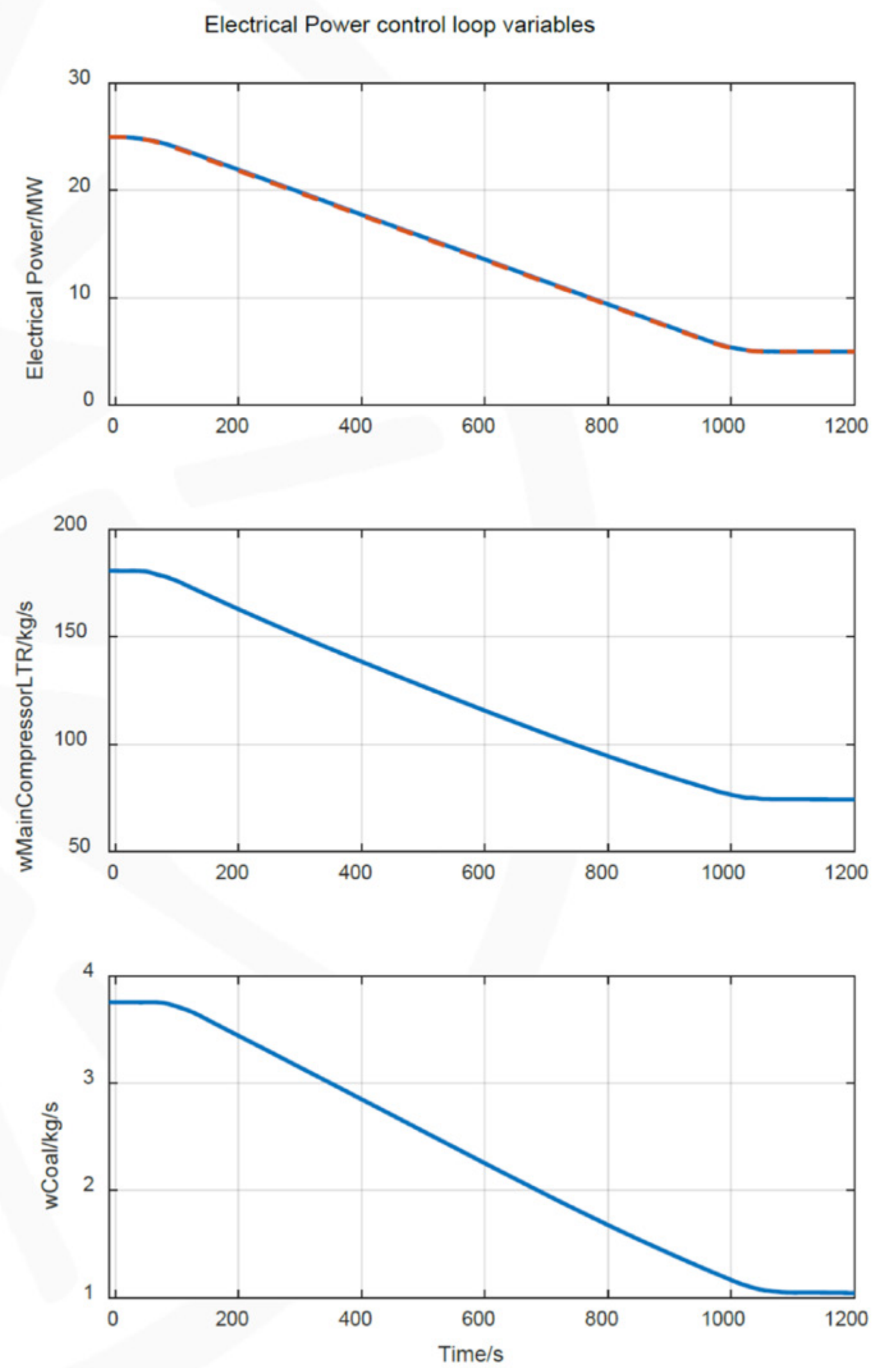

Figure 21: 5%/min load reduction transient: Power set point and output, main compressor flow rate, coal flow rate

Figure 22 shows the results of a +5% step increase of the load set point, such as could be caused by the primary frequency control in case of loss of a large generator on a grid, which causes a fast drop of the grid frequency and thus a small, but fast, increase of the power demand due to the droop.

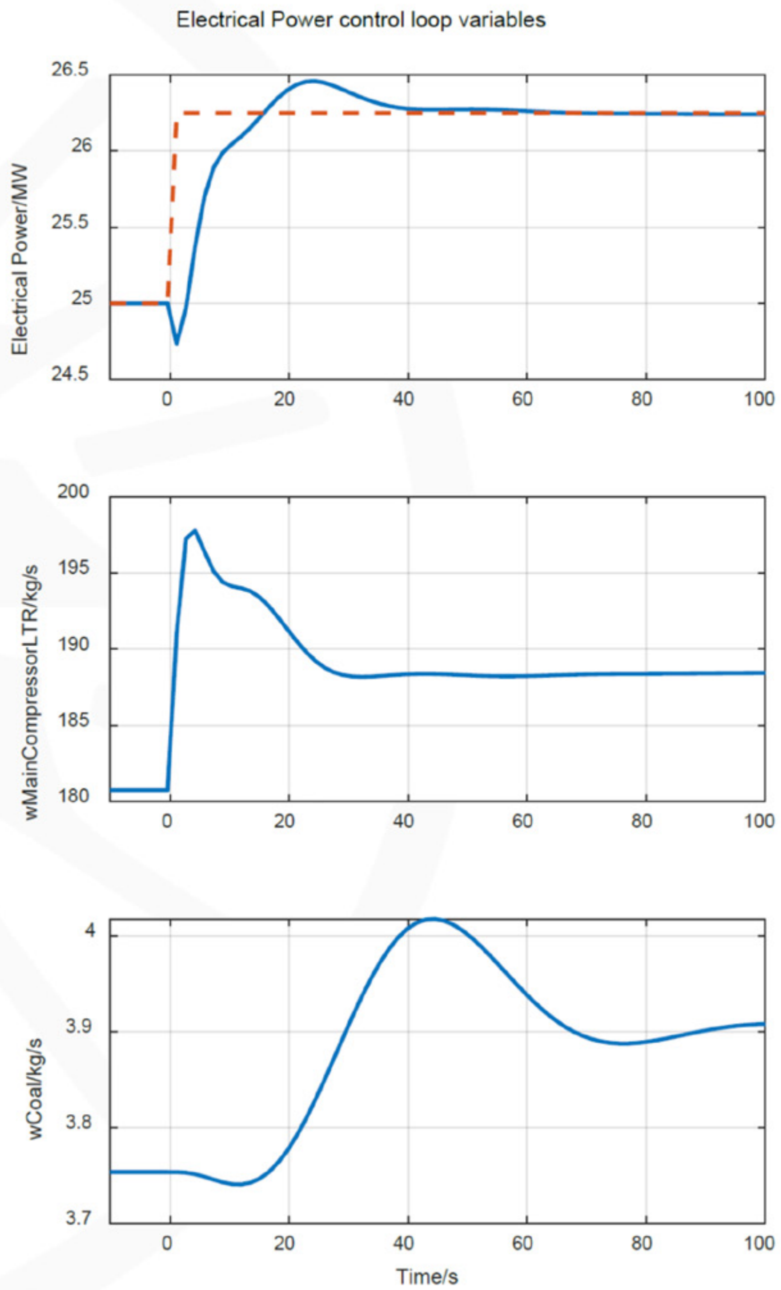

Figure 22: 5%/step increase of the load: Power set point and output, main compressor flow rate, coal flow rate

European grid codes typically require power plants to provide extra power output for primary control within 15 seconds from request that comes from the grid frequency drop; as it can be seen from the first plot in figure 7, 80% of the required power is already available after less than 10 seconds from the set point change, and the power output is completely stabilized in less than one minute. Such a fast transient understandably requires some overshoot of the two main controlled variables, namely the main compressor flow rate and the coal flow rate, as shown in the second and third plots of figure 22. However, these overshoots have a reasonable amplitude, about 5% of the design compressor mass flow rate and about 2.5% of the coal mass flow rate, and can thus be managed effortlessly. It is remarkable how this result can be obtained without the use of any kind of turbine control valves.

4. Potential impact

4.1. Environmental and social impact

The environmental impact of any object must be estimated considering its full lifecycle. As a result, the impact of a coal plant usually takes into account the construction and commissioning phase, the operation phase and the deconstruction phase.

Several indicators can be considered. The best-known and the most consensual is the Global Warming Potential (GWP), defined by the IPCC. It aggregates all the greenhouse gas emissions throughout the plant's lifecycle into one single figure. A more innovative one is the Abiotic Resource Depletion (ADP), which measures the impact of the plant regarding its use of non-renewable materials (fuel excluded). These two indicators are considered the most relevant ones as the project's main objective is two decrease the GWP of the plant, and as the plant is built with more noble materials than usual fossil-fuelled plants, which is expected to have a negative impact on the ADP indicator.

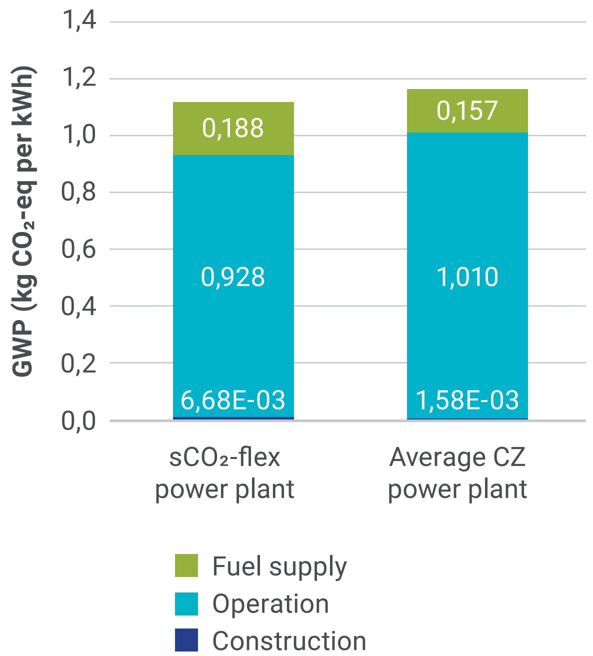

Figure 23: Global Warming Potential impact of the sCO_2-Flex plant compared to a reference water/steam plant

As the sCO_2-Flex plant's efficiency is better than the average European plants', and as most greenhouse gas emissions occur in operation, the sCO_2-Flex plant offers a lower environmental impact regarding this indicator: with its 1.123 kg CO_2 equivalent per kWh, it produces 8% less than the reference plant in operation, and still 4% less overall during its entire life cycle. The higher emissions of the fuel supply phase are due to the fuel that was selected for boiler design and are not specific to the cycle.

To the contrary, sCO_2-Flex plant's ADP is expected to be worse than average, because of the use of noble materials for construction. However, this cannot be quantified yet, as the detailed design is not yet finalized.

Concerning social impact, the improvement of the coal-fired power plants will allow to maintain the jobs related to the coal industry in the short term and give time to train workers for the adaptation to new and green energy production forms. A well-defined energy transition may be key to avoid new sources of social vulnerability and poverty that may arise due to the sudden loss of jobs.

Also, the sCO_2-Flex partners have elaborated a social code of conduct for responsible research in the field of energy, which proposes guidelines to avoid any harmful impact of energy research activities and promote positive social impacts of this activities.

The code of conduct suggests specific actions to be implemented by energy researchers in order to ensure that societal needs and concerns are considered during the different research stages. Following a practical approach, the code of conduct proposed concrete activities for each TRL

4.2. Cost estimates and scale-up projections

For a very small size (25 MWe gross, 23.6 MWe net), the sCO_2-Flex cycle is significantly more efficient than an equivalent steam cycle (around 42,5% versus 37%), which overall induces lower variable production costs as well as a smaller environmental footprint. The detailed design performed in sCO_2-Flex allowed the accurate estimation of its CAPEX, around 98.5 M€ (soft costs included), that is 4200 €/MWe, comparable to a reference water/steam plant of the same power output and location. The breakdown is summarized in figure 24. The boiler house accounts for around a third of the total costs. It is worth mentioning that its cost is itself very sensitive to the price of nickel-based alloys and may vary depending on their market price.

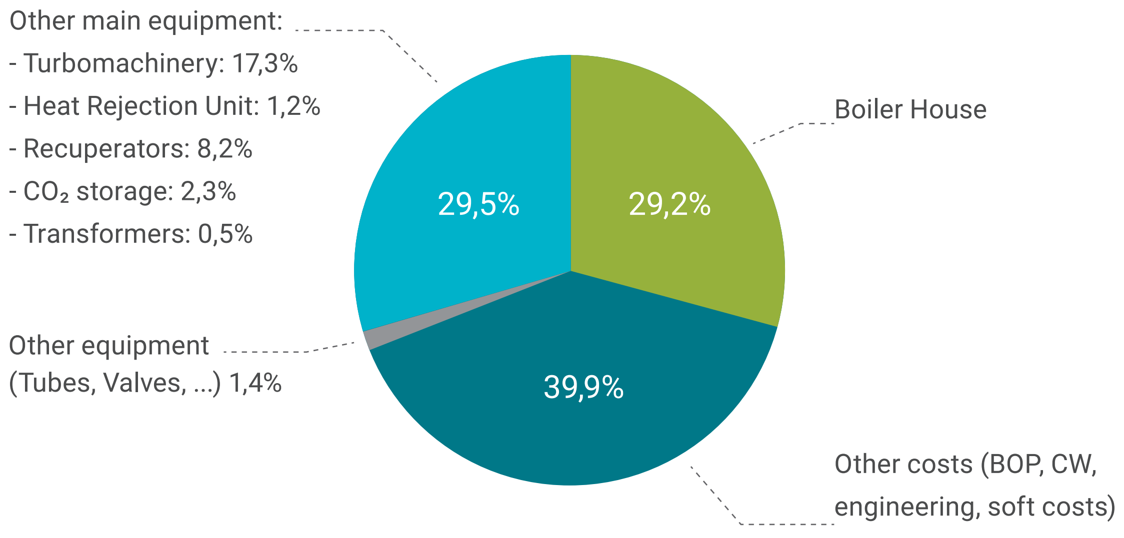

Figure 24: CAPEX breakdown of the 25 MWe plant developed in sCO2-Flex

The projections made on a 100 MWe plant reveal roughly the same cost structure (see figure 25) with a still higher sensitivity to boiler costs, for a total CAPEX around 2.3 times higher. The specific CAPEX (in €/kW installed) is thus divided by 1.7 when increasing the size of the plant. Such considerable scale effects are expected to be highly beneficial to sCO_2 technologies' development.

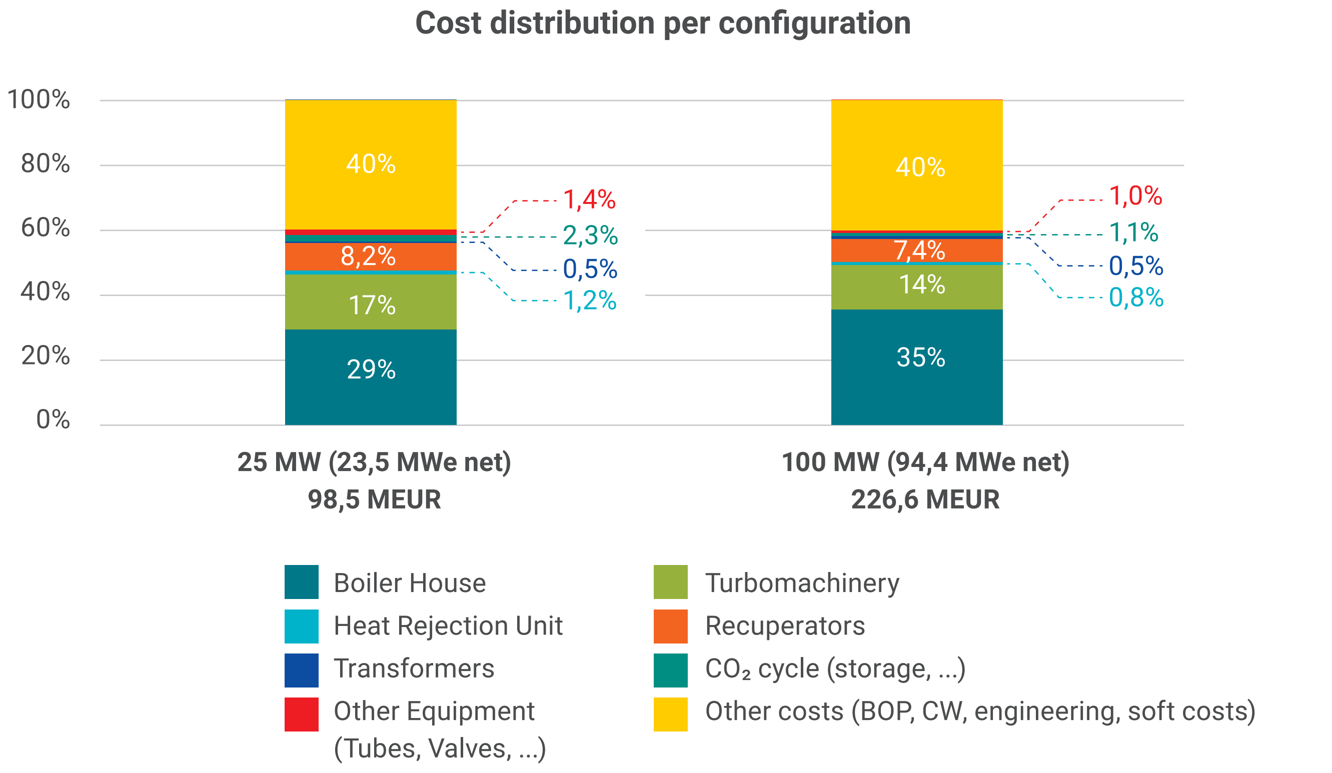

Figure 25: Comparison between the cost structure of the 25 MWe plant developed in sCO_2-Flex and a 100 MWe projection

5. Conclusion

The supercritical CO_2 cycle developed in sCO_2-Flex shows promising efficiency, flexibility and environmental footprint. The expected gain in component compactness has been widely confirmed by the project's developments, leading to the competitiveness of the selected sCO_2 cycle versus its water/steam counterpart in terms of CAPEX, with a significantly better efficiency and flexibility. sCO_2-Flex has successfully shown that sCO_2 cycles are a cost-effective, flexible and performant solution for small-size coal plants, with potential applications to concentrated solar power, waste heat recovery and biomass, as well as a promising scalability up to 100 MWe.

The next step towards the commercial availability of a full-scale sCO_2 cycle would be the construction and testing of a medium-scale demonstrator (> 10 MWe), in order to confirm the actual behavior of every component in the relevant environment and be able to test new equipment and layouts.

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