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SCUOLA DI INGEGNERIA INDUSTRIALE
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EXECUTIVE SUMMARY OF THE THESIS

Preliminary design and part load analysis of supercritical carbon dioxide power cycles for gas turbine bottoming applications

TESI MAGISTRALE IN ENERGY ENGINEERING – INGEGNERIA ENERGETICA

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1. Introduction

Renewables are increasingly growing in the energy mix, causing regulation and control problems for the grid due to their intermittent and unpredictable nature. Conventional fossil fuels power plants are required to provide the surplus of demand not covered by the former, therefore they must work at partial loads and ramp up and ramp down in a fast and efficient way. Nowadays, combined cycle power plants are usually based on a steam bottoming cycle. They are employed for base load operations with remarkable efficiencies. However, in the next future they will have to shift towards peaker operations, struggling to follow wide load variations due to the high thermal inertias due to water phase change. In this context, it is interesting to study an alternative bottoming cycle based on supercritical CO_2 [1].

sCO_2 is attractive for this type of applications because it involves layouts with compact turbomachinery and heat exchangers due to the high density compared to steam. Furthermore, it only operates in the cycle in single phase achieving lower temperature differences in the heat exchangers and reducing irreversibility [2].

Nevertheless, the main drawback of working close to the critical point is that small changes of temperature or pressure lead to sharp variation of working fluid thermodynamic and transport properties, requiring accurate and precise control strategies [3].

2. Case study and methodology

This thesis work is developed around the premises of the CO_2OLHEAT project, an EU-funded H2020 project to enhance waste heat recovery in the industrial and power generation sector by developing a sCO_2 power cycle demonstration plant in an operational environment [4].

One of the replication sites of the project is in the Paris region, where EDF is proposing to convert a power plant consisting of two gas turbines into a combined cycle with a sCO_2-based-bottoming configuration.

The objectives of this thesis are to study the possible sCO_2 configurations that can be applied as bottoming cycles to this system and to study their performance under nominal and off design operations.

First of all, using the commercial software Thermoflex [5], the gas turbine's exhaust conditions are derived. A numerically modelled

design criterium is implemented through NASA polynomials to vary the flue gases specific heat so that it is no longer constant but a function of temperature and composition of the exhaust.
After having performed a literature review about the most promising sCO_2 power cycle architectures for waste heat recovery, the initial objective is to study different possible configurations that can be implemented for the sCO_2 bottoming system.
Thereafter, a thermodynamic optimization is conducted with a numerical code developed in Matlab within the sCO_2 flex project by Politecnico di Milano [6]. All the layouts are optimized in nominal conditions, deriving characteristics quantities such as net output powers and efficiencies and evaluating the best performing configuration(s) from a thermodynamic point of view.
Additionally, a brief analysis is carried out looking at the investment costs for each configuration to understand the techno-economic potential of the investigated solution.
Finally, by varying the ambient conditions and the gas turbine load, it is investigated the off design behavior of the simple recuperative cycle, which has shown a good compromise between design efficiency, system complexity and specific cost. The analysis at partial loads is carried out considering two different cooling fluids, water and ambient air. Therefore two different resolution methodologies are adopted since in the air case the temperature of the cooling medium is strictly dependent on the ambient one. The results are shown in the form of maps of the cycle's optimal working point, the one with the highest net power output, for each ambient temperature and gas turbine load variation.

3. Nominal plant design methodology

The topping cycle gas turbine model is a General Electric GT13E2 with 210 MW of power output [4]. In nominal conditions the exhaust gases have a mass flow rate of 592.2 kg/s and a maximum temperature of 521 °C, while the molar composition is shown in Table 3.1.
The total net electrical power output in ambient conditions is 198.22 MW, while the heat power inlet is 525 MW, considering a natural gas with LHV of 50 MJ/kg and a flow rate of 10.5 kg/s. Hence, the gas turbine's total efficiency is roughly 37.8%.

Table 3.1: Exhaust molar composition at the turbine's outlet

xO_2	xCO_2	xH_2O	xN_2	xAr
0.136	0.032	0.081	0.742	0.009

Regarding the bottoming system, seven sCO_2 power cycle configurations are studied and optimized: (i) the simple recuperative cycle (SRC), (ii) the recompressed recuperative cycle (RRC), (iii) the simple recuperative cycle with recuperator bypass (SRCB), (iv) the recompressed recuperative cycle with high temperature recuperator bypass (RRCB), (v) the turbine split flow cycle (TSF), (vi) the single heated cascade cycle (SHC) and (vii) the dual heated cascade cycle (DHC). The last two configurations are particularly novel architectures tailored for WHR application as resulted from the literature review done in the first part.
The optimization methodology is based on varying: (i) the CIP, (ii) the TIT and (iii) the temperature differences between streams in the mixing processes.
The maximum cycle pressure and the minimum cycle temperature are set constant and equal respectively to 250 bar and 33 °C, because the optimizer would push them respectively to the upper and to the lower bound of the simulation interval.
For all the recuperator's pinch point a 10 °C value is set, while the PHE's one is computed from the analysis considering a minimum of 25 °C. They should be optimized from a technical-economic point of view, but as a preliminary study it is decided to keep them constant, as the optimizer would push their value to the lower bound.
The optimization objective function is the cycle's total efficiency, defined as:

$$\eta_{tot} = \eta_{cycle}\eta_{rec} = \frac{\dot{W}_{net}}{\dot{Q}_{in,cycle}} \frac{\dot{Q}_{in,cycle}}{\dot{Q}_{max,hs}} = \frac{\dot{W}_{net}}{\dot{Q}_{max,hs}}$$

Where $\dot{Q}_{max,hs}$ is computed as:

$$\dot{Q}_{max,hs} = \dot{m}_{hs}\left(h_{hs,max} - h_{hs,min}\right)$$

$h_{hs,min}$ is a function of $T_{hs,min}$, the minimum temperature at which the flue gases can be discharged into the environment, set at 90 °C, in order to avoid any formation of acid condenses and fouling on heat transfer surfaces.

4. Nominal plant design results

Starting with the nominal power plant design the main results are here proposed and commented, where the combined cycle's efficiency is defined with the following equation:

$$\eta_{cc} = \frac{\dot{W}_{net} + \dot{W}_{net,gas\ turbine}}{\dot{Q}_{in,gas\ turbine}}$$

In Table 4.1 are presented the net powers and all the efficiencies for each layout studied.

	W_{net} [MW]	*η cycle* [%]	*η rec* [%]	*η tot* [%]	*η cc* [%]
SRC	55.4	27.1	72.8	19.7	48.3
RRC	49.4	28.6	61.5	17.6	47.2
SRCB	71.8	25.7	99.5	25.5	51.4
RRCB	53.9	27.9	68.6	19.2	48.0
TSF	71.0	30.6	82.6	25.3	51.3
SHC	70.1	25.3	98.6	24.9	51.1
BHC	77.7	27.9	98.9	27.6	52.6

Table 4.1: Net powers and efficiencies results for each cycle in nominal condition

It is evident how the dual heat cascade cycle is the best performing one, as it shows not only the highest power output but also a greatest recovery efficiency achieving a remarkable combined cycle's efficiency of 52.6%. However, its main issues are the large number of components and the complexity in operation due to the double heater. Furthermore, from the economic analysis reported in Table 4.2, it results the one with highest investment cost.

	Investment cost [M€]	Specific cost [M€/MW]
SRC	40.5	0.73
RRC	54.6	1.11
SRCB	67.0	0.94
RRCB	64.3	1.19
TSF	68.5	0.96
SHC	67.6	0.96
BHC	83.2	1.07

Table 4.2: Total investment and specific costs for each cycle in nominal condition

The other new configuration implemented has quite comparable performance and cost to the TSF. The number of components is the same, but the heat exchangers are rearranged differently, leading the SHC to have lower cycle efficiency but better heat recovery from the exhaust gases.
It can be concluded from the on design analysis, that considering both thermodynamic and economic aspects, the configurations offering the greatest potential and recommended for future developments are the turbine split flow and the single heated cascade cycles.

However, looking at the costs, it cannot be recognized how the simple recuperative cycle lowers and in some cases even halves the investment cost compared to the others. This last configuration is selected for the part-load analysis as it represents the optimal solution for the studied application: in fact, a peaker plant will have to work a limited amount of time per year, not justifying the increased investment cost of more complex configurations. Secondly, the bottoming cycle will have to follow the load curve in a fast and prompt way, thus a more compact cycle configuration make sense for this kind of application.
These reasons along with the simplicity of the layout (Figure 4.1) and its operations motivated the choice of SRC for the off-design assessment.

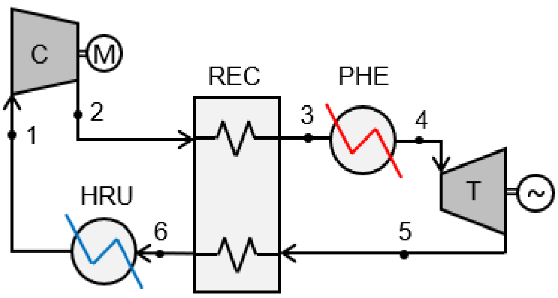


Figure 4.1: SRC layout

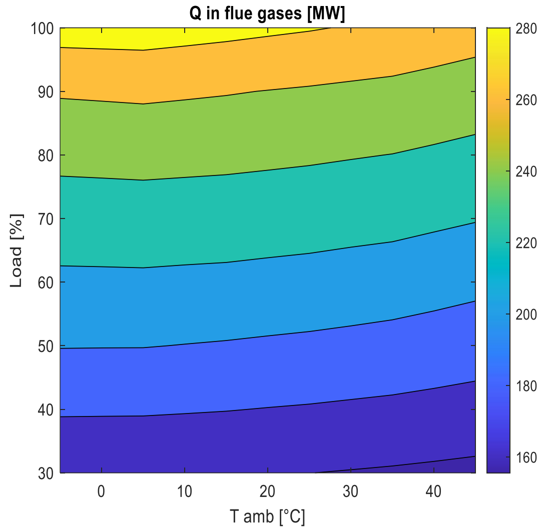
5. Part load analysis methodology

Temperatures analysis over a reference year at the replication site showed a range from -5 to 45 °C, while the gas turbine selected is high flexible and can operate up to partial loads of 30% [4].
Knowing these two ranges, it is possible to derive the heat power output from the topping cycle, which is also the heat source of the bottoming one (Figure 5.1).
Additionally, regarding cycle's components in off design conditions, all the heat exchangers are modelled adopting the same heat transfer coefficient and pressure drops of the nominal analysis. Compressors are controlled adopting IGV only at low or high volumetric flow rates, while turbines are not directly controlled as they follow the sliding pressure operations. HRUs are dependent on the choice of the cooling fluid. In the water scenario, it is made the assumption of water widely available at a temperature of 20 °C. In the

other case, air conditions are linked to the off-design ambient temperature analyzed.

Figure 5.1: Flue gases heat power inlet variation as a function of the gas turbine load and the ambient temperature

Having determined the heat source and the cycle components' behavior at partial loads, it is then possible to study the SRC cycle, considering the



two possible cooling fluid alternatives, water and air.

In the first scenario, two variables are analyzed: the CIP, in an interval from the design minimum pressure (i.e. 78.91 bar) to 100 bar with a number of 25 pressure values equidistant, and the TIT, in a range from -20 to 20 °C with a step of 2 °C from the optimal temperature of the precedent load iteration. CIT is not investigated because, even trying to optimize it, the variable remains fixed at the nominal condition in order to maximize the fluid density.

In the second scenario, three variables are considered: the CIP, that is kept with the same interval of the previous case, the TIT, in a range from -40 to 20 °C with a step of 1 °C from the design temperature of 375.66 °C, and the CIT, varied from nominal compressor inlet temperature of 33 °C to 40 °C with a step of 0.5 °C.

In order to control the variables just listed, two possible operating strategies are considered. The first one allows the CO_2 inventory variation within the cycle and optimizes all the selected variables. The second one fixes the CO_2 inventory, so TIT and CIT are optimized (only in the air case), but CIP can no longer be optimized but it is function of the CO_2 density variation.

6. Part load analysis results

Starting with the water-cooled case, it is interesting to show the variation in cycle's net electrical power as a function of the predetermined variables as depicted in Figure 6.1 where the variables are the TIT and the gas turbine load, while the ambient temperature and the CIP are kept constant to their nominal values. The optimal power output shifts towards negative ΔTIT, where $\Delta TIT = TIT_{studied} - TIT_{design}$, therefore TIT decreases compared to the nominal case. In the off design optimization the PHE pinch point is variable and no longer constant as in the previous analysis where it is fixed at 25 °C.

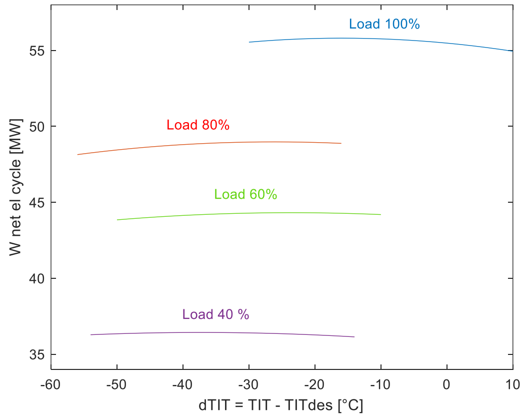


Figure 6.1: Trend of the net electrical power of the cycle against the variation of TIT and of the ambient temperature

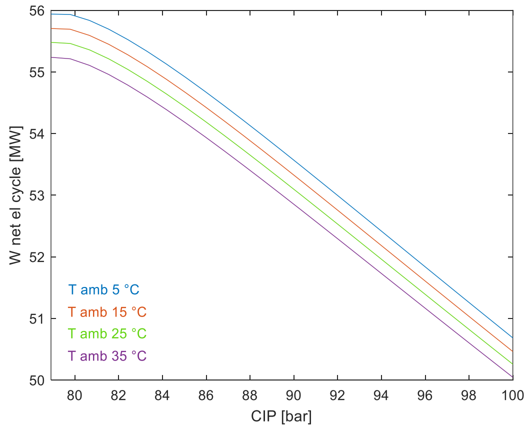


Figure 6.2: Trend of the net electrical power of the cycle against the variation of CIP and of the ambient temperature

Hence, the optimal value from the figure has a pinch point lower than 25 °C, leading to a higher recovery efficiency and a lower cycle efficiency due to the lower TIT. As a result the total cycle efficiency is slightly increased when TIT is about 15 °C lower than the nominal case.

Instead, considering the CIP and the ambient temperature change, keeping the TIT and the gas turbine load fixed. From Figure 6.2, it can be seen how the optimal CIP is in the range of nominal conditions. Lowering it was not tried to avoid entering the two-phase fluid area.
Moving on to the air-cooled scenario, the considerations made above about CIP and TIT still apply. Nevertheless, the additional variable studied here is the CIT. The HRU inlet temperature of the cooling air is closely dependent on the ambient one. Therefore, by increasing the latter, many operational points will become impossible as depicted in Figure 6.3.

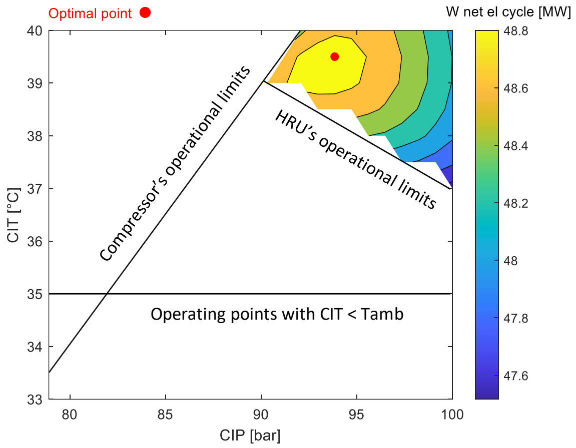


Figure 6.3: Trend of the net electrical power of the cycle at 35 °C of ambient temperature and 100% of gas turbine load against the variation of CIP and CIT with the optimal point and three exclusion areas

The unfeasibility are due to three main reasons:

- At high CITs and low CIPs some thermodynamic points result outside the compressor's operating map. Hence, the turbomachine can't work with them.
- When the ambient temperature is increased until 35 °C, there are some points with the CIT below this value, leading to a flows' intersection in the HRU.
- The maximum fans rotational speed is set at 125% of the rated value to avoid excessive mechanical stress. However, there are thermodynamic points that need fan speeds above that threshold in order to cool down to the defined CIT. Therefore, these points are removed.

For both cases, it is also interesting to analyze the cycle inventory. If it is assumed the presence of a CO_2 storage, under off-design operations it is possible to pressurize, by increasing the CO_2 mass in the cycle, or depressurize by decreasing it, depending on the desired operating conditions. However, the application of variable inventory involves additional costs due primarily to the presence of the storage and then to the control and compression system that regulates the CO_2 within the plant.

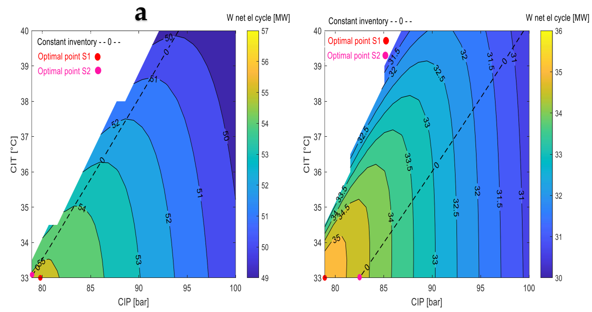


Figure 6.4: Trend of the net electrical power of the cycle against the variation of CIP and TIT with the optimal points for strategy S1 and S2, and the line at constant inventory with an ambient temperature of (a) 35 °C and (b) 5 °C

In the water-cooled scenario it seems that variable inventory may bring some benefit to the cycle's net power increase. It can be observed in Figure 6.4 how the points line at constant inventory is far from the optimum S1 strategy point, both considering an ambient temperature of 35 °C and one of 5 °C, with a difference that can reach more than 1 MW. Hence, it may be useful to provide a CO_2 storage.

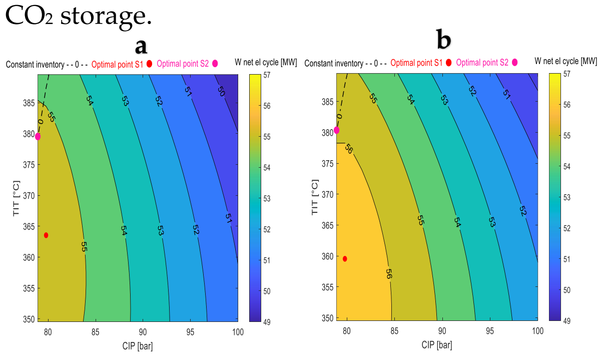


Figure 6.5: Trend of the net electrical power of the cycle at 25 °C of ambient temperature against the variation of CIP and CIT with the optimal point (red) and the line at constant inventory (black dashed) with a gas turbine load of (a) 100% and (4) 40%

In the air cooled case, it is obtained how the line of points at constant inventory moves further away from the optimum point both by decreasing the load (Figures 6.5) and by decreasing the ambient temperature. Therefore, even in this scenario, it

might be useful to include an inventory variation system.
However, under both conditions, it would be interesting to compute the specific cost of the power plant, defined as the ratio of total cost to net power, in order to compare the two solutions and evaluate if adding this option is really cost-effective.
Finally, the maps of the cycle's optimal working point as a function of gas turbine load and ambient temperature are here displayed comparing the two methodologies and considering.
In Figure 6.6 it is displayed the trend of the combined cycle efficiency. With both methodologies the maximum efficiency achieved is just under 50%. However, in the air scenario, it is evident the loss of performance when the ambient temperature increases from nominal conditions.

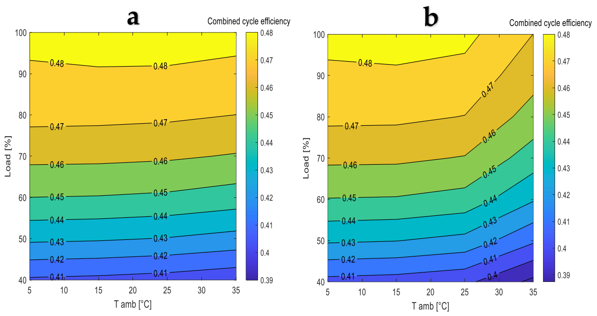


Figure 6.6: Trend of the combined cycle efficiency against the variation of gas turbine load and ambient temperature in the (a) water and (b) air cooled case

In the end a preliminary annual economic analysis is conducted considering three different power plants load distributions. The not discounted payback time is computed obtaining high values for the peaker type systems. However, conservative assumptions on the electricity price were assumed. Therefore, it is reasonable to expect a payback period lower than 10 years.

7. Conclusions

In this thesis project, two analyses were performed. The first one considered various layouts suitable for sCO_2 bottoming cycle applications and evaluated their performance in on design. It was obtained how the best performing configuration is the dual heated cascade cycle. Nevertheless, by taking into account also the investment costs, the choice shifts to less thermodynamically efficient configurations, but certainly simpler and cheaper such as the turbine split flow or the single heated cascade.
In the second one, the SRC layout was chosen for a partial load evaluation, since the configuration has shown modest thermodynamic performance, but reduced cost and simplicity that make it an excellent case study for the off-design analysis. It was found how it may make sense to lower the TIT for increasing the net power output, while the optimal CIP remained around the nominal point for all cases examined. Besides, in the air-cooled scenario, the ambient temperature has a great influence since many thermodynamic points become unfeasible above temperatures of 30 °C and the operating map has to move towards higher CIPs and CITs. A final consideration was brought regarding the possibility of varying the cycle inventory, discussing two possible operating strategies for the system and it was illustrated how useful it may be to include the option of variable inventory.
In the end, some maps of the cycle's optimal working point considering characteristic quantities were derived and a preliminary annual economic analysis was conducted computing payback periods high. However, assuming less conservative values, it is reasonable to expect values lower than 10 years.

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