OFF-DESIGN PERFORMANCE OF CSP PLANT BASED ON SUPERCRITICAL CO_2 CYCLES

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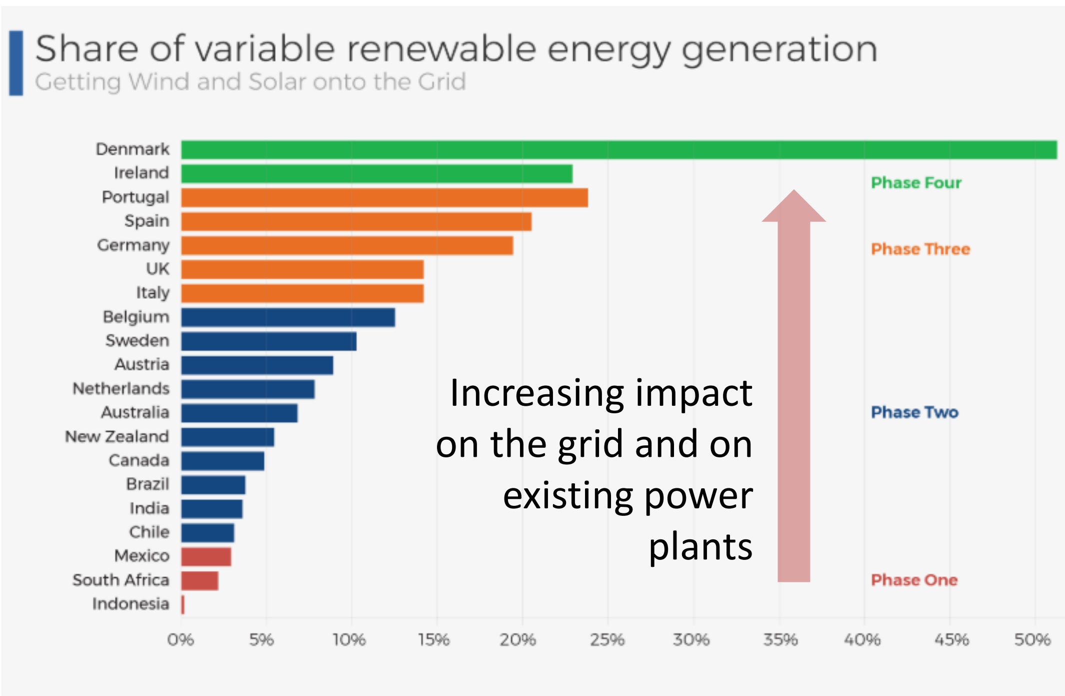
Outline

- Introduction
- The H2020 sCO2-Flex Project
- Scope of the work
- Cycle and Heat Exchangers Design
- Off-design Simulation: Methodology and Assumptions
- Part Load Analysis
- Variable Ambient Temperature Analysis
- Conclusions and Future Steps

Introduction – Future Energy Scenario

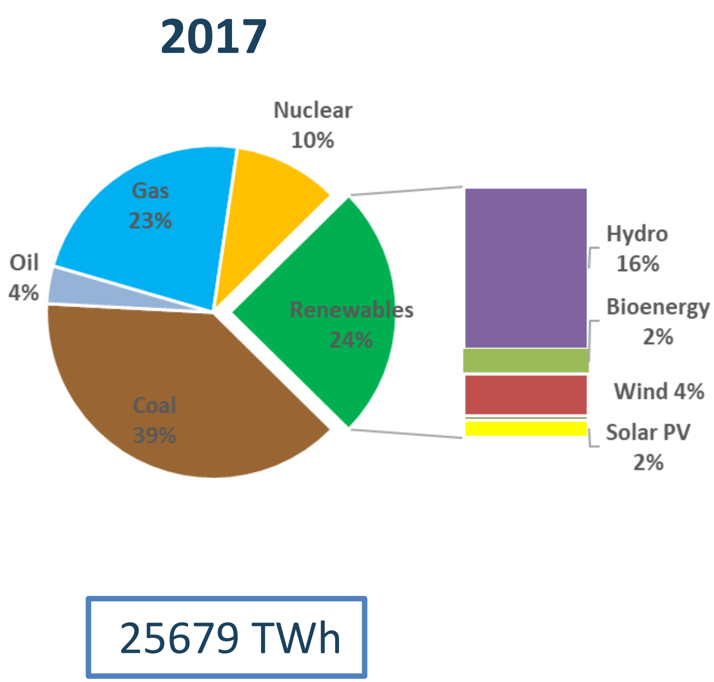
- The fraction of **Variable Renewable Energy** (VRE) in the EE generation has already exceeded 10% in 9 countries

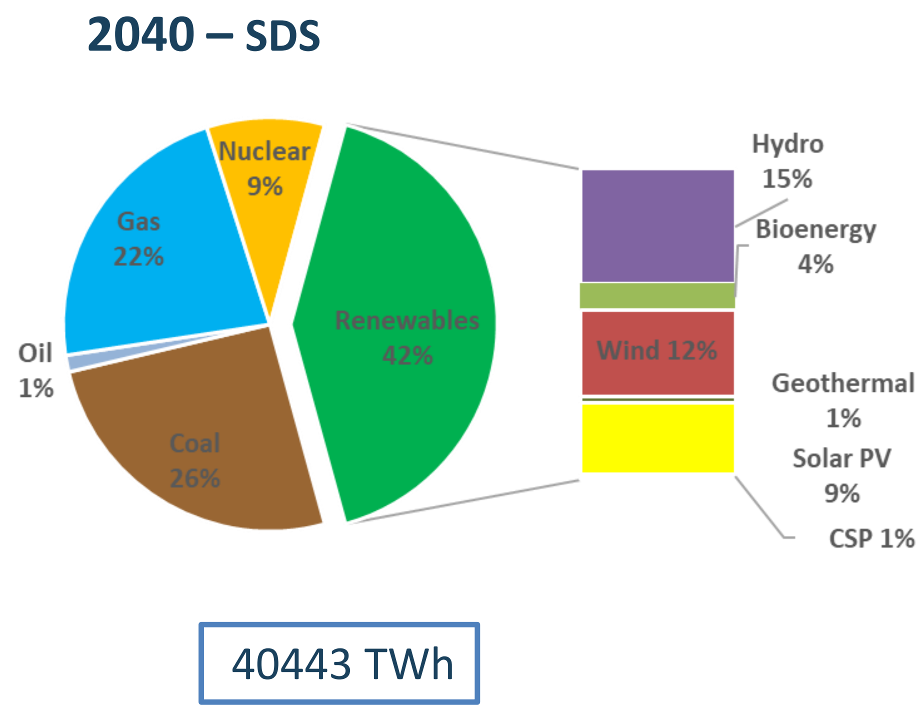


Source: https://www.iea.org/newsroom/energysnapshots/share-of-vre-generation.html

	Phase Three	Phase Four
Characterisation from a system perspective	Flexibility becomes relevant with greater swings in the supply/demand balance	Stability becomes relevant. VRE capacity covers nearly 100% of demand at certain times
Impacts on the existing generator fleet	Greater variability of net load. Major differences in operating patterns; reduction of power plants running continuously	No power plants are running around the clock; all plants adjust output to accommodate VRE
Impacts on the grid	Significant changes in power flow patterns across the grid, driven by weather condition at different locations; increased two-way flows between high and low voltage parts of the grid	Requirement for grid-wide reinforcement, and improved ability of the grid to recover from disturbances
Challenges depend mainly on	Availability of flexible resources	Strength of system to withstand disturbances

- The fraction of **Variable Renewable Energy** (VRE) in the EE generation has already exceeded 10% in 9 countries
- In 2040 according to IEA SDS **38%** of the electricity will come from VRE
- In 2050 according to the REmap scenario (IRENA) **58%** of the electricity will come from VRE





Source IEA, Key World Energy Outlook. 2018

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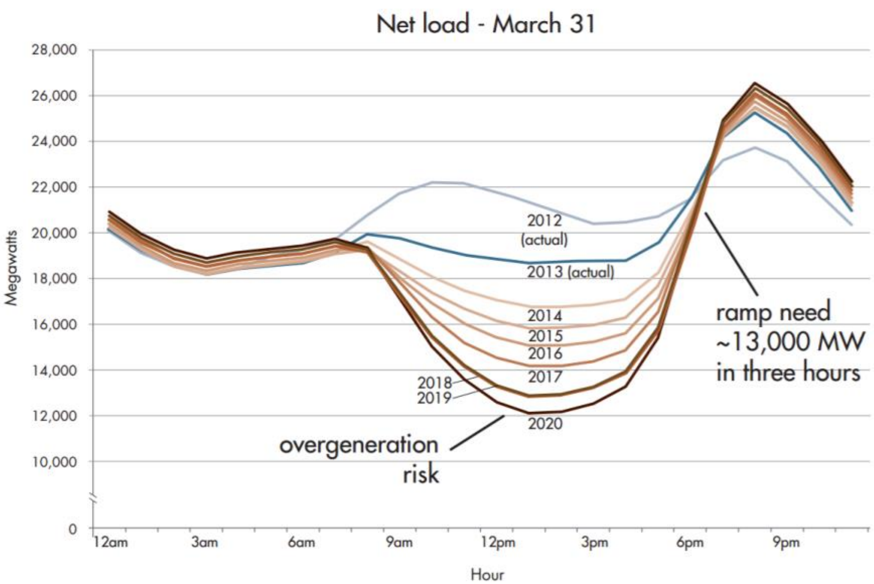
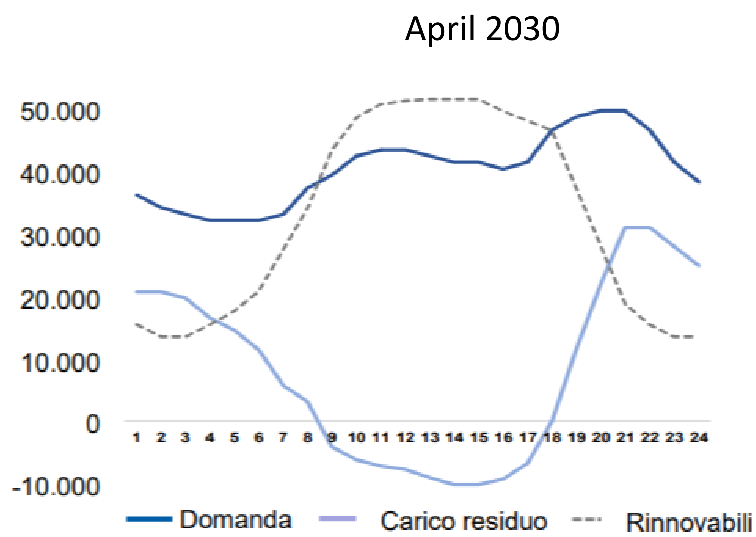
- In 2040 according to IEA SDS **38%** of the electricity will come from VRE
- In 2050 according to the REmap scenario (IRENA) **58%** of the electricity will come from VRE
- Conventional power plant (and CSP?) will have to provide flexible power in order to compensate VRE production



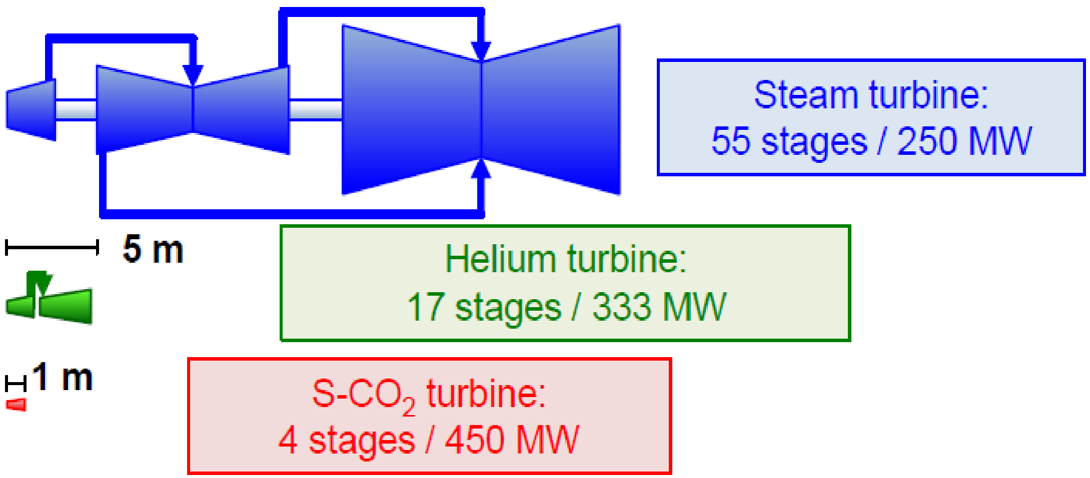


[Left] The *"duck curve"* by California Independent System Operator (CAISO) showing the potential for "overgeneration" occurring at increased penetration of solar photovoltaics (PV) in California.

[right] The *"Italian duck"* a forecast by Terna for an April 2030 day, considering the "overgeneration" due to PV and Wind

Compact Turbomachinery

Low compression/expansion ratio + high MM
→ limited number of stages



Note: Compressors are comparable in size

Adapted from Dostal (2004)

Introduction – Supercritical CO_2 Cycles Features

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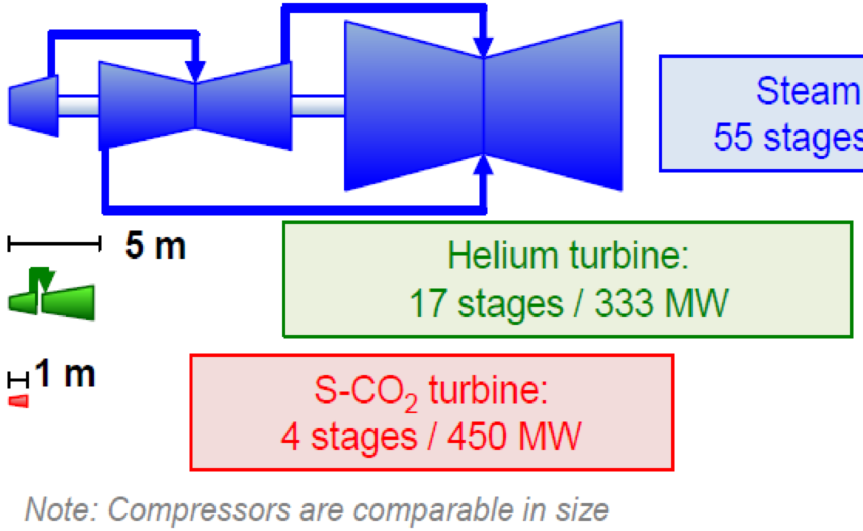
Compact Dry Cooling System

No water use → No water treatment
High pressure system → limited volumes
→ no air infiltration



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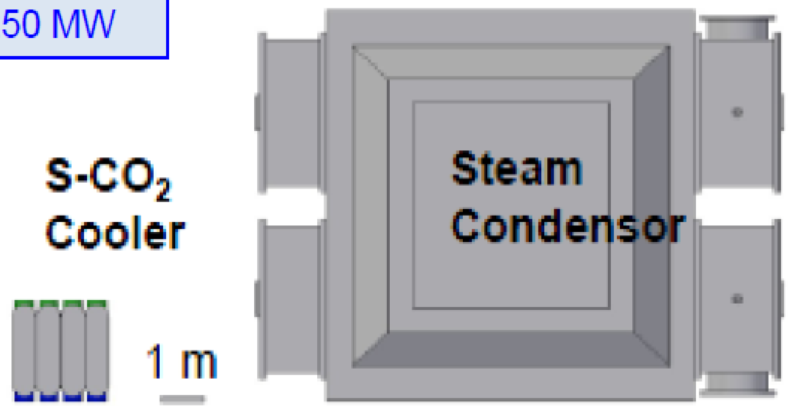

Source: Wright (2011)

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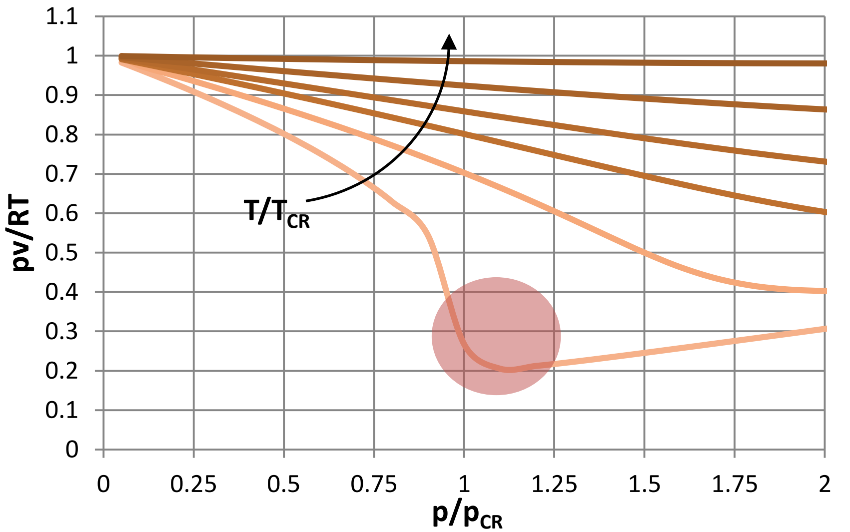
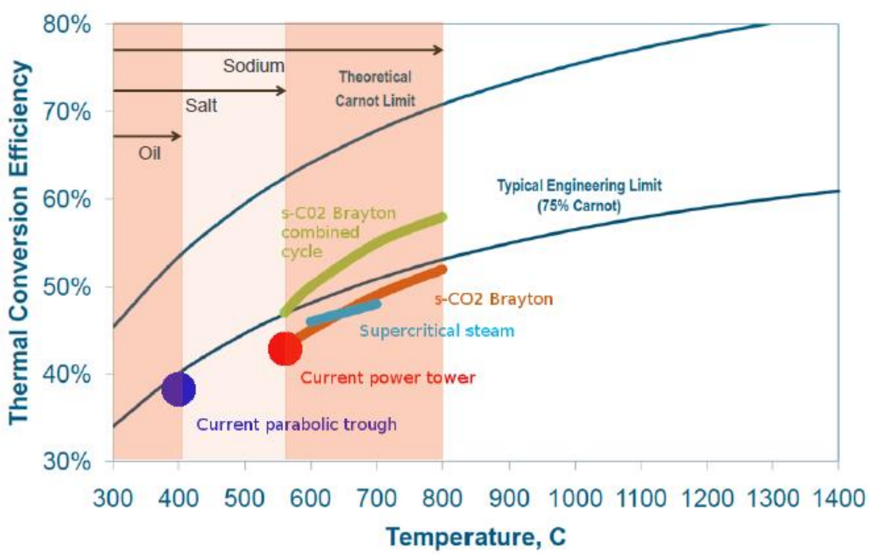
High Efficiency

Proximity of the compressor inlet to the CO_2 critical point → strong real gas effect
→ low compression work





J. Coventry et al. *Sodium receivers for solar power towers: a review*

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Inert & Inexpensive Gas
Cheaper and less harmful than other gases
Less material compatibility problems @high T

Can sCO_2 cycles be the solution for flexibility in the power sector?

(?) High Part Load Performance

(?) Fast Transient – Low Inertia

https://www.sco2-flex.eu/

sCO2- Flex aims at developing and validating the scalable/modular design of a 25MW$_e$ Brayton cycle using supercritical CO_2, able to increase the operational flexibility and the efficiency of existing and future coal/lignite power plants

Bringing the sCO_2 cycle to TRL6

- global cycle assessment at simulation level
- boiler, heat exchangers and turbomachinery tested at relevant environment

Paving the way

- to future demonstration projects (from 2020)
- to commercialization of the technology (from 2025).

THE SCO_2-FLEX PROJECT HAS RECEIVED FUNDING FROM THE EUROPEAN UNION'S HORIZON 2020 RESEARCH AND INNOVATION PROGRAMME UNDER GRANT AGREEMENT N° 764690

The H2020 sCO2-Flex Project – Consortium

1. **EFD:** Project Coordinator
2. **Zabala:** Dissemination & communication
3. **BHGE:** Turbine/compressor design & testing
4. **UJV REZ:** Pulverized coal boiler development
5. **CSM:** Corrosion & thermal test on materials
6. **CVR:** sCO_2 test loop validation
7. **Fives Cryo:** Recuperators development and manufacturing
8. **University of Stuttgart:** Heat exchanger test
9. **University of Duisburg-Essen:** Turbomachinery numerical validation

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10. POLIMI:
- Optimization of system design at nominal point
- Definition of optimal part-load operation strategy
- Evaluation of system dynamic
- **Cycle applications to other energy fields (CSP)**

Scope of the Work

Evaluate the off-design performance of sCO_2 cycles for CSP application considering:

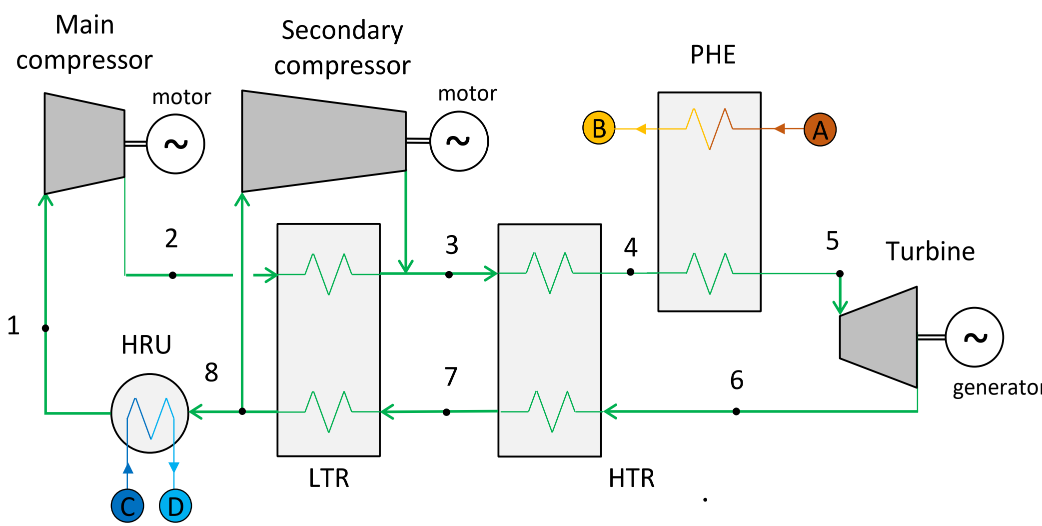
- **Part Load Operation**
  - Relevant for the future energy scenarios with high VRE contribution
- **Variable Ambient Temperature Operation**
  - Relevant for sCO_2 cycles using ambient air as heat sink

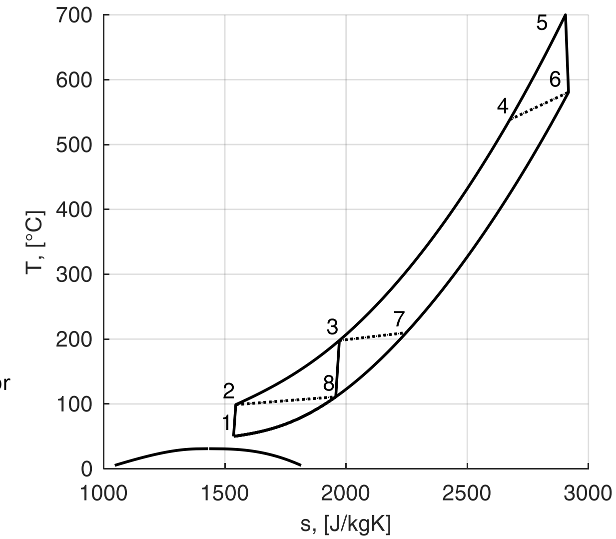
Obtain guidelines for turbomachinery design considering the system off-design operation

Cycle Design – Assumptions

A sCO_2 **recompression cycle** was selected as power conversion cycle
→ good compromise between performance and simplicity (1 Compr, 1 Turb).

Sodium at 715°C was considered as Heat Transfer Fluid providing heat to the cycle





Main Cycle Assumptions			
Parameter	**Value**	**Parameter**	**Value**
Cycle net electric power W_{net}, MW_{el}	**50**	HTR/LTR pinch point $\Delta T_{HTR}/\Delta T_{LTR}$,°C	12
Maximum cycle temperature T_5,°C	**700**	HRU CO_2 pressure drop $(\Delta p/p_{in})_{HRU}$	0.5%
Maximum cycle pressure p_2, bar	**250**	HTR/LTR hot side pressure drop $(\Delta p/p_{in})_{Hot,Rec}$	0.5%
Minimum cycle temperature T_1,°C	**50**	PHE CO_2 pressure drop Δp_{PHE}, bar	2.5
Turbine isentropic efficiency, η_{turb}	93%	HRU electric consumption ξ, MW_{el}/MW_{th}	0.0085
Main/secondary compressor efficiency, η_{comp}	89%	Design ambient temperature T_{amb}, °C	**40**

Cycle Design – Thermodynamic Results

- A numerical code that solves mass and energy balances has been developed in MATLAB + Refprop
- Optimization variables: $p_{\min}$ and $p_{\max}$
- Objective function: η_{cycle}

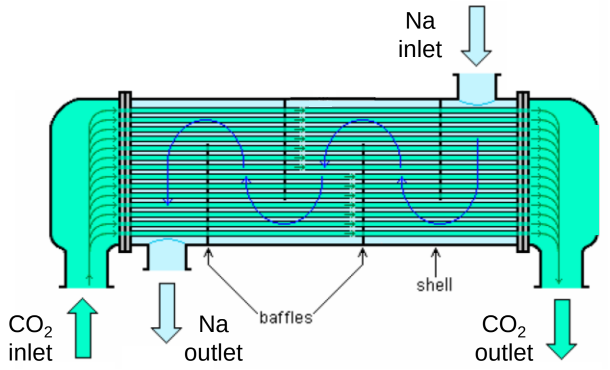
Main Cycle Design Results			
CO_2 mass flow at turbine inlet, kg/s	531.9	Main compressor electric power, MW_{el}	12.1
CO_2 mass flow at HRU, kg/s	375.1	Secondary compressor electric power, MW_{el}	10.2
Minimum cycle pressure p_1, bar	**104.1**	Heat rejection auxiliaries consumption, MW_{el}	0.47
HTR cold side $(\Delta p/p_{in})_{Cold,HTR}$	0.08%	Heat recuperated in the HTR $\dot{Q}_{HTR}$, MW_{th}	232.7
LTR cold side $(\Delta p/p_{in})_{Cold,LTR}$	0.04%	Heat recuperated in the LTR $\dot{Q}_{LTR}$, MW_{th}	66.7
Net heat input $\dot{Q}_{in,cycle}$, MW_{th}	108.7	Heat rejected to environment $\dot{Q}_{HRU}$, MW_{th}	54.8
Turbine electric power, MW_{el}	72.8	Net Cycle efficiency	**46 %**

Heat Exchangers Design – Assumptions

Primary Heat Exchanger

- Shell and tube heat exchanger
- Seban-Shimazaki correlation for h_{Na}
- Gnielinski correlation for h_{CO2}
- Material: INCONEL 617
- CO_2 pressure drop: 2.5 bar

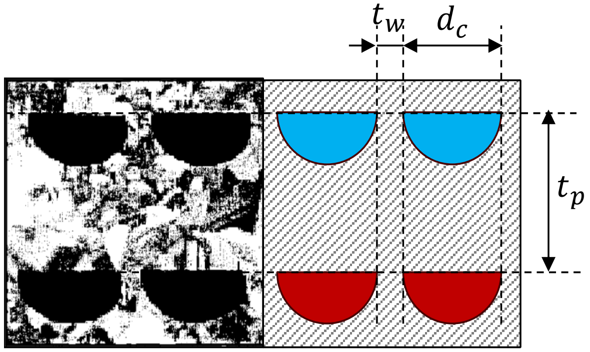
Parameter	Value
Tubes d_{int}, mm	16
Tube pitch, t_p	$1.25d_{ext}$
Na max velocity v_{Na}, m/s	0.25



Recuperators

- Printed circuit heat exchanger
- Model based on Dostal PhD thesis
- Gnielinski correlation for h_{int}
- Material: INCOLOY 800
- CO_2 pressure drop: 0.5%

Parameter	Value
Plate thickness t_p, mm	1.5
Channel diameter d_c, mm	2
Channel spacing t_w, mm	0.4



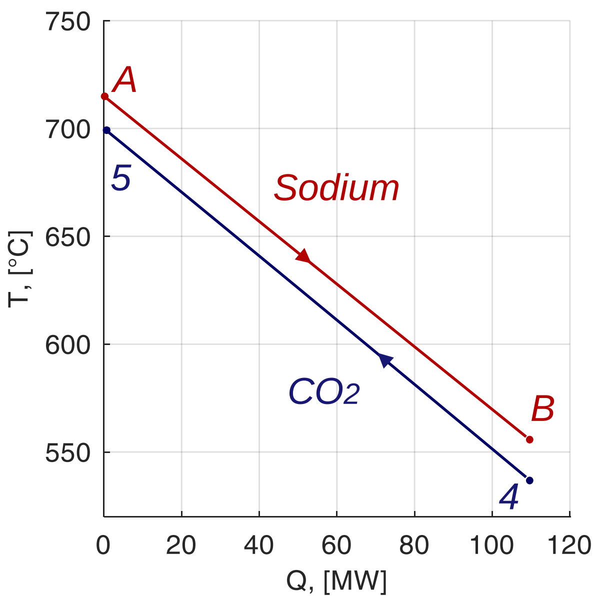
Heat Rejection Unit

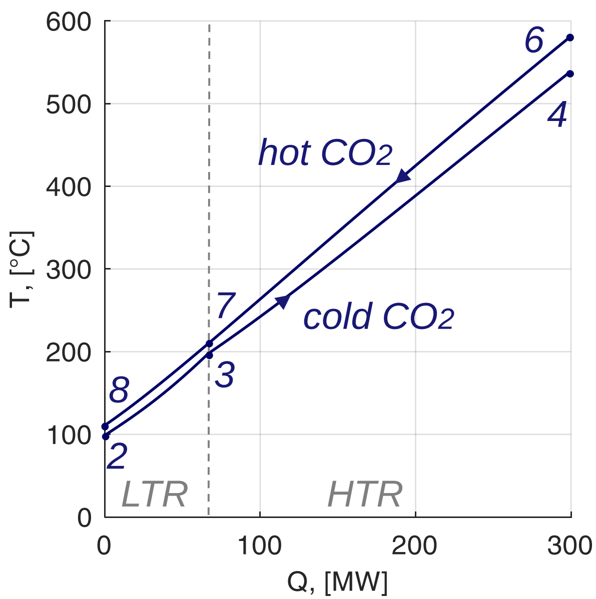
- Dry cooler tuned on manufacturer data
- Modified Gnielinski correlation for h_{int}
- Material: copper + aluminum
- Fans with variable speed EM
- CO_2 pressure drop: 0.5%

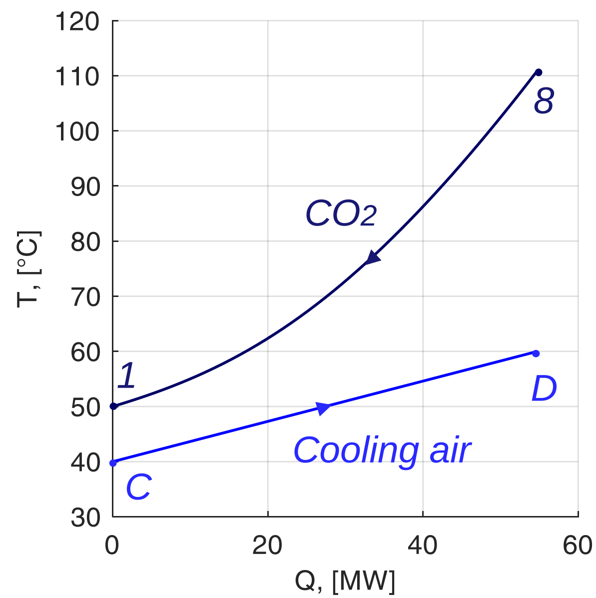
Parameter	Value
ΔT_{air}, °C	16
Tube d_{int}, mm	7.5
Fin thickness, mm	0.12
Fin spacing, mm	2.1
Fin height, mm	9.1

Heat Exchangers Design – Results

Parameter	HRU	LTR	HTR	PHE
Heat duty, MW	54.8	66.7	232.7	108.7
Hot side h.t.c., W/m^2K	3821.9	2163.3	1637.4	14291.5
Cold side h.t.c., W/m^2K	75.8	2143.7	1741.9	3392.5
Global h.t.c., W/m^2K	**1079.2**	**1033.1**	**820.0**	**2078.8**
Internal surface, m^2	2713.2	4666.5	9796.7	3160.1
HX metal mass, kg	21406.8	29261.1	61429.6	140873.0
CO_2 inventory, kg	**2353.2**	**390.6**	**427.6**	**5400.1**







HX sizing results (heat transfer areas) are used as input for the off-design study

Off-design Simulation: Methodology and Assumptions

- A MATLAB + Refprop code to simulate off-design operation of the selected sCO_2 cycle has been developed

- The codes requires as input:
  - Sodium mass flow rate
  - Sodium temperature
  - Ambient conditions
  - Components' constrains (A_{HX},…) from the design simulation
  - Information about the selected control strategy

- The code solves a system of non-linear equations and obtaines off-design thermodynamic operating points and performances

Off-design Simulation: Methodology and Assumptions

The following hypotheses/constraints have been also assumed:

- Δp and h.t.c. varied according to exponential laws for all HXs:

$$\Delta p = \Delta p_{design} \left(\frac{\rho_{design}}{\rho}\right)\left(\frac{\dot{m}}{\dot{m}_{design}}\right)^2 \qquad h_X = h_{X,design}\left(\frac{\dot{m}_X}{\dot{m}_{X,design}}\right)^{\alpha}$$

- HRU fan consumption computed through exponential correlation:

$$\dot{W}_{HRU,aux} = \dot{W}_{HRU,aux,design}\left(\frac{\dot{m}_{air}}{\dot{m}_{air,design}}\right)^{2.78}$$

- Isothermal mixing at the HTR inlet
- Turbine in chocked flow conditions and sliding pressure (no VIGV) and with constant $T_{in,turb}$:

$$\frac{\dot{m}_{in,turb}}{p_{in,turb}} = const$$

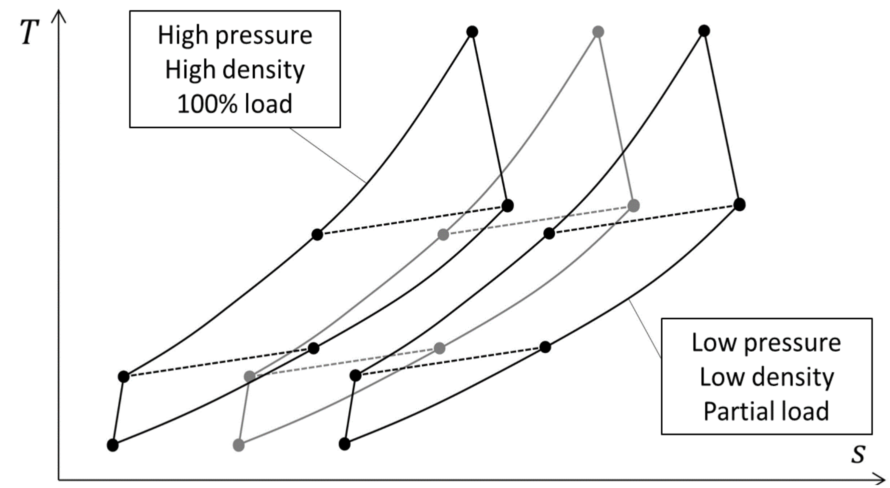
- **Constant turbomachinery efficiency at part-load**
- **No specific constraints for the compressors**: the obtained operating points in off-design are meant to provide useful preliminary design criteria

Part Load Analysis – Control Strategy of Closed Gas Cycles

Closed Ideal gas cycle control

It can be obtained by cycle de-pressurization removing a fraction of the working fluid mass with constant compression ratio (β) and thus with constant efficiency.

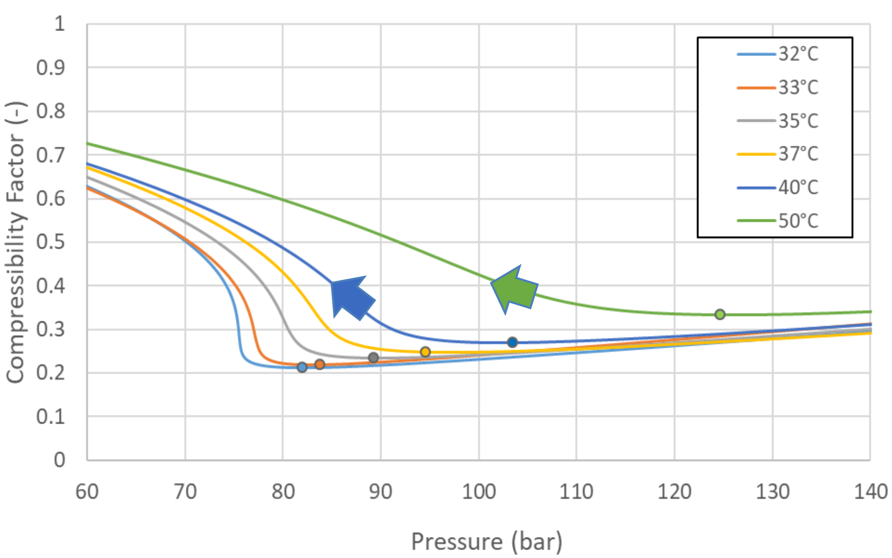
$$P = \dot{m}(w_t - w_c); \qquad \eta = 1 - \beta^{-\theta}$$



sCO_2 cycles control

Equal reduction of maximum and minimum cycle pressure (constant β) may not be convenient for sCO_2 cycles due to the compressor proximity to critical point.

A reduction of the minimum pressure may imply strong increase of the compressor work and thus $\eta \downarrow$



Part Load Analysis – Operating Strategies

As the Sodium mass flow rate decreases the maximum cycle pressure decreases (sliding pressure). Three operating strategies have been investigated for the minimum cycle pressure:

- CASE 1: minimum cycle pressure kept equal to the design value

- CASE 2 : minimum cycle pressure optimized to obtain maximum cycle efficiency

- CASE 3: minimum cycle pressure varied in order to guarantee a constant ΔT_{Na} across the PHX

Further assumption:

- No ambient temperature variation

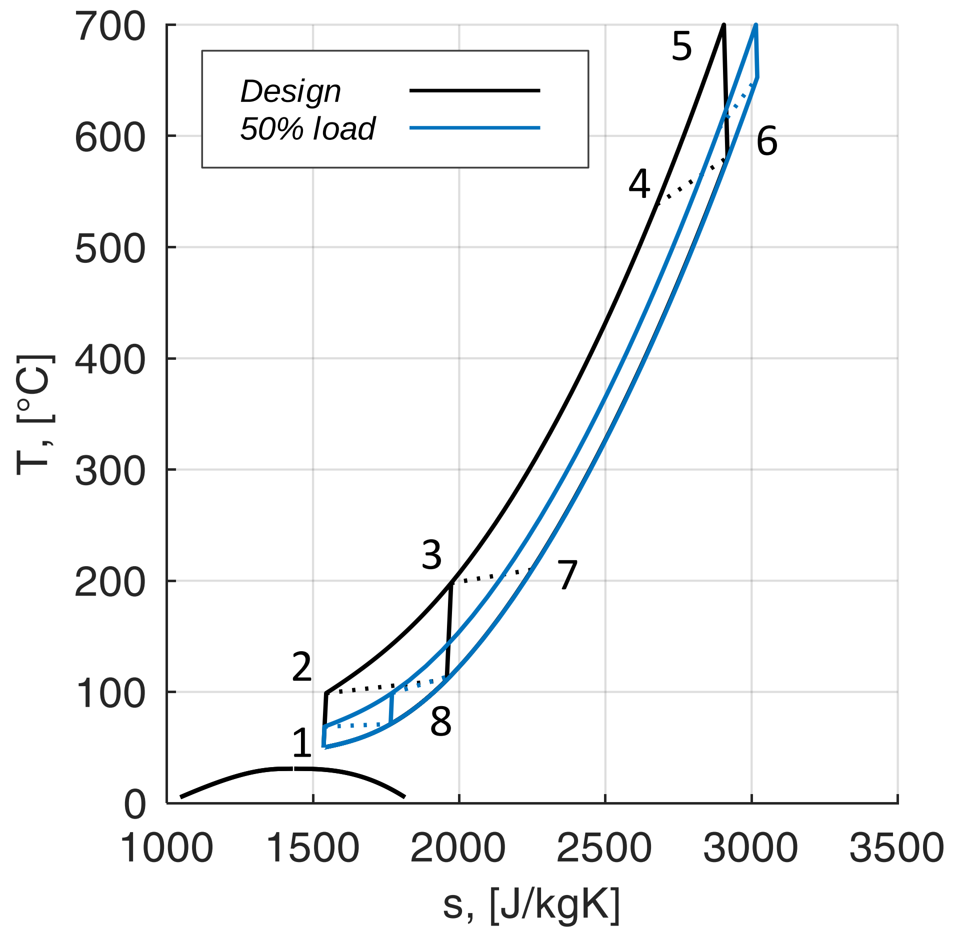
CASE 1: p_{min} equal to the design value

When the HTF mass flow rate is reduced:

Maximum cycle pressure is reduced leading to higher temperature at PHE (point 4) but also lower temperature at HTR bypass inlet (point 3)

Cycle pressure ratio reduces with penalizing effects on cycle efficiency

Heat exchangers show higher effectiveness because of the lower duty and temperature differences reduce with positive effects on cycle performance



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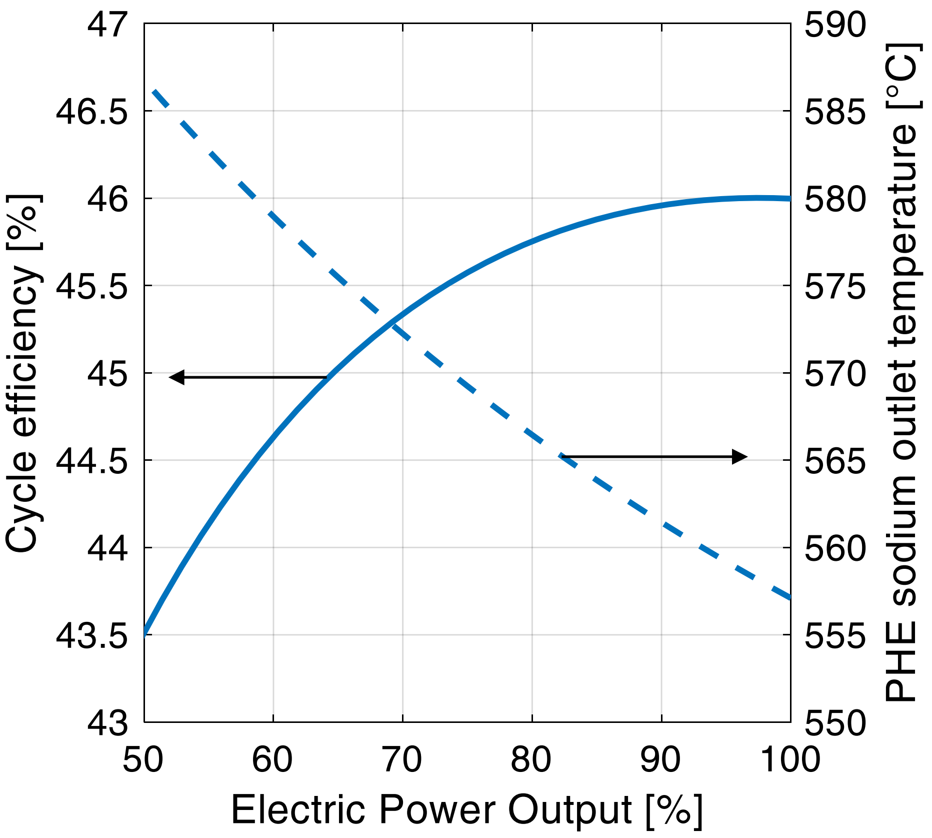
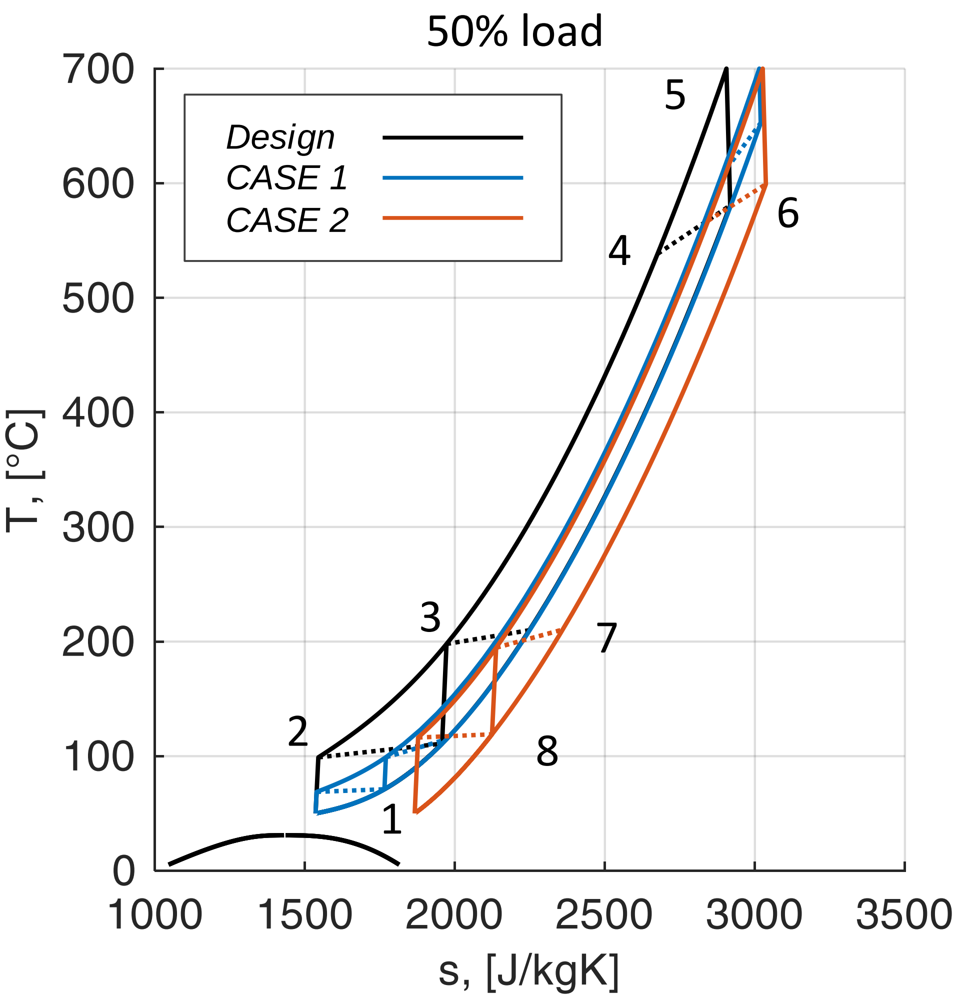


CASE 2: optimized p_{min}

CASE2 is obtained by optimizing for each part-load the cycle minimum pressure to obtain the maximum cycle efficiency.

When the HTF flow rate is reduced:

A reduction of the cycle minimum pressure limits the penalization related to the cycle compression ratio reduction



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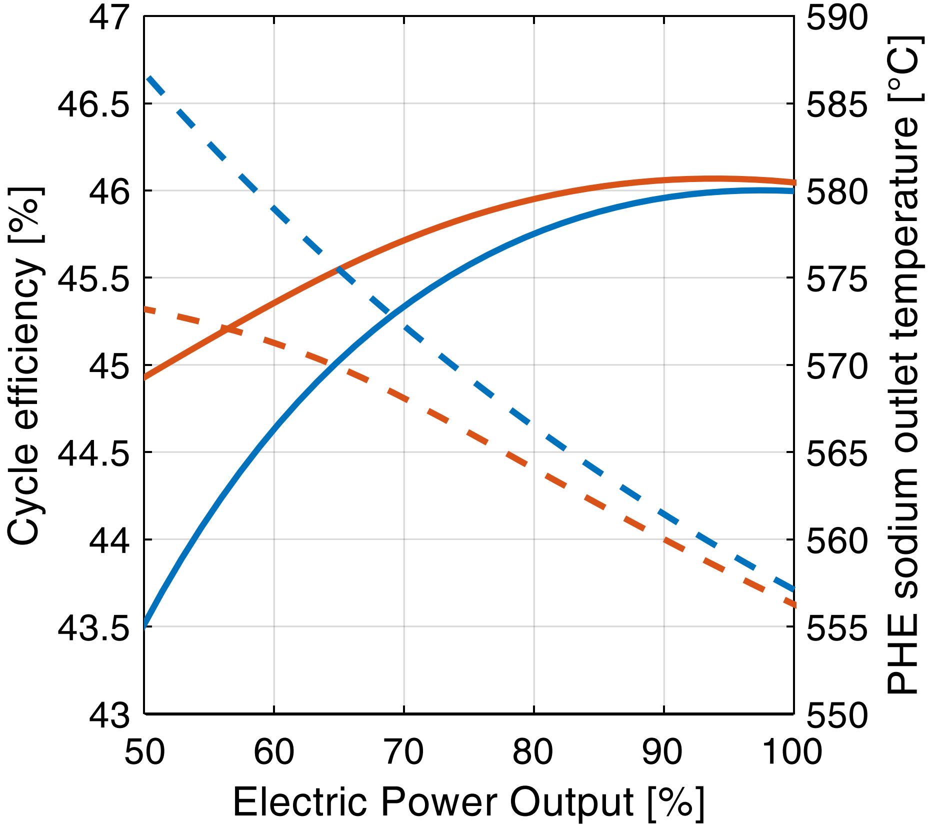
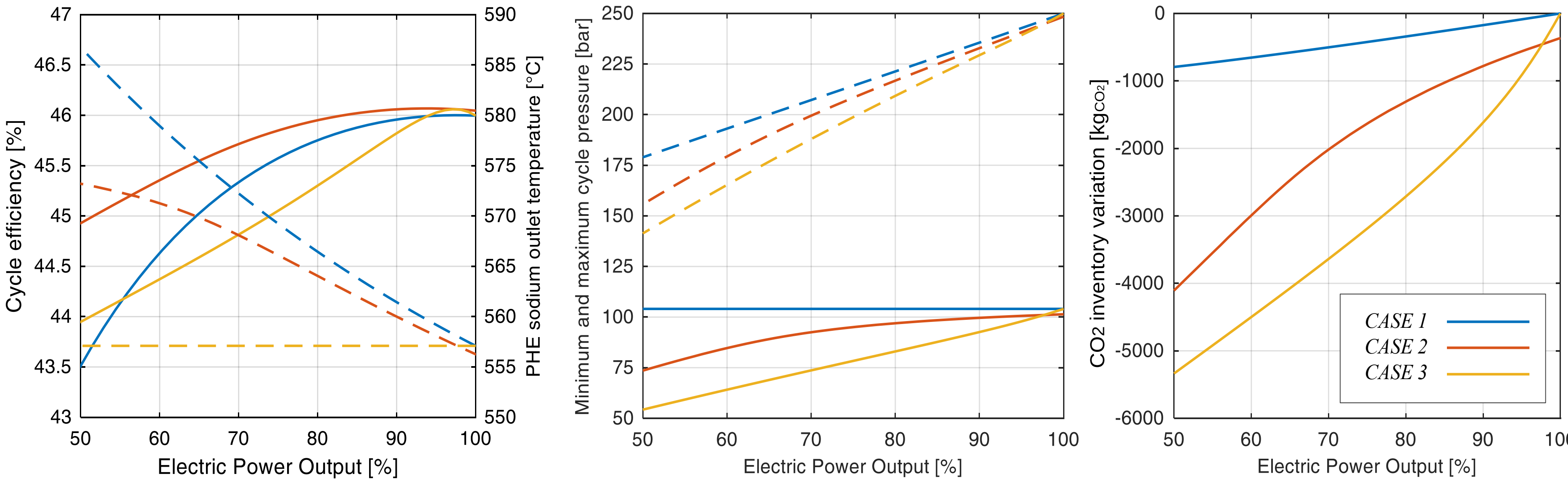
The cycle efficiency at 50% load can be increased by 1.5 percent point with respect to CASE 1



CASE 3: ΔT_{Na}=const

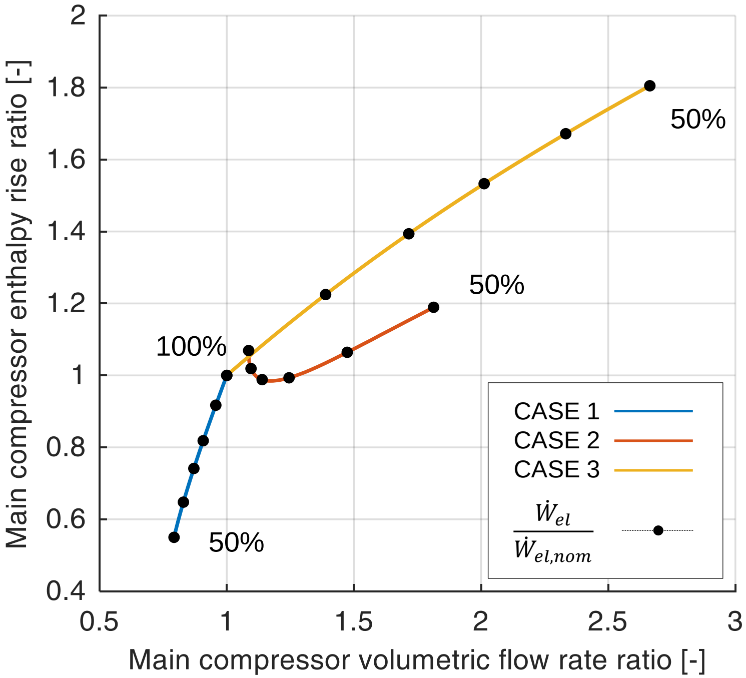
The minimum cycle pressure is obtained through a sensitivity analysis in order to keep a constant HTF temperature variation across the PHX.



CASE 3 performs poorly as keeping ΔT_{Na}=const requires a continuous reduction of the minimum cycle pressure immediately shifting the compressor inlet away from the critical point.

All CASEs require a tank to store the excess CO_2 extracted from the cycle. CASE 3 implies also the maximum CO_2 inventory variation.

The main compressor operating points for CASE1-3 reported in a dimensionless $\Delta h - \dot{V}$ diagram show strongly different trends.



In CASE 1 both $\dot{V}$ and Δh reduce with the load (HTF mass flow rate):

- cycle $\beta \downarrow \rightarrow \Delta h \downarrow$
- $\rho_1(T,P) = const \rightarrow \dot{V}_1 = \frac{\dot{m}_{CO2}}{\rho_1} \downarrow$

In CASE 2-3 the compressor inlet shifts away from the critical point thus both $\dot{V}$ and Δh increase

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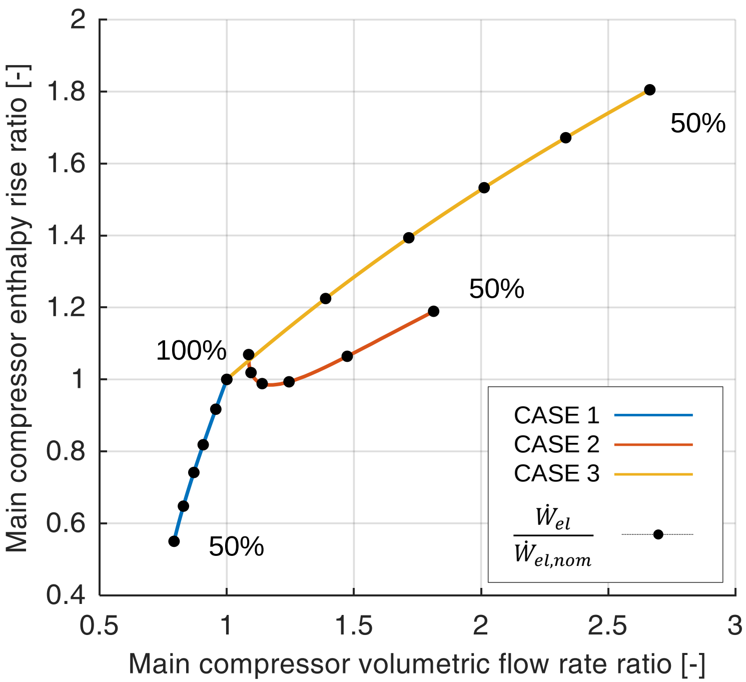

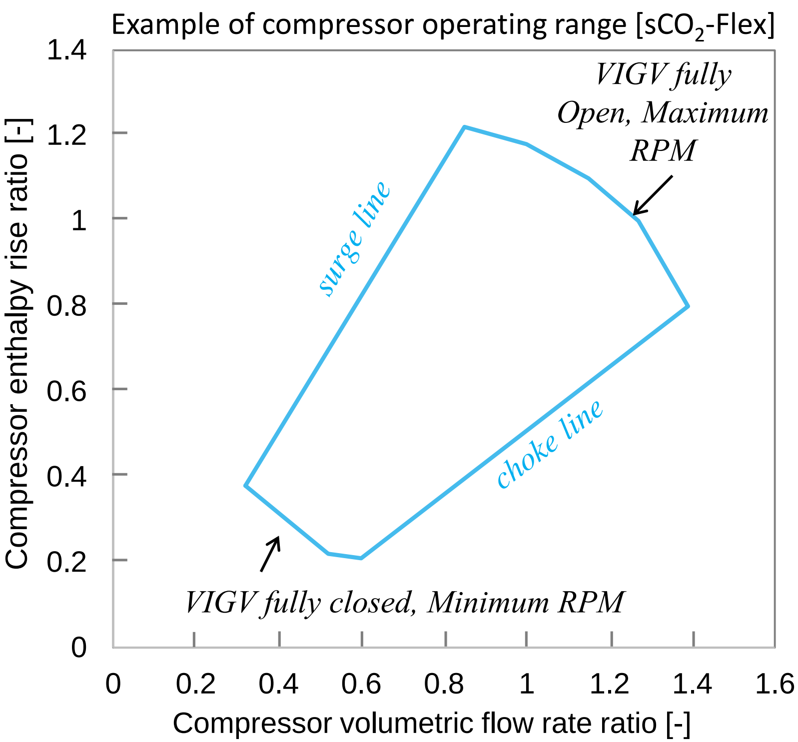

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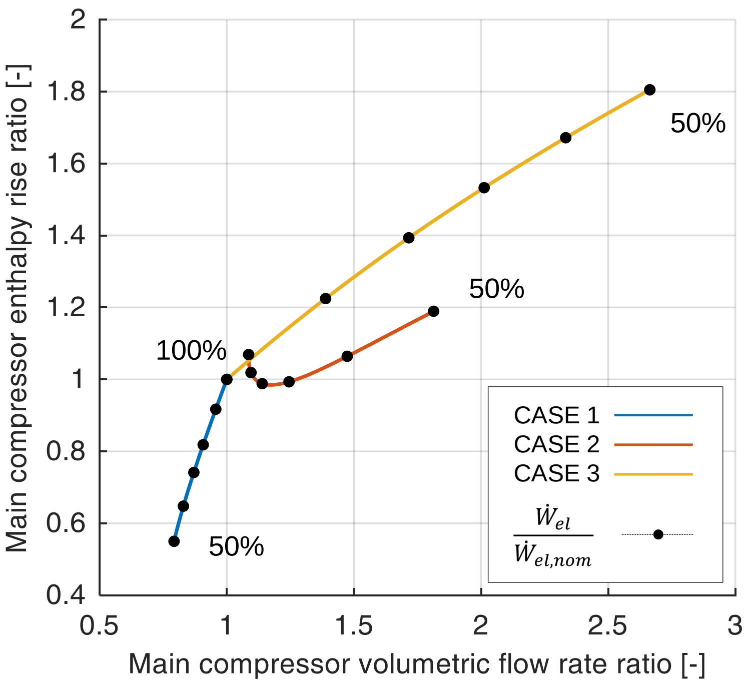

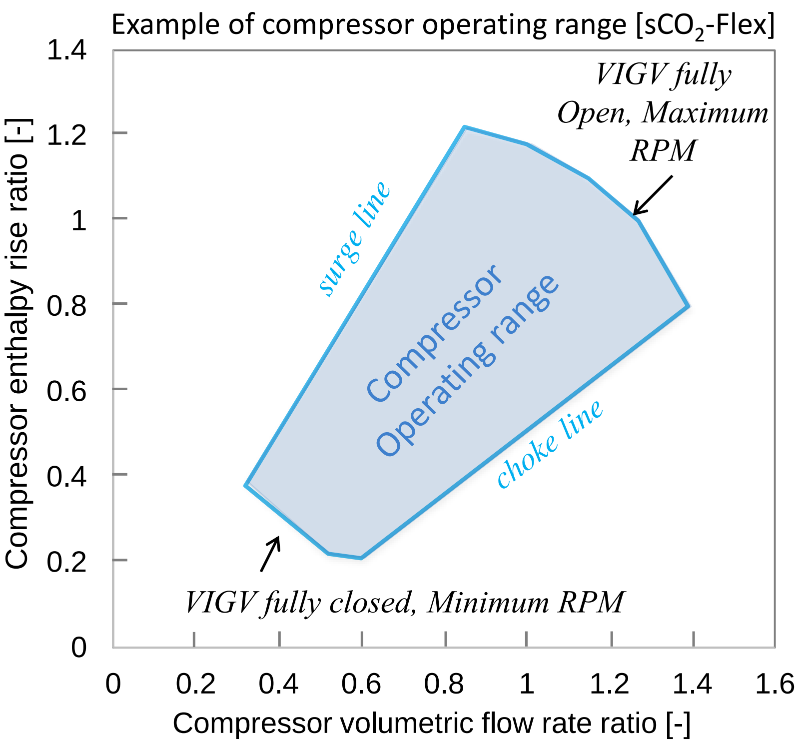

- The operating point of the compressor can be varied acting on VIGV and on RPM.

Part Load Analysis – Considerations on Compressors Design

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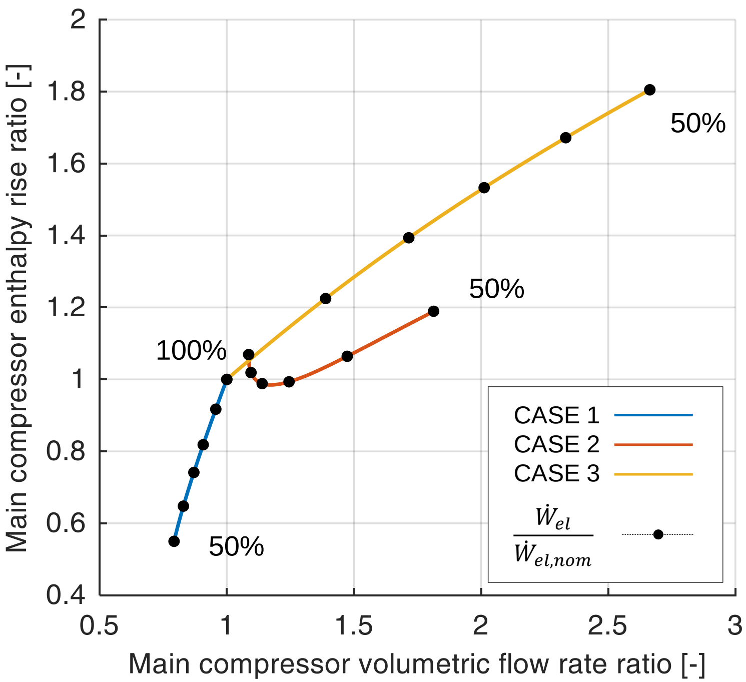
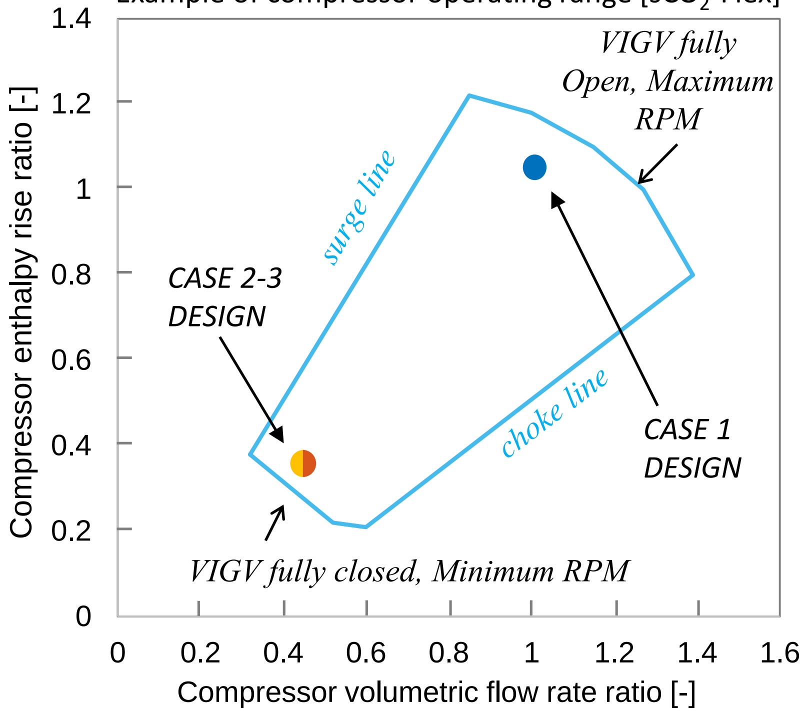


Example of compressor operating range [sCO_2-Flex]

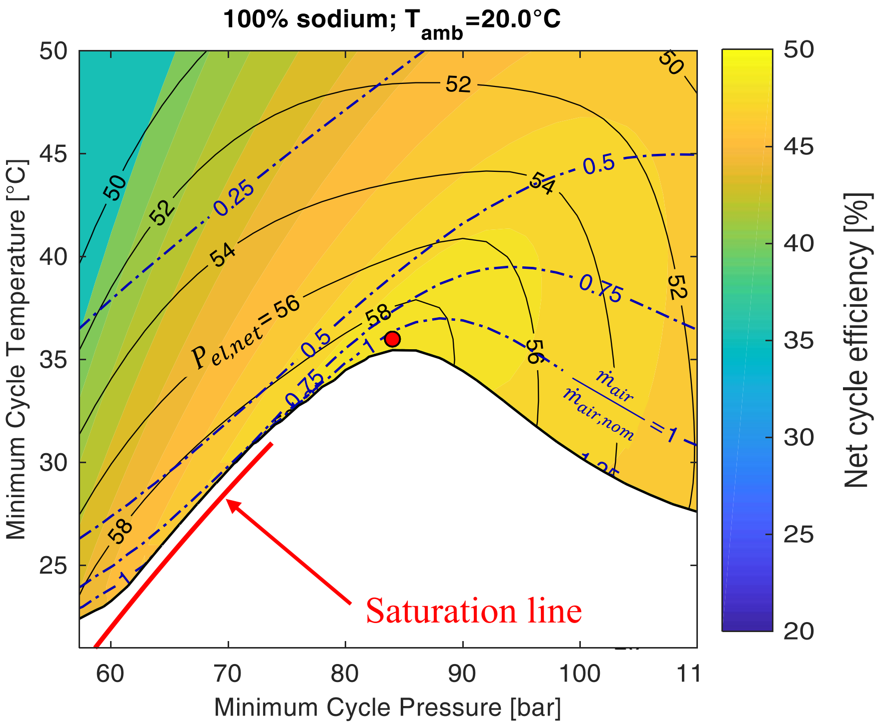


- The operating point of the compressor can be varied acting on VIGV and on RPM
- CASE1-3 design point needs to be significantly different.
- Further degrees of freedom may be obtained using 2 series/parallel compressors or acting on the split ratio between primary and secondary compressor.

Variable Ambient Temperature Analysis

For a given ambient temperature minimum cycle T and P have been parametrically investigated to find the best operating point (maximum efficiency).

The selected T and p can be obtained by varying **HRU fan speed** and **fluid inventory.**



Design: T_{amb} = 40°C, T_1=CIT=50°C, P_1=104.1

With lower ambient temperature it is preferred to keep the HRU fan speed at maximum reducing the compression work (CIT ↓).

As T_{amb} reduces the best operation moves towards the critical point minimizing the compression work.

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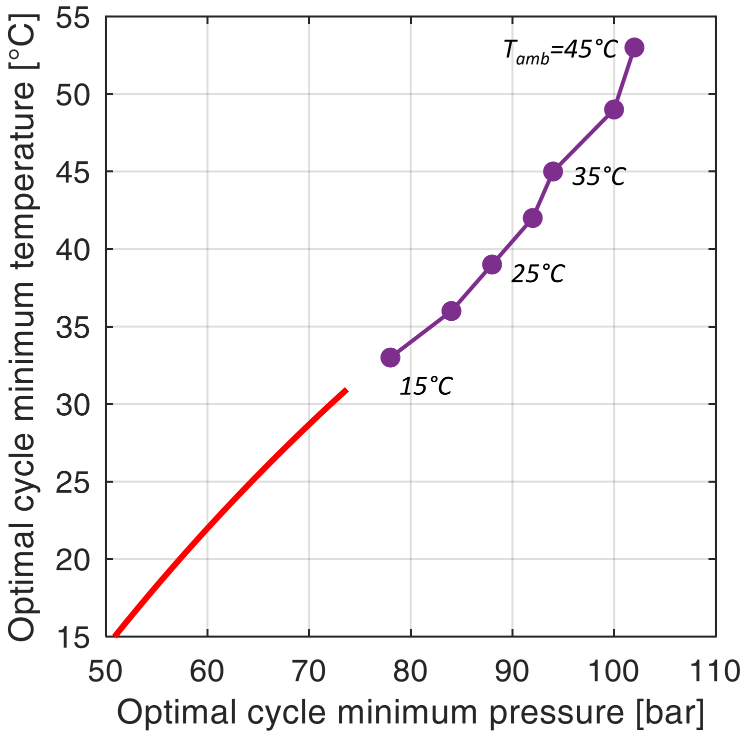

Design: $\begin{cases} T_{amb} = 40°C \\ T_1 = CIT = 50°C \\ P_1 = 104.1 \end{cases}$

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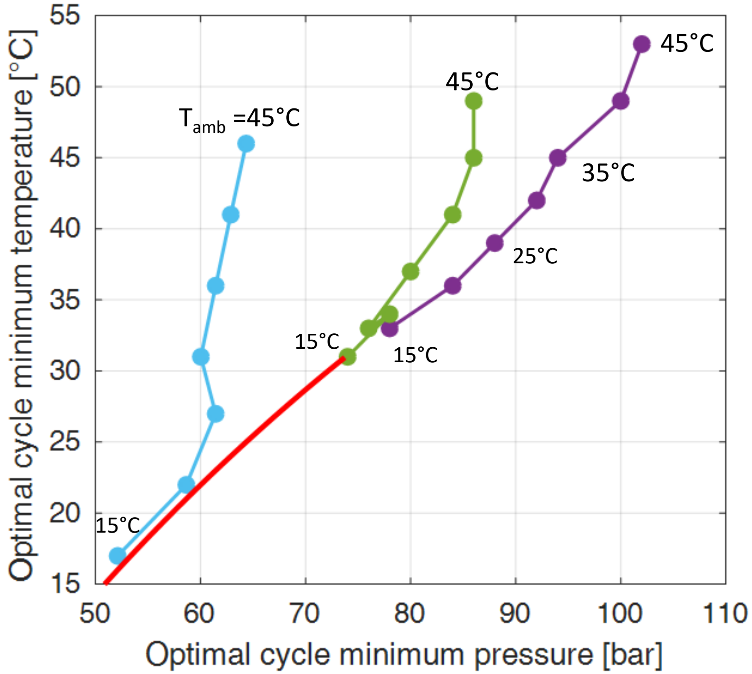
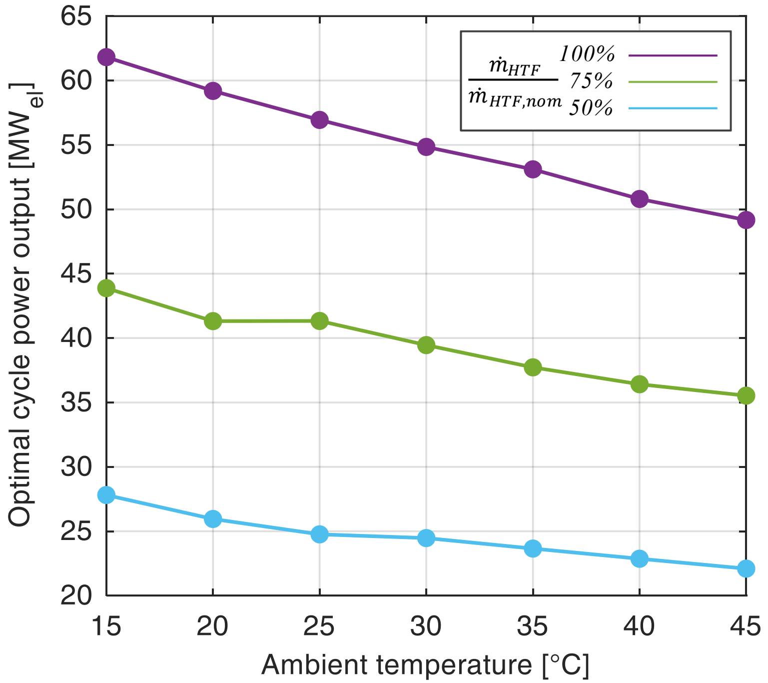
Variable Ambient Temperature AND Part Load Analysis

- The combined effect of **variable ambient temperature** and **variable sodium mass flow rate** was finally investigated.
- At 50% load a stronger CIT reduction is obtained, but no condensation occurs.
- The best operating point is thus obtained for **every ambient temperature** and **every load**.





Conclusions

- Part load analysis of sCO_2 power plants is a topic poorly investigated in literature and guidance and/or best strategies are still missing
- A tool for the evaluation of the off-design performance of sCO_2 cycles has been developed
- The minimum cycle pressure optimization allows to increase the plant efficiency at part-load, with a stronger inventory variation (larger sCO_2 storage)
- The design of the main compressor should allow an increase of Δh and $\dot{V}$ at part-load operation
- The design of the Heat Rejection Unit should consider the variation of the ambient temperature for the selected site

Future Steps

- Implementation of performance maps for turbomachinery in order to account for components efficiency variation
- Introduction of other possible control options such as anti-surge valves, split ratio variation in off-design operation, reduction of the maximum cycle temperature
- Evaluate yearly performance of CSP plants using sCO_2 cycle maximizing the energy output or the system economic performances
- Evaluate the best HRU and compressor design to maximize the yearly energy output

Thanks for your attention!

https://www.sco2-flex.eu/

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