PART-LOAD OPERATION OF COAL FIRED sCO_2 POWER PLANTS

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The sCO_2-Flex H2020 European project

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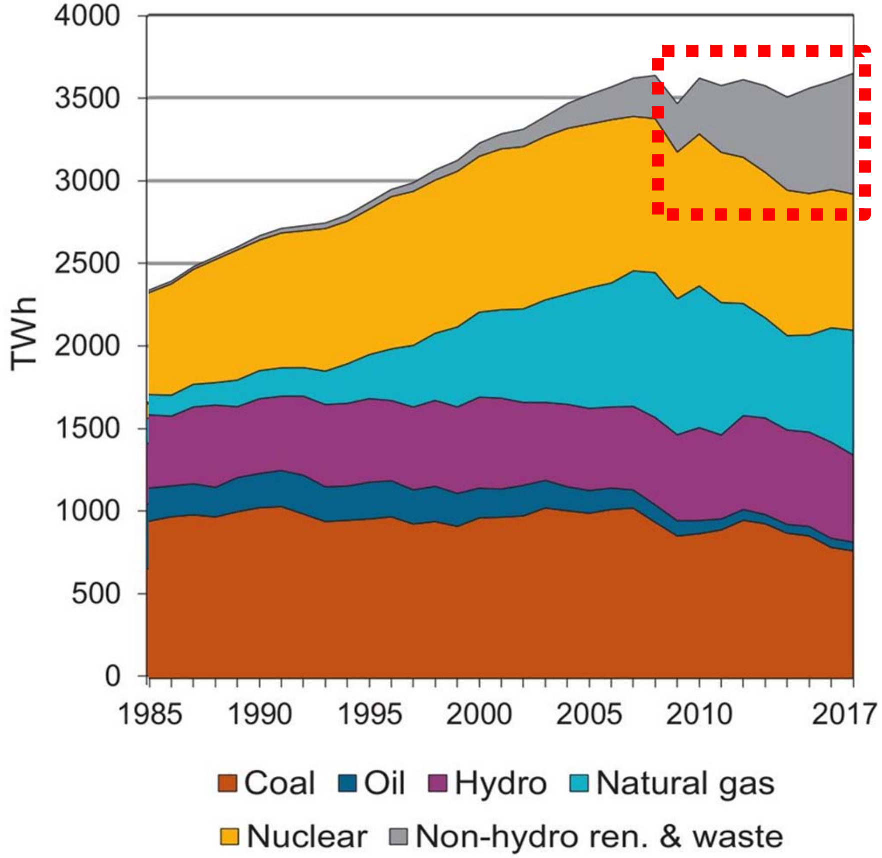
Why coal fired sCO_2 power plants?

Big changes are coming…

In recent years, the share of electricity produced from **renewable sources** has been growing rapidly

The electrical output of wind and solar power technologies is highly variable and intrinsically **unpredictable**

There is **lack** of an affordable and competitive large-scale electricity storage systems



OECD E.U. countries electricity production by renewable energy source.
IEA - World Energy Outlook 2018

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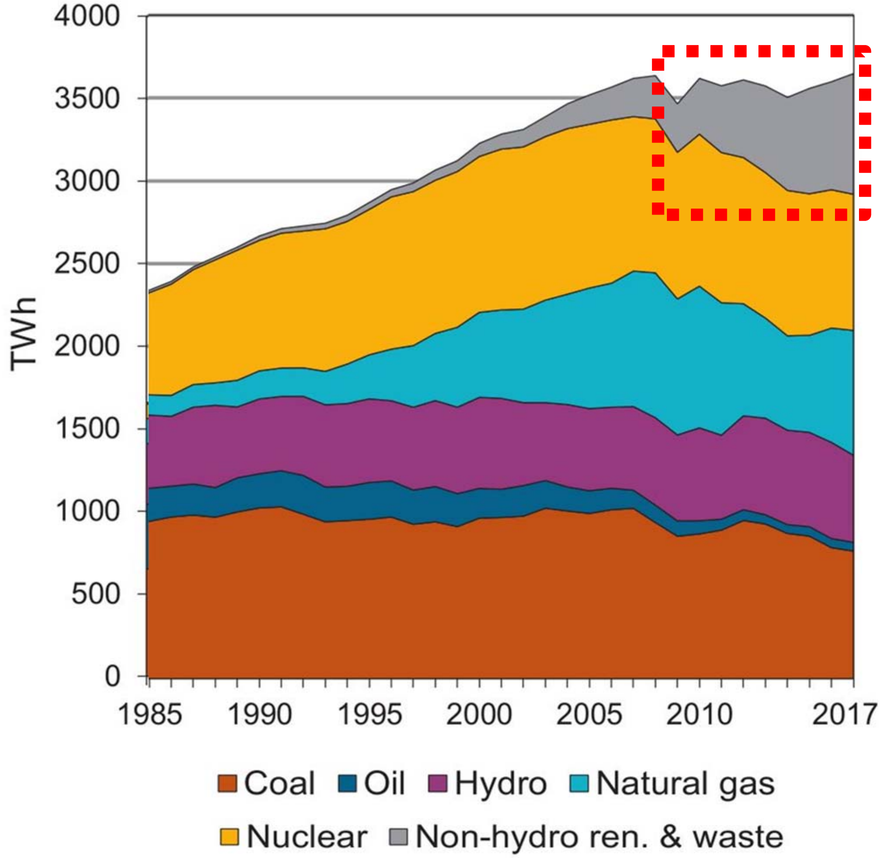
This involve stress and difficulties in operating the transmission electrical grid in stable conditions



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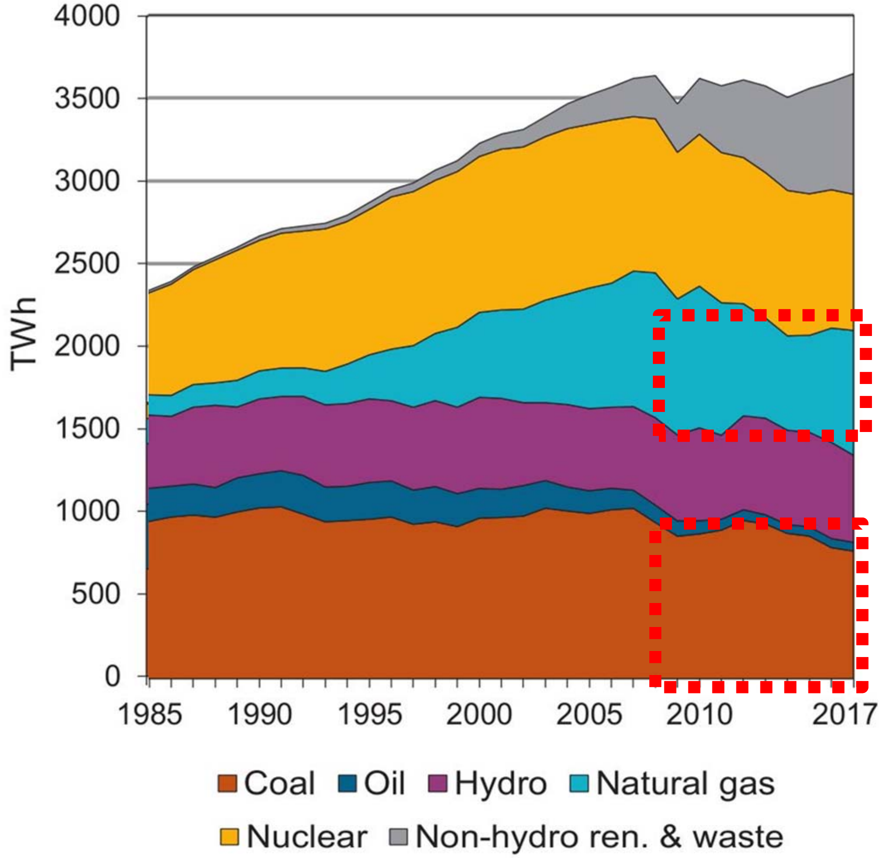
… but fossil fuels firmly resist

Fossil fuels fired power plants still play an **important role** in large scale power production in the E.U. and worldwide

Current coal and lignite fired power plants are **not optimized for part-load** operation and enhanced flexibility

A shift in the role of fossil fuel power plants is required moving from base-load operation to providing fluctuating **back-up power** to meet unpredictable and short-noticed load variations.



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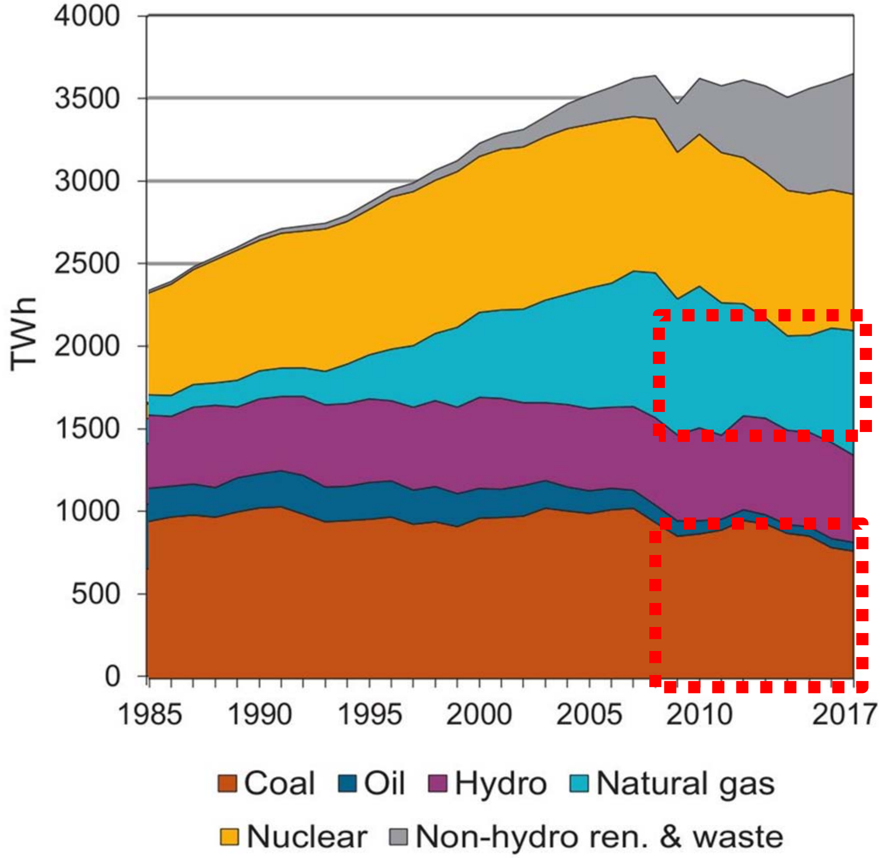
Which solution for the near future?



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Can sCO_2 be the solution for fossil fuel power sector?

Safe and economic

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Compact turbomachines

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Compact turbomachines

Reduced inertia

sCO_2-Flex H2020 European project

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

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

Goals and Activities

H2020 sCO_2-Flex project is dedicated to:

The 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 and lignite power plants

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Bringing the sCO_2 cycle to TRL6

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

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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).

Consortium and Polimi Role

The consortium is made of 10 partners from 5 European countries

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1.	EFD:	Project leader
2.	Zabala:	Dissemination and communication
3.	Nuovo Pignone BHGE:	Turbine and compressor design and testing
4.	UJV REZ:	Pulverized coal boiler development
5.	RINA – CSM:	Corrosion and 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

Aim of the study

Main questions are:

Are sCO_2 power cycles more efficient than steam Rankine cycles in part-load operation?

Which is the best part-load operating strategy for sCO_2 coal fired power plants?

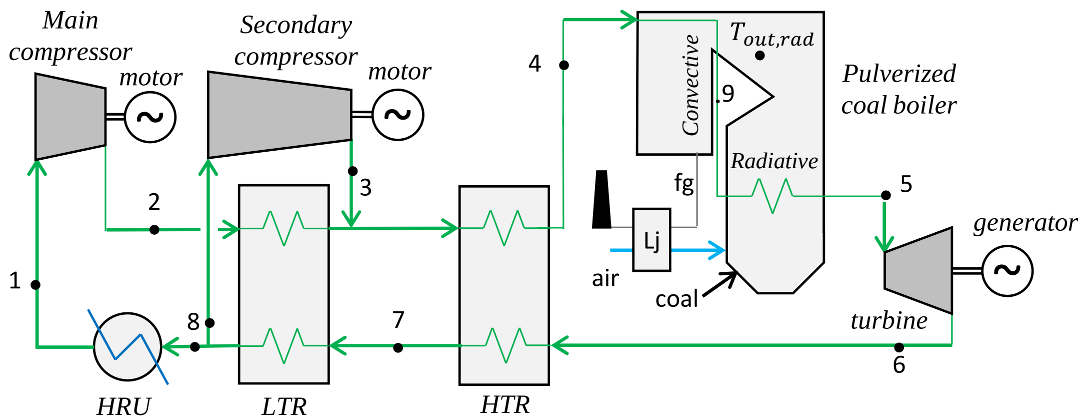
What is the optimal nominal design of the turbomachinery considering the part-load operation?

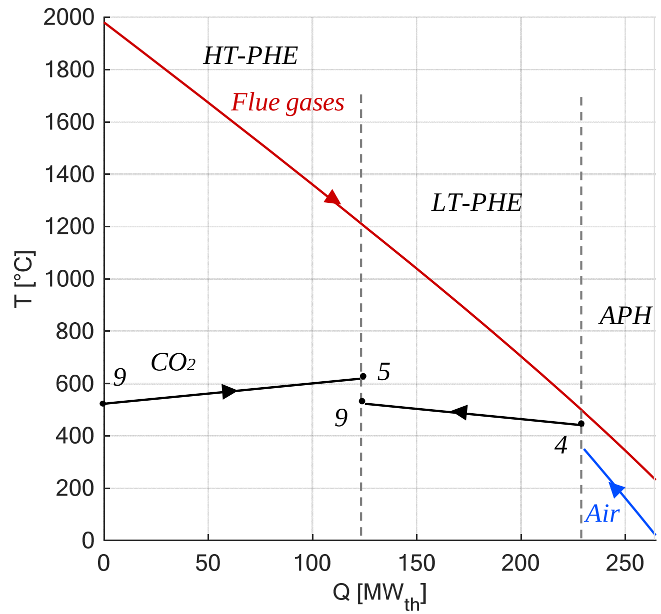
Nominal design and sizing of the cycle components

Selected cycle layout

Cycle selected: Recompressed cycle

Great cycle thermodynamic efficiency thanks to the heat capacity match on both LTR and HTR **but penalized boiler efficiency** due to the high boiler stack temperatures and stack losses





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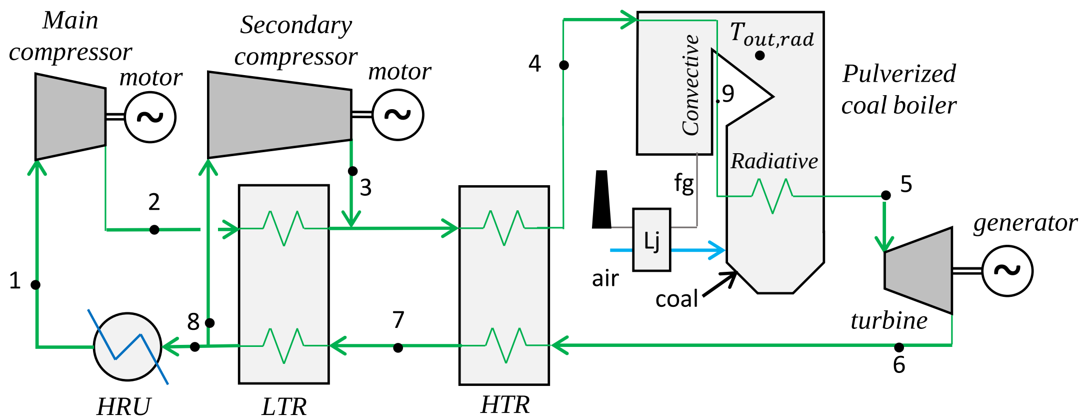
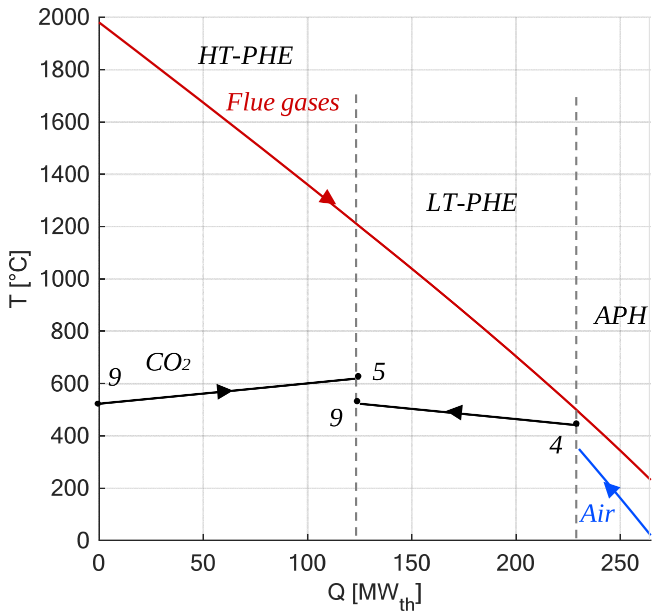
Possible solution?



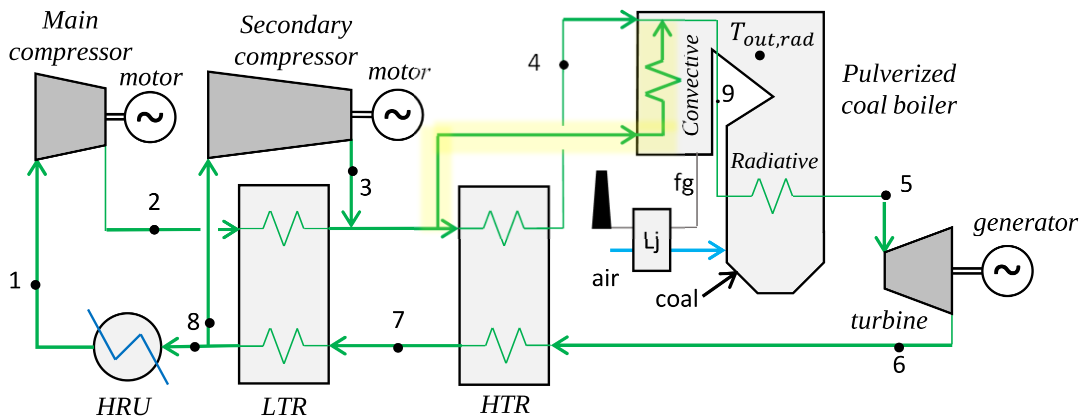


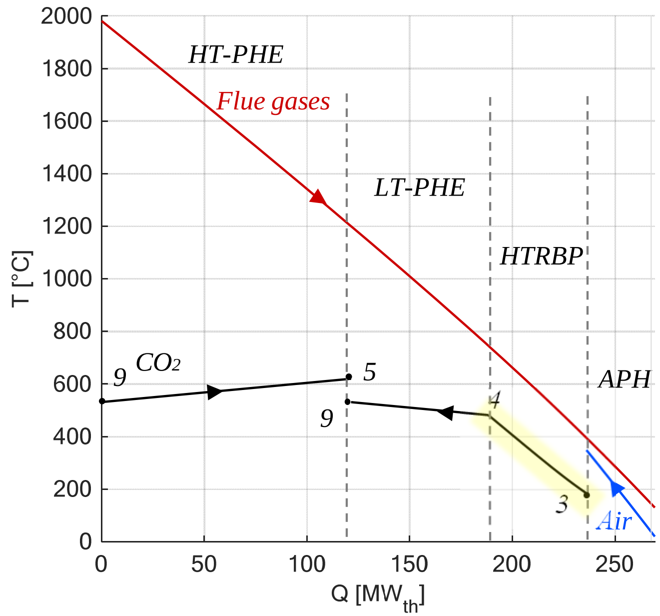
Selected cycle layout

Cycle selected: Recompressed cycle **with HTR bypass**

Great cycle thermodynamic efficiency thanks to the heat capacity match on both LTR and HTR **but penalized boiler efficiency** due to the high boiler stack temperatures and stack losses

Increase in the overall plant efficiency thanks to the **reduction of the stack losses** by means of the introduction of the HTR bypass





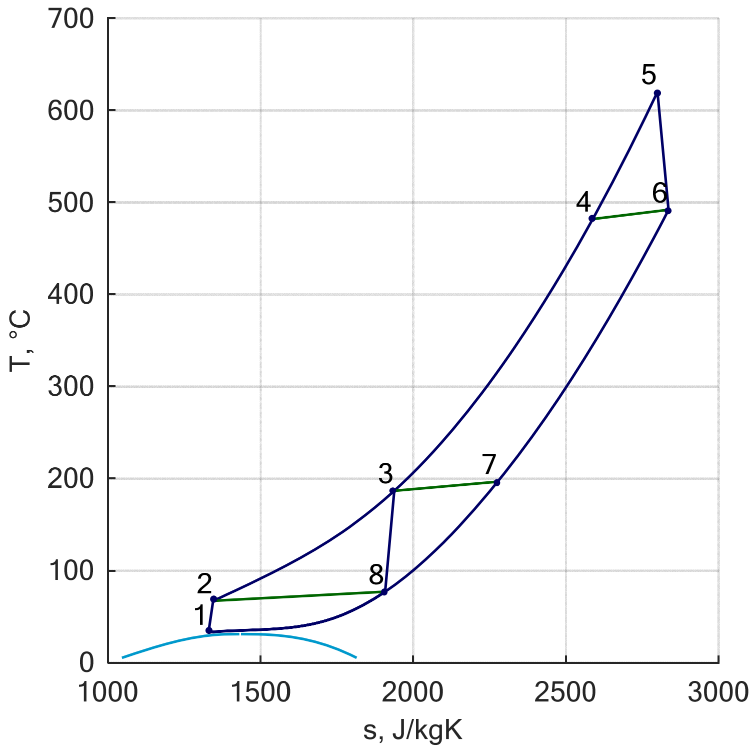
Code description and assumptions

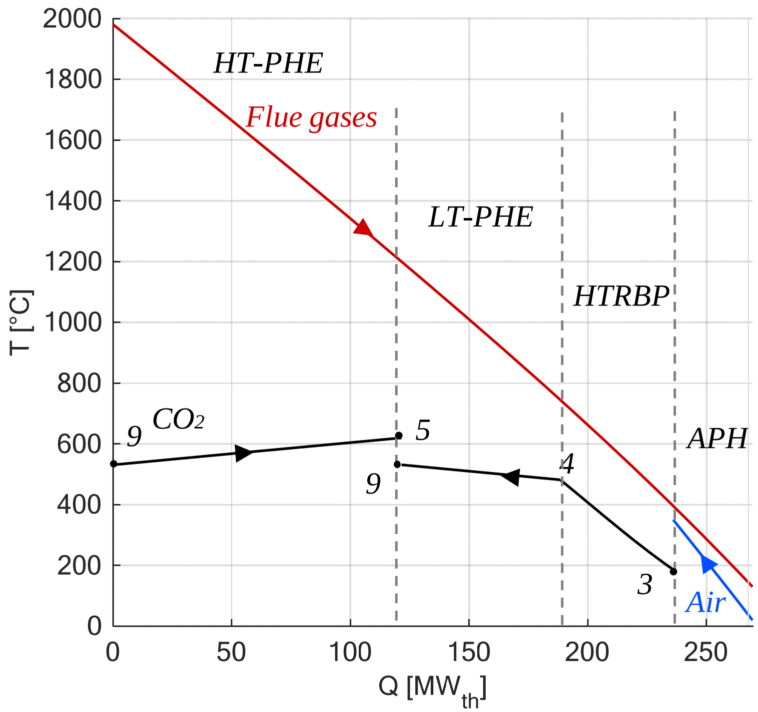
- A numerical code has been developed in MATLAB + REFPROP
- Optimization variables: p_{min} and p_{max}
- Objective function: $\eta_{plant} = \eta_{boiler}\,\eta_{cycle}$

Cycle parameter	Value
Minimum cycle temperature T_1,°C	33
Maximum cycle temperature T_5,°C	620
Maximum admissible cycle pressure p_2, bar	250
Turbine adiabatic efficiency, η_{turb}	84.3 %*
Main compressor efficiency, η_{comp1}	82.2 %*
Sec. compressor efficiency η_{comp2}	82.4 %*
Generator/motor efficiency $\eta_{\mathrm{me},t}/\ \eta_{\mathrm{me},c}$	96.4 %
Recuperators pinch point ΔT_{REC},°C	10
Boiler pinch point $\Delta T_{\mathrm{pp},boiler}$,°C	50
Ambient temperature T_{amb},°C	20

*= The reported turbomachinery efficiency value is just indicative

Thermodynamic optimization results

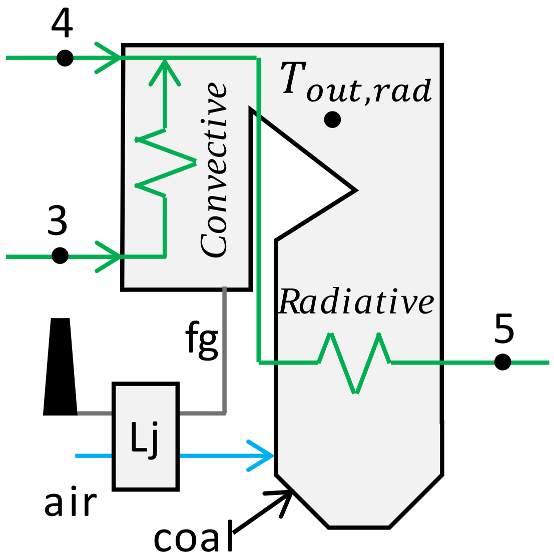




Optimization results	Value	Optimization results	Value
Minimum cycle pressure p_1,bar	33	Sec. comp. electric power, MW_{el}	31.28
Maximum cycle pressure p_2, bar	250	HRU auxiliaries, MW_{el}	1.09
CO2 mass flow at turbine, kg/s	1087.12	Cycle efficiency	41.92%
Turbine electric power, MW_{el}	152.27	Boiler efficiency	94.37%
Main comp. electric power, MW_{el}	20.98	**Overall plant efficiency**	**39.56%**

Components description – Coal Boiler

Parameter	Value
Δp, bar	2
Excess air	20%
$\Delta T_{\mathrm{pp},Lj}$,°C	30



Radiative section

- $T_{out,rad} = 1200°C$
- From Thermoflex:

$$q''_{rad} = f(T_{out,rad})$$

Parameter	Value
d_i, mm	20
$Pitch/d_{ext}$	1.45
Tube and memb.	INCONEL 617

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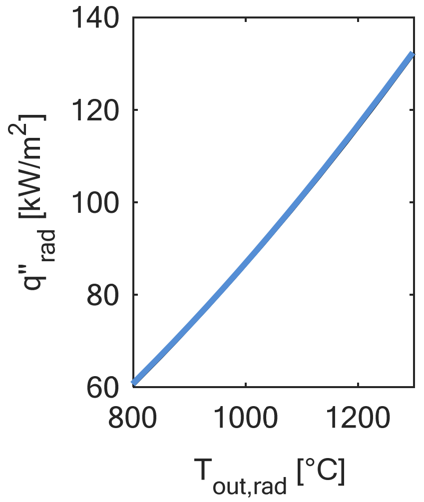
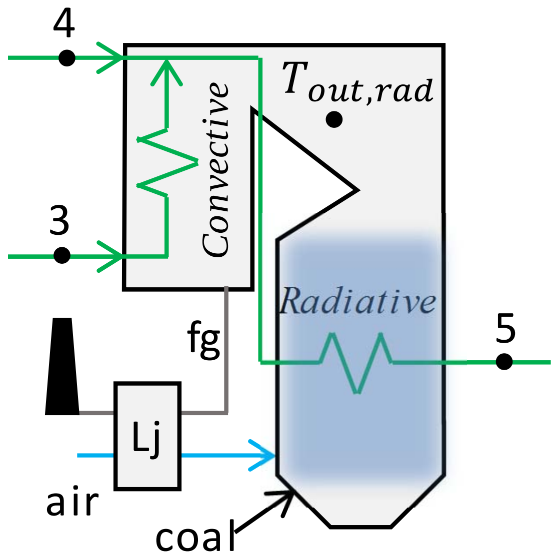




Components description – Coal Boiler

Radiative section

- $T_{out,rad} = 1200°C$
- From Thermoflex:

$$q''_{rad} = f(T_{out,rad})$$

Convective section

- To avoid acid condenses:

$$T_{stack} \geq 130°C$$

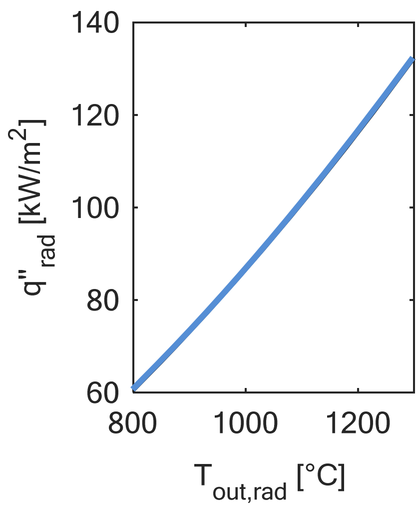
- Flue gas heat transfer coefficient:

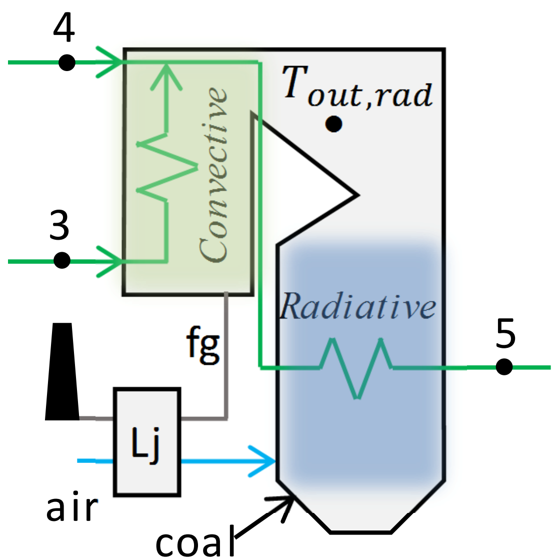
$$h_{fg} = h_{fg}^{rad} + h_{fg}^{conv}$$

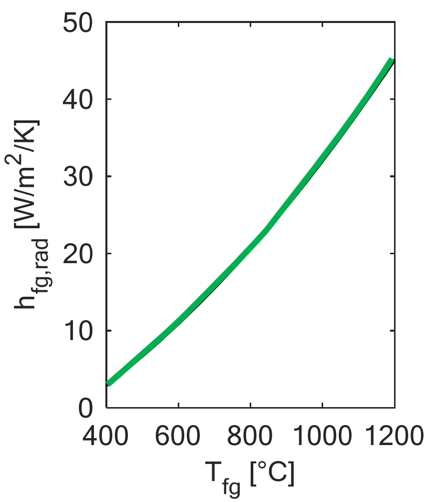
Parameter	Value
d_i, mm	20
$Pitch/d_{ext}$	1.45
Tube and memb.	INCONEL 617

Parameter	Value
Δp, bar	2
Excess air	20%
$\Delta T_{pp,Lj}$,°C	30

Parameter	Value
d_i, mm	40
L-T $Pitch/d_{ext}$	1.2 - 10
Tube material	INCONEL 617







Heat Rejection Unit

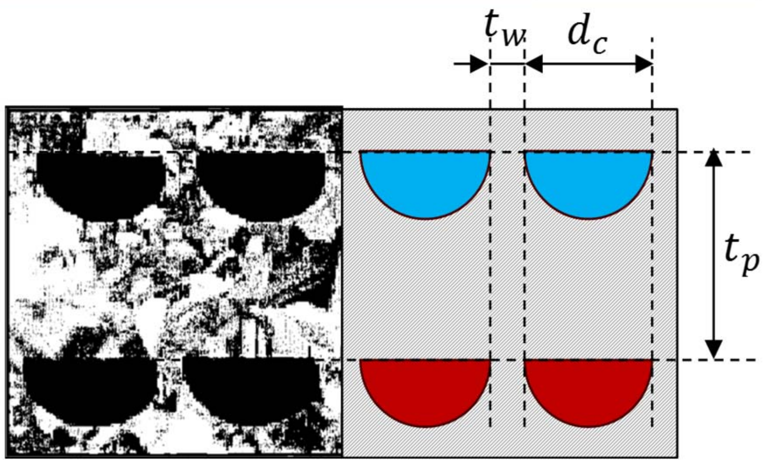
- Dry cooler derived from refrigeration field
- Model tuned on manufacturer data
- Modified Gnielinski correlation for h_{int}
- Material: copper tubes + aluminum fins
- Fans provided by variable speed EM
- CO_2 pressure drop: 0.5%

Parameter	Value
ΔT_{air}, °C	20
h_{ext}, W/m²K	75.8
Tube int. diameter d_i, mm	7.5
Circular fin thickness, mm	0.12
Circular fin spacing, mm	2.1
Circular fin height, mm	9.1

Recuperators

- Printed circuit heat exchanger (HEATRIC)
- 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
Internal volume/metal volume	0.387

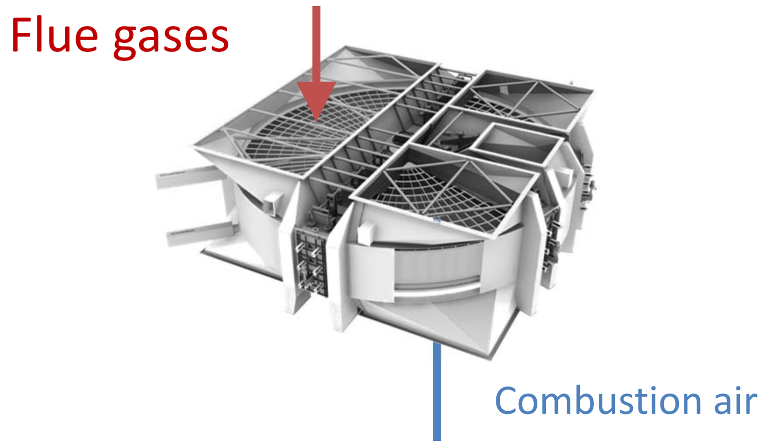


Air preheater

- One-stage Ljungström rotating air preheater
- Only primary air at combustor
- Component sized through Thermoflex commercial software

Parameter	Value
Global h.t.c.,W/m2K*	1.5

*= referred to matrix porous area



Main results from heat exchangers sizing

Parameter	HRU	LTR	HTR	PHE rad	PHE conv	HTR bypass
Heat duty, MW	188.4	162.4	369.3	121.5	66.9	47.5
Hot side h.t.c., W/m^2K	4880.7	4398.9	3682.7	-	105.7	84.7
Cold side h.t.c., W/m^2K	75.8	4503.5	3717.4	-	3419.6	4247.1
Global h.t.c., W/m^2K	**1093.59**	**2039.5**	**1737.6**	**80.2**	**98.2**	**80.3**
Internal surface, m^2	8912	6898.7	14345.5	1595	1203	1876
HX metal mass, kg	70316	43258	89953	77331	73573	113787
CO_2 inventory, kg	**7962.3**	**10178.6**	**11398.2**	**1170.2**	**1972.4**	**4495.53**

Turbomachinery

Compressors and turbine are designed by:

The main compressor is a centrifugal compressor equipped with IGV and variable speed motor (-60%/+10%)

The expander is an axial turbine rotating at fixed speed → no possibility to control it at part load

Turbomachinery	Control	η_{iso}*
Main compressor	IGV+RPM	82.2 %
Secondary compressor	IGV+RPM	82.4 %
Turbine	-	84.3 %

Part-load cycle modelling and operational strategies

Part load simulation methodology and assumptions

The part load simulation have been run with the following hypothesis:

- Turbine in chocked flow conditions (1)
- Δp and h.t.c. varied according to exponential laws for all HXs (2-3)
- HRU fan consumption computed through exponential correlation (4)

$$\frac{\dot{m}\sqrt{T_{in,turb}}}{p_{in,turb}A_{in,turb}} = const \qquad (1)$$

$$\Delta p = \Delta p_{design}\left(\frac{\rho_{design}}{\rho}\right)\left(\frac{\dot{m}}{\dot{m}_{design}}\right)^2 \qquad (2)$$

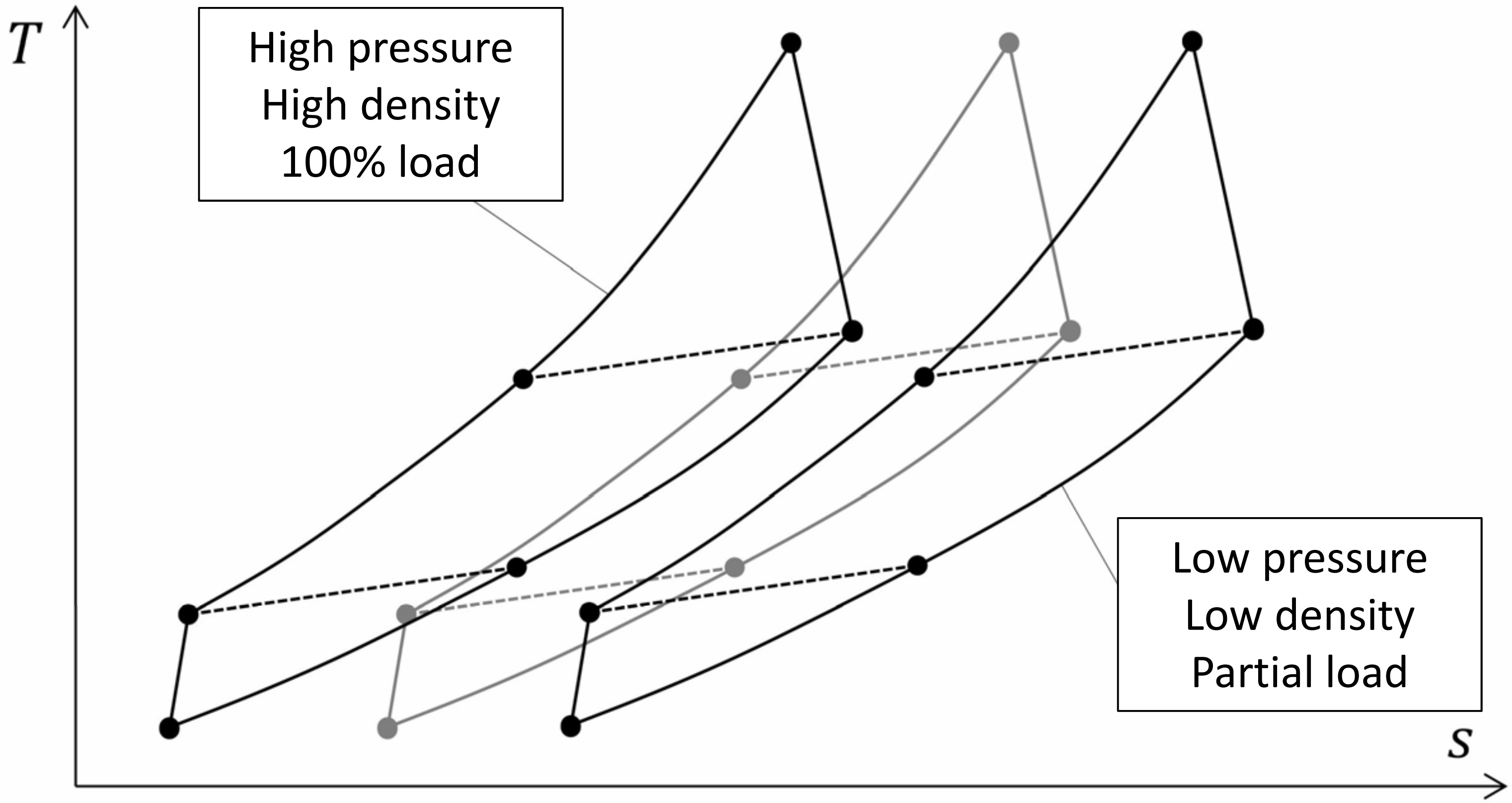
$$h_X = h_{X,design}\left(\frac{\dot{m}_X}{\dot{m}_{X,design}}\right)^\alpha \quad with \begin{cases} X = CO_2 & \alpha = 0.8 \\ X = gas & \alpha = 0.6 \end{cases} \qquad (3)$$

$$\dot{W}_{HRU,aux} = \dot{W}_{HRU,aux,design}\left(\frac{\dot{m}_{air}}{\dot{m}_{air,design}}\right)^{2.78} \qquad (4)$$

Control strategy of closed gas cycles

The part load control of a closed ideal gas cycle can be obtained by cycle de-pressurization removing a fraction of the working fluid mass.

This control strategy is not directly applicable to sCO2 cycles due to the proximity of critical point.



Operating strategy definition

As pulverized coal mass flow rate is reduced:

1) the air excess is kept equal to the nominal value $$\varepsilon = \frac{\dot{m}_{air}}{\dot{m}_{air,stoich}} = const$$

2) Turbine inlet temperature is controlled by acting on the main compressor operating point $$\frac{\dot{m}\sqrt{T_{in,turb}}}{p_{in,turb}A_{in,turb}} = const$$

3) Main compressor inlet temperature is maintained constant acting on HRU fan speed

4) Split fractions are set in order to keep ΔT_{mix}=0

5) Main compressor inlet pressure is set acting on fluid inventory

Operating strategy definition

Three operating strategies have been investigated:

- CASE 1: turbine in sliding pressure and minimum cycle pressure kept equal to the design value

- CASE 2 : turbine in sliding pressure and minimum cycle pressure optimized to obtain maximum plant efficiency

- CASE 3: constant maximum cycle pressure using a turbine provided by IGV and minimum cycle pressure equal to the design value

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- CASE 3: constant maximum cycle pressure using a turbine provided by IGV and minimum cycle pressure equal to the design value

For all the cases the **turbomachinery efficiency** at part-load has been considered equal to the **nominal value** and **no specific constraints are considered for the compressors**: the obtained operating points in terms of volumetric flow and enthalpy rise variation in off-design are meant to provide useful preliminary design criteria.

Results

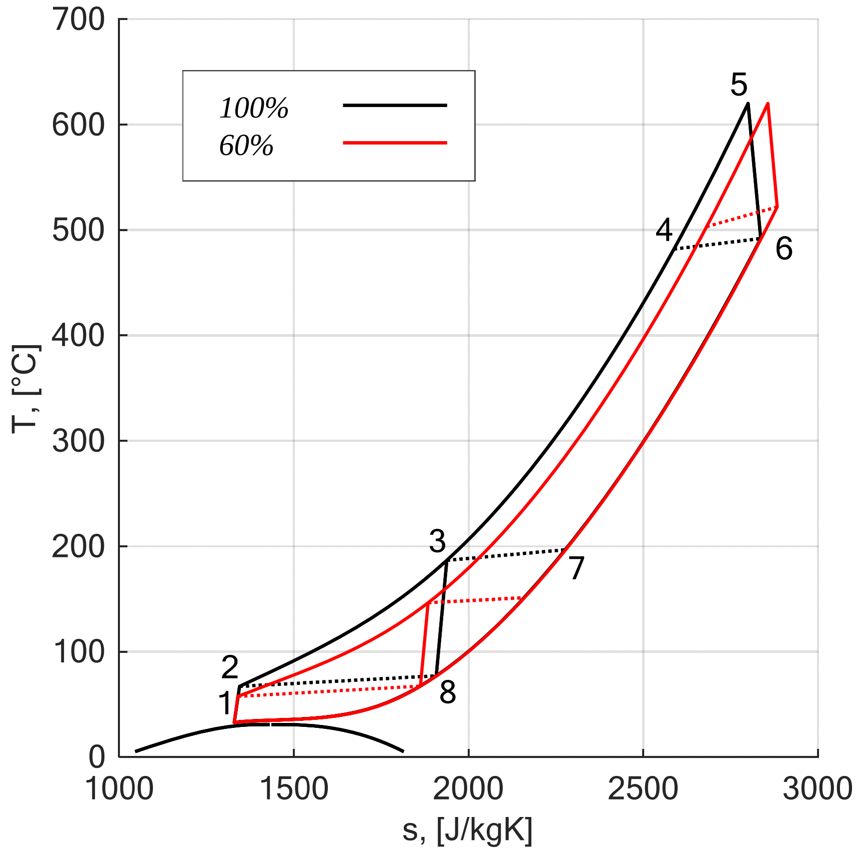
CASE 1: turbine in sliding pressure and p_{min} equal to the design value

When coal mass flow rate is reduced:

Maximum cycle pressure is reduced leading to higher temperature at the boiler inlet (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 plant overall performances

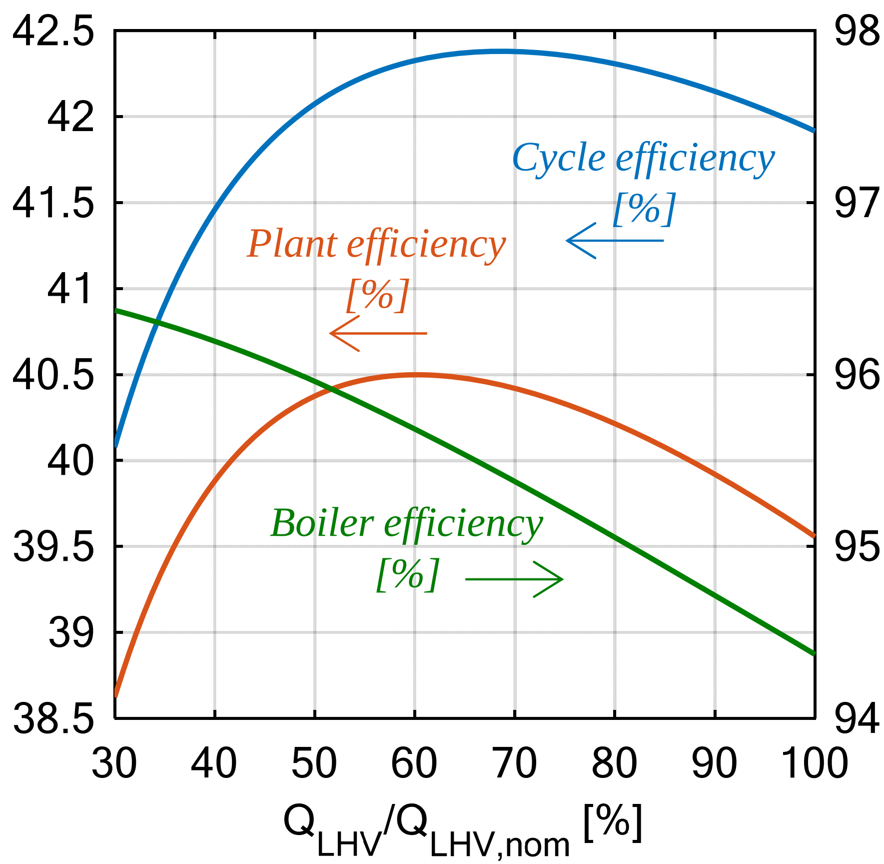


CASE 1: turbine in sliding pressure and p_{min} equal to the design value

When coal mass flow rate is reduced:

For coal mass flow rates **from 100% to 60%** the effect of lower cycle pressure ratio is compensated by the lower temperature differences and the higher effectiveness of the recuperators and boiler HX banks.

The increase of CO_2 temperature at boiler convective section inlet (point 4) due to the reduction of the pressure ratio is balanced by the decrease of the temperature at the exit of the LTR (point 3).

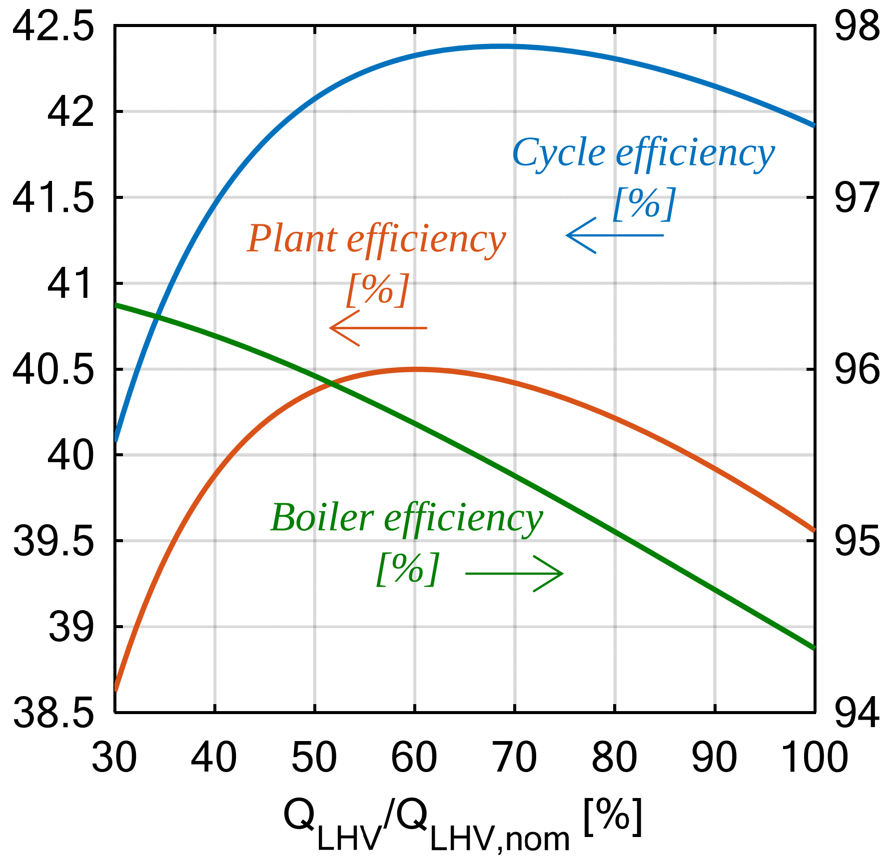


CASE 1: turbine in sliding pressure and p_{min} equal to the design value

When coal mass flow rate is reduced:

For **lower loads (60% to 30%)** the cycle efficiency is strongly penalized by the excessive reduction in the cycle pressure ratio, which limits the turbine specific work and the net output of the cycle.

Furthermore, the strong reduction of the CO_2 mass flow rate penalizes the HX heat transfer coefficients limiting the internal heat regeneration of the cycle.



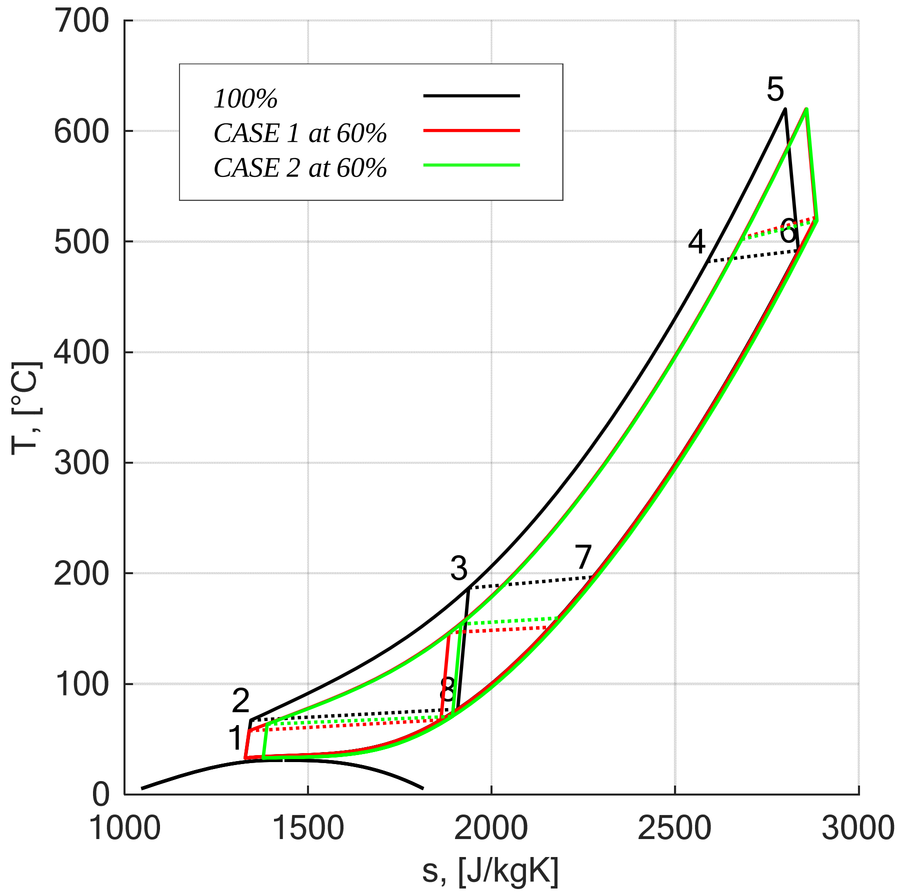
CASE 2: turbine in sliding pressure and optimized p_{min}

The optimal minimum cycle pressure is obtained through a sensitivity analysis reducing p_{min} down to 95% of the nominal value.

When coal mass flow rate is reduced:

It is convenient to progressively reduce also the cycle minimum pressure

The overall plant efficiency can be increased by 1 percent point with respect to CASE 1 at minimum load



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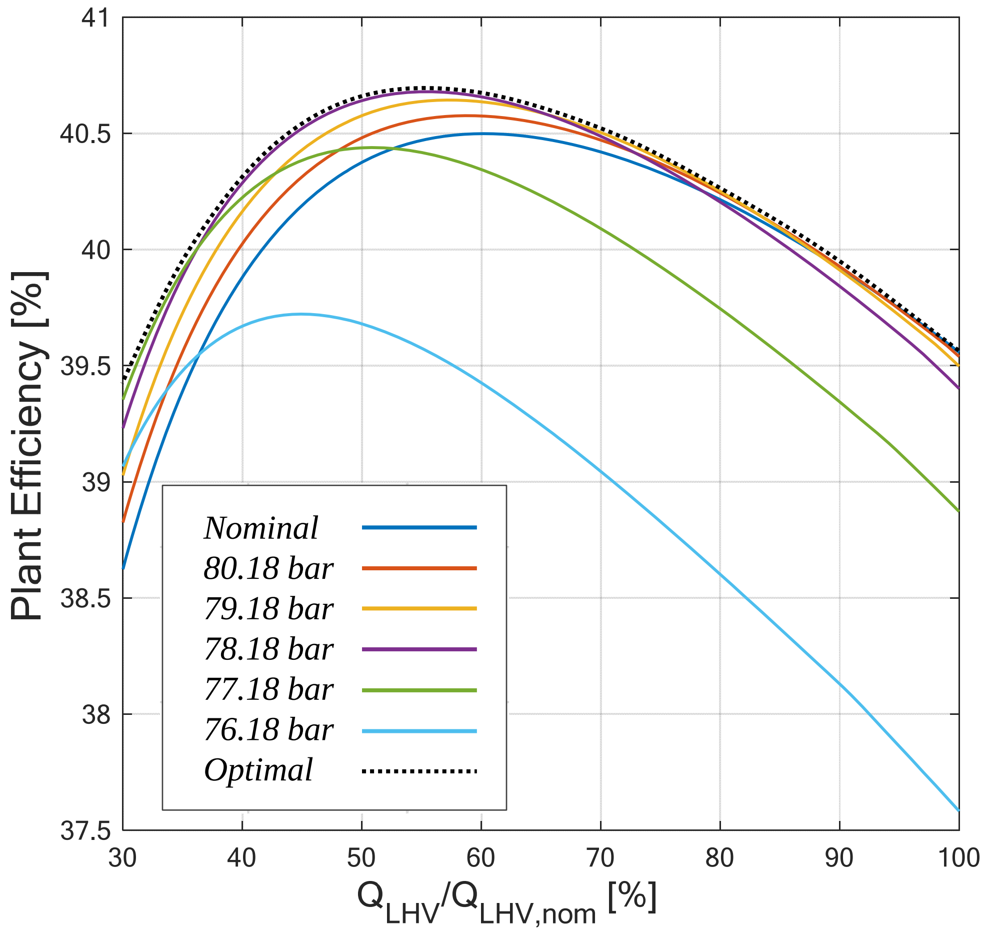


CASE 3: turbine with IGV and p_{min} equal to the design value

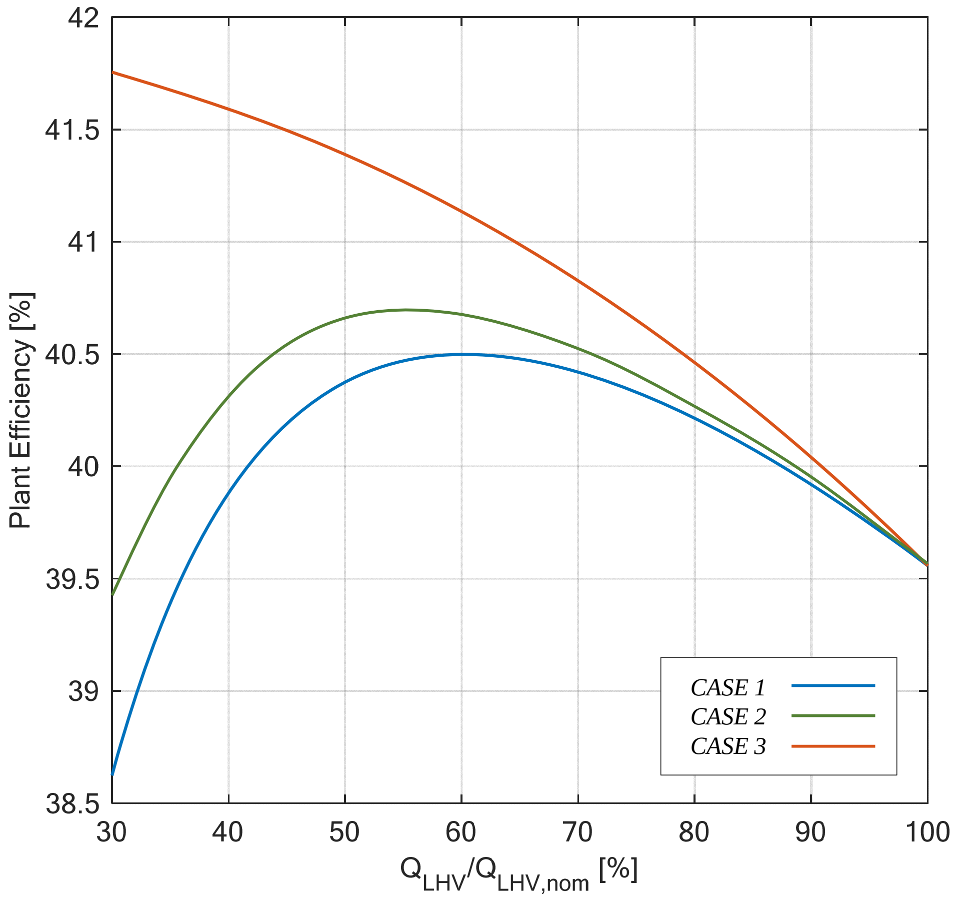
The adoption of IGV at turbine inlet would be possible thanks to the relatively low turbine inlet temperature (620°C).

When coal mass flow rate is reduced:

Acting on IGV allows to maintain the maximum pressure of the cycle

The pressure ratio remains unchanged and equal to the optimal value according to the minimum and maximum temperature

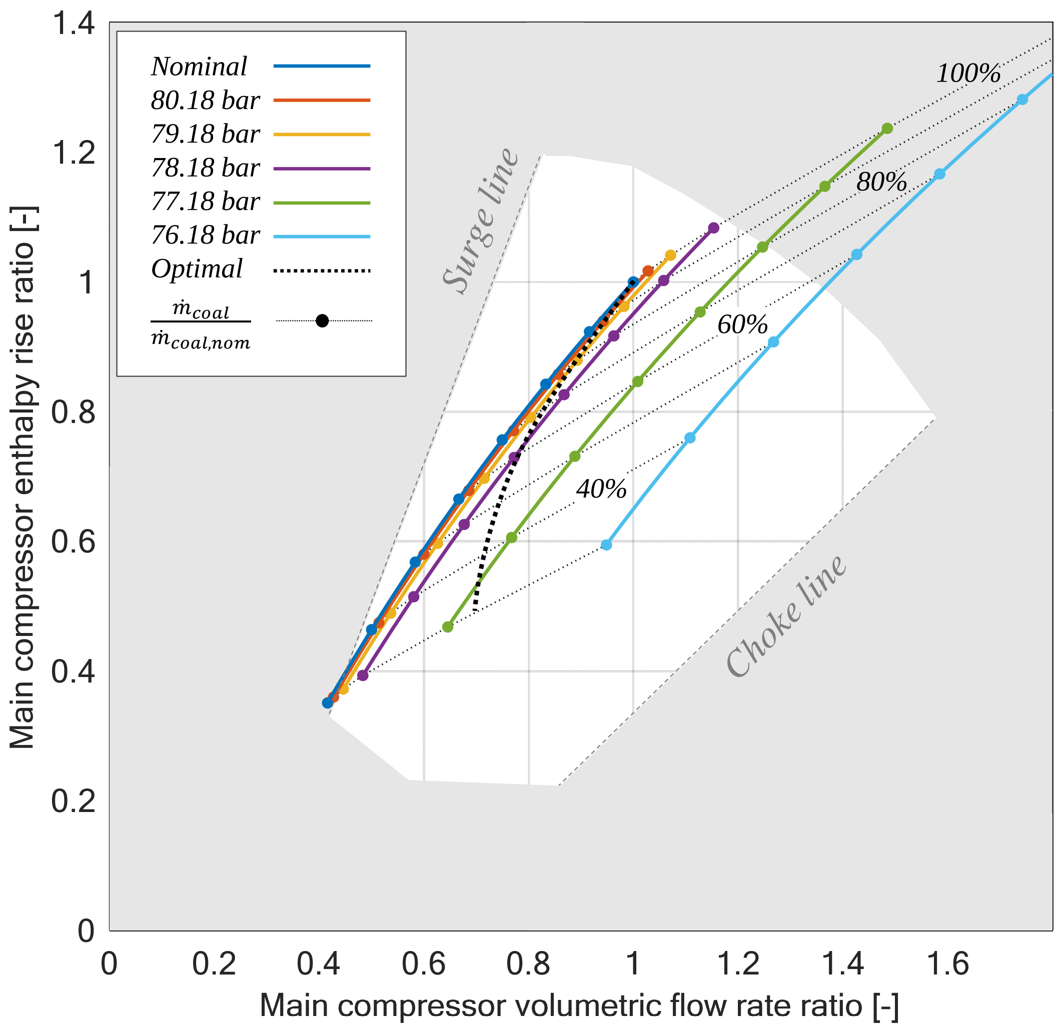
The efficiency increases because of the higher effectiveness of the heat exchangers and the lower fan consumption



Main compressor:

CASE 1 (dark blue line) shows a locus of operative points almost parallel to the surge line in a region of high efficiency with a slight departure towards the choke line with possible penalization of compressor efficiency only for very low coal mass flow rates

CASE 2 (dotted black line) mitigates this problem keeping the point in a region of high efficiency

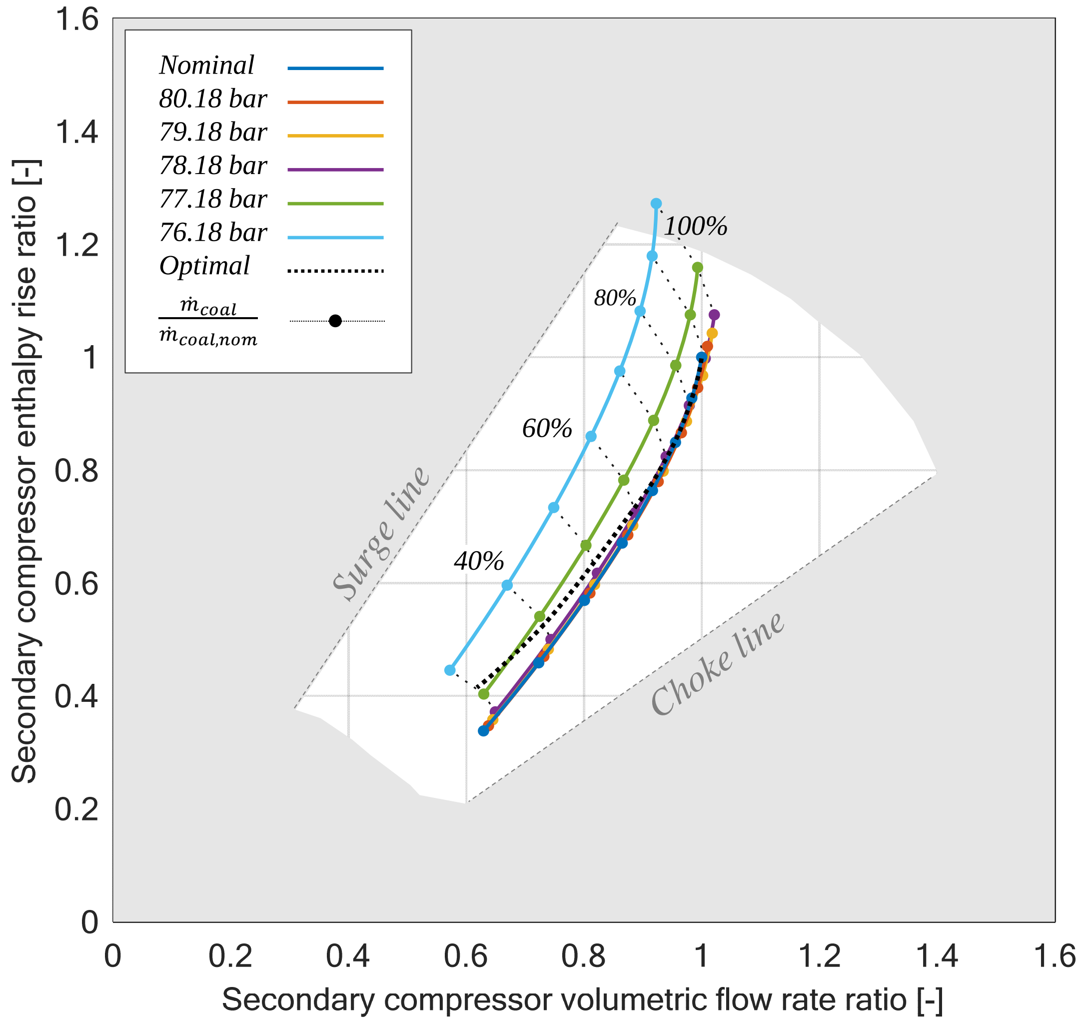


Considerations on secondary compressor design

Secondary compressor:

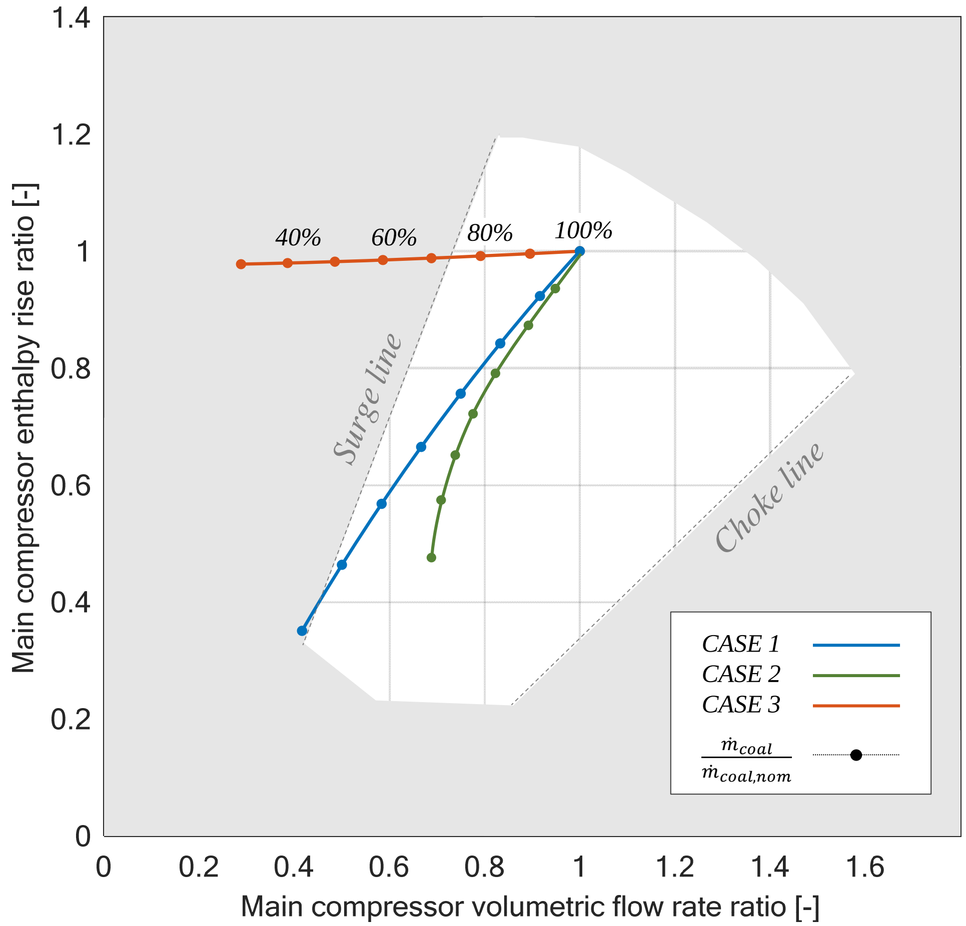
Adopting strategy of CASE 1 or CASE 2 does not affect appreciably the operative point of the turbomachine

The secondary compressor working point never fall out of the operative region but get closer to a low efficiency region below 50% of the load → this problem is mitigated by CASE 2



Considerations on compressors design for CASE 3

For CASE 3 the pressure ratio remains almost unchanged (minor effect of pressure drops) and operative points are rapidly pushed towards the surge line → impossibility to operate the compressor as they are



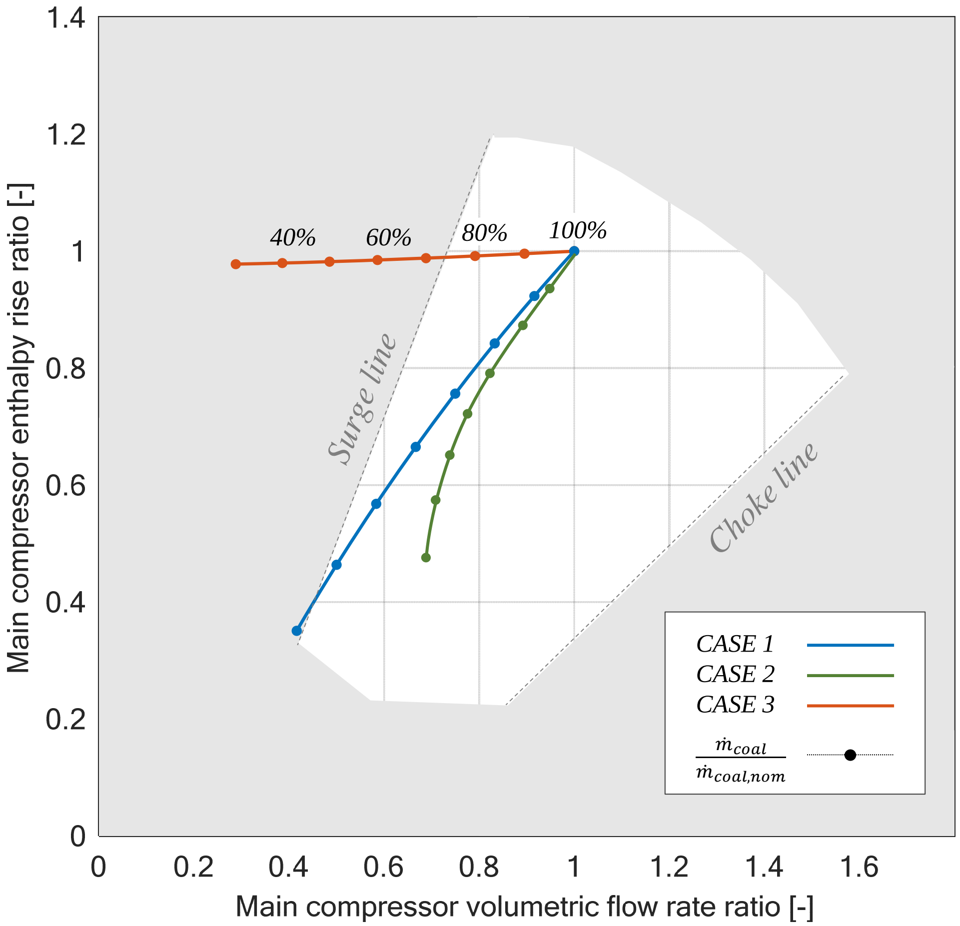
Considerations on compressors design for CASE 3

For CASE 3 the pressure ratio remains almost unchanged (minor effect of pressure drops) and operative points are rapidly pushed towards the surge line → impossibility to operate the compressor as they are

A possible solution is to use more than one compressor in parallel

↓

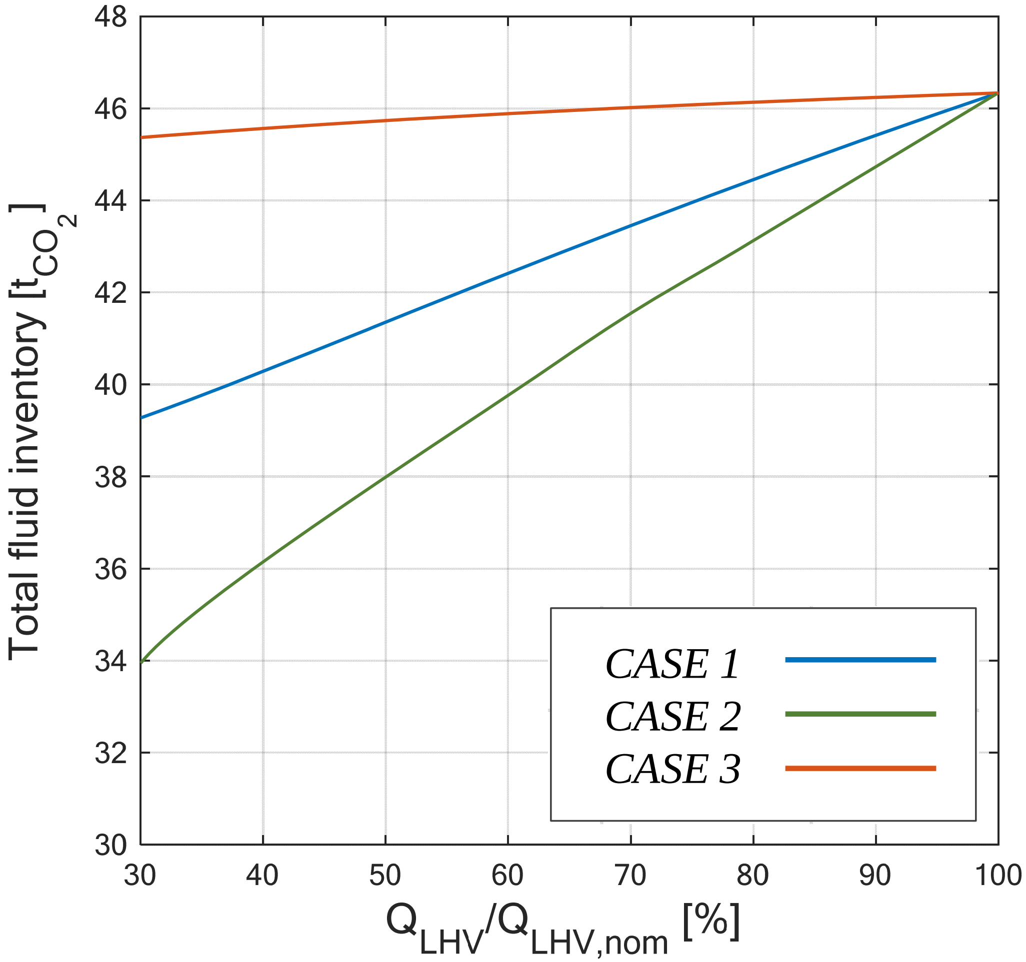
Smaller compressors with lower efficiency!



Considerations on CO_2 inventory

The adoption of a CO_2 storage tank is necessary in all the control strategy but the size of the tank can be very different

If the maximum pressure is kept constant (CASE 3) the inventory variation just slightly decreases at partial load because of operative temperature variation, with potential advantages in terms of dynamic control of the system and in terms of investment costs
→ Small storage tank needed

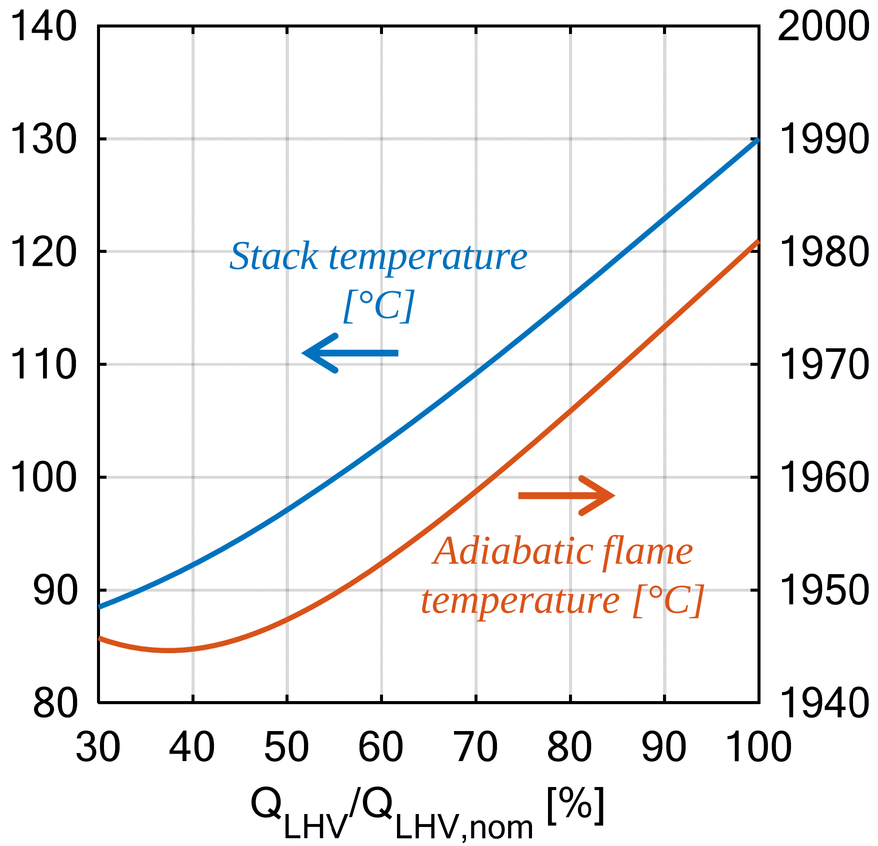


Considerations on flue gases stack temperature

At part load the flue gases stack temperature decreases below the limit value due to the increased efficiency of the heat exchangers

Possible solutions:

- Ljungstrom bypass of a fraction of the combustion air → this solution is often not permitted
- A decrease of Ljungstrom rotational speed to reduce the global htc of the heat exchangers



Conclusions and future steps

Conclusions

Part load analysis of sCO_2 power plants is a topic poorly investigated in literature and guidance and/or benchmarks are still missing

In all the investigated cases it is necessary to solve the problem of flue gases stack temperature

For this specific case the use of IGV on the turbine is not recommended since it involves difficulties in the compressors operation due to surge limitation

The part load optimization of the cycle minimum pressure allows to increase the plant efficiency at low loads, but it results in a stronger inventory variation and the need of a larger sCO_2 storage

Future steps

Investigation of the impact of maximum cycle temperature variation and split fractions in part load

Implementation of performance maps for turbomachinery in order to account for components efficiency variation in part load strategy definition

Introduction of other possible control options such as anti-surge valves, air preheater bypass and RPM regulation, …

Broaden the study to other possible cycle layouts and to other possible applications (WHR, solar tower CSP, …)

Thanks for the attention!

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

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Compact turbo-machines	Low pressure ratios and high molecular mass allows to design compact turbomachines order of magnitude smaller than steam turbines for the same rated power
Reduced inertia	Possibly reduced thermal inertia of the heat exchangers and low rotational inertia for turbomachinery allowing to operate the plant with fast transients