DESIGN CONSIDERATIONS OF SCO_2 TURBINES DEVELOPED WITHIN THE CARBOSOLA PROJECT

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Forschungsprojekt CARBOSOLA

Technische Universität Dresden
HZDR Helmholtz Zentrum Dresden Rossendorf
DLR

Gefördert durch:
Bundesministerium für Wirtschaft und Energie
aufgrund eines Beschlusses des Deutschen Bundestages

Design considerations of sCO_2 turbines developed within the CARBOSOLA project

Agenda

1. CARBOSOLA

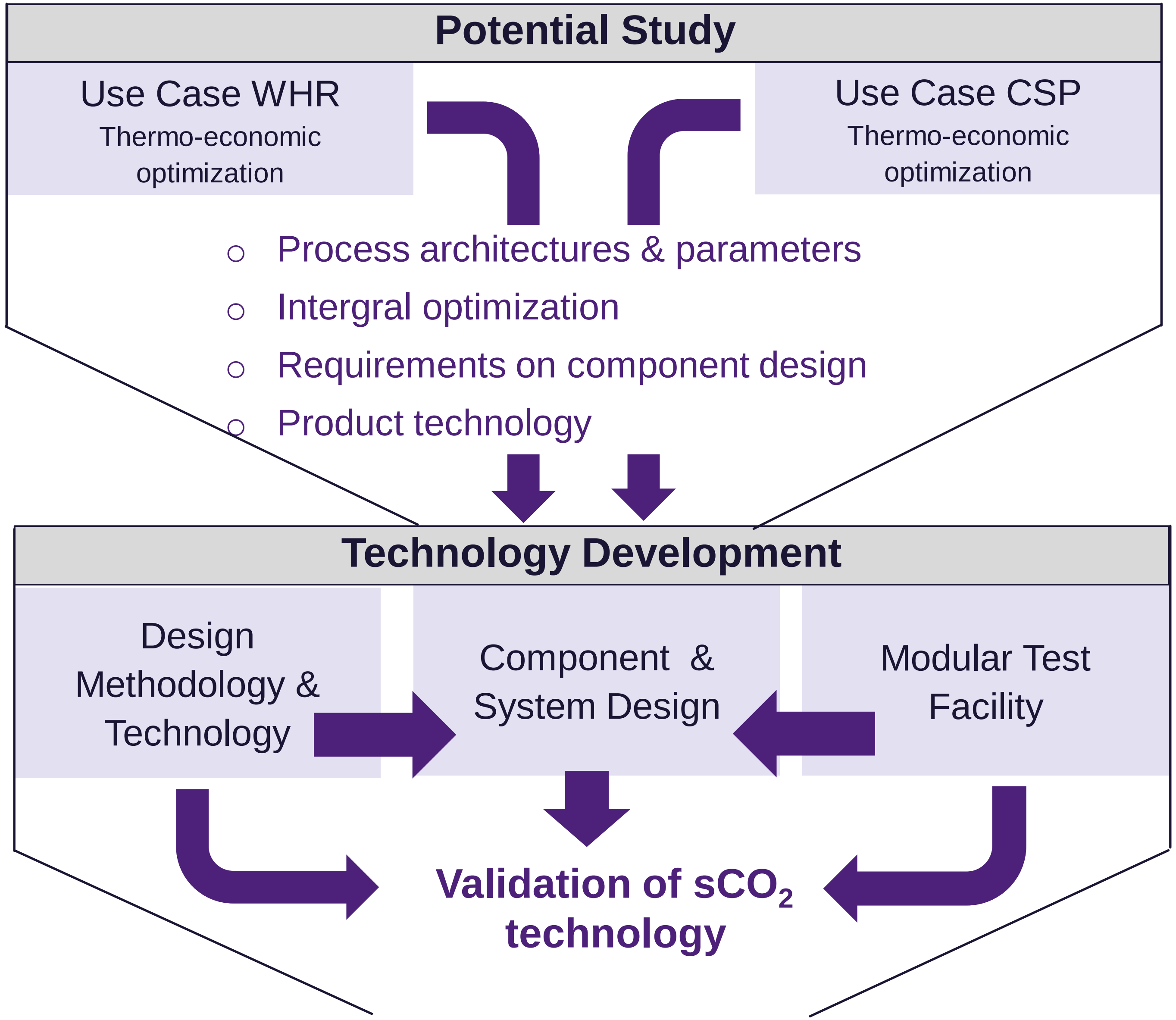
   The german R&D initiative on sCO_2 as an alternative working fluid

2. sCO_2 cycles for waste heat recovery

   Basic considerations, cycle layouts

3. Scaling and optimization of CO_2 turbine
4. Outlook
5. Summary

CARBOSOLA project



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Use case & boundary conditions
Heat source: 2 x SGT-A65

Exhaust gas characteristics (Heat source):

• Pressure:	1,04 bar
• Temperature:	432 °C
• Cold flue gas temp.	≥ 75 °C
• Mass flow:	337 kg/s

Heat exchanger parameters:

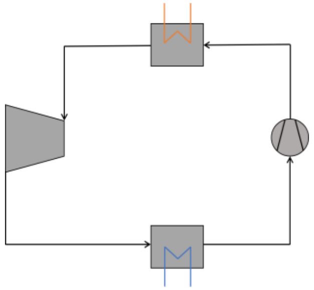
• Pinch heater	≥ 10 K
• Pinch recuperator	≥ 10 K
• Pinch cooler (gaseous)	≥ 10 K
• Approach temp. cooler	≥ 5 K

Wet cooling tower parameters:

• Approach temperature	≥ 3 °C
• Ambient temperature	15 °C (ISO)

Potential sCO_2 cycle architectures

1 Heater 0 Recuperator



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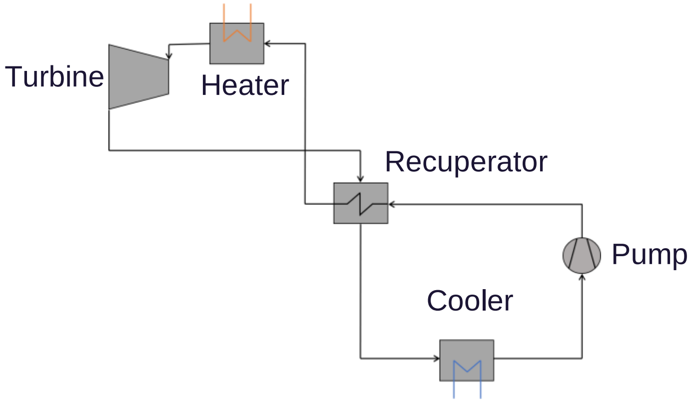
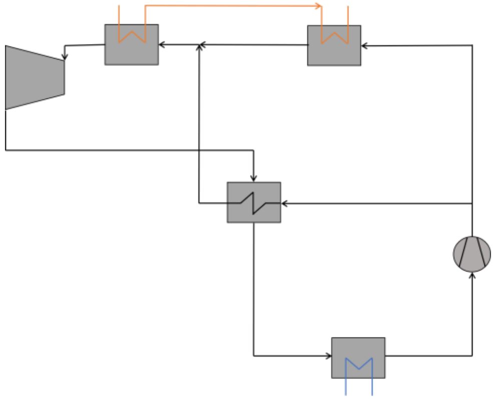
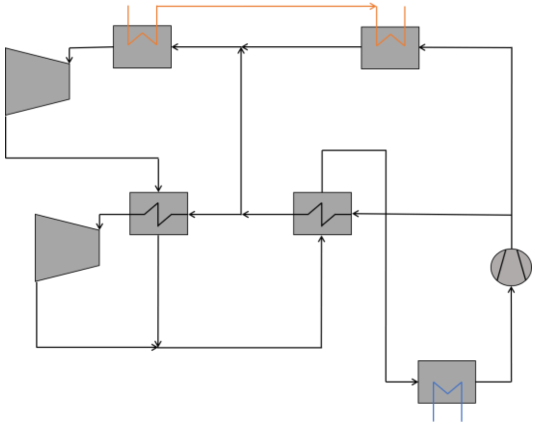


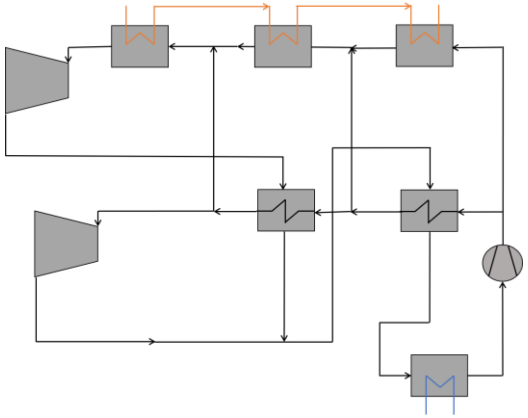
2 Heater 1 Recuperator



2 Heater 2 Recuperator



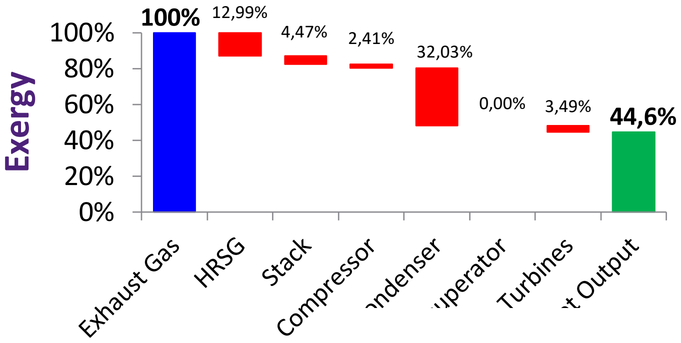
3 Heater 2 Recuperator



Exergy loss analysis

Example: $P_{Turb.\ inlet/outlet}$=300 bar/75 bar; $\Delta T_{Heater/Recu/Cooler}$ = 10K/10K/5 K

1 Heater 0 Recuperator



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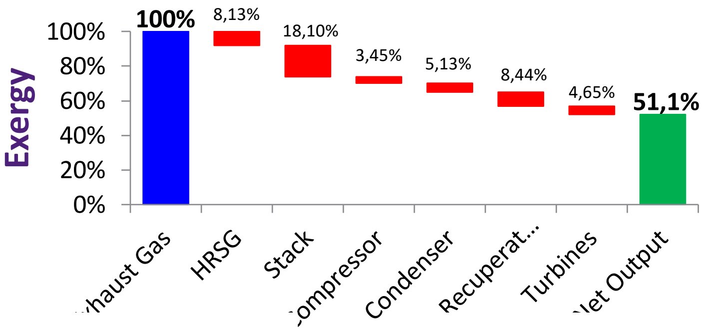

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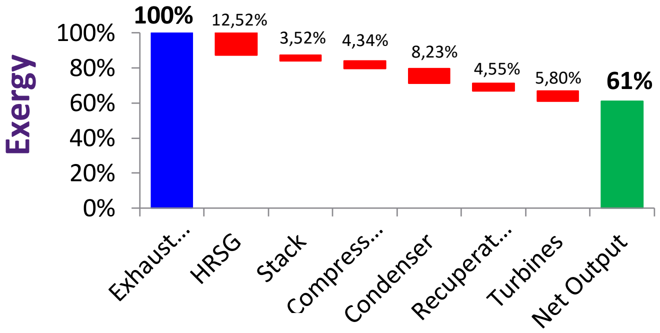

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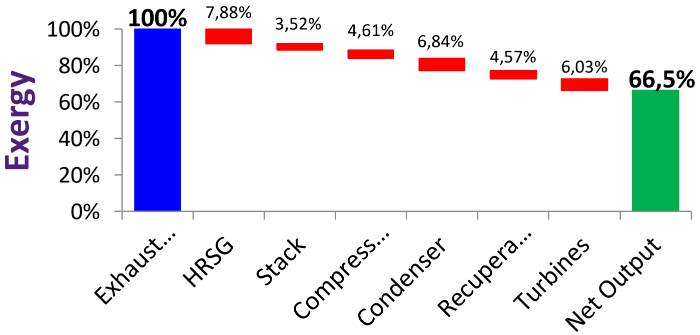

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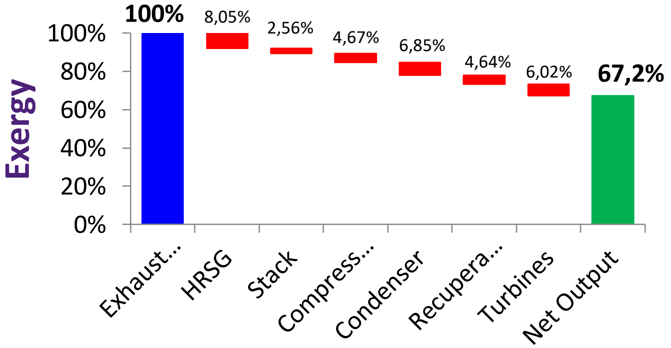

→Most complex cycle architecture leads to max net power output and efficiency but

What is the most economic cycle configuration ?

→ **Answered by Dr. Thiago Gotelip & Prof. U. Gampe**

And L. Heller & R. Buck for CSP-application

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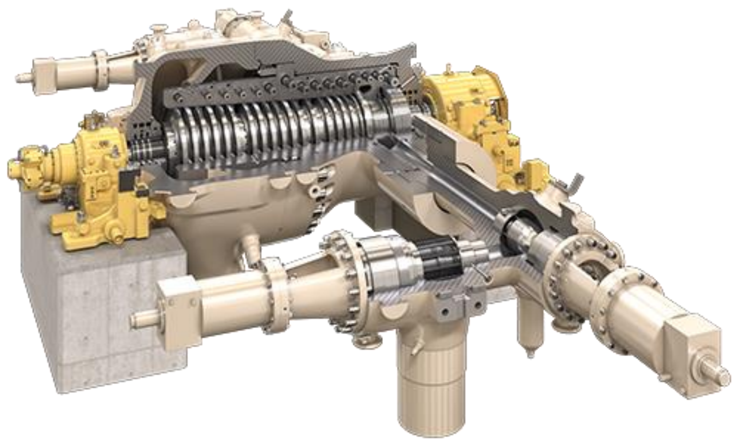
Agenda

Initial design approach:
Scaling of high pressure barrel type turbine topology



Baseline: High efficient barrel type steam turbine

$$f_s = \sqrt{\frac{\dot{V}_{Steam}}{\dot{V}_{CO2}}}$$

$$c_{Steam} = c_{sCO2}$$

Thermal boundary condition	Value
Inlet pressure [bar]	240
Inlet temperature [°C]	350
Mass flow [kg/s]	427
Outlet pressure [bar]	64
Nominal shaft power [MW]	51
Inlet volume flow [m³/s]	2,1
Scaling results	
Speed [1/s]	96
Shaft diameter [mm]	385
Max. length of bladepath [mm]	1300

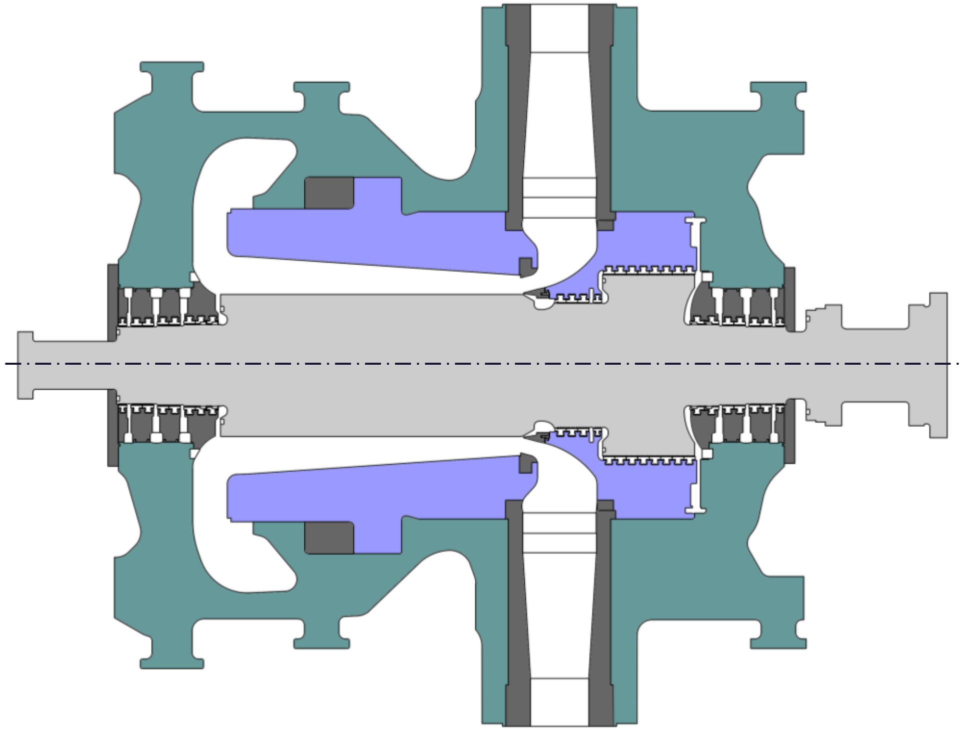
Scaling approach

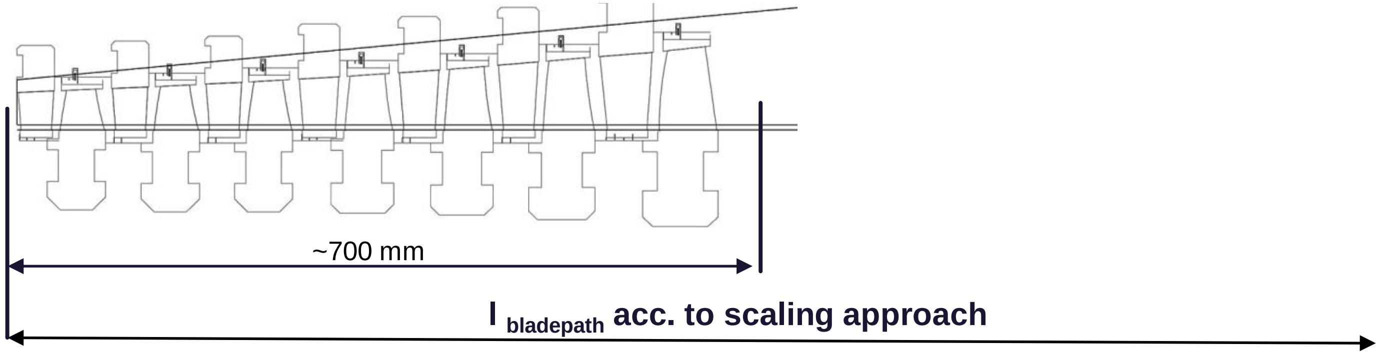
Same flow velocities as reference turbine

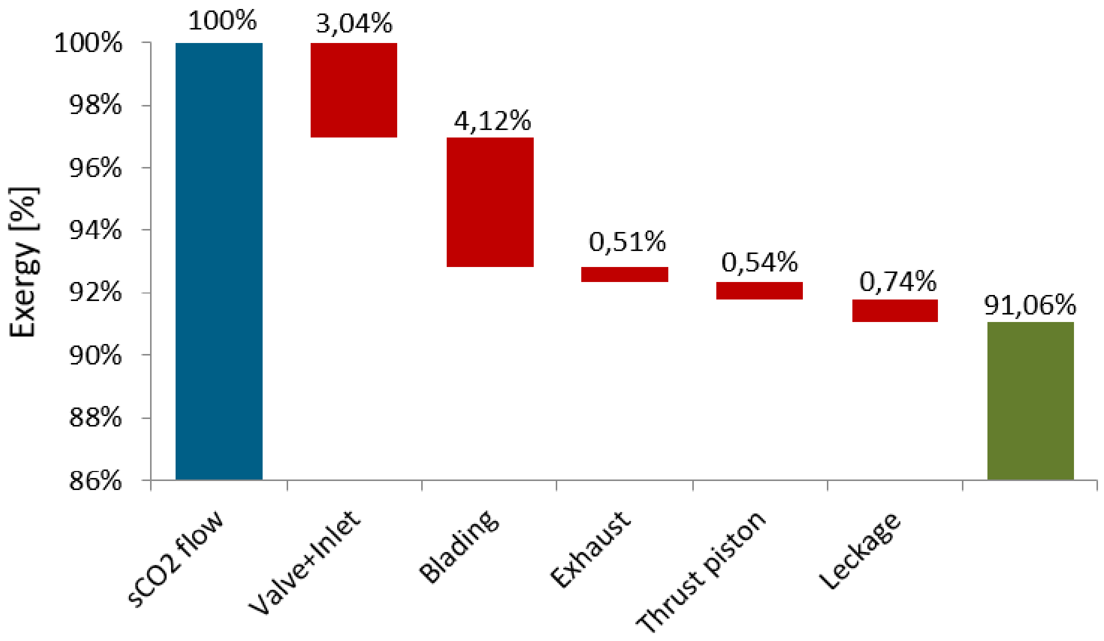
→Approx. same stage efficiencies (~93 %), same centrifugal stresses

→Starting point for further optimization

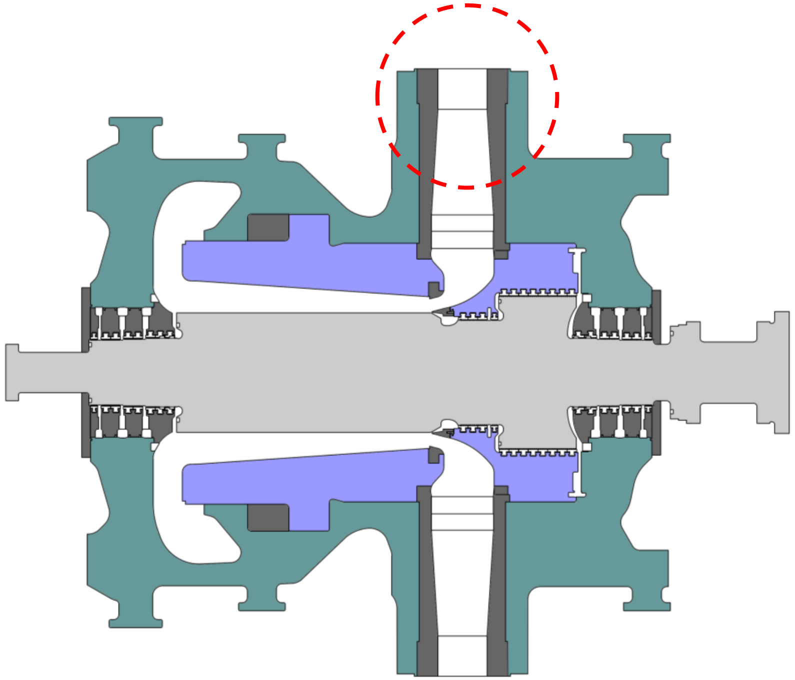
sCO_2 turbine design study
Result of scaling approach

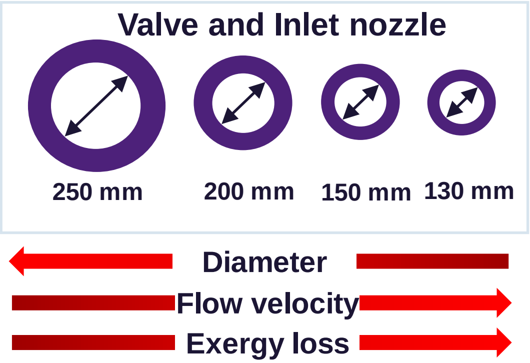


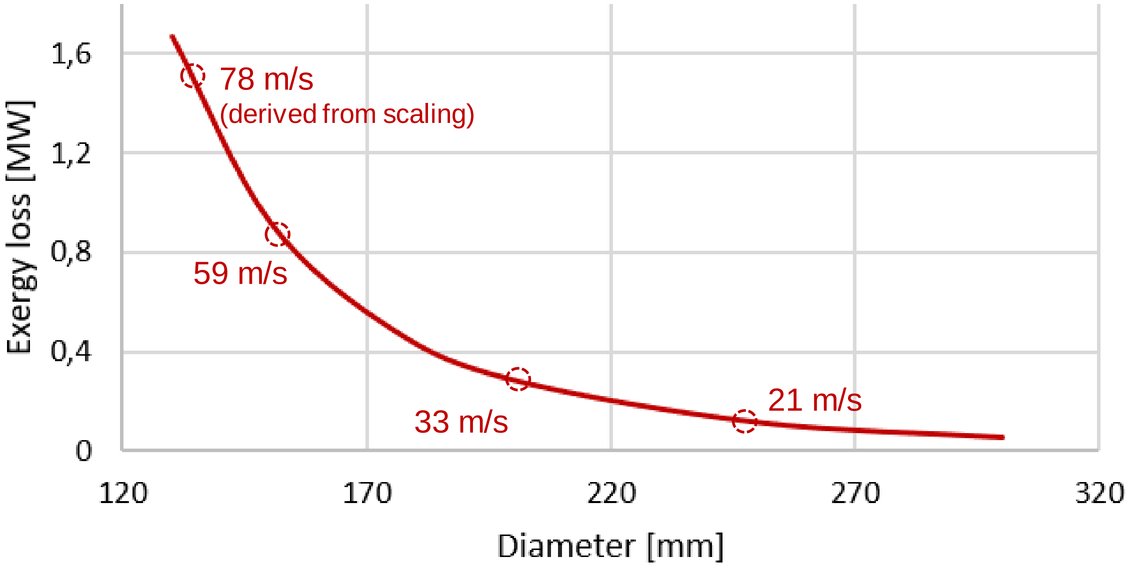




sCO_2 turbine design study
Inlet flow optimization





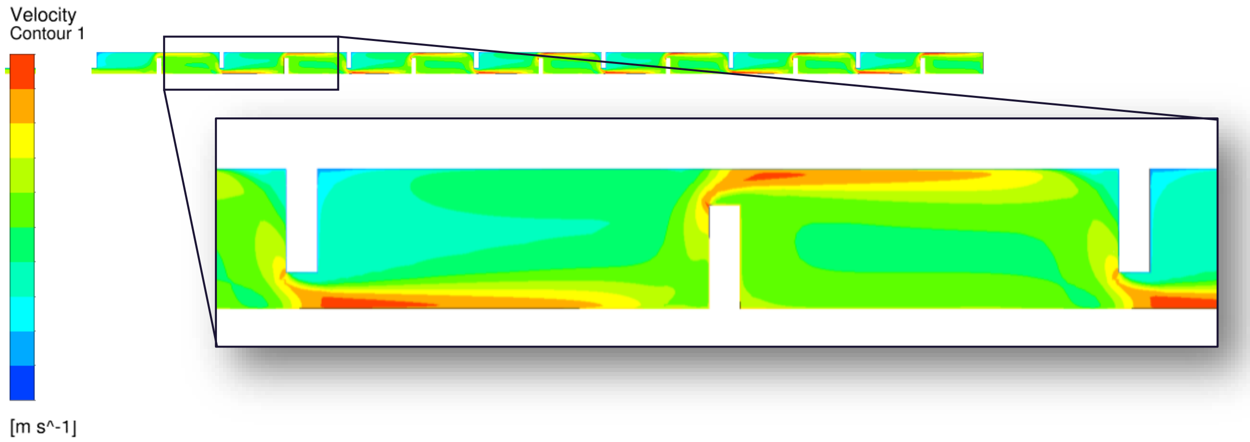


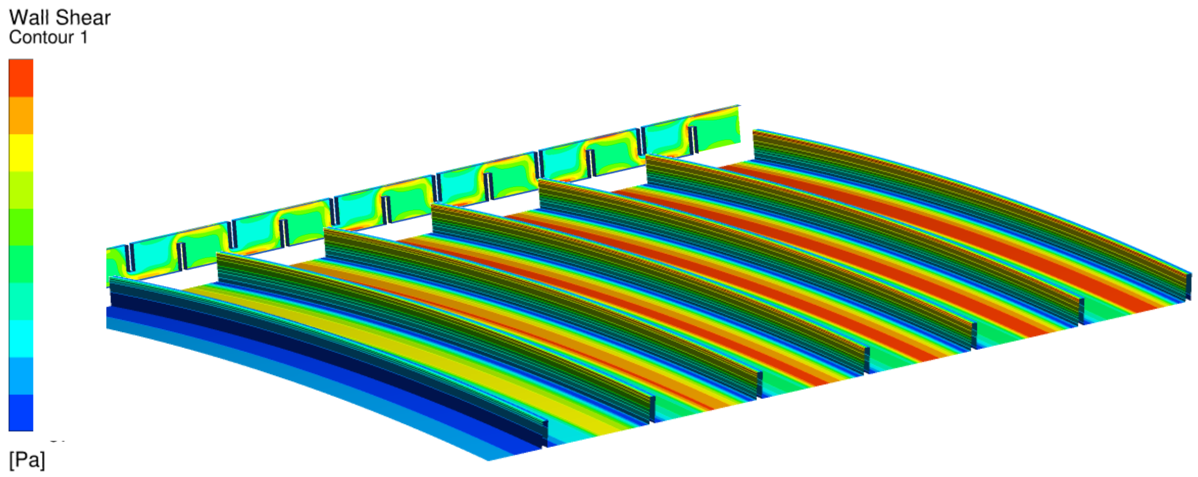
→Performance sensitive to pressure losses

→Thermo-economic optimization will lead to larger flow diameters than known from conventional steam cycles

→200 mm corresponding to 33 m/s was chosen within this project

Digression: Labyrinth seal flow physics
Leakage and windage heating effects





- Pressure difference drives mass flow $\dot{m}$

$$\dot{m} = \underline{a} \cdot \underline{A} \sqrt{\frac{(p_1{}^2 - p_2{}^2) \cdot \rho_1}{p_1 \cdot \underline{n}}}$$

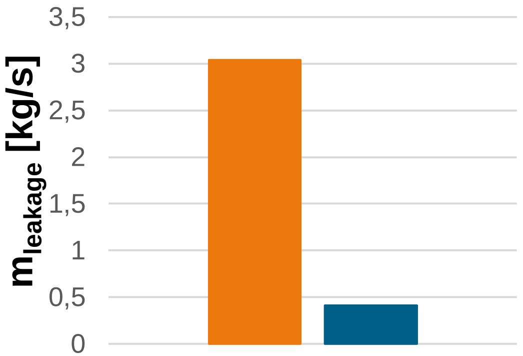
- Circumferential wall shear at rotating shaft is reason for fluid heat up Δh_t

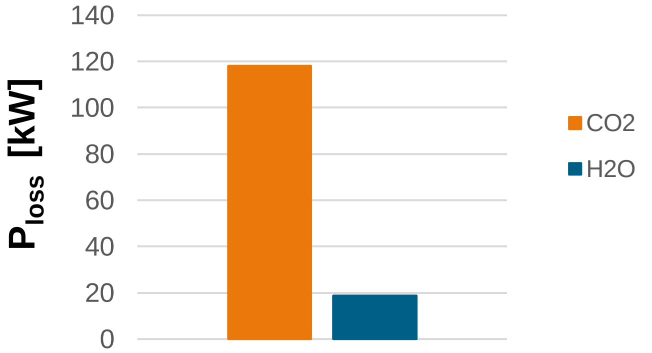
$$\underbrace{M}_{\tau_u \cdot r\,dA} \cdot \omega = \dot{m} \cdot \Delta h_t$$

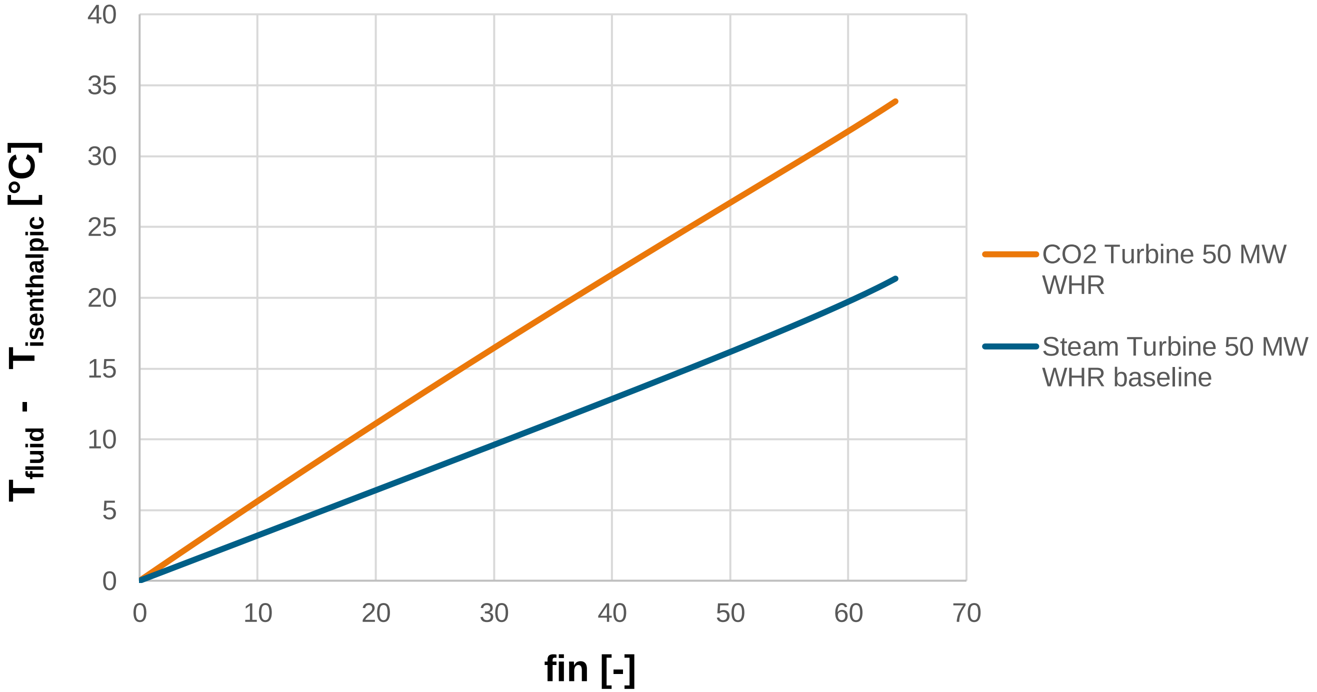
- Empirical model acc. to Hecker et al.:

$$\boldsymbol{\tau} = \text{Const.} \cdot Re^{-0.2} \times 0{,}5\, \boldsymbol{\rho}\, \boldsymbol{w}^2$$

Digression: Labyrinth seal flow physics
Comparison CO_2 vs Steam







Parameters	
Mean labyrinth diameter D [mm]	430
Length of labyrinth L [mm]	384
Rotor frequency f [1/min]	5790

sCO_2 turbine design study
Thrust piston optimization

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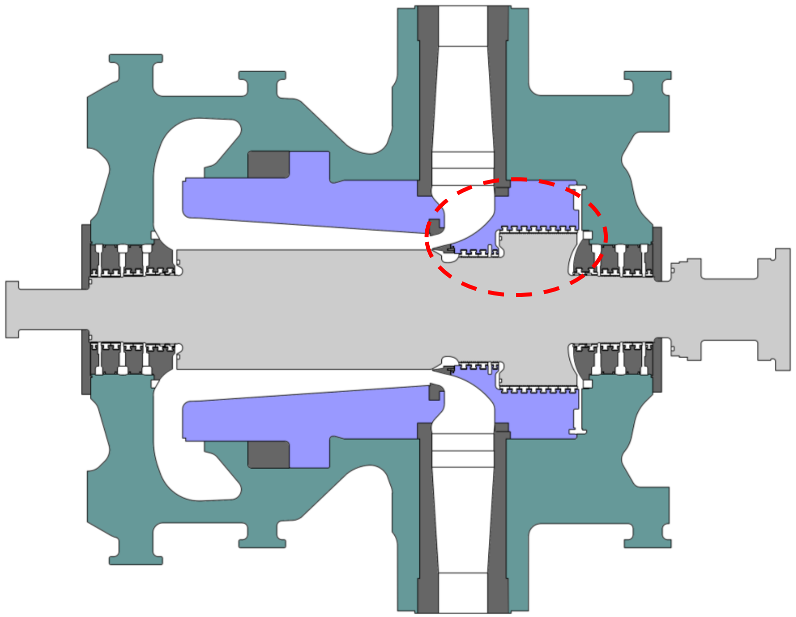
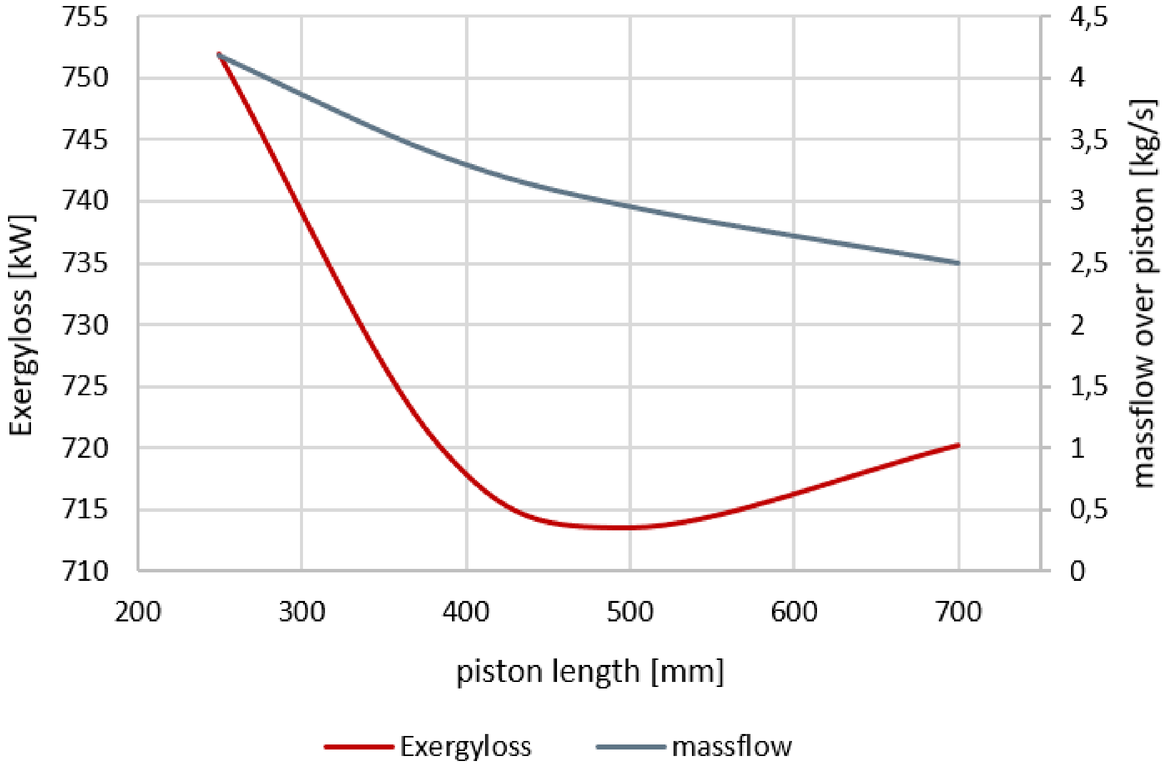


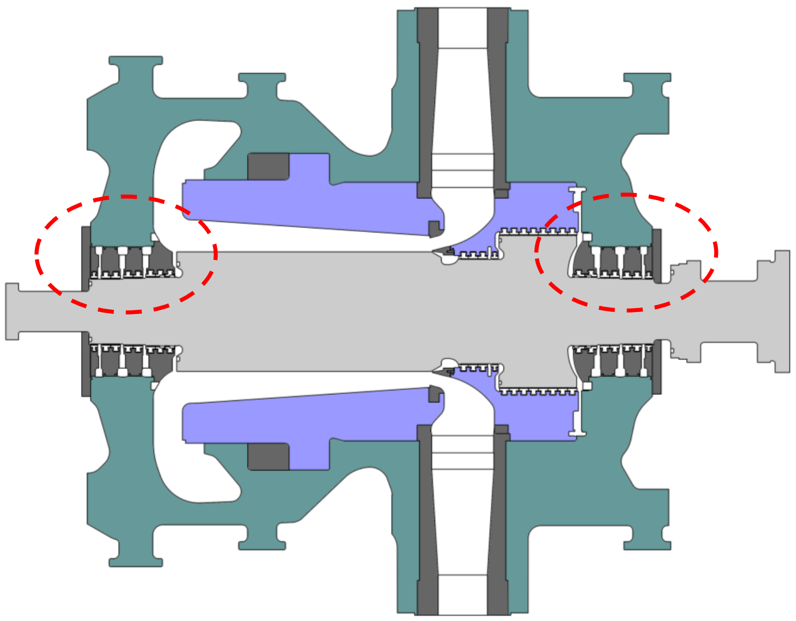


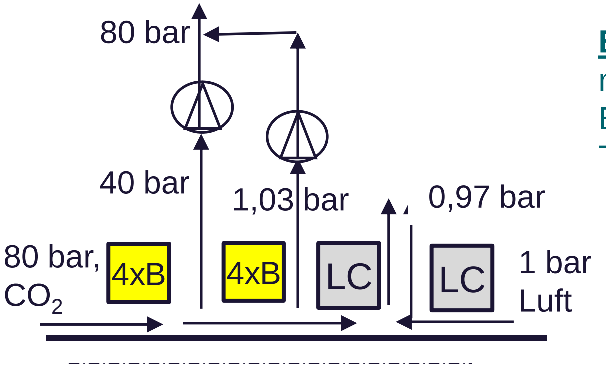
→Minimizing the leakage flow does not correspond to minimizing the exergy losses

→Balancing the thermo-economic optimization with design requirements

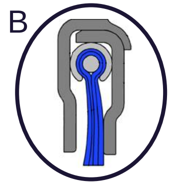
→Piston length of approx. 350 mm was chosen within the design study

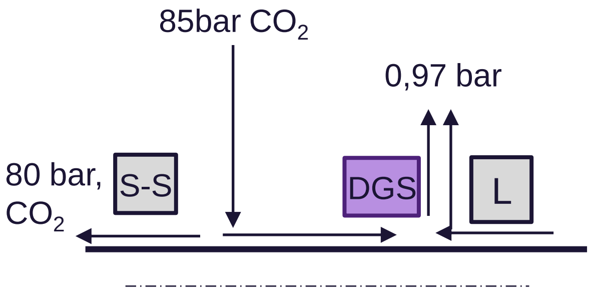
sCO_2 turbine design study
Shaft sealing concepts



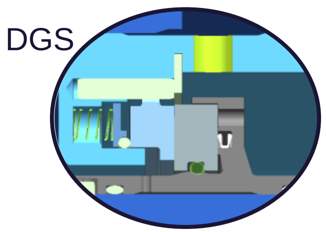


Brushes Labyrinth:
m leak (total) = 1296 kg/d
Exergy loss = 0,478 MW
Turbine efficiency* = 88,78%





DGS:
m leak (total) = 518,4 kg/d
Exergy loss = 0,107 MW
Turbine efficiency* = 89,54%

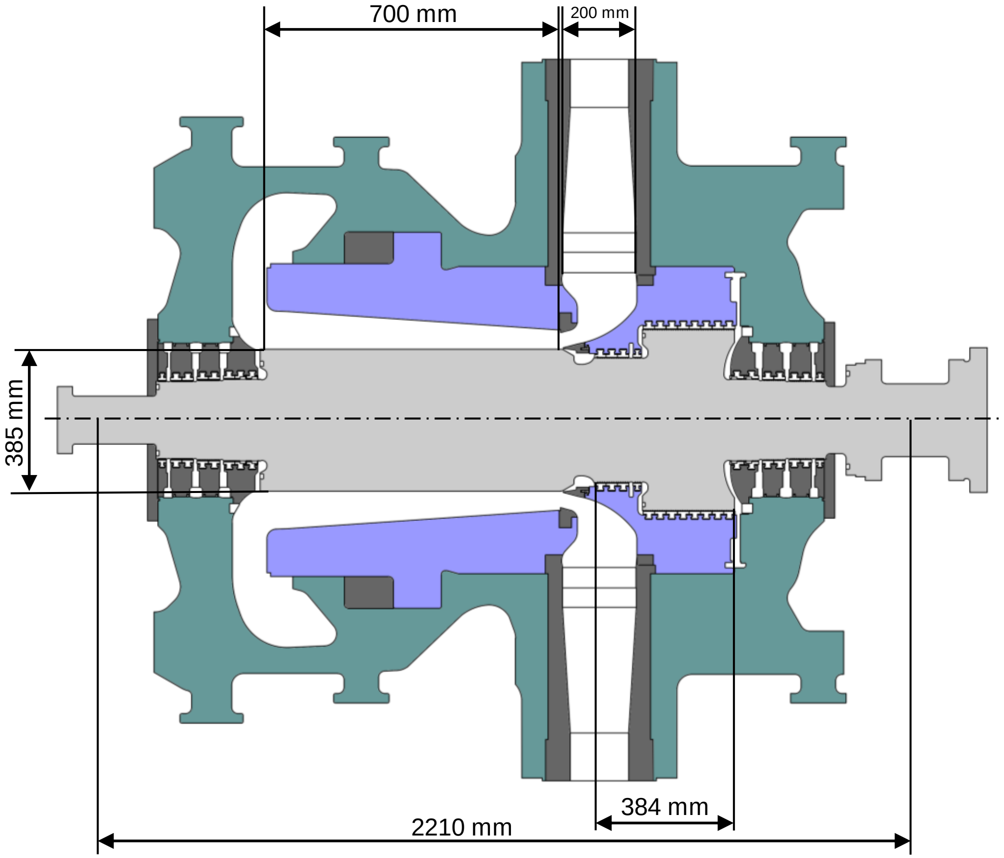


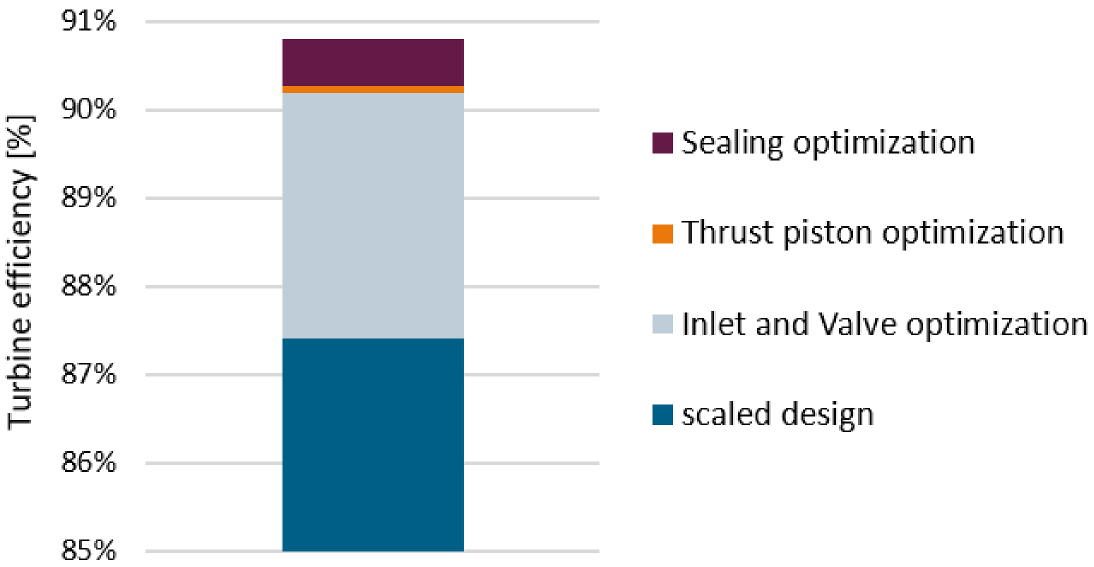
→Dry Gas Seals reduce leakages/losses significantly

→Even with DGS recompression of leakages might be necessary

*Isentropic efficiency with Turbine outlet pressure of 80 bar

sCO_2 turbine design study
Optimization result overview





Design considerations of sCO_2 turbines developed within the CARBOSOLA project

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CARBOSOLA project

Potential Study

Use Case WHR
Thermo-economic optimization

Use Case CSP
Thermo-economic optimization

- Process architectures & parameters
- Intergral optimization
- Requirements on component design
- Product technology

Technology Development

Design Methodology & Technology

Component & System Design

Modular Test Facility

Validation of sCO_2 technology

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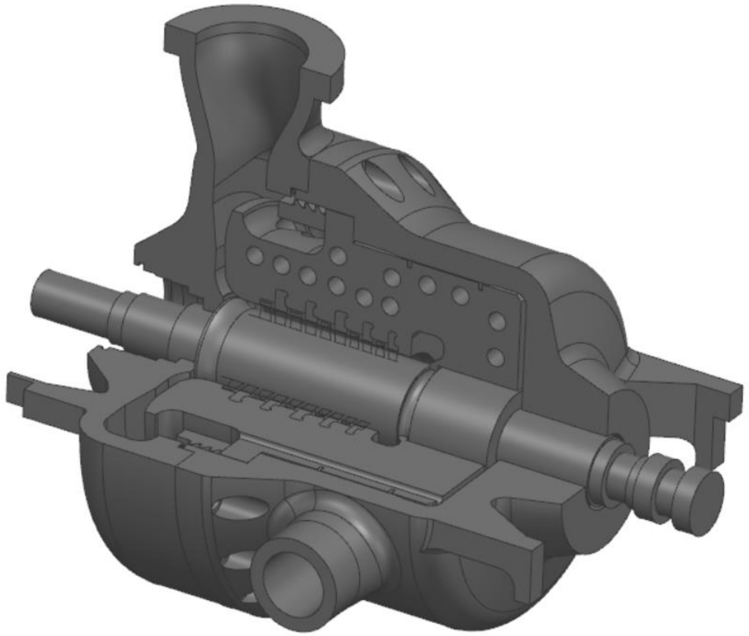
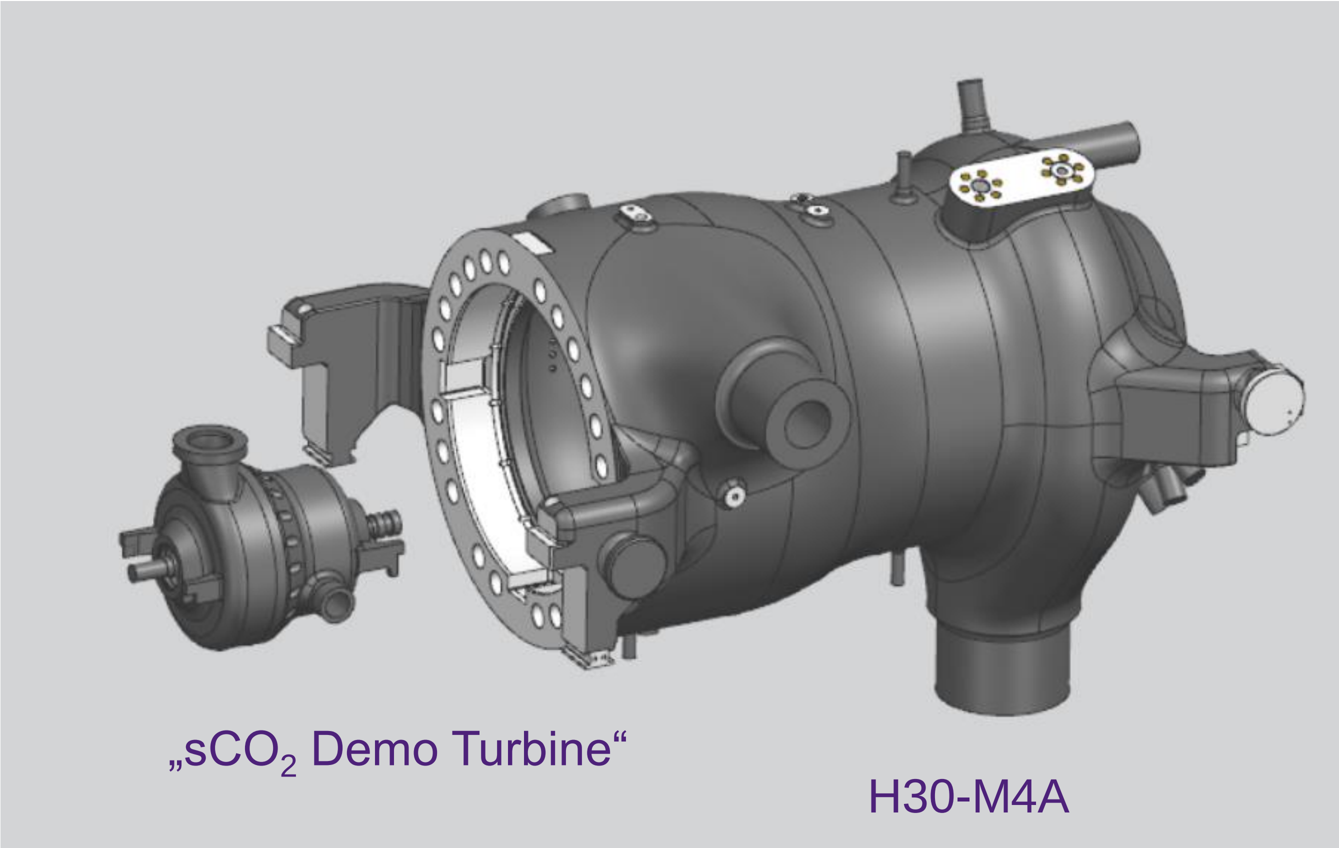
CARBOSOLA
Outlook





→Component & system design of demo plant

→Detailed design of small scale demo turbine has been started

Summary

- CARBOSOLA project has been initiated to drive the sCO_2 technology development in Germany
- Different aspects of the turbine design have been investigated leading to a first design concept
- Optimization of scaled design approach leading to high efficient barrel type turbine
- Due to the specific fluid properties, specific phenomena such as fluid friction, pressure losses, asymmetrical flow conditions and leakages are of greater importance and need to be considered during design process

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Thank you for your attention

Stefan Glos   Patrik Rene Lippe

Dominic Schlehuber   Simon Kobler   Michael Wechsung