Faculty of Mechanical Engineering, Chair of Thermal Power Machinery and Plants

Supercritical CO_2 (sCO_2) as alternative working fluid for wide range of operating temperature – turbomachinery design aspects

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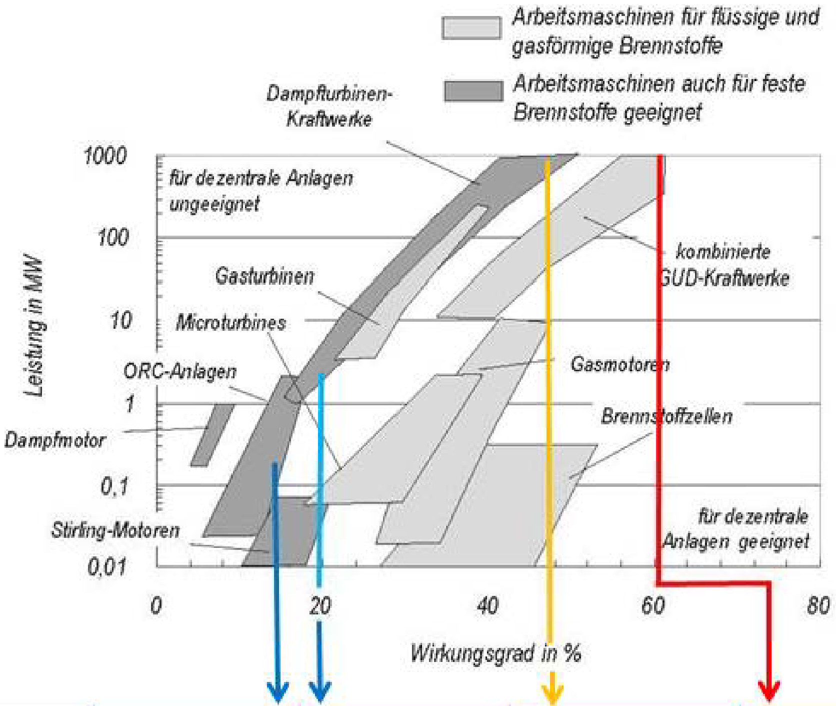
Outline

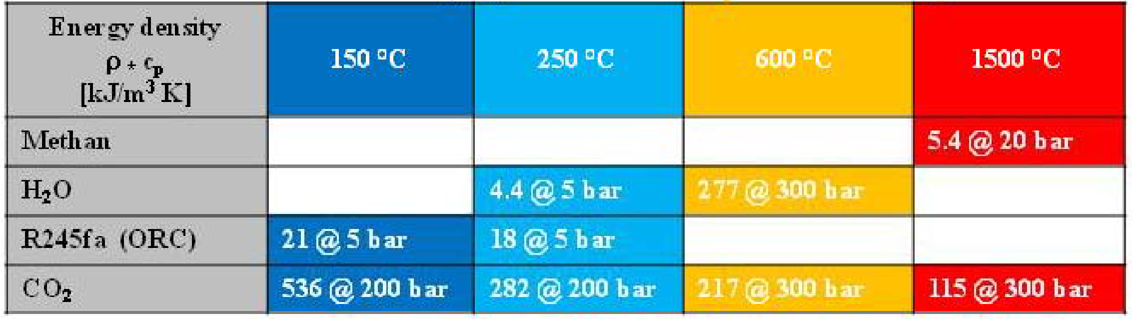
1. Introduction
2. Comparison of thermodynamic cycles
3. Turbine size comparison study
4. Other machinery related aspects and development needs
5. Integrated concept for turbine/generator arrangement
6. Summary

Existing power generation technologies and potential of sCO_2 as alternative working fluid



Range of application of power plant technologies

Ref.: Karl, J. (2012). Dezentrale Energiesysteme. Oldenbourg Verlag München.



Energy density $\rho * c_p$ [kJ/m^{3}K]	150 °C	250 °C	600 °C	1500 °C
Methan				5.4 @ 20 bar
H_2O		4.4 @ 5 bar	277 @ 300 bar	
R245fa (ORC)	21 @ 5 bar	18 @ 5 bar		
CO_2	536 @ 200 bar	282 @ 200 bar	217 @ 300 bar	115 @ 300 bar

Comparison of energy density of working fluids at turbine inlet

Development trends in power generation

- Transformation processes in the European power generation sector characterized by

  - **smaller units** (distributed power generation),
  - **higher location independence** (distributed power generation) and
  - **higher operational flexibility**

- Supercritical CO_2 as alternative working fluid opens up new opportunities to meet the changing requirements profile.

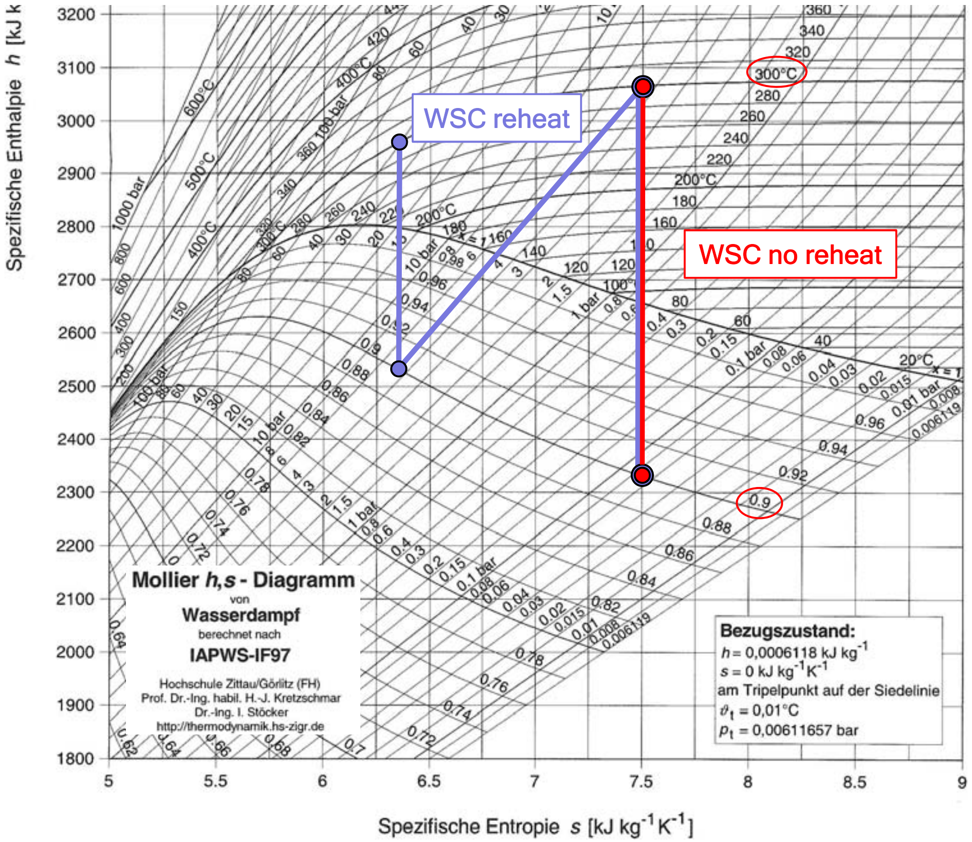
- **Supercritical CO_2** enables to cover a much **wider operating range with the same fluid and at high efficiency**.

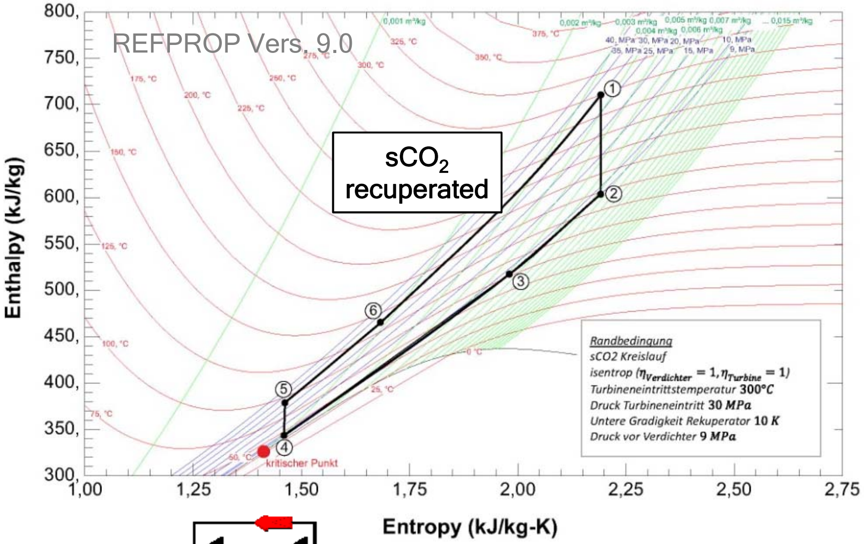
  → Opportunity of **stepwise development** starting with lower fluid pressure, temperature (reducing development risks)
  → Waste heat recovery (WHR), exhaust heat recovery (EHR)

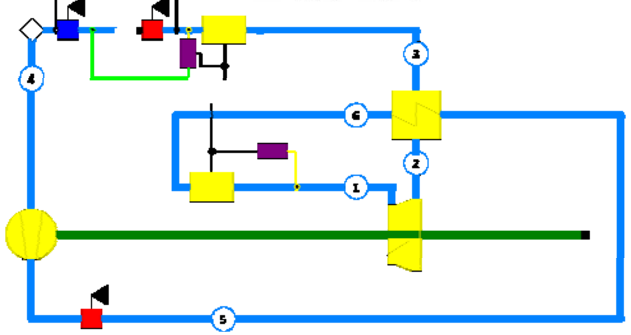
Comparison of thermodynamic cycles

sCO_2 vs. water steam cycles (WSC):

Example for comparison of thermal efficiency η_{th} for perfect cycles
WHR/ EHR application with max. cycle temperature 300 °C

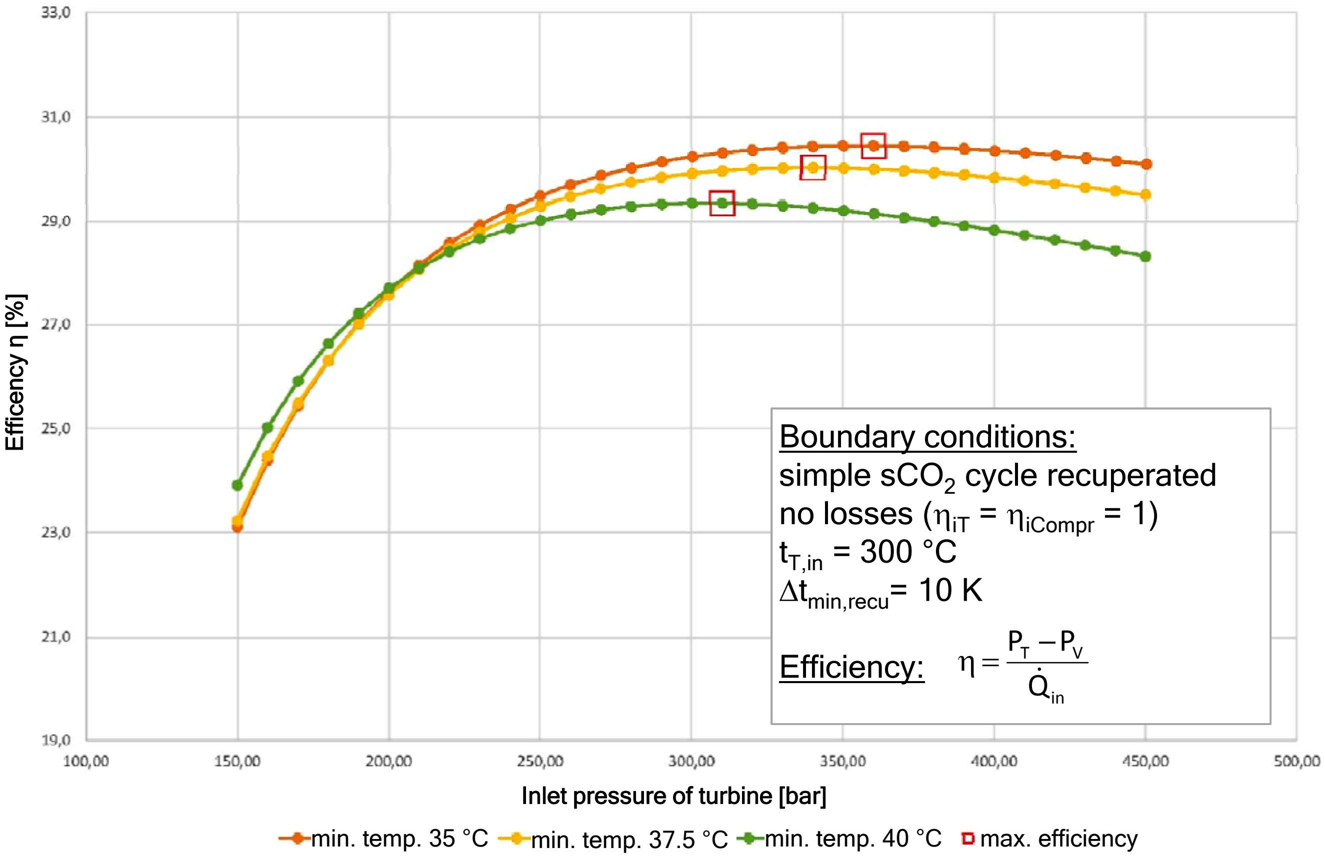






Recuperated sCO_2 cycle:

Optimum turbine inlet pressure for perfect cycle
max. cycle temperature 300 °C



Boundary conditions:
simple sCO_2 cycle recuperated
no losses ($\eta_{iT} = \eta_{iCompr} = 1$)
$t_{T,in} = 300$ °C
$\Delta t_{min,recu} = 10$ K

Efficiency: $\eta = \frac{P_T - P_V}{\dot{Q}_{in}}$

Comparison of thermodynamic cycles

sCO_2 vs. water steam cycles (WSC):

Example for comparison of thermal efficiency η_{th} for perfect cycles
Max. cycle temperature 300 °C

Cycle	Fluid parameters	η_{th}
WSC, no reheat	p_{in} = 0.46 MPa p_{ex} = 7.3 kPa, x = 0.9	25.2 %
WSC, reheat	p_{in} = 4.1 MPa, p_{reheat} = 0.46 MPa t_{reheat} = 300 °C p_{ex} = 7.3 kPa, x = 0.9	34.5 %
sCO_2 recuperated	$p_{T,in}$ = 30 MPa (optimum for assumed boundary conditions) $p_{T,ex}$ = 9 MPa, $t_{T,ex}$ = 172 °C $p_{Cond,in}$ = 9 MPa, $t_{Cond,in}$ = 104 °C $p_{Compr,in}$ = 9 MPa, $t_{Compr,in}$ = 40 °C $p_{Compr,ex}$ = 30 MPa, $t_{Compr,ex}$ = 94 °C	29.3 %

Turbine size comparison study

sCO_2 vs. WSC:

Comparison of **mass flow rate** for **same effective output P_N**

$$\boxed{\left(\frac{\dot{m}}{P_N}\right)_{sCO_2}} : \left(\frac{\dot{m}}{P_N}\right)_{\text{WSC no reheat}} : \left(\frac{\dot{m}}{P_N}\right)_{\text{WSC reheat}} = \boxed{1015} : 100 : 64$$

Comparison of **volume flow rates** for **same effective output** and **flow velocity**

Turbine inlet $$\boxed{\left(\frac{\dot{V}_{in}}{P_N}\right)_{sCO_2}} : \left(\frac{\dot{V}_{in}}{P_N}\right)_{\text{WSC no reheat}} : \left(\frac{\dot{V}_{in}}{P_N}\right)_{\text{WSC reheat}} = \boxed{6.1} : 100 : 6.4$$

Turbine outlet $$\boxed{\left(\frac{\dot{V}_{ex}}{P_N}\right)_{sCO_2}} : \left(\frac{\dot{V}_{ex}}{P_N}\right)_{\text{WSC no reheat}} : \left(\frac{\dot{V}_{ex}}{P_N}\right)_{\text{WSC reheat}} = \boxed{0.5} : 100 : 63.6$$

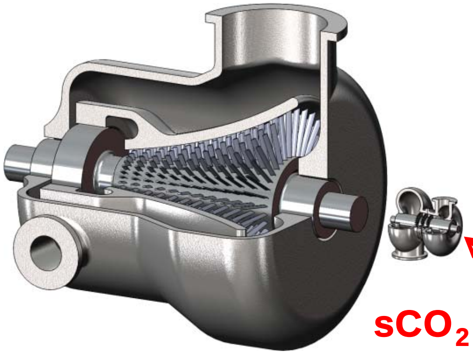
sCO_2 vs. WSC:

Turbine size comparison

(Simplification: isentropic expansion)

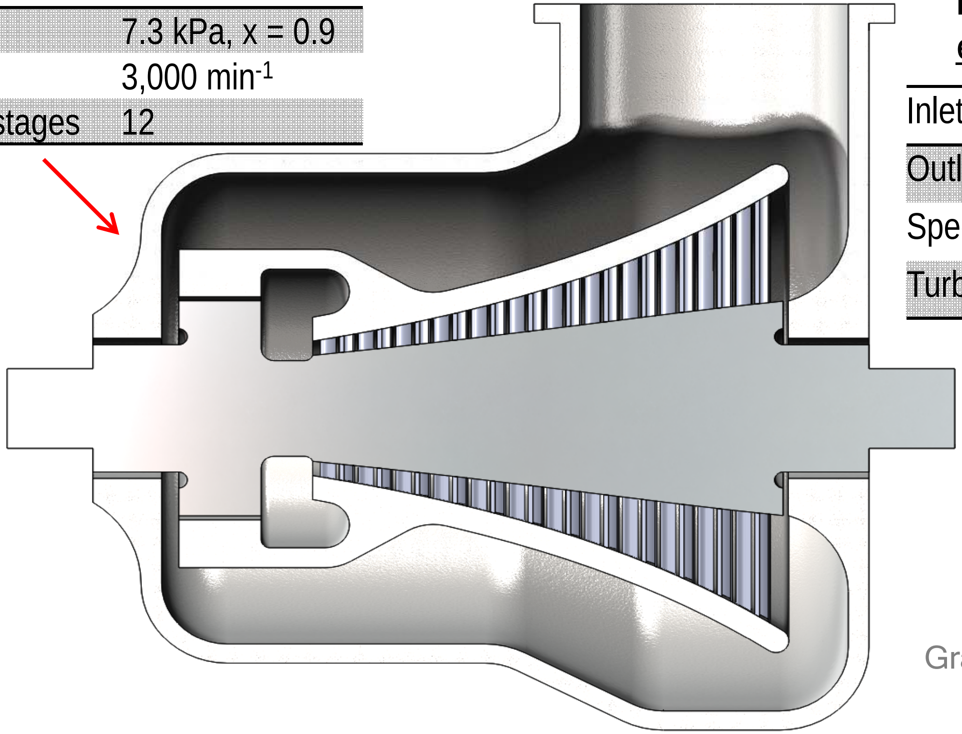
Steam turbine, no reheat
WSC with effective output
P_N = 10 MW

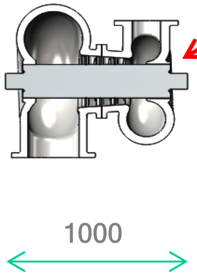
Inlet	0.46 MPa/300 °C
Outlet	7.3 kPa, x = 0.9
Speed	3,000 min^{-1}
Turbine stages	12



sCO_2 turbine
Recuperated sCO_2 cycle with
effective output P_N = 10 MW

Inlet	20 MPa/300 °C	30 MPa/300 °C
Outlet	9 MPa/ 214 °C	9 MPa/ 172 °C
Speed	10,000 min^{-1}	20,000 min^{-1}
Turbine stages	3	2





Graphs: Gampe, Spura (2016) TU Dresden

Other machinery related aspects and development needs

Thrust and radial bearings

Technology	Survey	References
Gas foil	• Small turbomachines (< 3 MW, ≥ 30,000 rpm) • Working fluid as lubricant, lower viscosity than oil • Test loops like those of Sandia supporting also gas foil bearing development • Gas foils → promising solution for high temperature sCO_2 turbomachinery application, **so far no experience for application with sCO_2**	[1] [2] [3]
Hydrodynamic	• Medium/ large turbomachines (> 3 MW, ≤ 30,000 rpm) • Necessary separation of the lubricant oil from working fluid	[1]
Magnetic	Medium turbomachines (1 - 20 MW, 30,000 – 10,000 rpm)	[1]
Hydrostatic	Medium/ large turbomachines (> 20 MW, ≤ 10,000 rpm)	[1]

[1] Musgrove et al. (2015). Fundamentals of Supercritical CO_2. Panel session of ASME Turbo Expo 2015. Montreal.

[2] Wright et al. (2011). Overview of Supercritical CO_2 Power Cycle Development at Sandia National Laboratories. Univ. Turbine Systems Research Workshop. Columbus/ Ohio. Oct. 2011.

[3] Thatte (2011). Performance and Life Characteristics of Hybrid Gas Bearing in a 10 MW Supercritical CO_2 Turbine. Proc. of ASME Turbo Expo 2016, GT2016-57695. Seoul.

Other machinery related aspects and development needs

Inner and outer seals

Technology	Survey	References
Labyrinth seals	• Application of labyrinth seals in compressor of the Sandia sCO_2 test loop • Test facility at UW-Madison for measurement of leakage rate through labyrinth seals and for validation of numerical simulation data • Geometry parameter studies for design of labyrinth seals for sCO_2 application	[4]
Dry gas seals (DGS)	• Investigation of the influence of fluid behaviour near critical point on design and performance of DGS • Challenge: Dynamic instabilities arising from sonic transition in thin sCO_2 films of DGS	[5, 6]

[4] Yuan et al. (2014). Experiment and Numerical Study of Supercritical Carbon Dioxide Flow Through Labyrinth Seals. 4th Int. Symposium – Supercritical CO_2 Power Cycles. Pittsburgh/ Pennsylviania. Sept. 2014.

[5] Zakariya and Jahn (2016). Performance of Supercritical CO_2 Dry Gas Seals Near the Critical Point. Proc. of ASME Turbo Expo 2016, GT2016-56537. Seoul.

[6] Thatte and Dheeradhada (2016). Coupled Physics Performance Predictions and Risk Assessment for Dry Gas Seal Operating in MW-Scale Supercritical CO_2 Turbine. Proc. of ASME Turbo Expo 2016, GT2016-57670. Seoul.

Other machinery related aspects and development needs

Material aspects

Survey	**References**
Key components from materials point of view = components subjected to high pressure and temperature → turbine, heater(s)	e.g. [7]
High-temperature strength and corrosion resistance as most important criterias for material selection	
Corrosion resistance of candidate materials in sCO_2 mainly depending on **operating temperature, sCO_2 purity and alloying elements**	[7, 8, 9, 10]
Little information on corrosion behaviour of candidate materials for higher temperatures (> 600 °C), Ongoing corrosion tests	[9, 10, 11, 12]

[7] Wright et al. (2013). Materials Considerations for Supercritical CO_2 Turbine Cycles. Proc. of ASME Turbo Expo 2013, GT2013-94941. San Antonio.

[8] Musgrove et al. (2015). Fundamentals of Supercritical CO_2 Panel session of ASME Turbo Expo 2015. Montreal. June 2015.

[9] Saari et al. (2014). Corrosion Testing of High Temperature Materials in Supercritical Carbone Dioxide. 4th Int. Symposium - Supercritical CO_2 Power Cycles. Pittsburgh/ Pennsylviania. Sept. 2014.

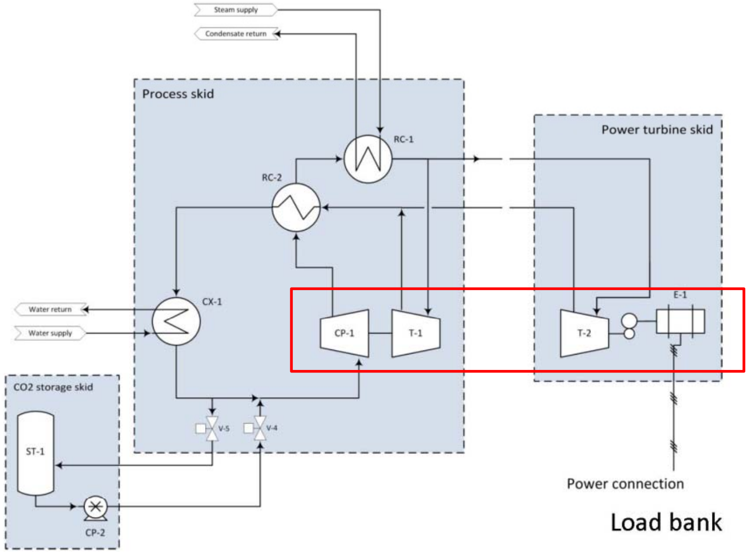
[10] Pint and Keiser (2014). The Effect of Temperature on the s CO_2 Compatibility of Conventional Structural Alloys. 4th Int. Symposium - Supercritical CO_2 Power Cycles. Pittsburgh/ Pennsylviania. Sept. 2014.

[11] Lee et al. (2014). Compatibility of Candidate Materials in High-Temperature s CO_2 Environment. 4th Int. Symposium - Supercritical CO_2 Power Cycles. Pittsburgh/ Pennsylviania. Sept. 2014.

[12] Mahaffey et al. (2014). Materials Corrosion in High-Temperature Supercritical Carbone Dioxide. 4th Int. Symposium - Supercritical CO_2 Power Cycles. Pittsburgh/ Pennsylviania. Sept. 2014.

Other machinery related aspects and development needs

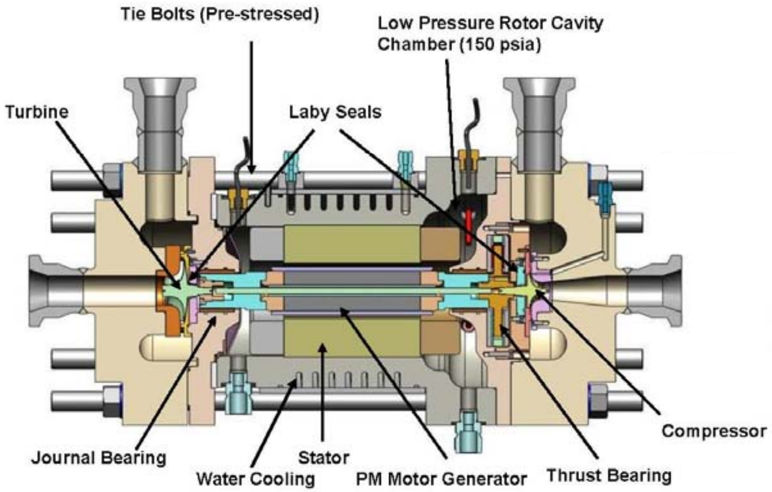
Selected machinery train concepts (simple recuperated sCO_2 cycle)



Source: Held (2014). Initial Test Results of a MW-Class Supercritical CO_2 Heat Engine. 4th Int. Symposium - Supercritical CO_2 Power Cycles. Pittsburgh/ Pennsylviania. Sept. 2014.

Echogen EPS100

Heat recovery engine 7.3 MW net
Twin-shaft concept:
var. speed compressor-turbine-unit;
const. speed power turbine (30,000 rpm)
+ gearbox & synchronous gen. (1800 rpm)



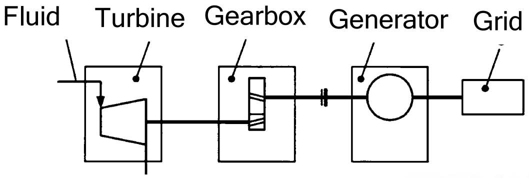
Source: Wright et al. (2011). Overview of Supercritical CO_2 Power Cycle Development at Sandia National Laboratories. Univ. Turbine Systems Research Workshop. Columbus/ Ohio. Oct. 2011.

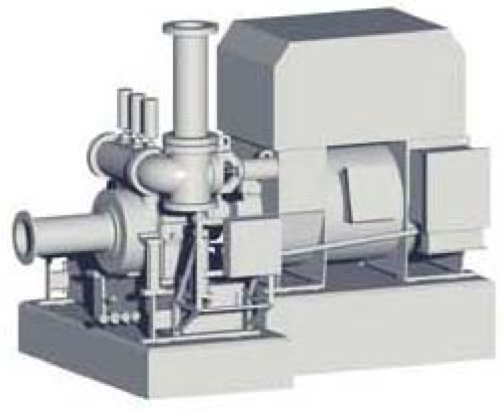
Sandia test loop

125 kW, 75,000 rpm
Single-shaft concept with permanent magnet generator in centre arrangement

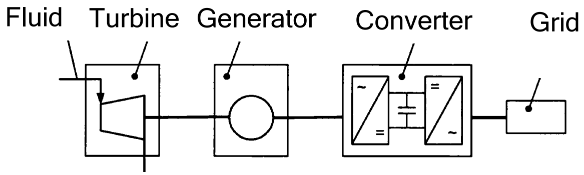
Integrated concept for turbine/generator arrangement

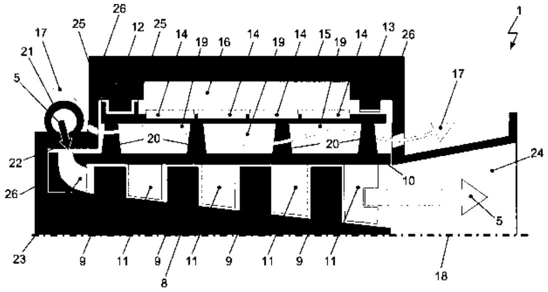
Traditional design of small capacity turbo-generator sets	Proposed integrated concept x)





Graph: Siemens, Pre-designed steam turbines, 2016



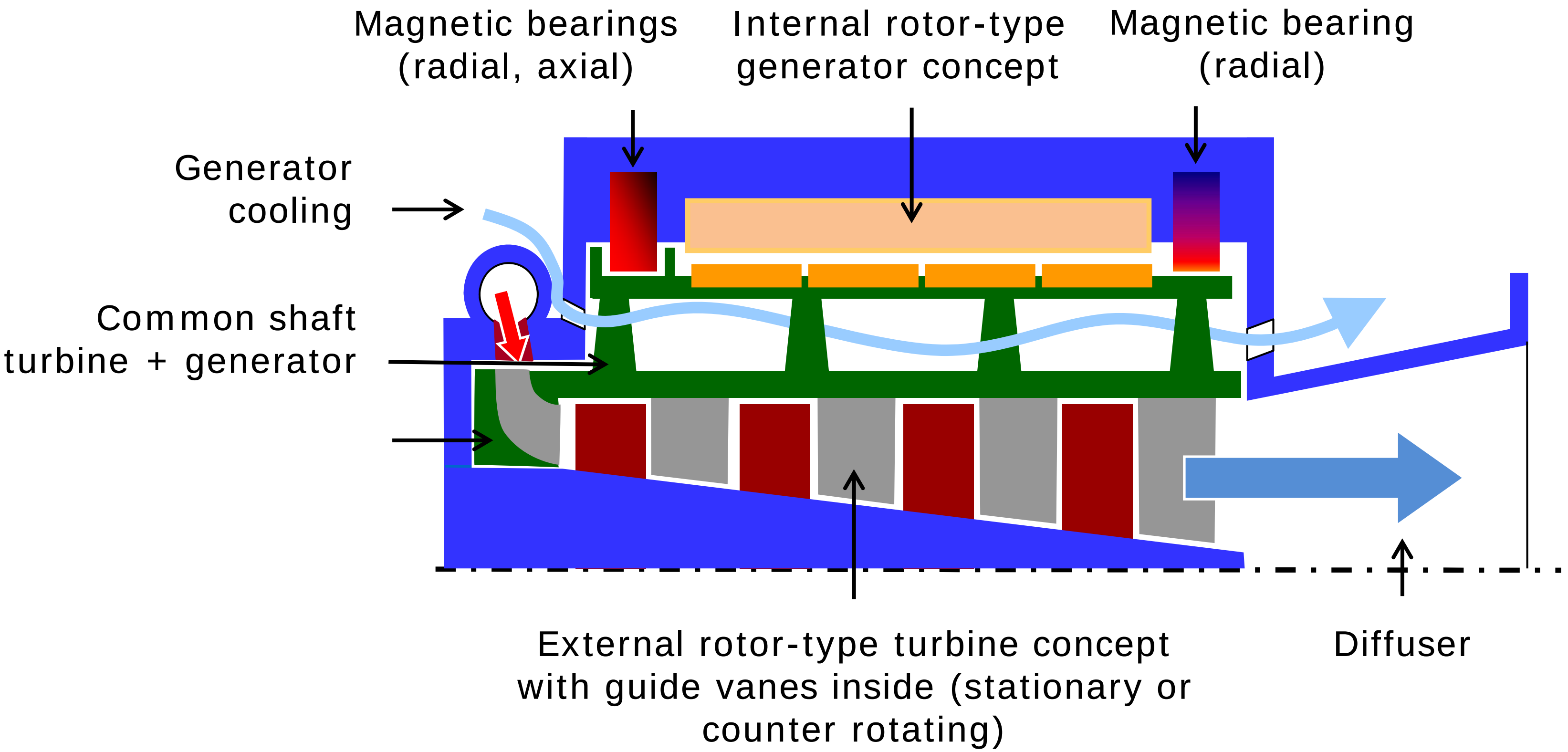


x) Gampe, U.; Hofmann, W.; Hildebrand, N.; Buchheim, G.; Bernet, St. (2012). *Turbo-Generator-Einheit*. Publication of the patent application DE 10 2011 000 420 A1 2012.08.02, patent appl. TU Dresden.

Large size and weight of power unit with • high speed turbine • gearbox • low speed generator • auxiliaries (oil supply system etc.)	• High speed turbine-generator set, no gearbox, variable speed at part load → expected increase of el. efficiency $\geq$ 2 pp @ DP and 10 pp for part load (turbine capacity size $\leq$ 5 MW) • „Oil-free“ design via magnetic bearings

Integrated concept for turbine/generator arrangement

Concept features


Magnetic bearings (radial, axial)

Internal rotor-type generator concept

Magnetic bearing (radial)

Generator cooling

Common shaft turbine + generator

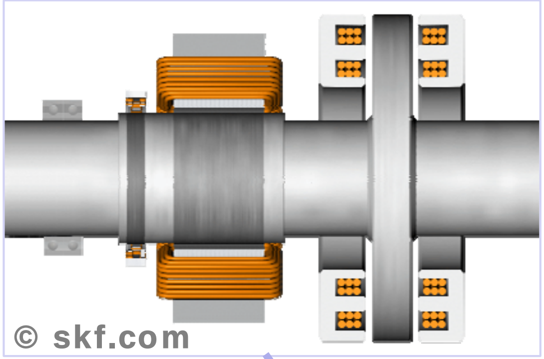
External rotor-type turbine concept with guide vanes inside (stationary or counter rotating)

Diffuser

Integrated concept for turbine/generator arrangement

Concept features

magnetic bearings



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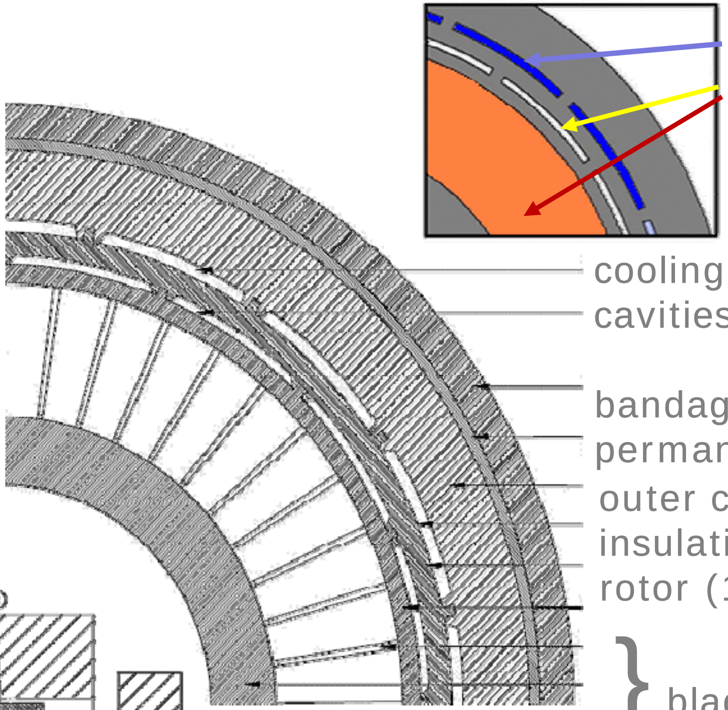
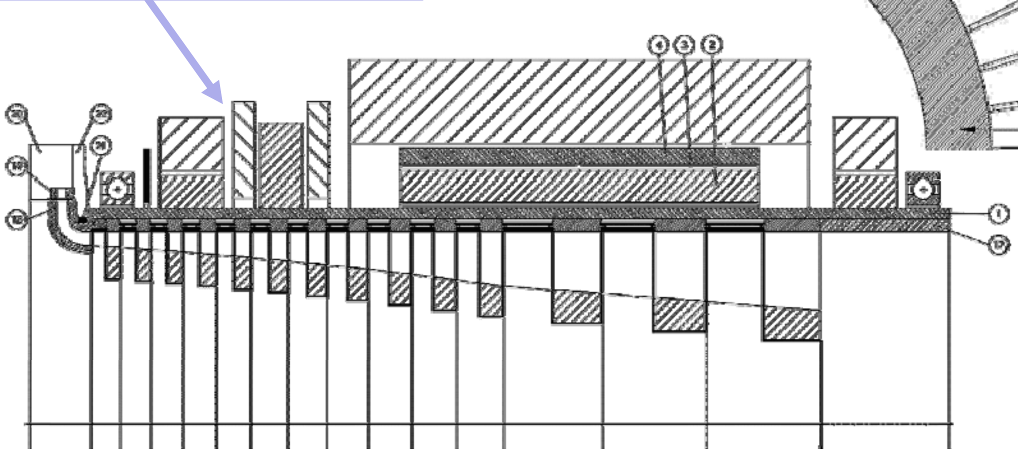


cooling fluid
cavities
steam

cooling fluid
cavities (isolation)

bandage (4)
permanent magnets (3)
outer cylinder (2)
insulating layer
rotor (1)

} blading



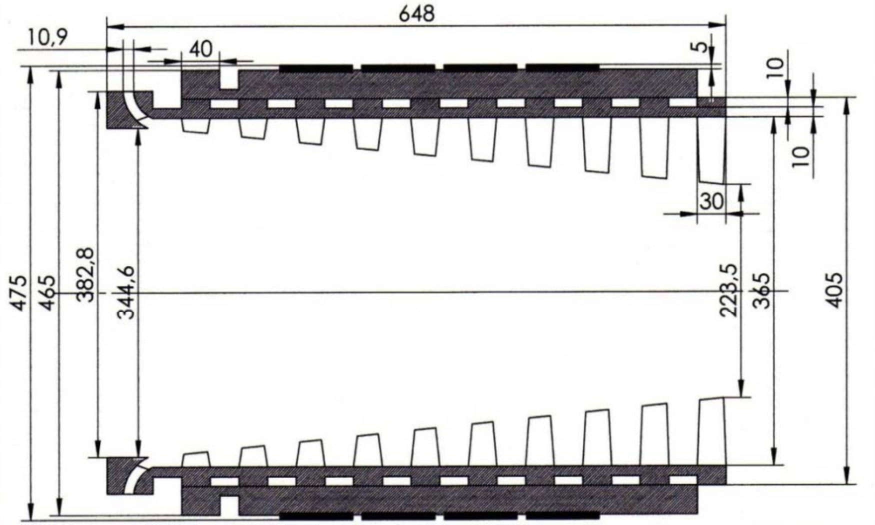
Concept study of backpressure turbine with integrated generator (WSC)

Boundary conditions:

- Life steam: 50 bar(a), 450 °C, 30,000 kg/h
- Backpressure: 3 bar(a)
- Speed: 10,000 rpm
- Output: 5,000 kW (decentralized feed into low-voltage grid)
  (additional technical limitations of frequency conversion to 50 Hz for higher output)

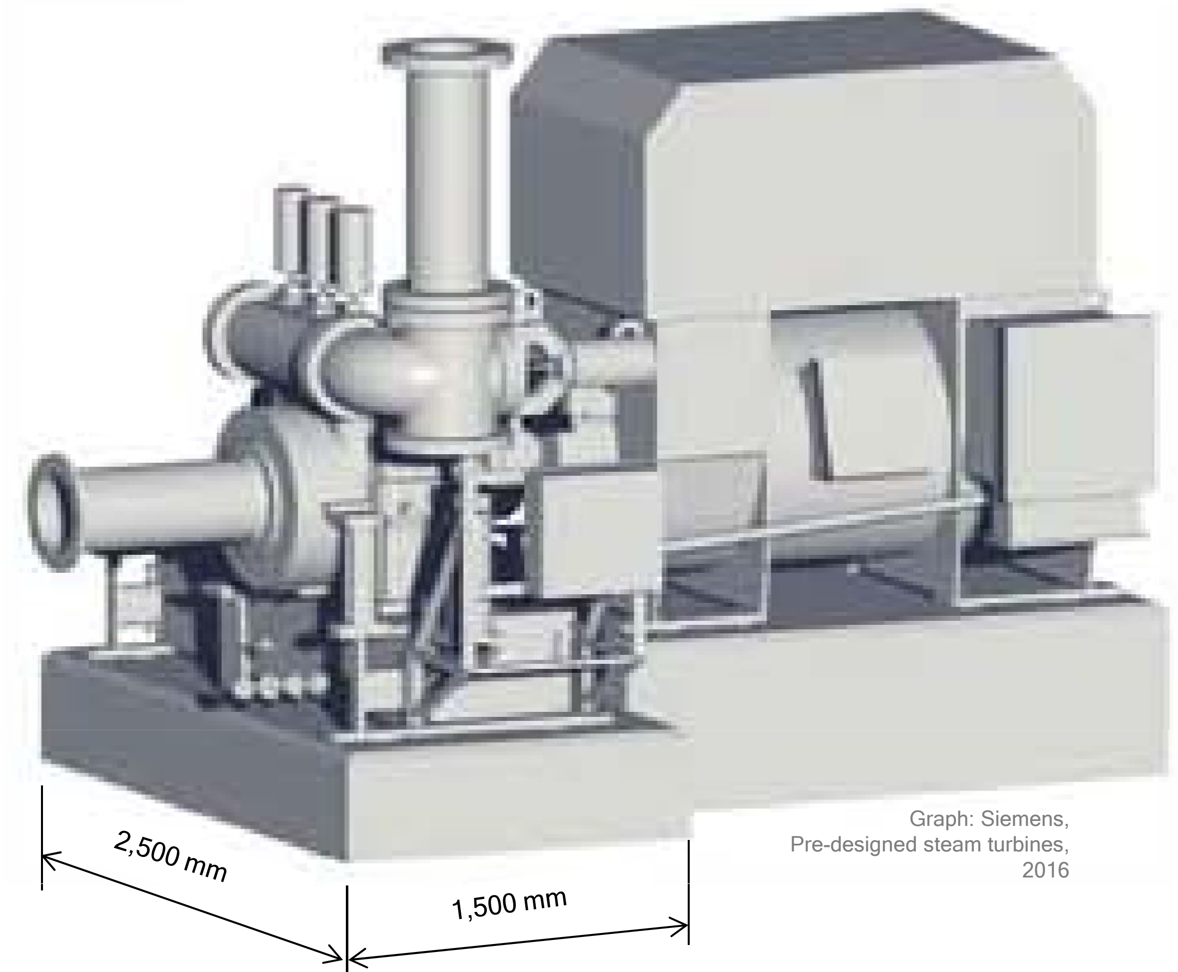
Result:

- Length (without diffuser): 650 mm
- Outer diameter
  (without outer casing): 500 mm

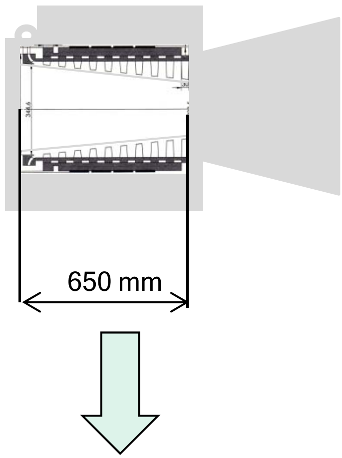


Integrated concept for turbine/generator arrangement
Turbine size comparison

Traditional design of small capacity turbo-generator sets	Proposed integrated concept



Graph: Siemens,
Pre-designed steam turbines,
2016



Significant size and mass reduction

Technical challenges

- **Thermal decoupling of the mechanically coupled system**:
  - limitation of thermal loading of the permanent magnets of generator
  - low heat flow turbine – generator
  - 3D blading design
  - thermal-mechanical behaviour and long-term reliability
- **Rotor dynamics** at **part load**
- **Short-circuit torques**
- **Long-term performance** of **permanent magnets** due to heating up (heat flow turbine – generator, electrical fields)

Summary

- **Trend** to more distributed power generation
  → demand for **smaller units of high efficiency and flexibility**
- Alternative cycles with sCO_2 as working fluid are promising due to high efficiency and energy density.
- **Small-scale units** are advantageous for **greater location independence**.
- **Thermal efficiency** of a **recuperated sCO_2 cycle** is about **4 pp higher than** the one of a **simple water steam cycle** (no reheat) for turbine inlet temperature of **300 °C** (example for heat recovery application)
- Although **mass flow rate** for same effective power is by factor 10 **higher** for sCO_2, **turbine inlet / outlet volume flow rates** are by factor 0.06/ 0.005 **lower** resulting in **much smaller turbine size**.
- **High-temperature strength** and **corrosion resistance** in **sCO_2** environment are **most important criteria for** selection of **materials**.
- **Higher energy density** resp. size reduction of mechanical components should necessarily be accompanied by the same of **electrical components (motor / generator).**
- The vision of an **integrated concept** for turbine/generator arrangement as presented is promising for size reduction, but technologically challenging.

»Wissen schafft Brücken.«

Thank you for listening.