3rd European Conference on supercritical CO_2

Modelling and performance analysis of a supercritical CO_2 system for high temperature industrial heat to power conversion at off-design conditions

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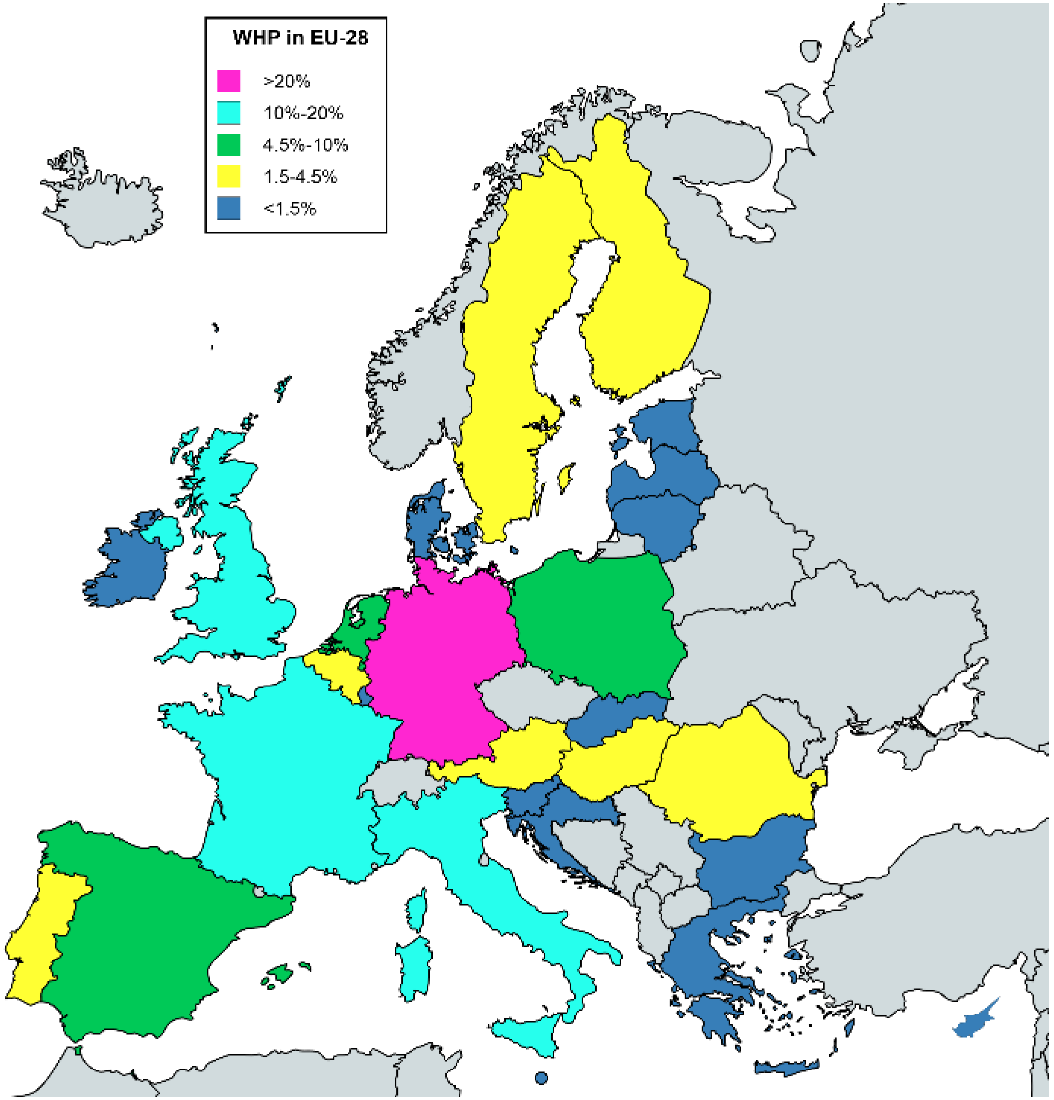
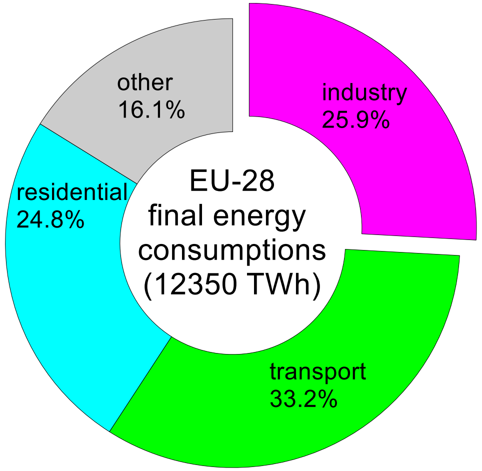
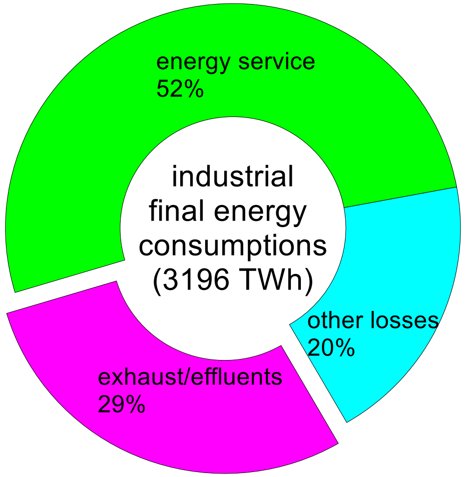
Outline

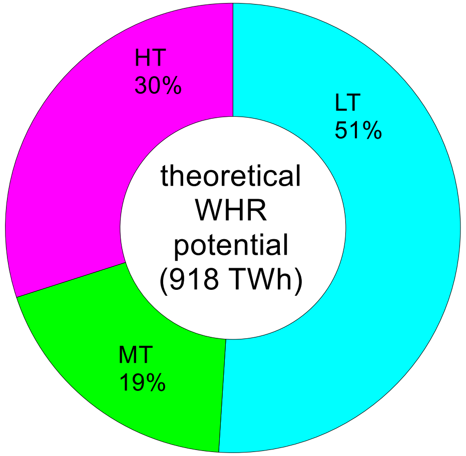
- Waste heat recovery – EU potential
- Why sCO_2?
- Modelling methodology
- Heat exchangers and turbomachines
- System model
- Results
- Conclusions

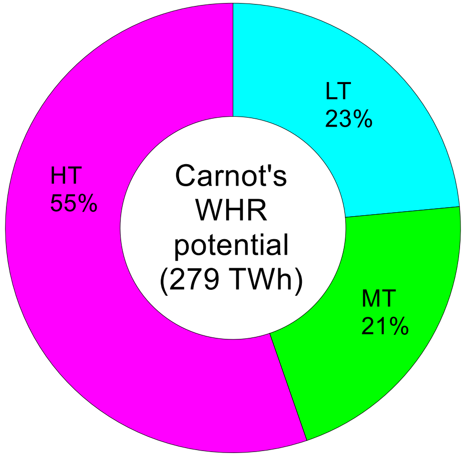
Waste heat recovery – EU potential



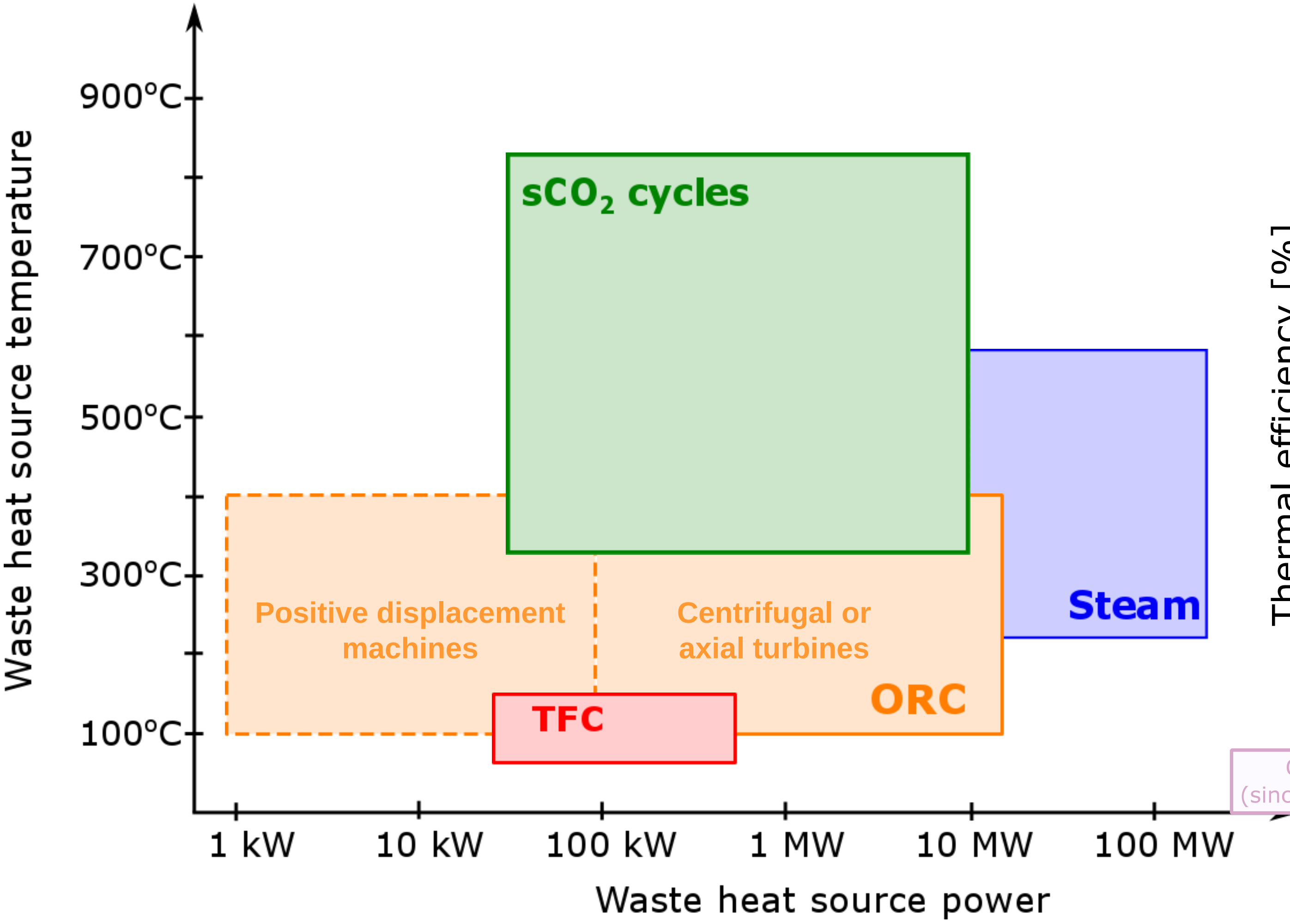




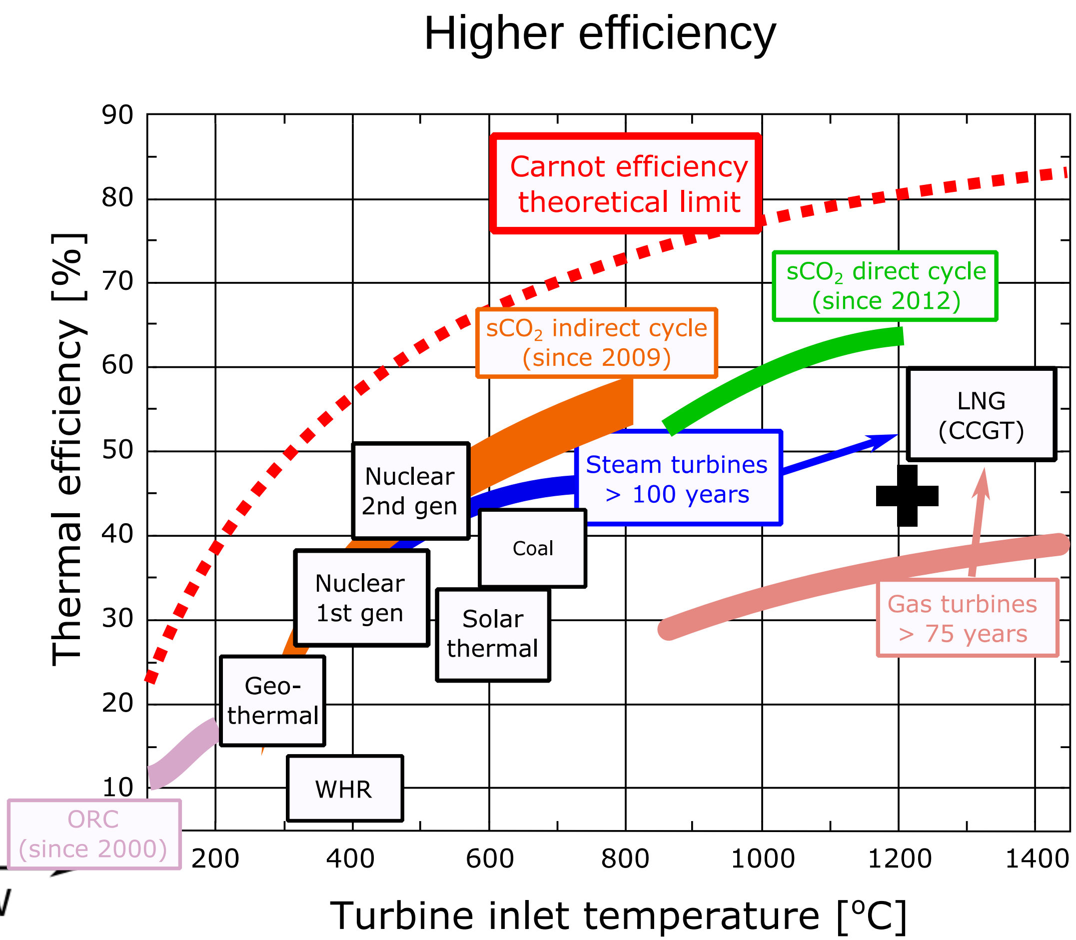




Why sCO_2?

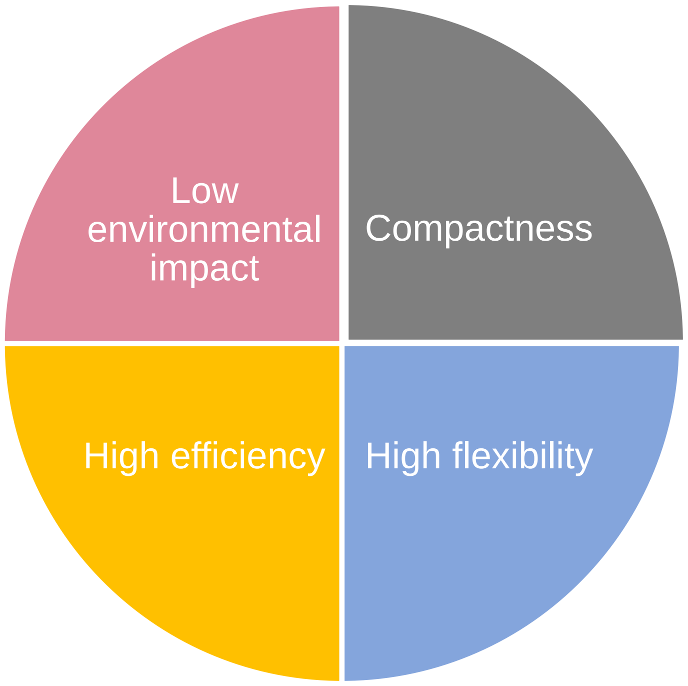


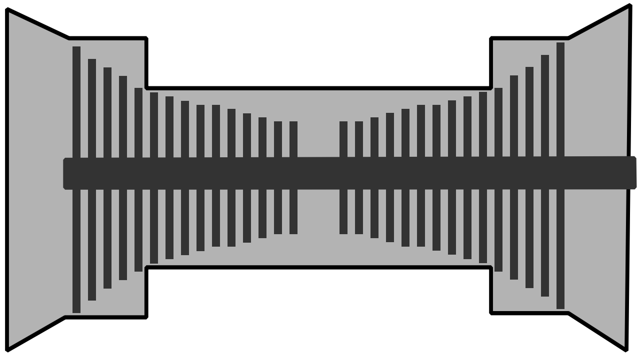
Waste heat source temperature
900°C
700°C
500°C
300°C
100°C
sCO₂ cycles
Steam
Positive displacement machines
Centrifugal or axial turbines
ORC
TFC
1 kW
10 kW
100 kW
1 MW
10 MW
100 MW
Waste heat source power

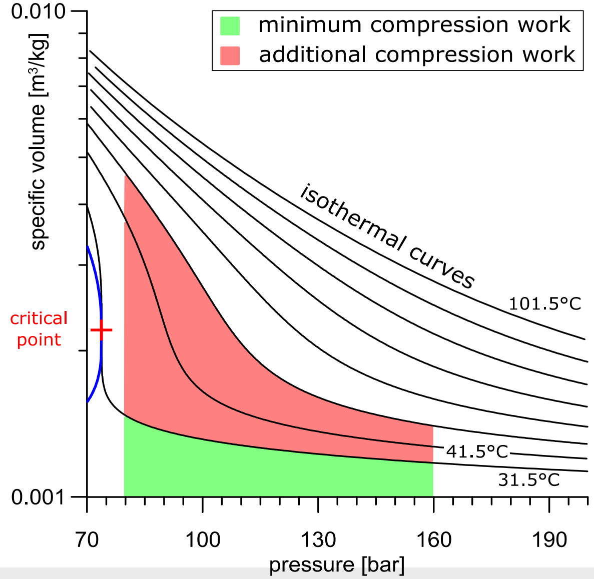


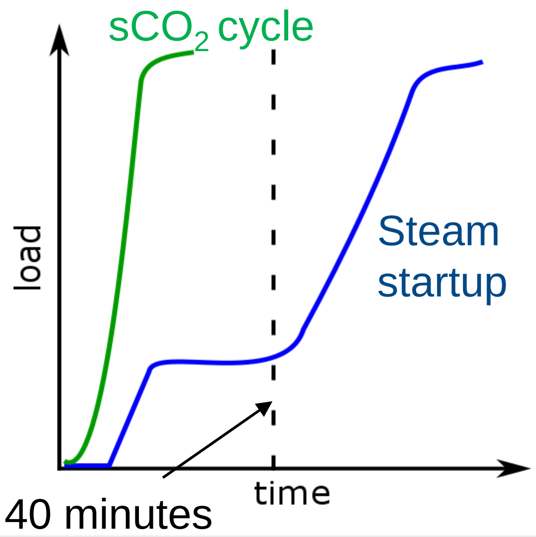
Higher efficiency
Thermal efficiency [%]
Carnot efficiency theoretical limit
sCO₂ direct cycle (since 2012)
sCO₂ indirect cycle (since 2009)
LNG (CCGT)
Steam turbines > 100 years
Nuclear 2nd gen
Coal
Nuclear 1st gen
Solar thermal
Gas turbines > 75 years
Geo-thermal
WHR
ORC (since 2000)
Turbine inlet temperature [°C]

Further advantages



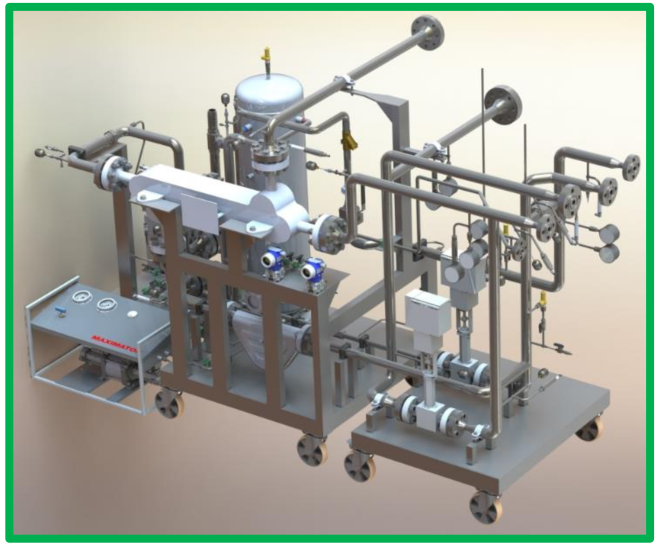


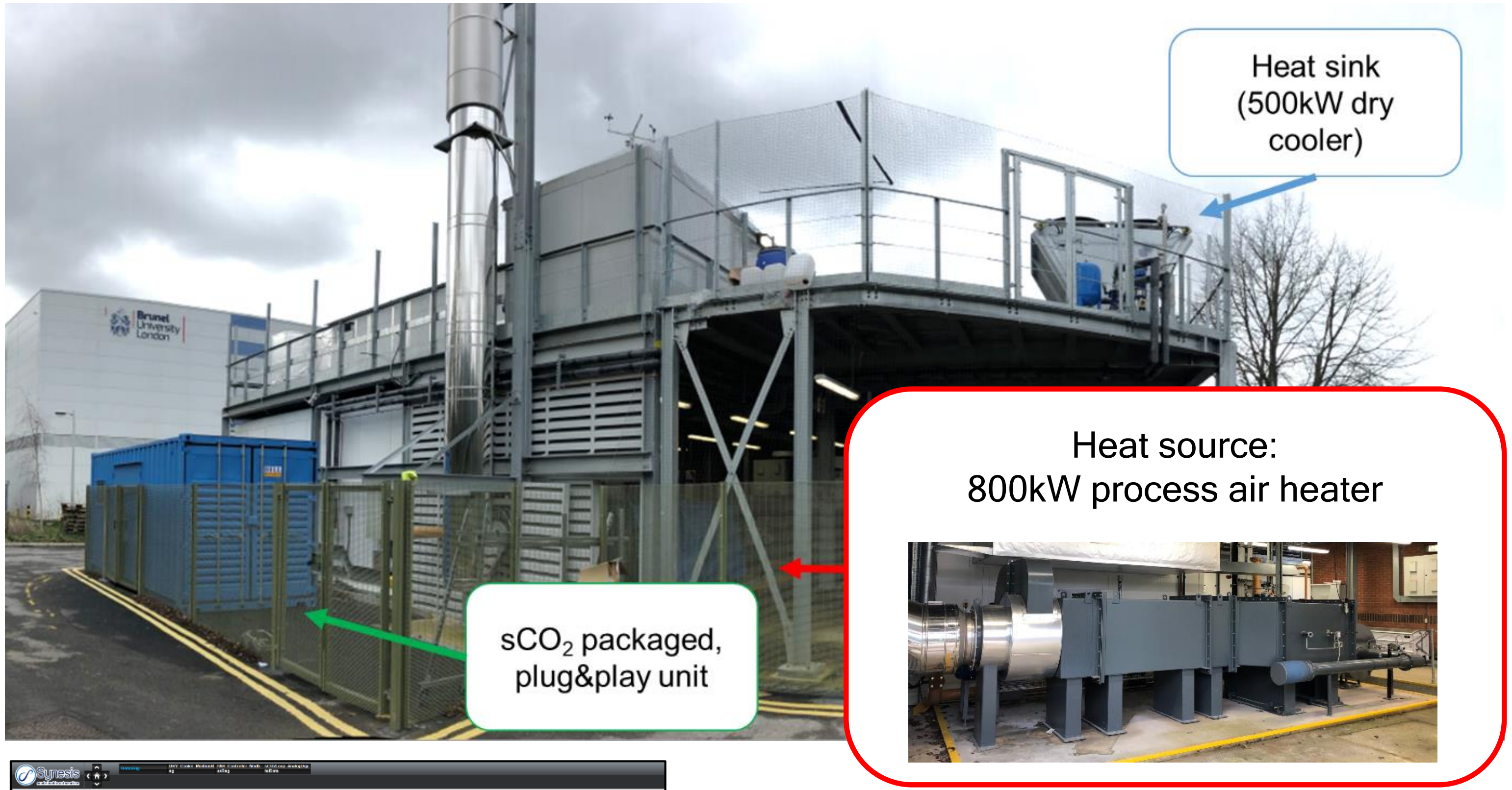


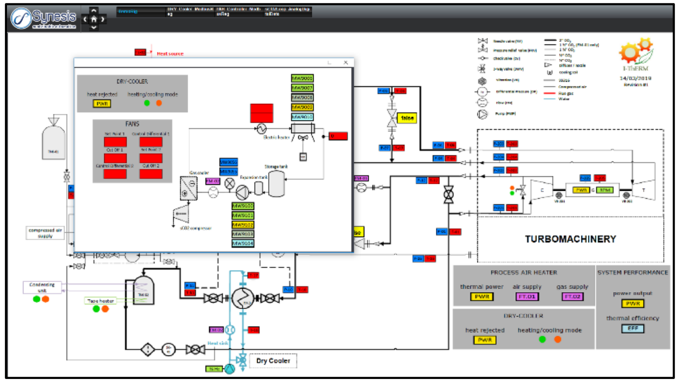


sCO_2 demonstrator in the I-ThERM project

- Simple regenerative architecture
- High temperature flue gas as heat source and water as heat sink
- Single shaft turbomachinery
- 50 kW nominal power
- Supervisory control system







Modelling methodology

Input data

- Experimental testing or from more complex models
- Off-design working points or performance maps are calculated

1-D standalone component model

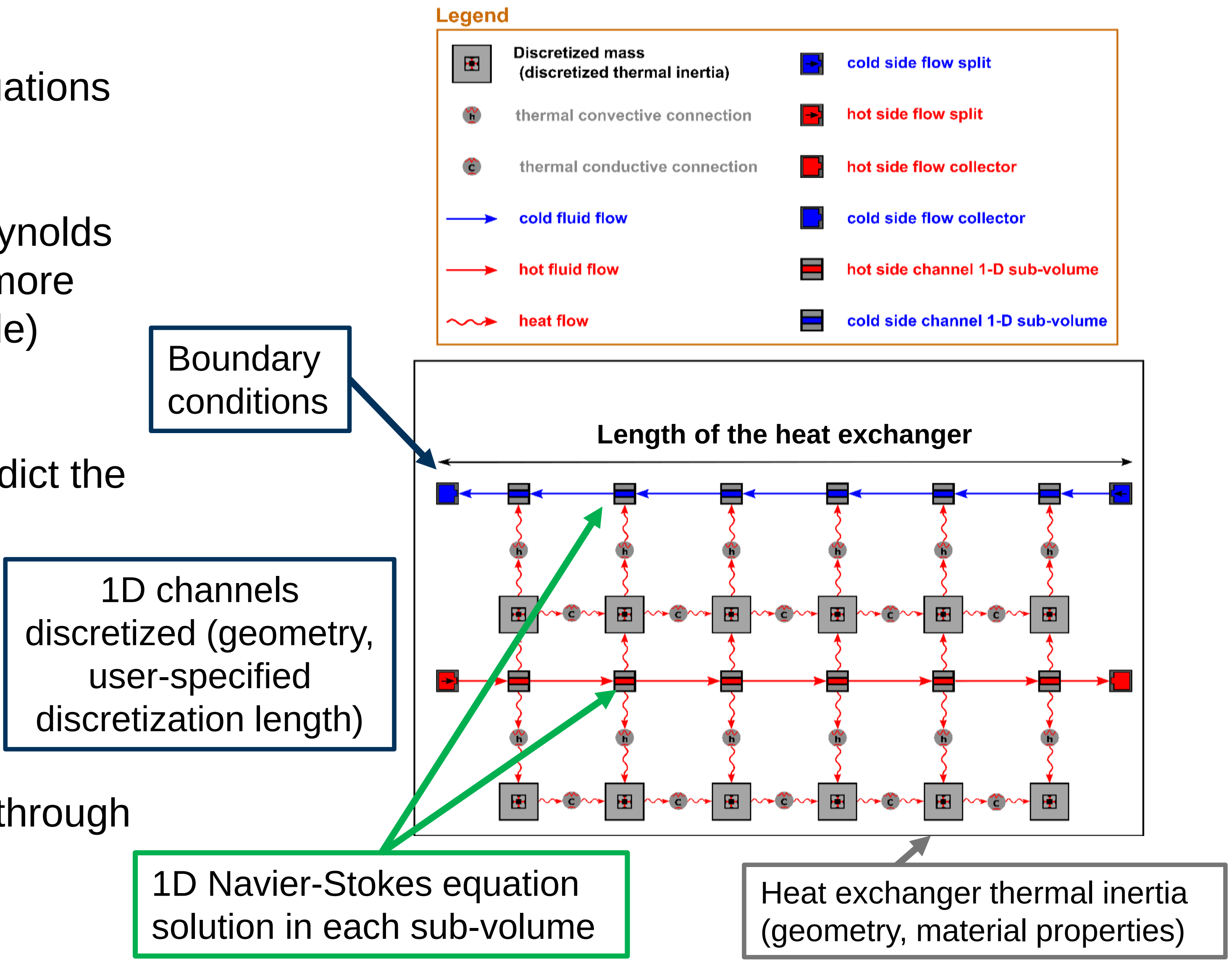
- GT-SUITE™ platform
- Calibration dependent
- Good trade-off between accuracy and computational effort
- Correlation required for pressure drops and heat transfer equipment

1-D system model

- Different models assembled and connected by pipe sub-models
- Steady-state and transient simulations

Heat exchangers

- 1-D formulation of the Navier-Stokes equations
- Best fitting coefficients of the Nusselt-Reynolds curves calculated from manufacturer or more complex models data (non refrigerant side)
- Gnieliski heat transfer correlations to predict the heat transfer coefficient (refrigerant side)
- Colebrook equation for pressure drops
- Heat exchanger inertia taken in account through geometrical and material properties



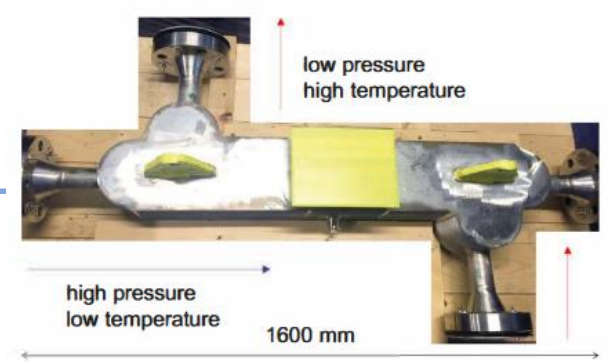
Heat transfer equipment

Cooler (CO_2/water)

- High p, low T (75 bar, 30ºC-100ºC)
- Plate heat exchanger

Recuperator (CO_2/CO_2)

- High p, medium T (up to 140 bar, 120ºC-320ºC)
- high-end industrial product
- Printed Circuit Heat Exchanger

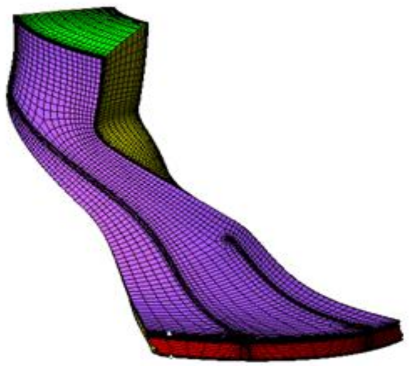


Heater (flue gas/CO_2)

- High p, High T (up to 140 bar, >350ºC)
- Low pressure drops required on both sides
- Low technological readiness level
- Micro-tube heat exchanger

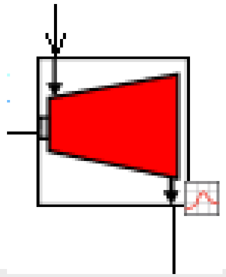
Turbomachines

3-D CFD model



- 3-D CFD models used to generate performance maps
- ANSYS (CCD, RTD and BladeGen)
- ANSYS CFX 17.1 solver (mesh of approx. 10^6 nodes)
- κ-ε model for flow turbulence and compressibility
- Span-Wagner Equation of State model for real gas properties

Map based model



Geometrical details

- Sandia experimental results used to validate the 3-D CFD modelling methodology
- Blade number and shape similar to Sandia turbomachines and impeller new design

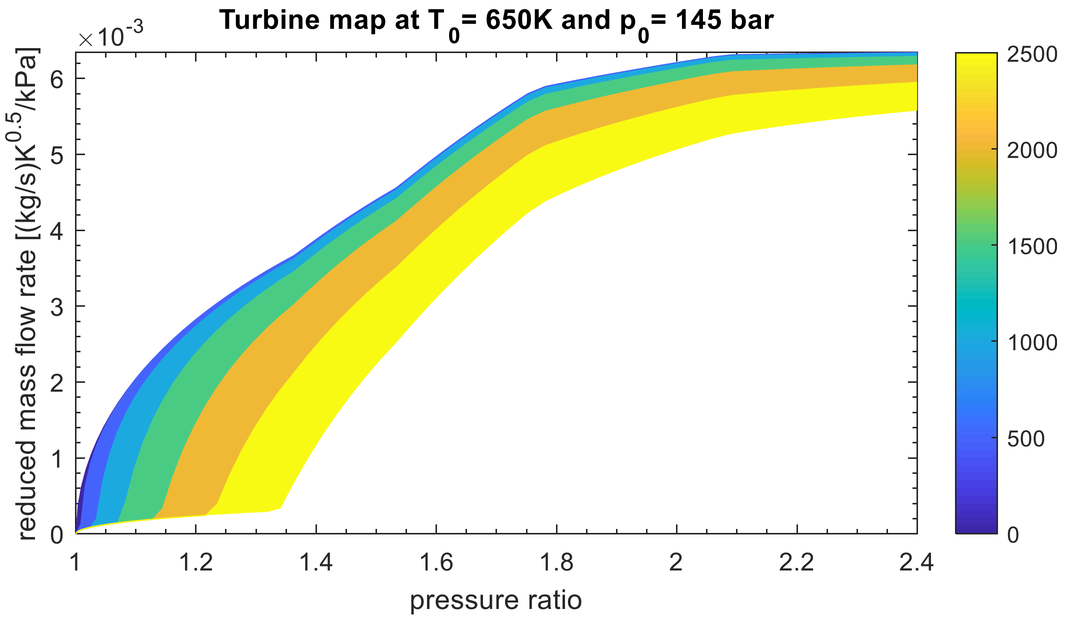
		Compressor	Turbine
Rotor	Diameter	55 mm	72 mm
	Number of blades	7	14
Nozzle	Number of blades	11	17
Isentropic efficiency (total-static)		75%	80%

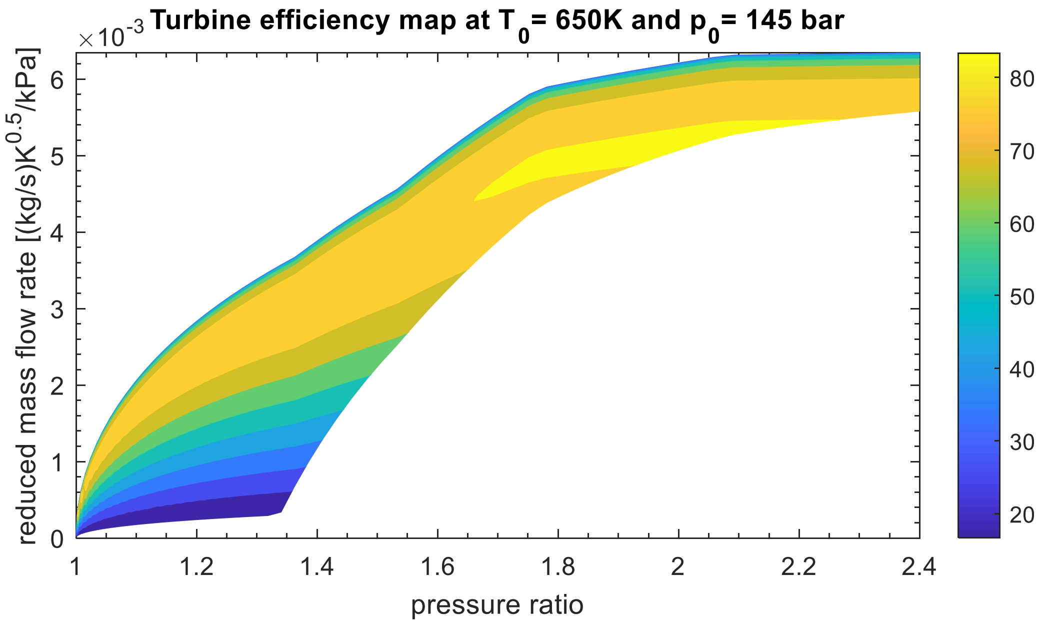
SNL compressor

SNL turbine

Turbine performance maps

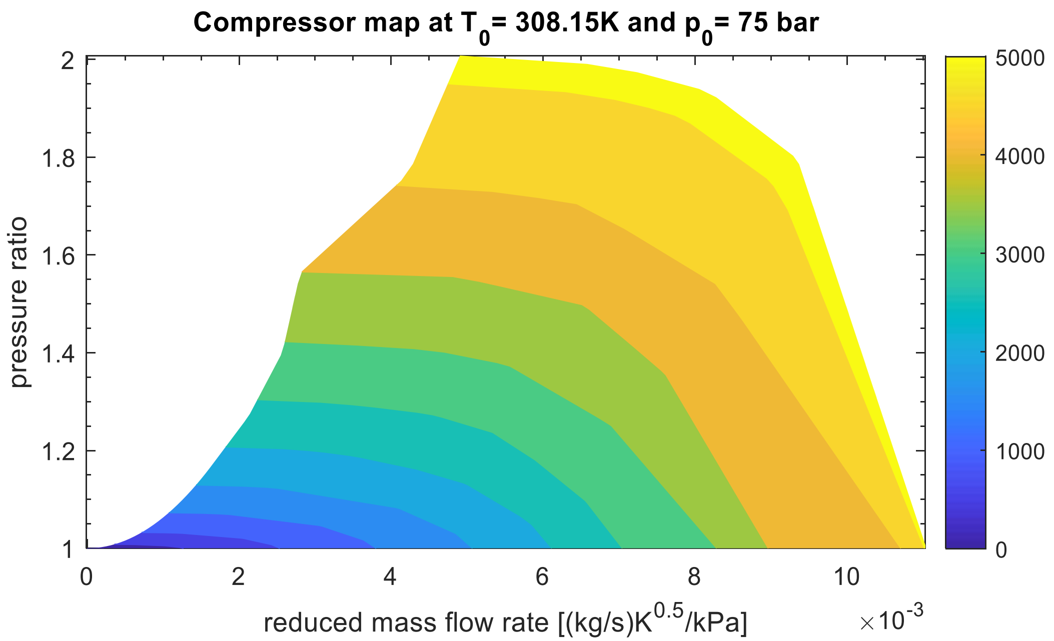
- Reduced mass flow rate and revolution speed
- Reference temperature and pressure of 377ºC and 145 bar respectively
- To obtain the maps several simulations carried out keeping constant CO2 inlet thermodynamic conditions (pressure and temperature) and changing outlet static pressure at different revolution speeds

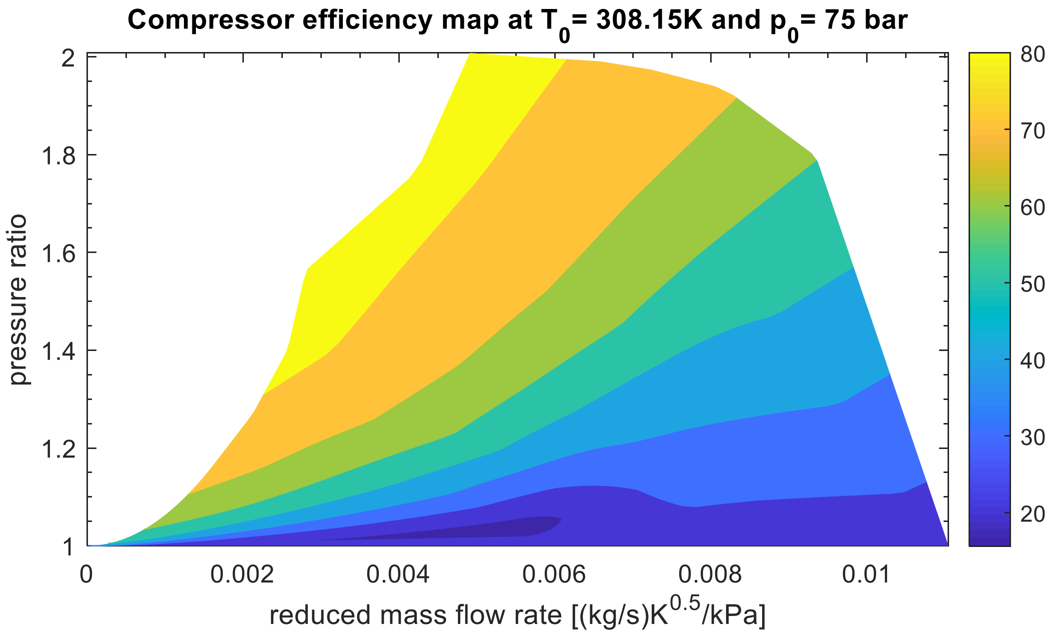




Compressor performance maps

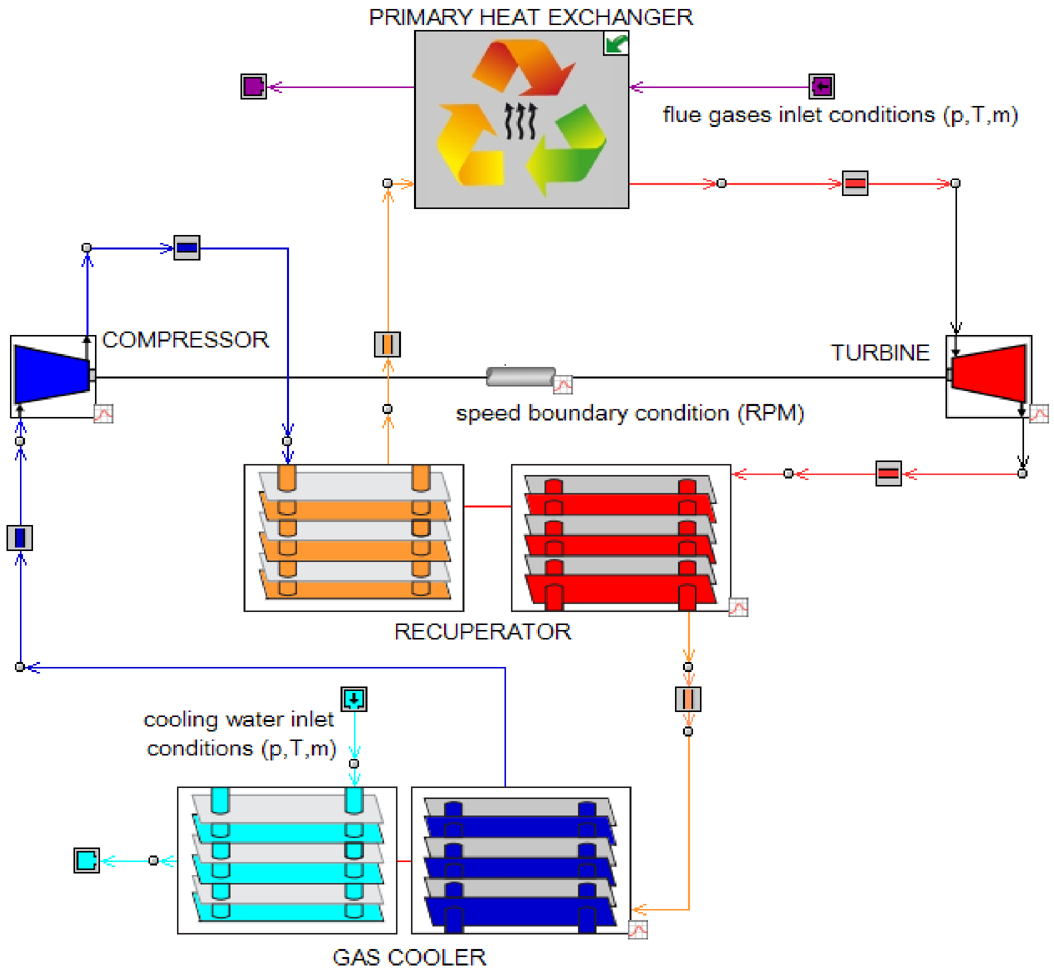
- Same procedure to generate the turbine maps
- Reference temperature and pressure of 35ºC and 75 bar respectively
- Small distortion in surge line due to the change in the pressure rise characteristics occurring between higher and lower rotational speeds.
- Thermo-physical properties of CO2 lead to a reduction of the choke margin in the compressor





System model

- Boundary conditions
  - heat sink/source mass flow rate
  - Heat sink/source inlet temperature
  - Revolution speed of the shaft
- Heat and pressure losses neglected in pipes
- Shaft inertia considered
- Refprop for the fluid thermo-physical properties
- Power quantities purely mechanical

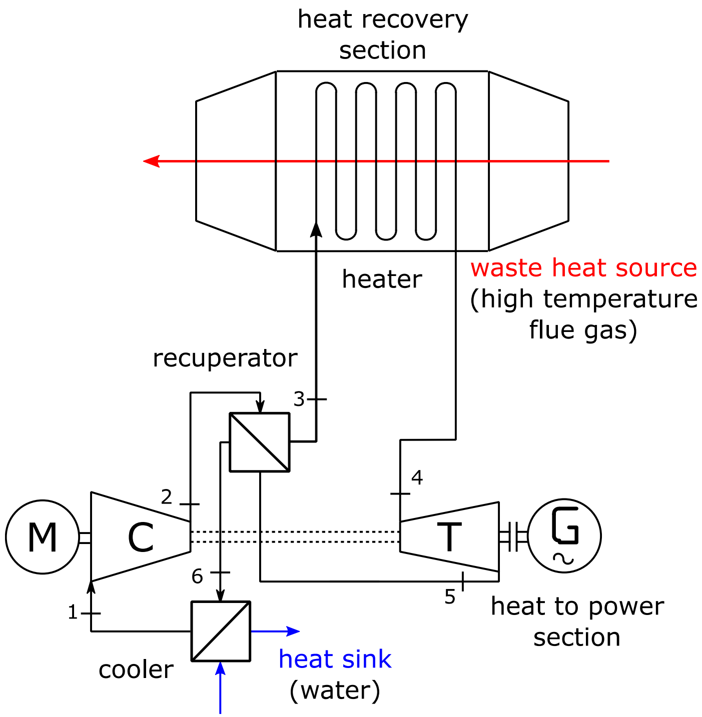


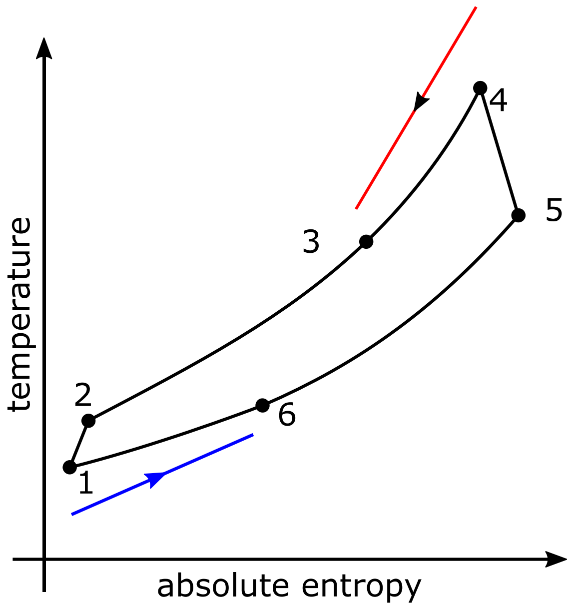
System design point – model input

Heat source		Design
Mass flow rate	[kg/s]	1.0
Inlet temperature	[°C]	650
Inlet pressure	[bar]	1

Cold source		Design
Mass flow rate	[kg/s]	1.6
Inlet temperature	[°C]	22
Inlet pressure	[bar]	3

Turbomachines		Design
Revolution speed	[RPM]	86000



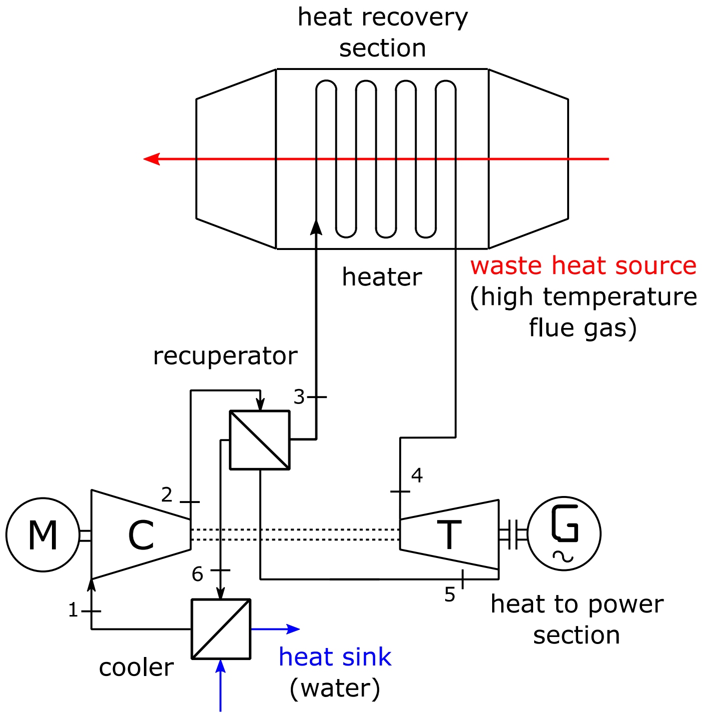


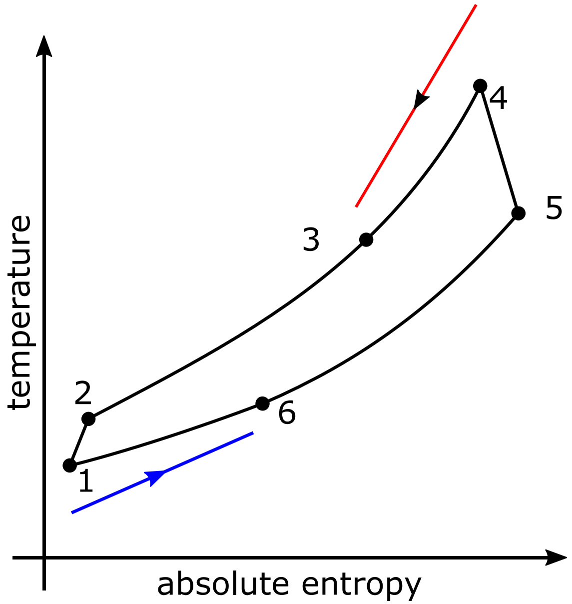
System design point – model output

Supercritical CO_2		Design
Mass flow rate	[kg/s]	2.1
Maximum pressure (2-4)	[bar]	128
Maximum temperature (4)	[°C]	400
Minimum pressure (5-1)	[bar]	75
Minimum temperature (1)	[°C]	36

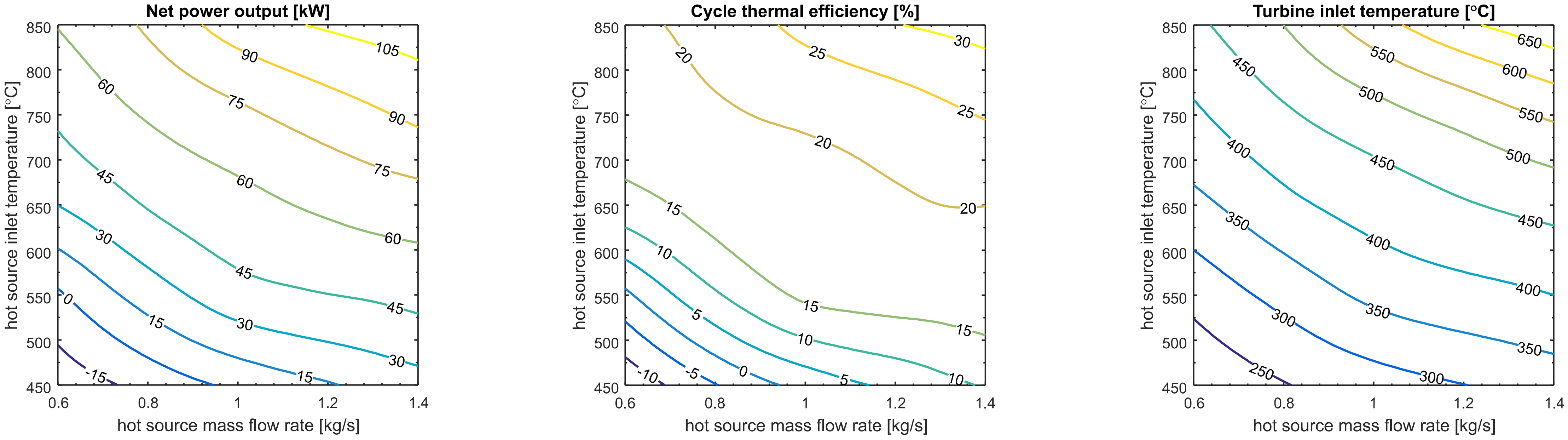
Turbomachines		Design
Compressor isentropic efficiency	[%]	75
Turbine isentropic efficiency	[%]	80

System performance		Design
Cycle pressure ratio	[-]	1.7
Net power output	kW	50
Overall efficiency	%	20





Results: heat source variation

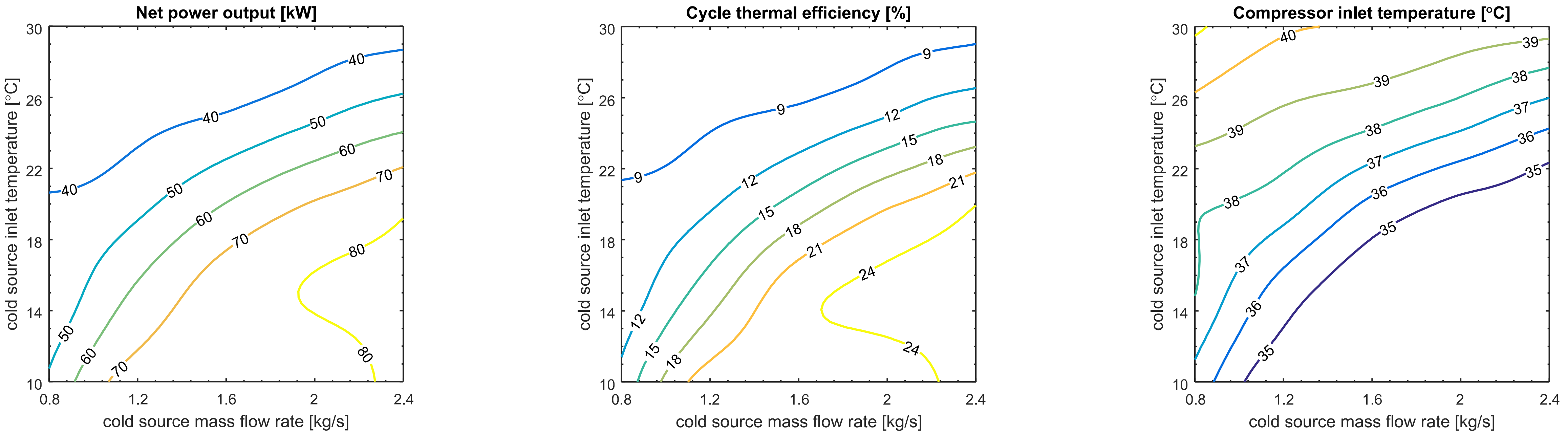


- Cycle low pressure ratio allows a not null power generation only for heat source inlet temperature and mas flow rate higher than 500°C and 0.8 kg/s

- For a heat source temperature higher than 800°C up to 100 kW and 30% of efficiency can be achieved

- A better thermal matching between the exhausts and the CO_2 may lead to higher turbine inlet temperatures and hence to greater system performance

Results: cold source variation



- Increasing the cooling load leads to higher unit performance thanks to the lower working fluid temperature at the compressor inlet
- When the cooling fluid inlet temperature is higher than 22°C no effect is shown on the system performance for a further decrease of the cooling load
- No CO_2 condensation at higher cooling loads

Conclusions

- One dimensional model of a sCO_2 heat to power conversion system has been developed in GT-SUITE™ and herein presented
  - ✓ Heat exchanger inertia considered through geometrical and material properties
  - ✓ Performance maps for turbomachines calculated with 3-D CFD models
  - ✓ Heat source/sink inlet conditions and turbomachines revolution speed as model boundary conditions
  - ✓ Exhaust gases as heat source and water as cooling mid
  - ✓ Heat and pressure losses neglected in pipes

- Performance maps of the sCO_2 unit calculated for heat source and sink variations
  - ✓ Operating range of the unit
  - ✓ Useful insights for system control and optimisation
  - ✓ For low cycle pressure ratio high temperatures of the heat source must be achieved for a positive power output
  - ✓ High cooling loads allow to improve the system performance without causing CO_2 condensation at the compressor inlet

Acknowledgements

I-ThERM

The Industrial Thermal Energy Recovery Conversion and Management project aims to

investigate, design, build and demonstrate innovative plug and play waste heat recovery solutions to facilitate optimum utilisation of energy in selected applications with high replicability and energy recovery potential in the temperature range 70°C - 1000°C

www.itherm-project.eu

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