University of Stuttgart
Institute of Nuclear Technology
and Energy Systems

Operational Analysis of a Self-Propelling Heat Removal System using Supercritical CO_2 with ATHLET

M. Hofer, M.Buck, J. Starflinger

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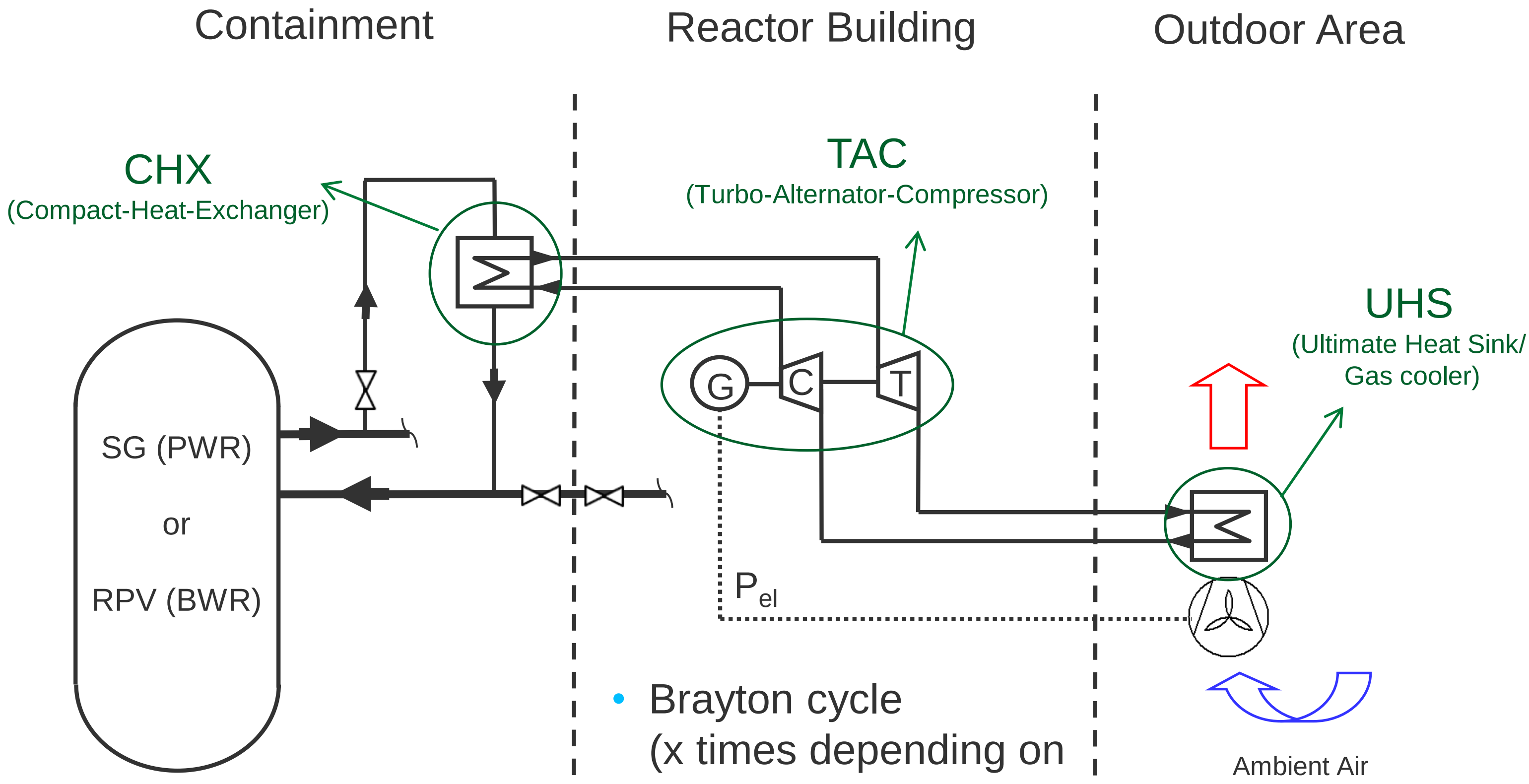
Outline

Introduction (1): Overview and motivation

- Motivation
  - Fukushima
  - Scientific Trend: new heat removal systems
- Active Heat Removal System with Turbo-Compressor
  - sCO_2 as a working fluid
  - Air as ultimate heat sink
  - Self-propelling
  - Very compact

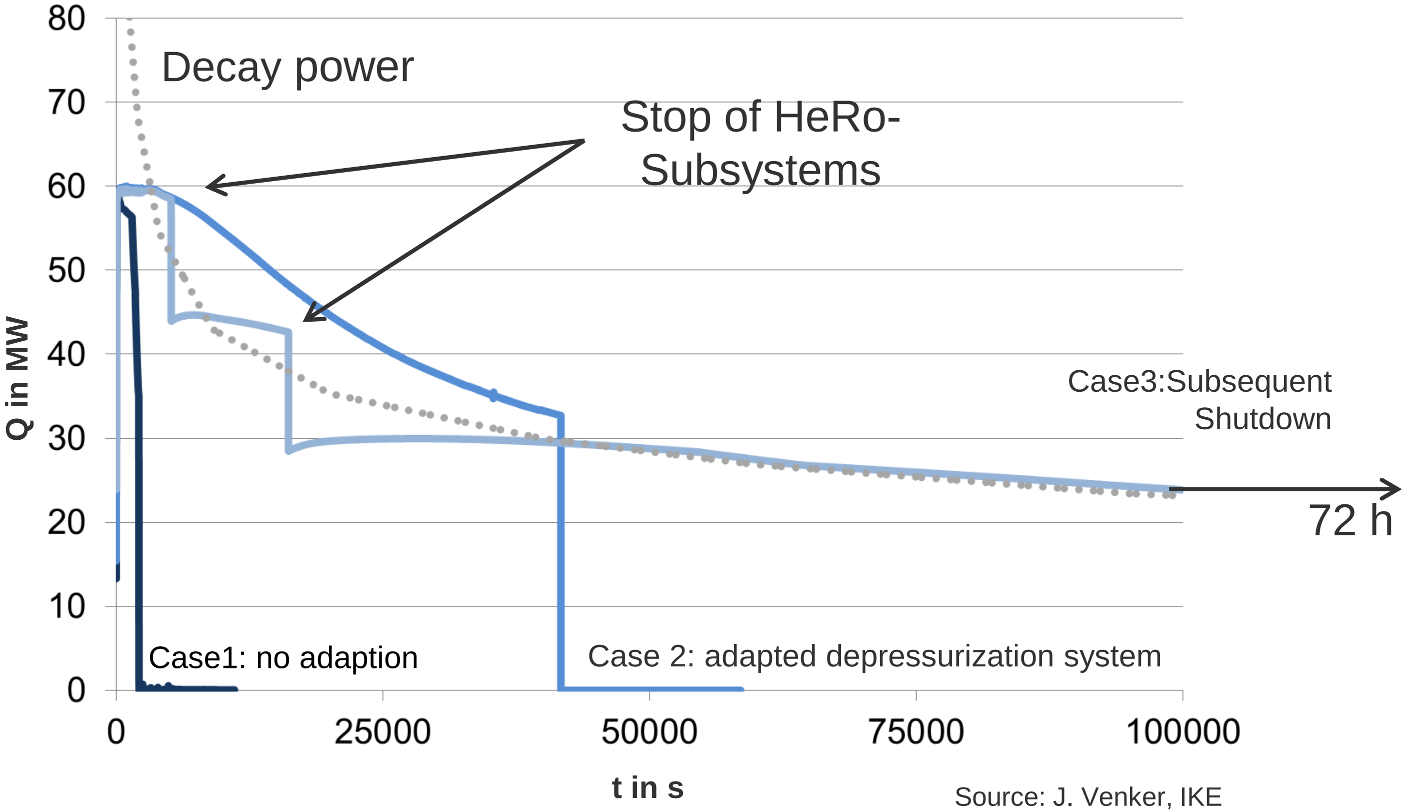
Introduction (2): Concept of heat removal system



- Brayton cycle
  (x times depending on NPP size)

Introduction (3): Former ATHLET Simulations

Decay power compared to thermal power of heat removal system

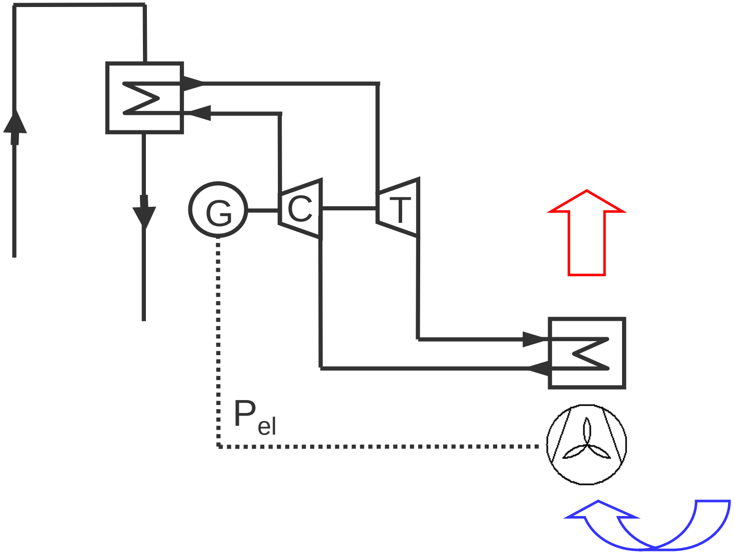


Source: J. Venker, IKE

Introduction (4): Overall Objective

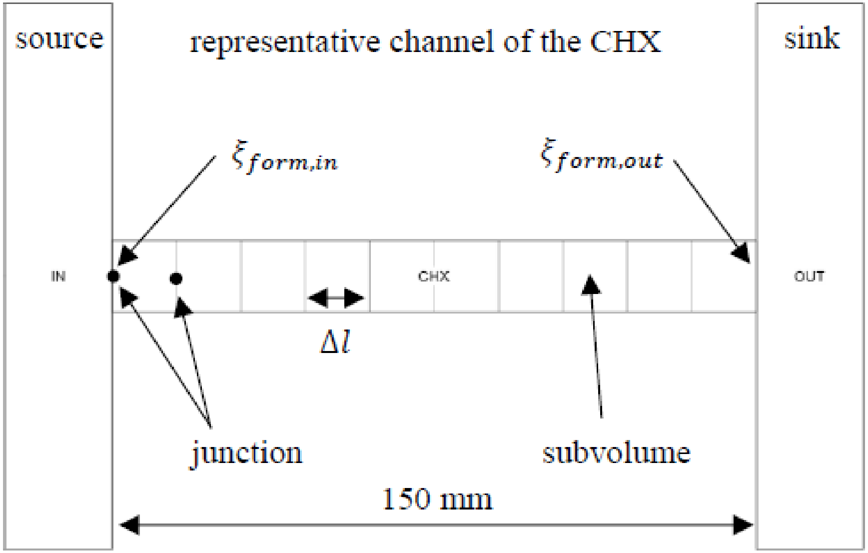
- Enable ATHLET to simulate sCO_2-Brayton-Cycles and their interaction with existing or future BWR, PWR, VVER, etc. for safety analyses
  - ATHLET code extensions
  - Validation experiments
  - NPP and cycle simulations

- Objective of this presentation
  - Overview of the code extensions
  - Operation strategies of the sCO_2-Brayton-Cycle (under varying steam side boundary conditions)



Compact heat exchanger: Representative channel model

- Only one representative channel pair is modelled
- Correlations for heat transfer coefficients and pressure drop (sCO_2: Gnielinski and Colebrook)
- Inlet and outlet $\Delta p_{plenum} = \xi_{Form} \dot{m}^2/(2\rho)$



Simulation of experimental CHX

Ultimate heat sink: Simplified model

- Modelling similar to CHX (only one pipe of the plate-fin HX is modelled)
- In this analysis the air side heat transfer is not modelled explicitly
- $\dot{Q}_{UHS}$ is controlled to keep the compressor inlet temperature constant
- In reality this is achieved by varying the fan speed

UHS at glass model:
Experimental test-loop
in Essen (Germany)
Source: A. Hacks, UDE

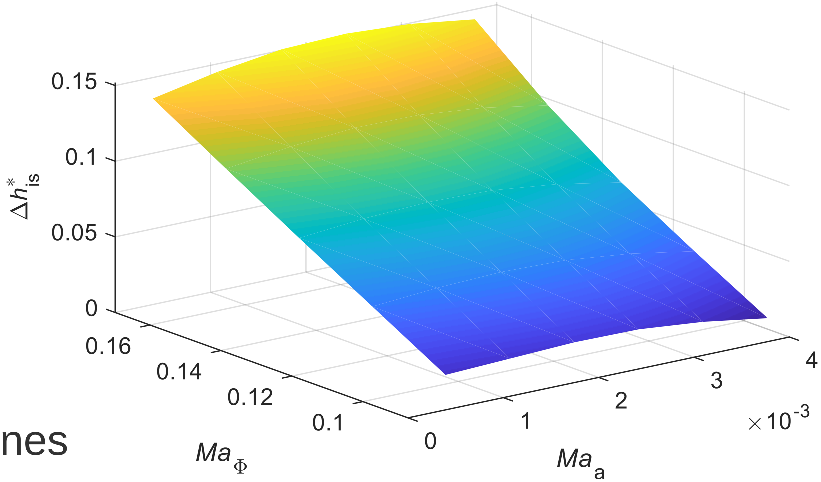
Radial turbomachinery (1): Real gas approach

- Performance map for specific thermodynamic inlet condition (e.g. design)
- Real gas similarity approach for different conditions [1]
- Dimensionless Performance Map: $(\eta, \frac{\Delta h_{is}}{c^2}) = f(Ma_a, Ma_\theta)$
- $M_a = \frac{\dot{m}}{\rho D^2 c}$ and $M_\theta = \frac{ND}{c}$ with speed of sound $c = \sqrt{\left(\frac{\delta p}{\delta \rho}\right)_s}$
- speed of sound c should be used instead of the heat capacity ratio $\gamma = \frac{c_p}{c_v}$ in the similarity approach (especially for liquid-like CO_2)

- [1]: Pham et al. (2016) An approach for establishing the performance maps of the sc-CO2 compressor: Development and qualification by means of CFD simulations. *International Journal of Heat and Fluid Flow*, 61, 379–94. https://doi.org/10.1016/j.ijheatfluidflow.2016.05.017
- Model in this work is mathematically identical to [1]

Radial turbomachinery (2): This analysis

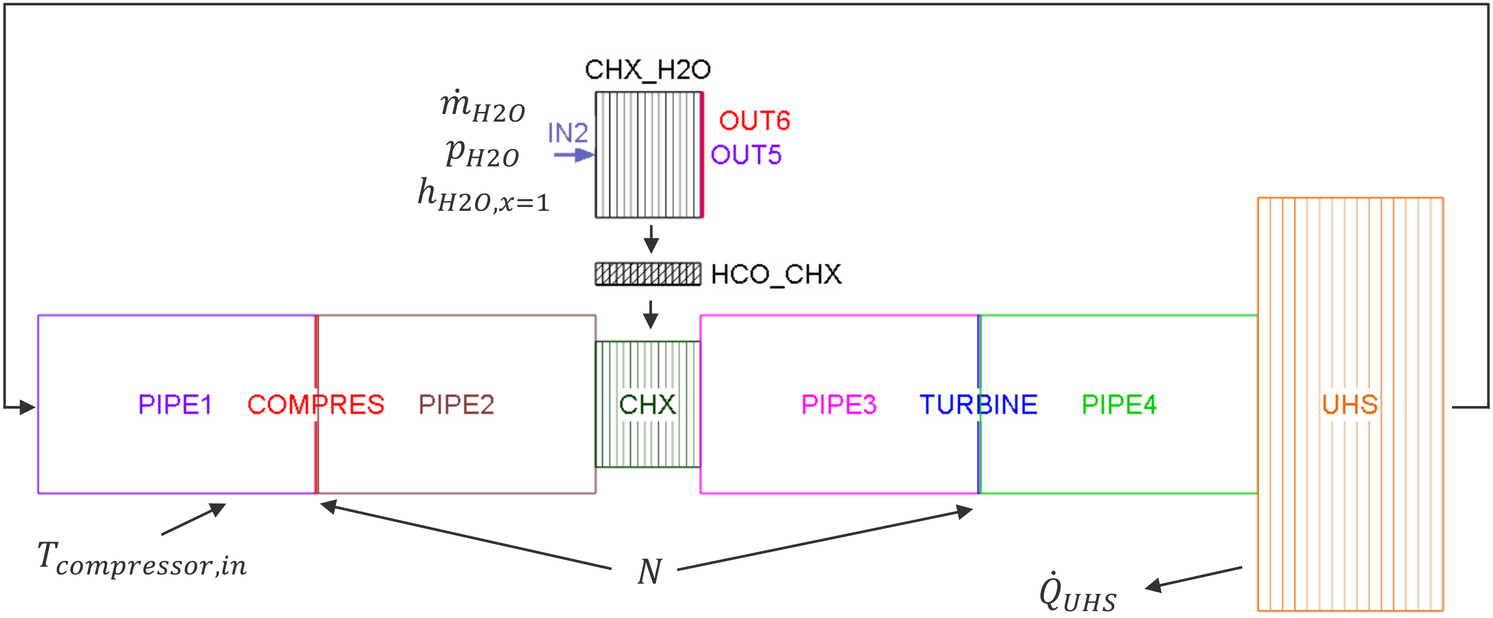
- Compressor
  - Dimensionless performance map of the glass model compressor →
  - Conservative approach because pressure ratio and efficiency are lower compared to large-scale machines



- Turbine
  - Stodola’s cone law with efficiency correlation for radial machines
  - Performance map of the glass model turbine cannot be used because at the design point of the large-scale cycle the rotational speed of both machines is different

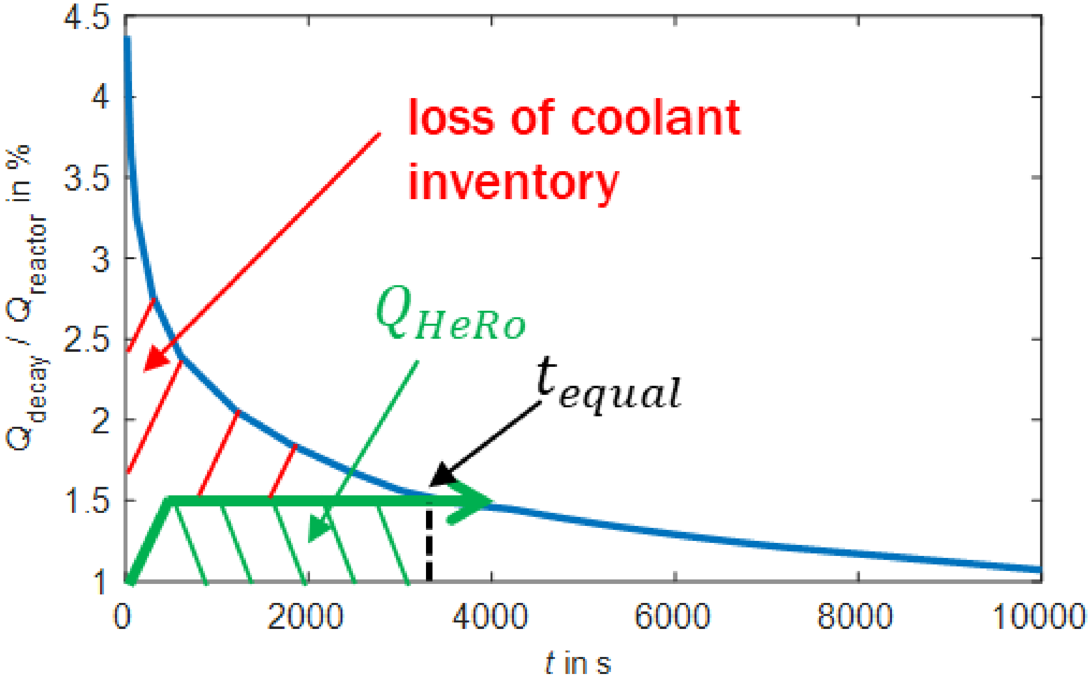
Operational Analysis (1): ATHLET nodalisation

- $\dot{Q}_{UHS}$ is controlled to keep $T_{compressor,in}$ at its design value
- Variation of H_2O side boundary conditions



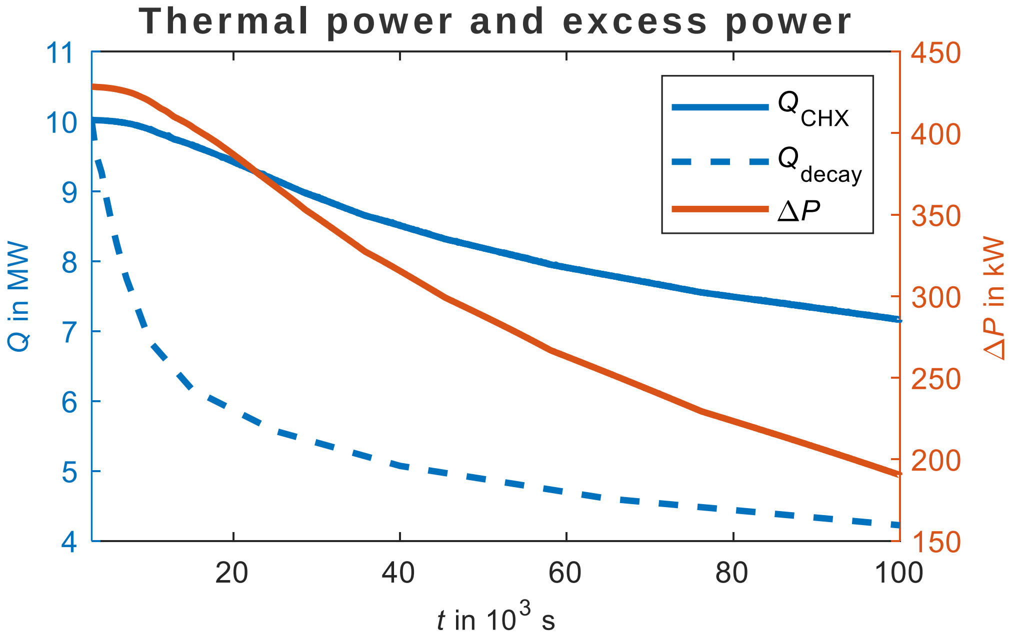
Operational Analysis (2): Start and end of analysis

- Simulation starts at t_{equal} $(\dot{Q}_{HeRo} = \dot{Q}_{decay})$
- Simulation end at 100 000 s = 27.8 h



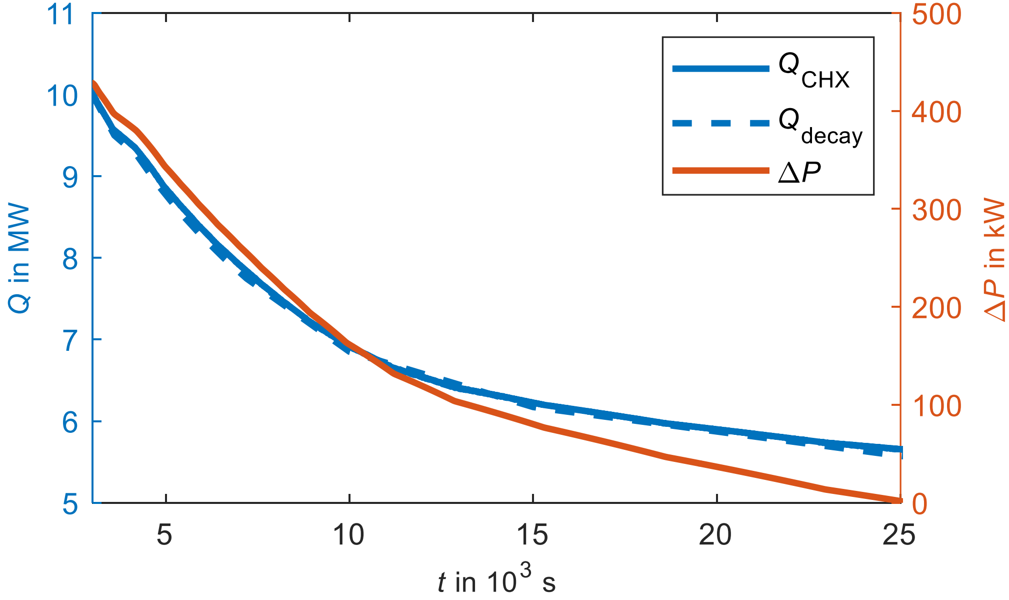
Operational Analysis (3): Case 1

- Boundary conditions:
  - $T_{H2O,sat}$ constant, $\dot{m}_{H2O} \sim \dot{Q}_{decay}(\mathrm{t})$ (with $\dot{Q} = \dot{m}\Delta h_{vap}$)
  - Turbomachinery speed N is kept constant at design value
- Results:
  - $\dot{Q}_{CHX} > \dot{Q}_{decay}$
  - $\Delta P > 0$
- Consequences:
  - Reactor will cool down
  - $T_{H2O,sat}$ is not constant



Operational Analysis (4): Case 2

- Boundary conditions:
  - $\dot{m}_{H2O}$ constant, $T_{H2O,sat}$ follows the decay heat curve ($\Delta h_{vap} \sim \dot{Q}_{decay}(\mathrm{t})$)
  - Turbomachinery speed N is kept constant at design value
- Results:
  - $\dot{Q}_{CHX} \sim \dot{Q}_{decay}$
  - $\Delta P = 0$ reached
- Consequences:
  - Cool down of reactor must be limited → control required

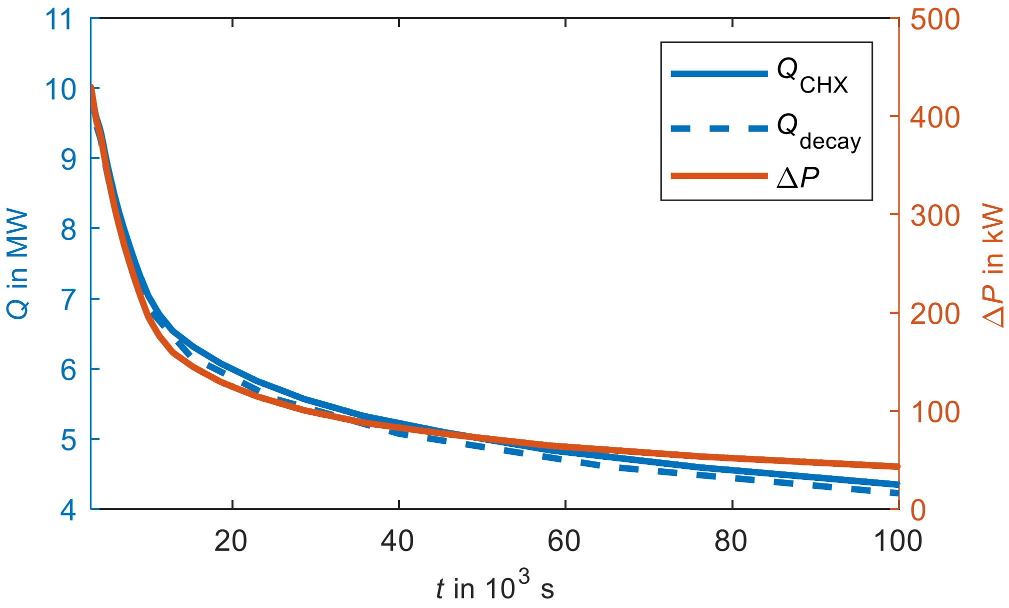


Operational Analysis (5): Case 3 and 4

- Case 3
  - $T_{H2O,sat}$ constant, $\dot{m}_{H2O} \sim \dot{Q}_{decay}$(t) (like case 1)
  - Turbomachinery speed N is controlled to keep $\dot{Q}_{CHX} \approx \dot{Q}_{decay}$
  - Result: $\Delta P > 0$, but compressor surge occurs
- Case 4
  - Turbomachinery speed is controlled to keep $\dot{Q}_{CHX} \approx \dot{Q}_{decay}$
  - $T_{H2O,sat}$ is gradually decreased to 200 °C (less than in case 2, to avoid $\Delta P < 0$)
  - $\dot{m}_{H2O}$ is calculated to match $\dot{Q}_{decay} = \dot{m}\Delta h_{vap}$

Operational Analysis (6): Case 4: Results + Consequences

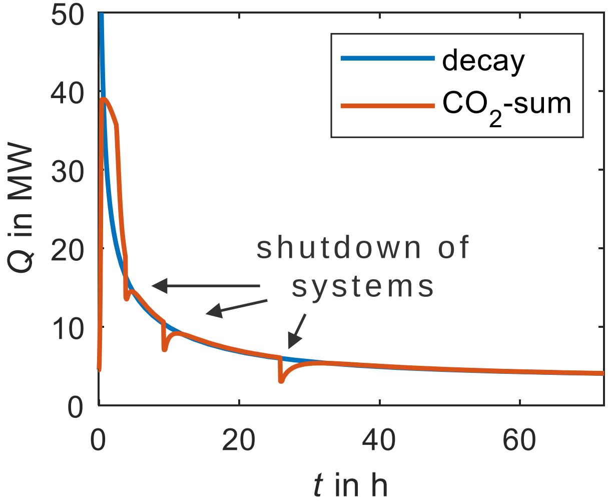
- $\dot{Q}_{CHX} \sim \dot{Q}_{decay}$ → it is possible to follow the decay heat curve
- $\Delta P > 0$ → always self-propelling
- In the long term single units must be switched off because $\Delta P \rightarrow 0$
- No compressor surge



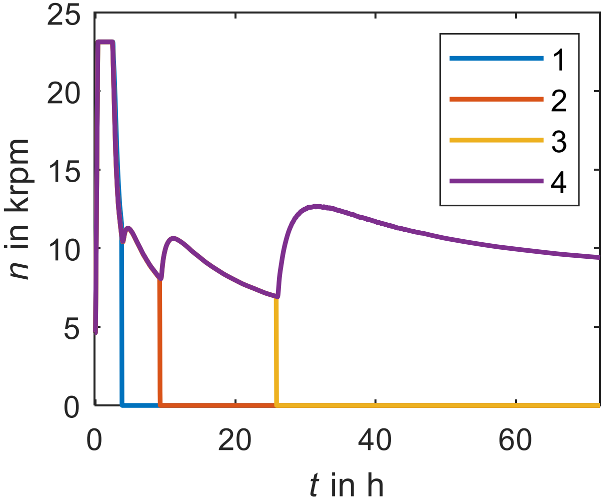
NPP simulations: First results

4 systems (with adaption to decay heat curve)

- Control of turbomachinery speed and subsequent shutdown
- Systems can run for more than 72 h ($\Delta P > 0$)



Decay power compared to thermal power of all systems



Shaft speed of all systems

Conclusion

- Modelling of components
- Operational analysis
  - Shaft speed control enables smooth operation
  - No compressor surge due to cool-down
- Future Work
  - Further improvement of models, input etc. (next presentation)
  - Analysis of varying ambient temperature (next presentation)
  - Simulation of the sCO_2-HeRo-System attached to the NPP (in progress)

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Federal Ministry
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and Energy

sCO$_2$-4-NPP

THANK YOU

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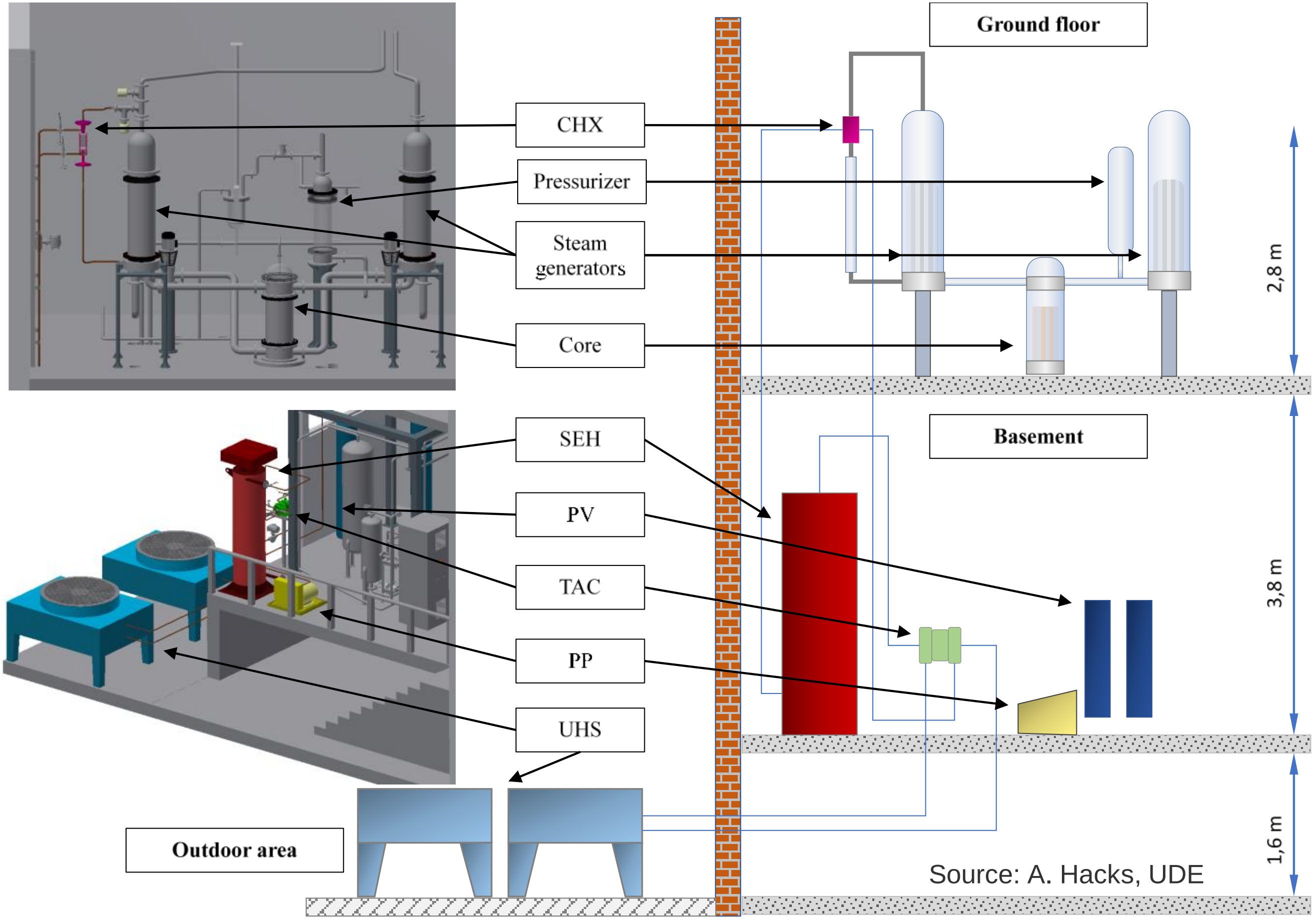
Thank you!

Markus Hofer

e-mail	markus.hofer@ike.uni-stuttgart.de
phone	+49 (0) 711 685-60855
fax	+49 (0) 711 685-62010

University of Stuttgart
Institut für Kernenergetik und Energiesysteme
Pfaffenwaldring 31 • 70569 Stuttgart

Introduction (4): Glass Model with sCO$_2$-HeRo-System



Ground floor

CHX

Pressurizer

Steam generators

Core

Basement

SEH

PV

TAC

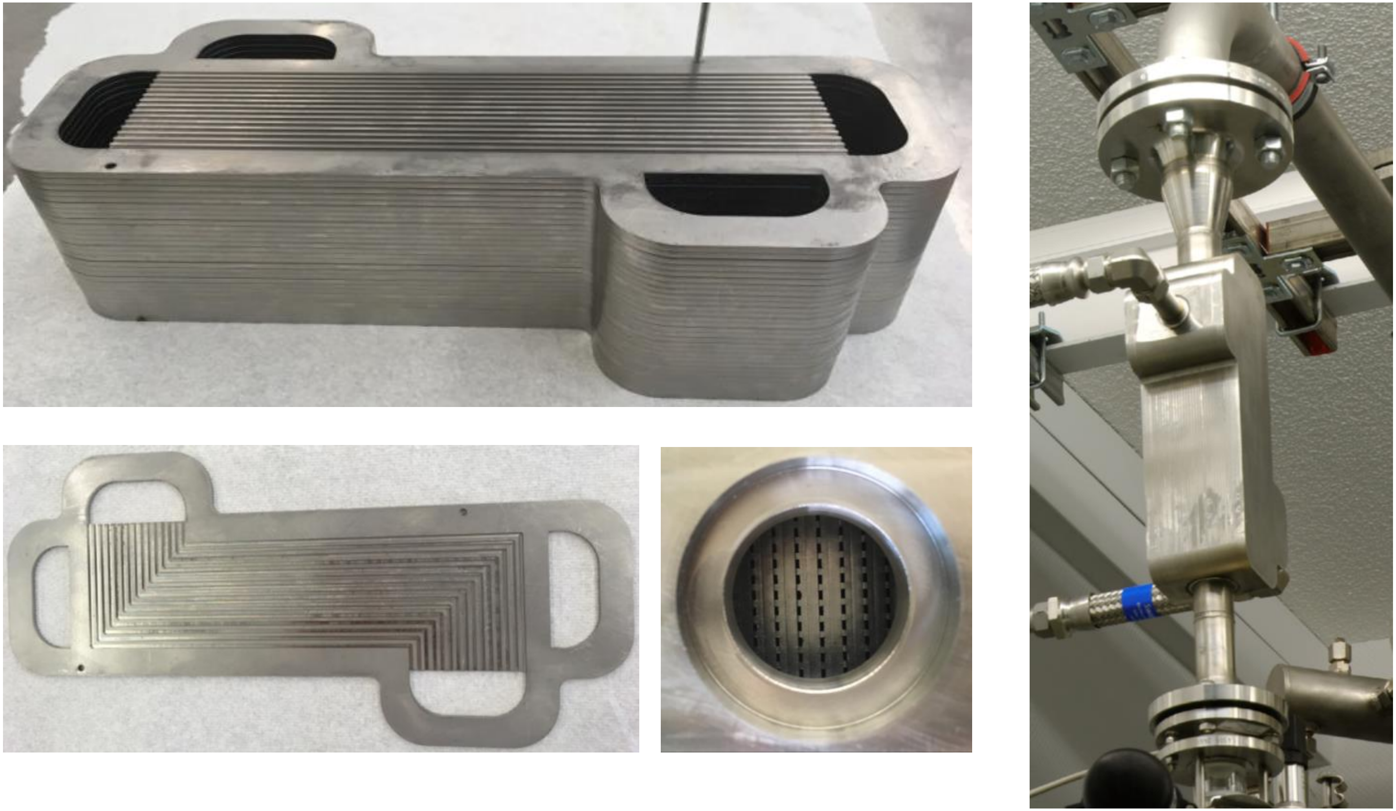
PP

UHS

Outdoor area

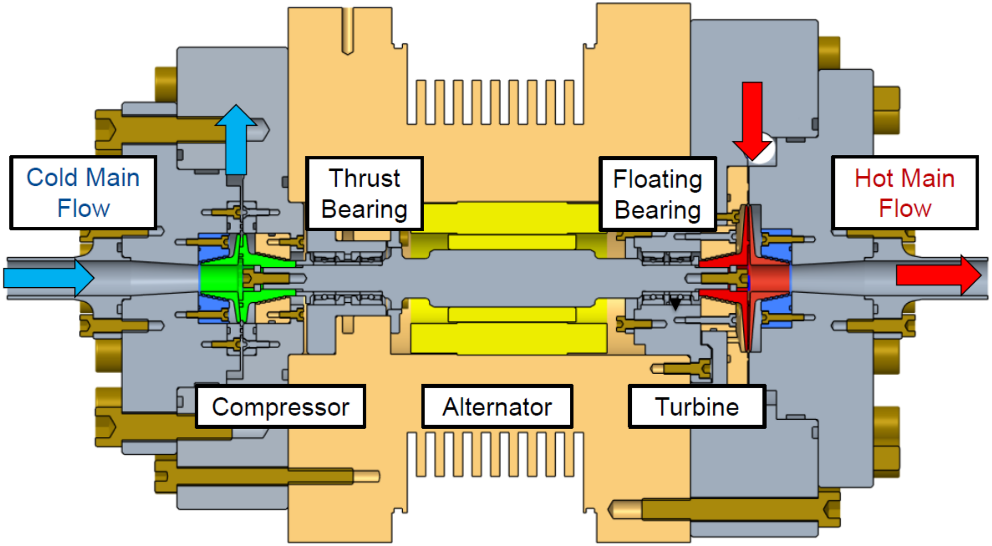
Source: A. Hacks, UDE

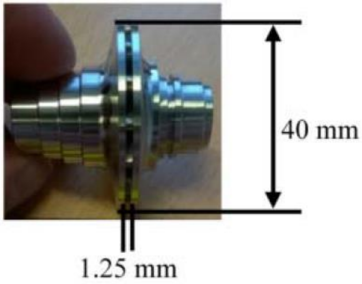
Modelling of the compact heat exchanger (CHX)



Source: M. Strätz, IKE

Modelling of the radial turbomachinery (RT)

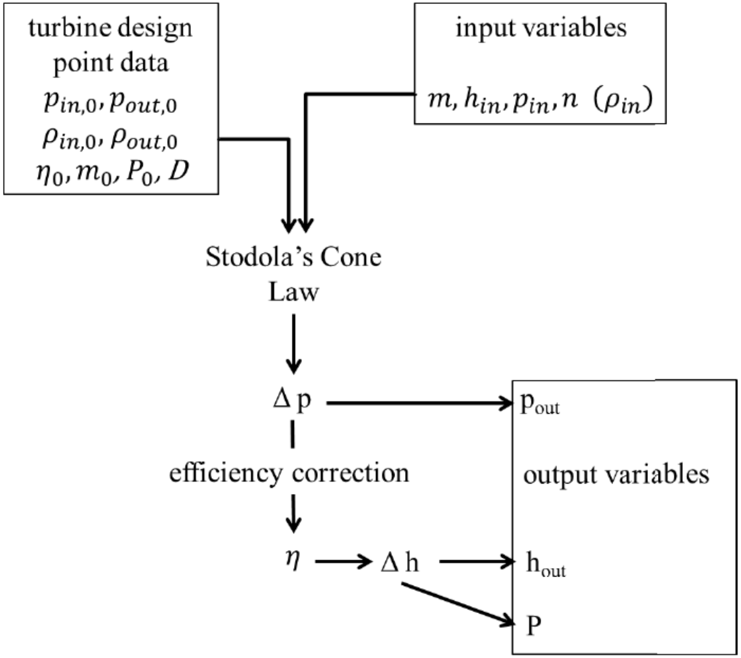




Source: A. Hacks, UDE

Radial turbomachinery (1): Previous status of ATHLET

- Junction related lumped parameter model
- Axial turbine model
- Representation in conservation equations: Δp and $\dot{Q}$
- $P = \eta_t \dot{m} \Delta h_{is} = \eta_t \dot{m} \frac{\Delta \mathrm{p}}{\tilde{\rho}}$
- Radial machines (previous modifications)
  - η for radial turbines
  - Δp not adapted (Stodola's cone law)
  - No suitable radial compressor model



Source: J. Venker, IKE

Radial turbomachinery (1): Improvement of ATHLET

$$Z_{cr}T_{cr} = Z_t T_t \left(\frac{1+\gamma}{2}\right)^{-1} \quad \text{(Eq. 1)} \quad P_{cr} = P_t \left(\frac{1+\gamma}{2}\right)^{-\frac{\gamma}{\gamma-1}} \quad a = \sqrt{n_s Z R T}$$

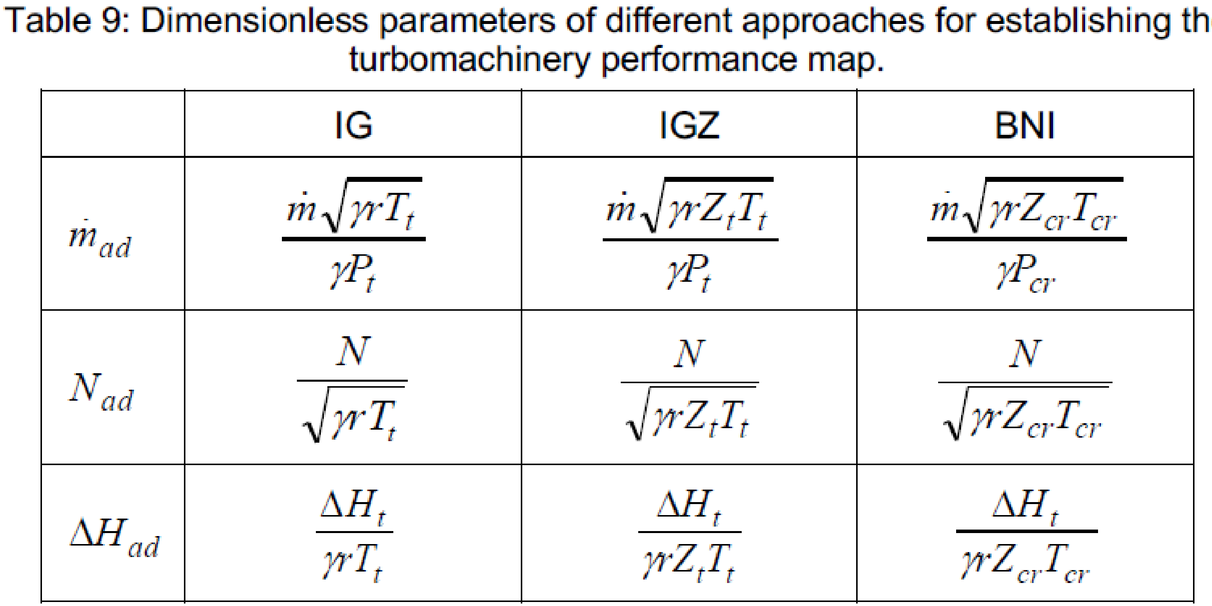


Table 9: Dimensionless parameters of different approaches for establishing the turbomachinery performance map.

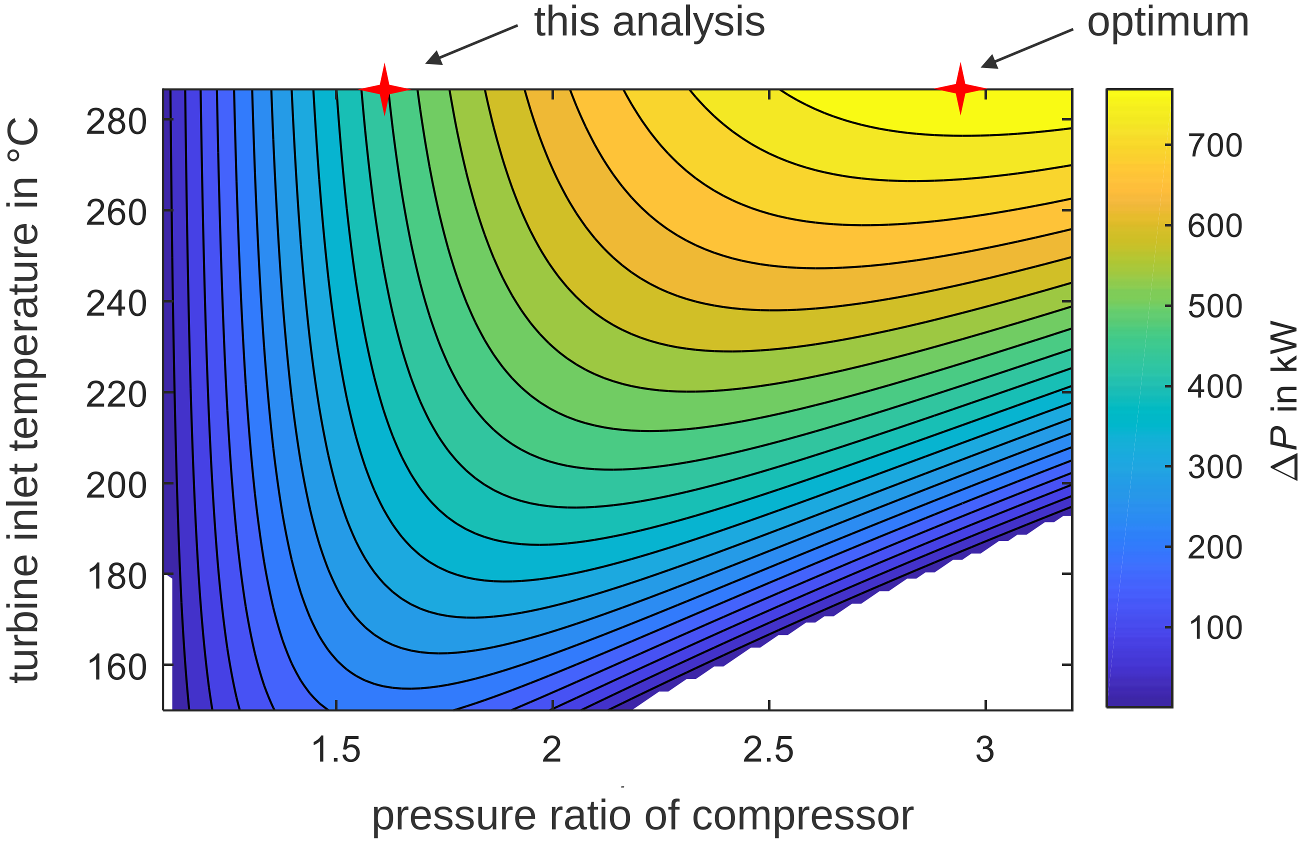
	IG	IGZ	BNI	New approach
$\dot{m}_{ad}$	$\frac{\dot{m}\sqrt{\gamma r T_t}}{\gamma P_t}$	$\frac{\dot{m}\sqrt{\gamma r Z_t T_t}}{\gamma P_t}$	$\frac{\dot{m}\sqrt{\gamma r Z_{cr} T_{cr}}}{\gamma P_{cr}}$	$\frac{\dot{m}\sqrt{n_s r Z_t T_t}}{n_s P_t}$
N_{ad}	$\frac{N}{\sqrt{\gamma r T_t}}$	$\frac{N}{\sqrt{\gamma r Z_t T_t}}$	$\frac{N}{\sqrt{\gamma r Z_{cr} T_{cr}}}$	$\frac{N}{\sqrt{n_s r Z_t T_t}}$
ΔH_{ad}	$\frac{\Delta H_t}{\gamma r T_t}$	$\frac{\Delta H_t}{\gamma r Z_t T_t}$	$\frac{\Delta H_t}{\gamma r Z_{cr} T_{cr}}$	$\frac{\Delta H_t}{n_s r Z_t T_t}$

- In terms of units, the diameter D is missing to be dimensionless ($m_{ad}: 1/D^2$, $N_{ad}: D$)
- [1]: Pham et al. (2016) An approach for establishing the performance maps of the sc-CO2 compressor: Development and qualification by means of CFD simulations. *International Journal of Heat and Fluid Flow*, 61, 379–94. https://doi.org/10.1016/j.ijheatfluidflow.2016.05.017

Radial turbomachinery (2): Improvement of ATHLET

- Similarity approach (Buckingham´s $\boldsymbol{\pi}$-Theorem)
- Dimensionless Performance Map: $(\eta, \frac{\Delta h_{is}}{c^2}) = f(Ma_a, Ma_\theta)$
- Dimensionless numbers are constant for similar operational points
- $Ma_a = \frac{\dot{m}}{\rho D^2 c}$ and $Ma_\theta = \frac{ND}{c}$ with speed of sound $c = \sqrt{\left(\frac{\delta p}{\delta \rho}\right)_s}$
- Entropy necessary as additional thermodynamic property
- $\gamma = \frac{c_p}{c_v}$ should not be used in similarity approach (especially for liquid CO_2)
- Power is calculated via the entropy (higher accuracy in the 2-phase region)
- Model is applicable for compressors and turbines as long as performance maps are available

Design point (DP): At maximum heat load



$DP_{opt} \neq DP_{analysis}$ due to application of compressor map

Boundary conditons

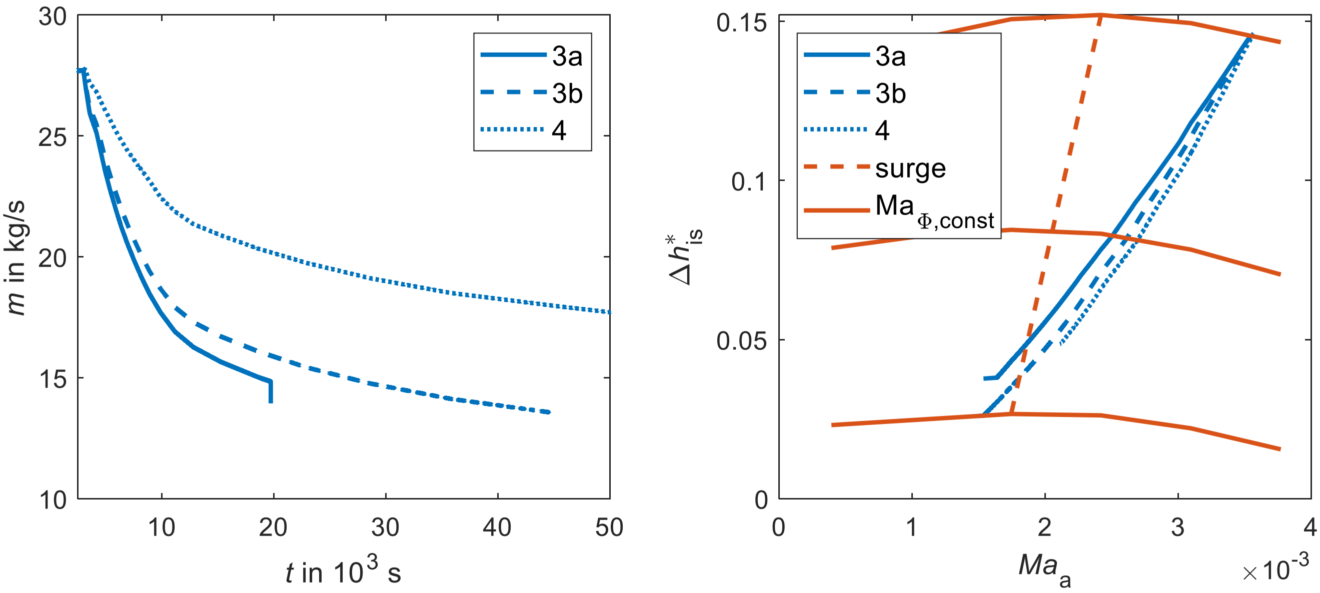
<table>
<tr><th></th><th>Case 1</th><th>Case 2</th><th>Case 3a</th><th>Case 3b</th><th>Case 4</th></tr>
<tr><td>$\dot{m}_{H2O}$</td><td>declining</td><td>constant</td><td>declining</td><td>declining</td><td>calculated</td></tr>
<tr><td>$h_{in,H2O}$</td><td colspan="5">at saturation point of steam (x=1) for all cases</td></tr>
<tr><td>$\vartheta_{in,H2O}$</td><td>constant</td><td>declining</td><td>constant</td><td>constant</td><td>declining</td></tr>
<tr><td>$\Delta T_{sub,H2O}$</td><td colspan="2">not constant (result)</td><td colspan="3">constant at design value</td></tr>
<tr><td>$\Delta T_{PP,UHS}$</td><td colspan="5">constant at design value except for 3b (increasing)</td></tr>
<tr><td>N</td><td colspan="2">constant at design value</td><td colspan="3">controlled to match $\Delta T_{sub,H2O}$</td></tr>
<tr><td>$\dot{Q}_{UHS}$</td><td colspan="5">controlled to match $\Delta T_{PP,UHS}$</td></tr>
<tr><td>$\dot{Q}_{CHX}$ / $\dot{Q}_{decay}$ (result)</td><td>>1</td><td>≈1</td><td colspan="3">1</td></tr>
</table>

Operational Analysis (4): Case 3 and 4: Boundary Conditions

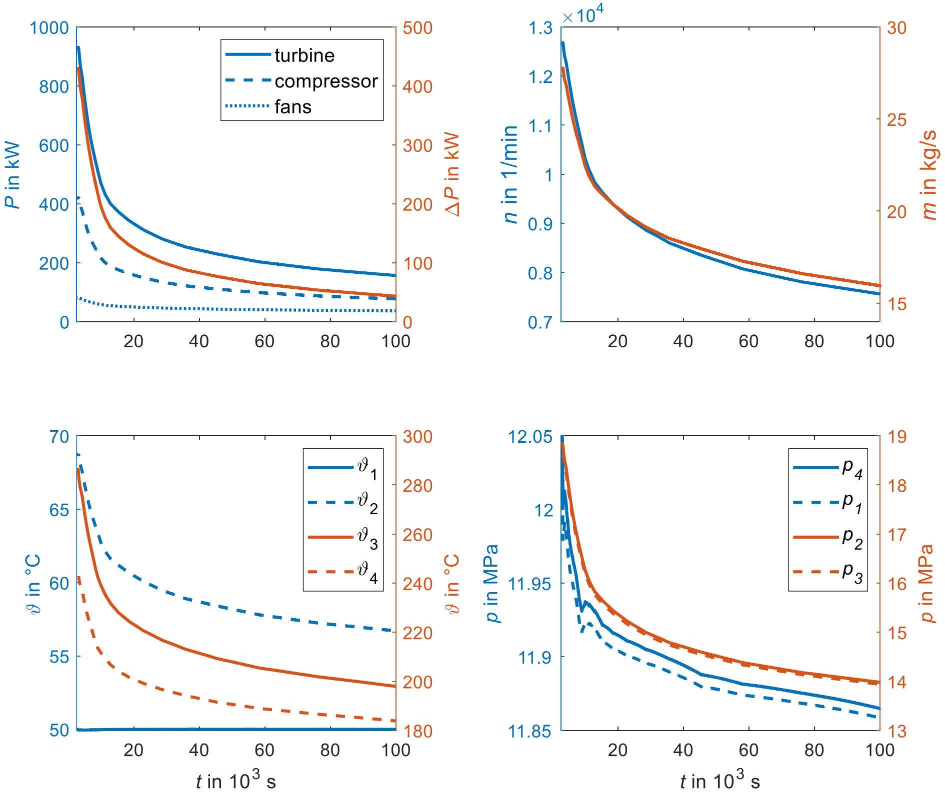
- Case 3 (a + b)
  - $T_{H2O,sat}$ constant, $\dot{m}_{H2O} \sim \dot{Q}_{decay}$(t) (like case 1)
  - Turbomachinery speed N is controlled to keep $\Delta T_{sub,H2O}$ constant
  - case 3b = case 3a + $T_{compressor,in}$ is increased
- Case 4
  - Turbomachinery speed is controlled to keep $\Delta T_{sub,H2O}$ constant
  - $T_{H2O,sat}$ is decreased to 200 °C (guessed) at end of the simulation (decrease is shaped like decay heat curve)
  - $\dot{m}_{H2O}$ is calculated to match $\dot{Q}_{decay} = \dot{m}\Delta h_{vap}$

Operational Analysis (5): Case 3 + 4: Results + Consequences

- $\dot{m}_{CO2}$ is decling: case 3a < case 3b < case 4
- Crossing of surge line and stop of simulation in case 3a and 3b



Case 4


P in kW
ΔP in kW
1000
800
600
400
200
0
500
400
300
200
100
0
turbine
compressor
fans
20
40
60
80
100
n in 1/min
m in kg/s
×10⁴
1.3
1.2
1.1
1
0.9
0.8
0.7
30
25
20
15
ϑ in °C
70
65
60
55
50
300
280
260
240
220
200
180
ϑ1
ϑ2
ϑ3
ϑ4
t in 10³ s
p in MPa
12.05
12
11.95
11.9
11.85
19
18
17
16
15
14
13
p4
p1
p2
p3