Numerical dimensioning of a Pre-Cooler for sCO_2 Power Cycles to utilize industrial Waste Heat

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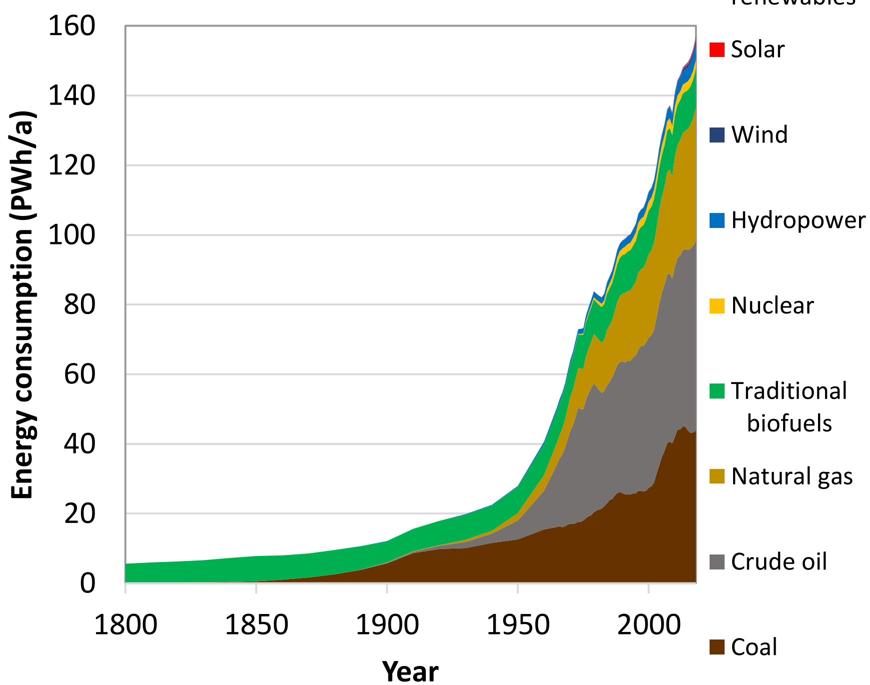
Outline

Motivation and Introduction

Industrial Waste Heat

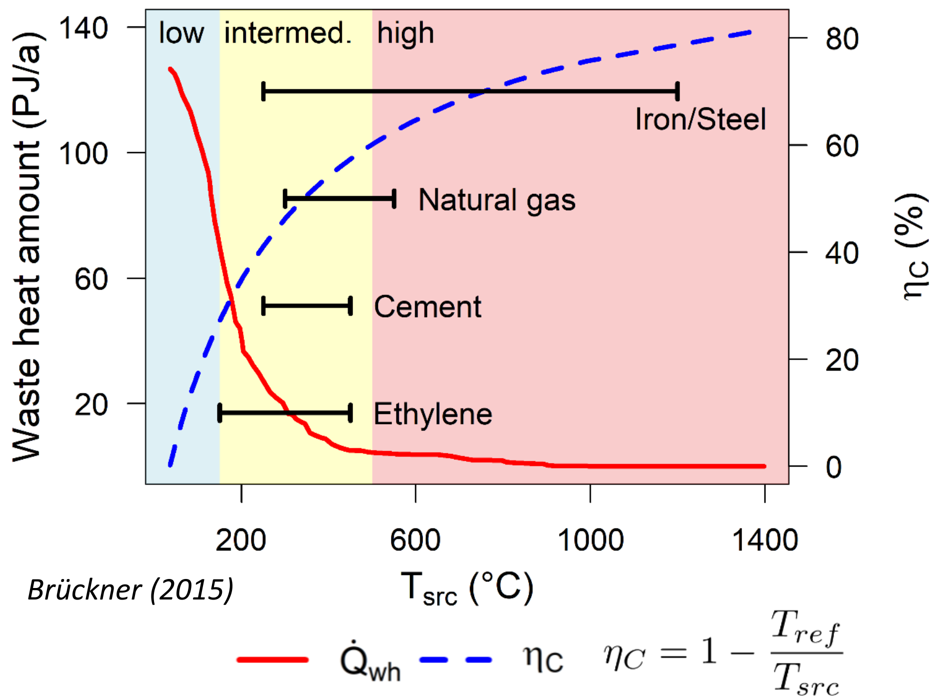
... is heat that arises both from equipment inefficiencies and from thermodynamic limitations on equipment and processes. DOE (2008)

Global primary energy consumption



Our World in Data (2018)

Industrial waste heat potential in Europe



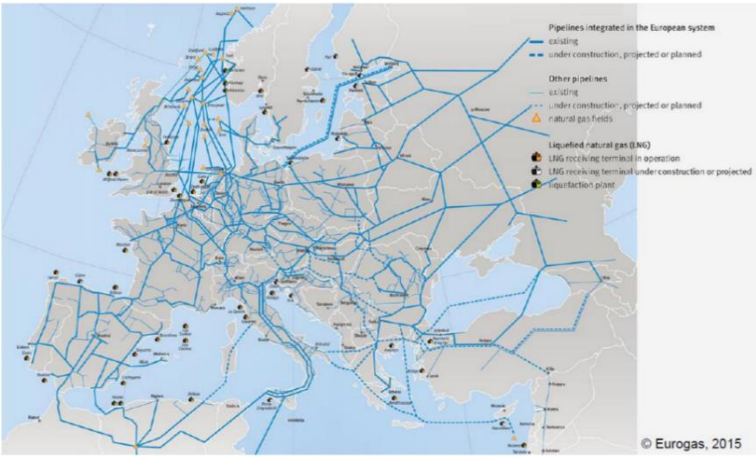
Brückner (2015)

→ **Waste heat utilization by sCO_2**

DRESDEN concept HZDR

Industrial Waste Heat

Gas grid Europe



Gas compressor stations

GASCADE Gastransport GmbH

Gas compressor stations:

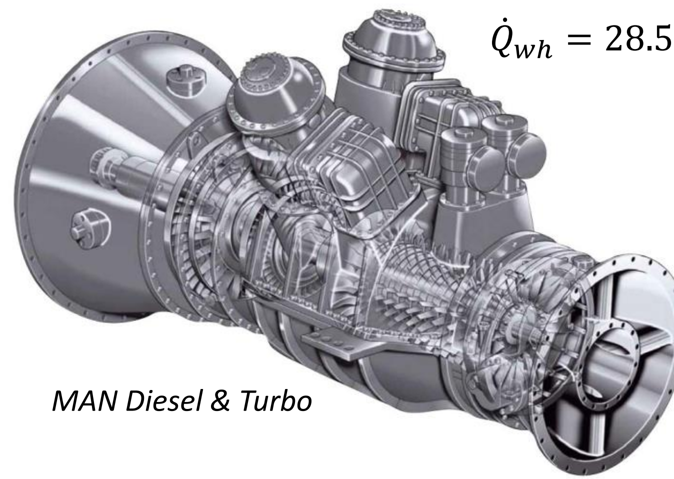
- Approximately each 100 km
- Compressor driven by gas turbine
- Exhaust gas stream potential waste heat
- Utilization by sCO2 power cycle

Industrial gas turbine

$$T_{wh} = 515\ °C$$

$$\dot{m}_{wh} = 49.1\ kg/s$$

$$\dot{Q}_{wh} = 28.5\ MW$$

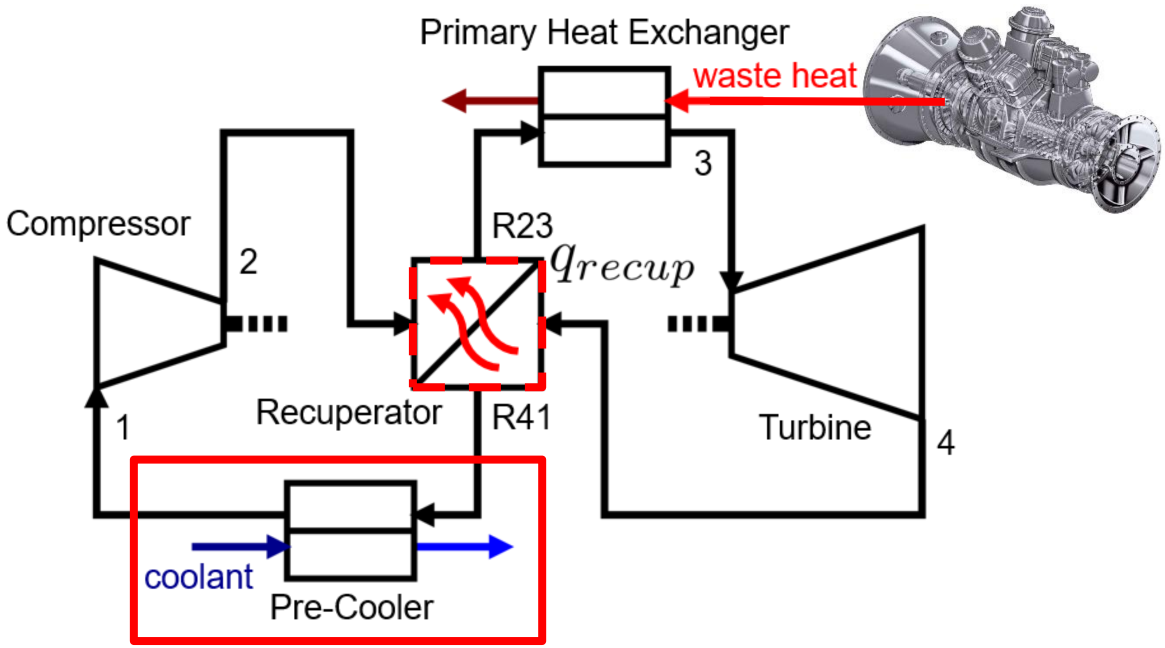


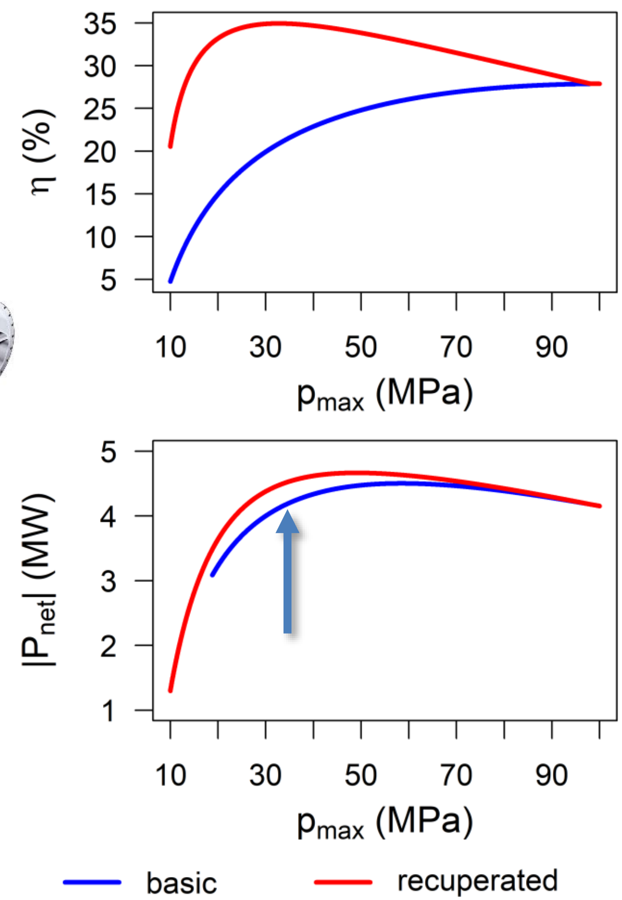
MAN Diesel & Turbo

Industrial Waste Heat

Cycle modelling

Component	Turbine	Compressor	Recuperator
Efficiency	90 %	85 %	90 %





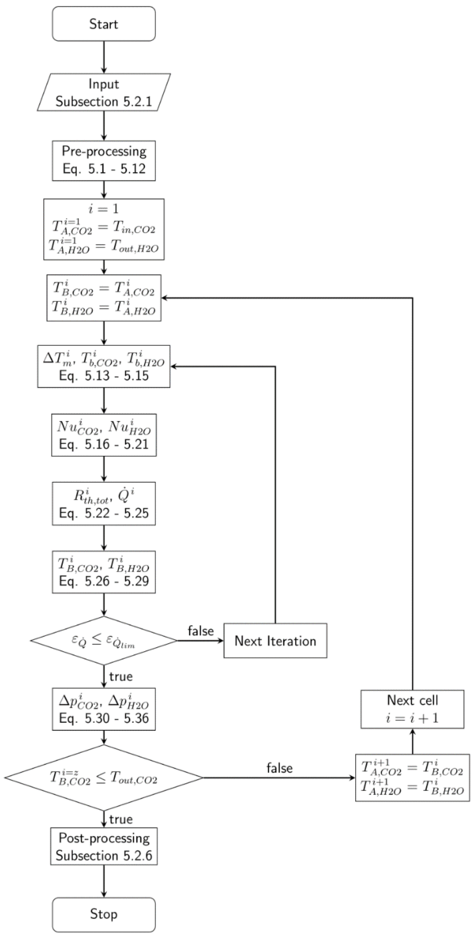
Both cases possible.

Liquid cooler for not recuperated cycle: p = 7.5 MPa, T = 352 °C

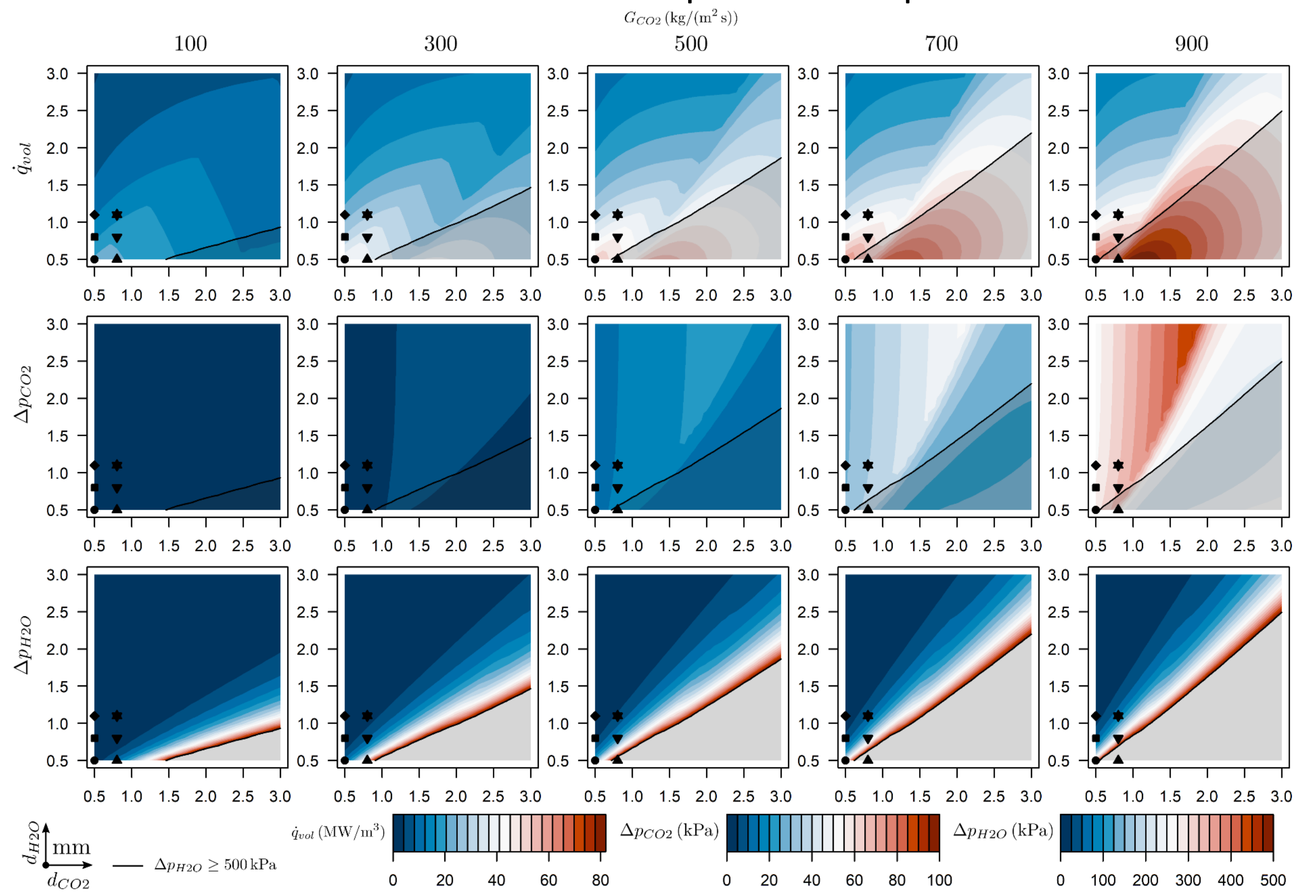
Numerical model

Analytical pre-calculation

Analytical pre-calculation



Heat transfer and pressure drop

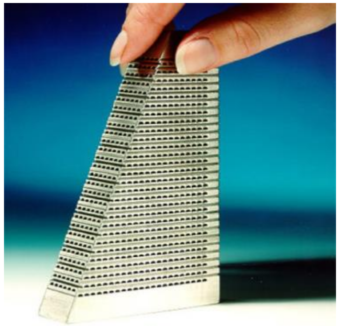


→ Prediction of pressure drop, heat transfer rate and channel length

Numerical model

Model and Boundary Conditions

Li (2011)



Conservation of mass

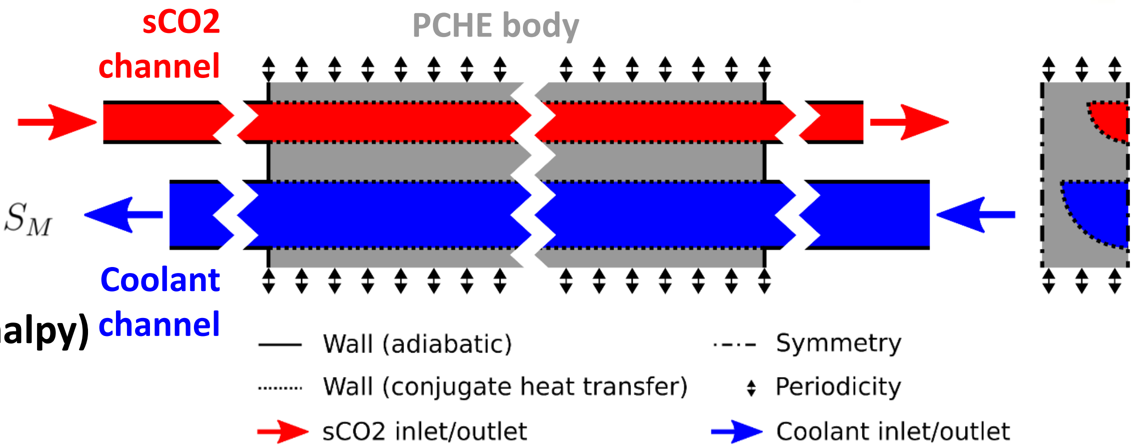
$$\frac{\partial \overline{u}_j}{\partial x_j} = 0$$

Conservation of momentum

$$\frac{\partial\left(\rho \overline{u}_j \overline{u}_i\right)}{\partial x_j} = -\frac{\partial \overline{p}}{\partial x_i} - \frac{\partial\left(\overline{\tau}_{ij} + \rho \overline{u_i' u_j'}\right)}{\partial x_j} + S_M$$

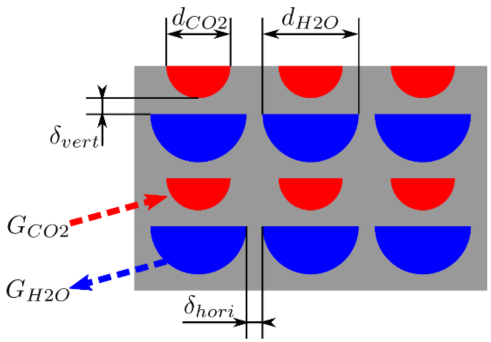
Conservation of energy (static enthalpy)

$$\frac{\partial\left(\rho \overline{u}_j \overline{h}\right)}{\partial x_j} = -\frac{\partial\left(\overline{\dot{q}}_j + \rho \overline{u_j' h'}\right)}{\partial x_j} + S_E$$



CFD Simulation

- solving steady-state RANS equations
- eddy viscosity modeled by SST turbulence model



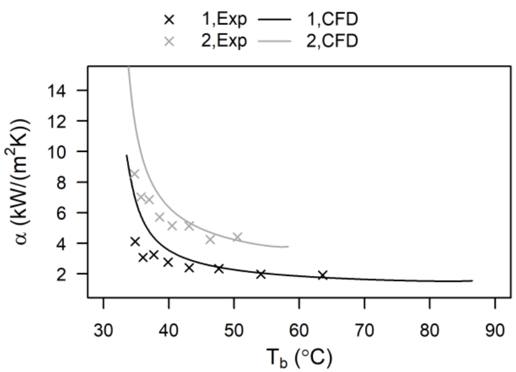
Configurations

- Different mass flow rates
- Different channel diameters
- Different internal fin heights

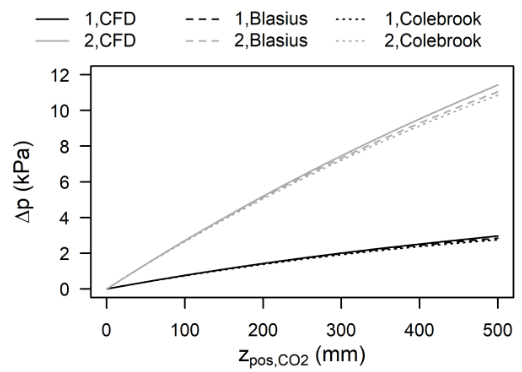
Numerical model

Mesh independency and model validation

Validation by experiments of Kruizenga A. et al. (2011):



(a) HTC

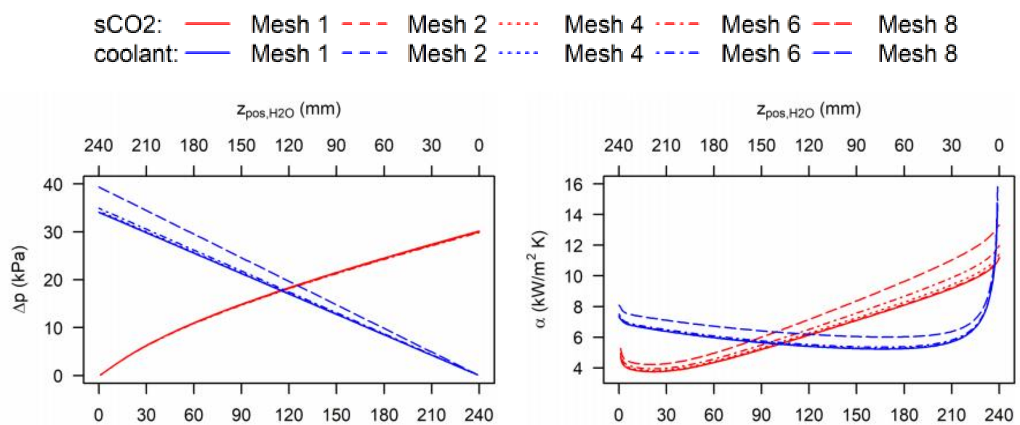


(b) Pressure drop

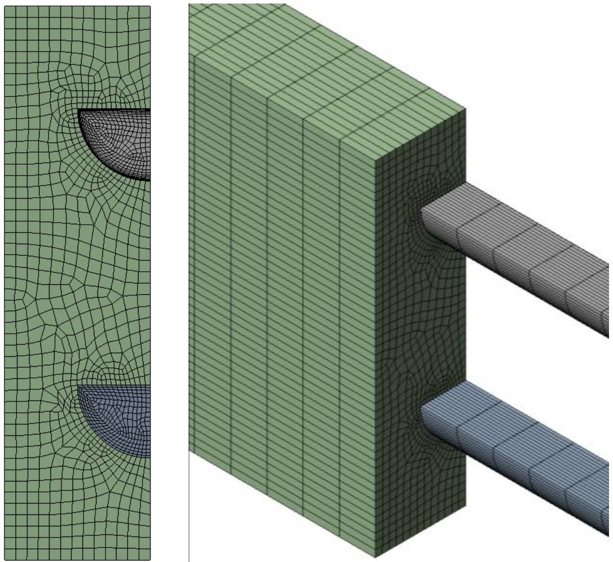
#	d_{CO2} mm	$l_{entr,CO2}$ mm	$l_{ch,CO2}$ mm	$l_{exit,CO2}$ mm	G_{CO2} kg/(m² s)	$\dot{q}_w$ kW/m²	$T_{in,CO2}$ °C	ρ_{ref} kg/m³
1	1.9	200	500	100	326	−23.2	90	217.0
2	1.9	200	500	100	762	−33.9	60	233.4

Kruizenga, A. et al., (2011).

Mesh independency for 1.5 to 9.6 million elements:



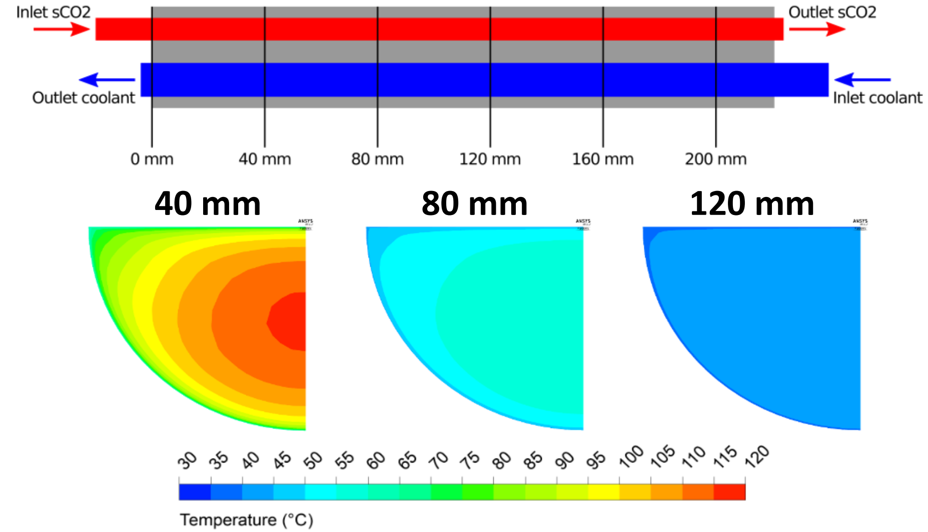
Inflation layer $y^+ < 1$



Motivation | Industrial Waste Heat | Numerical model | Results | Summary

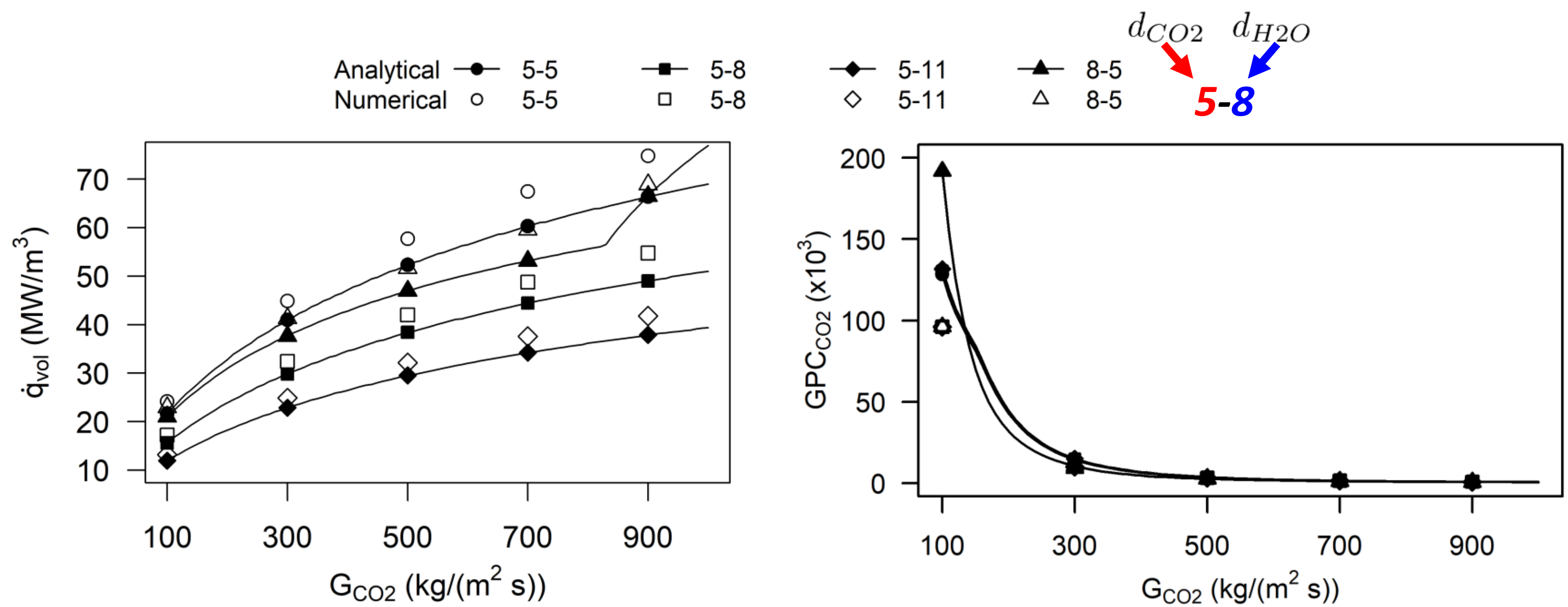
Results

Numerical results – channel diameter



Channel – reduced diameter increases

- Heat transfer surface and heat flow
- Higher pressure drop
- Small impact on global performance



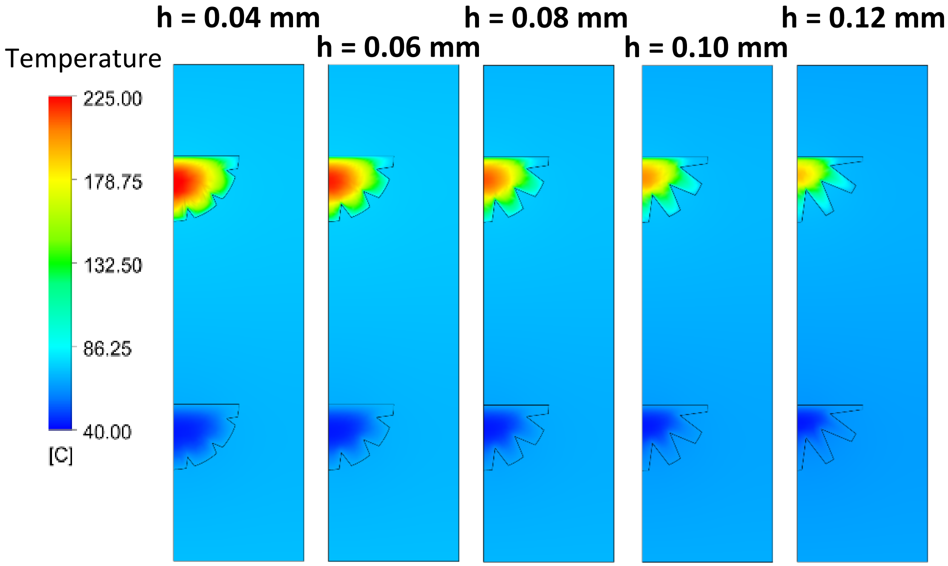
Member of the Helmholtz Association

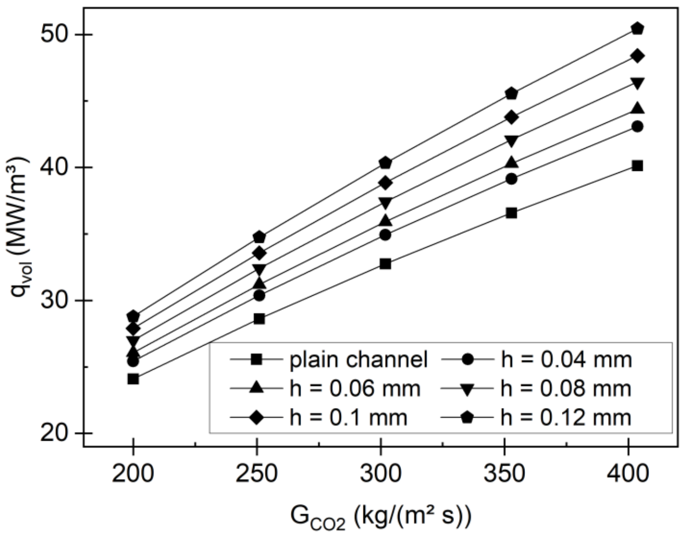
Results

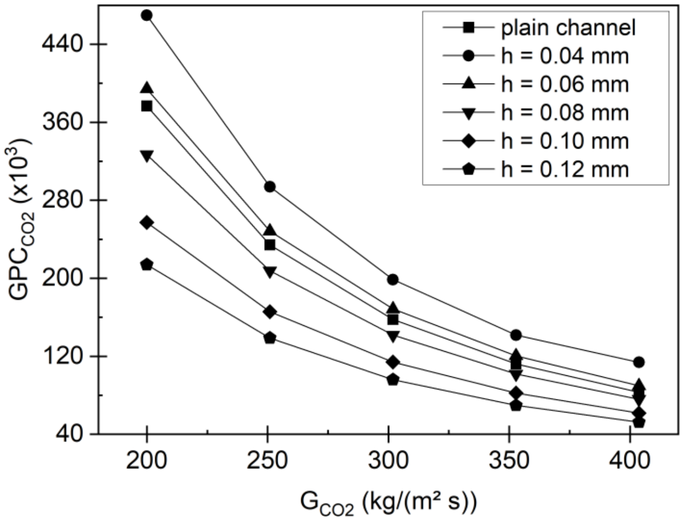
Numerical results – fin height

Internal fin design – fin height increases

- Heat transfer surface and heat flow
- Higher pressure drop
- Global performance optimum at h = 0.04 mm





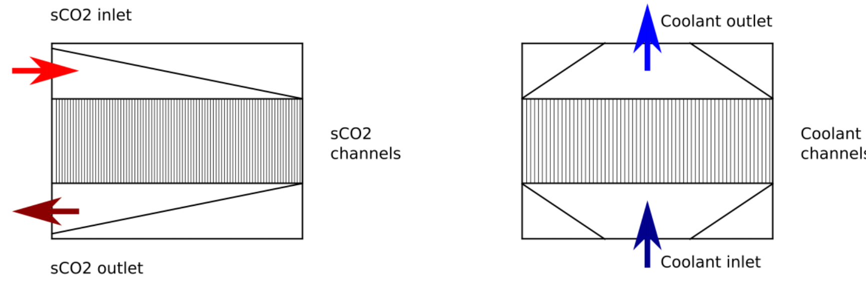


Results

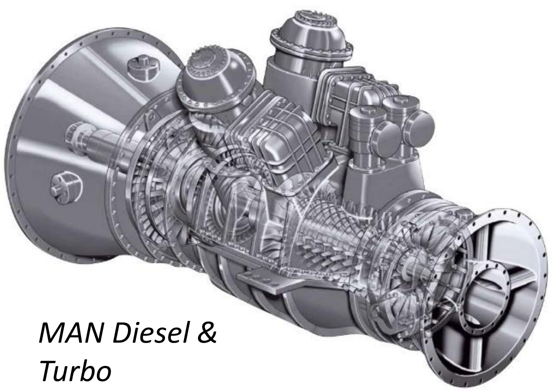
Design proposal

Modular design for industrial gas turbine

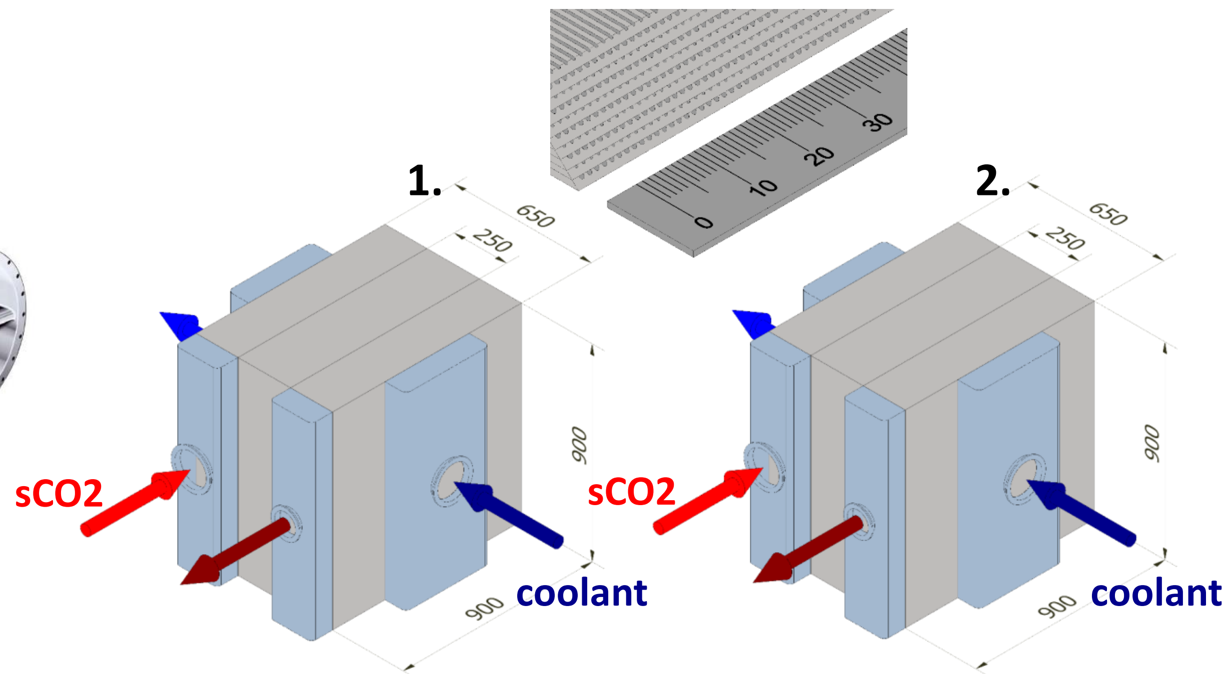
- m_{CO2} = 20 kg/s
- 860 plates, each 677 channels
- gas turbine WHR: 2 modules



(a) sCO2 plate (b) Coolant plate



MAN Diesel & Turbo



Summary and outlook

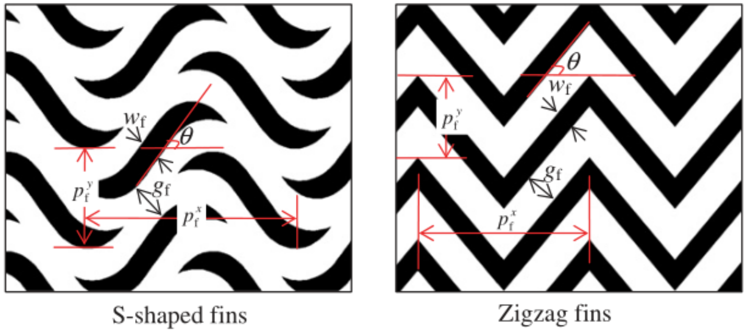
Design proposal

Summary

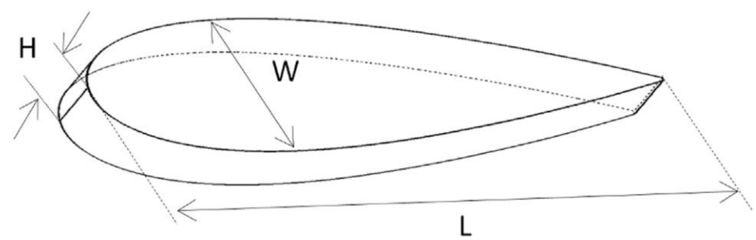
- Essential industrial waste heat sources identified
- Simple sCO2 power cycle model developed
- Analytical and numerical pre-cooler model developed and evaluated
- Numerical optimization of channel diameter and internal fin design
- Pe-cooler design proposal for application case

Outlook

- Extend model to further channel geometry
- Assess structural integrity
- Sophisticated flow arrangements (channel geometry)

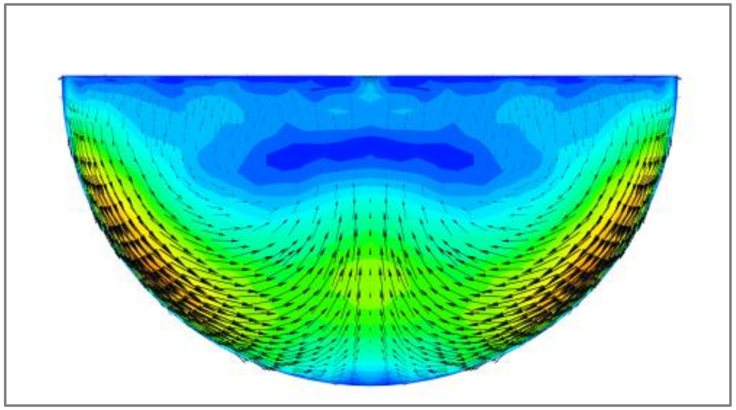


Ngo (2007)



Xu (2014)

Thank you for your attention.



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Ngo, T. L., Kato, Y., Nikitin, K. & Ishizuka, T. *Heat transfer and pressure drop correlations of microchannel heat exchangers with S-shaped and zigzag fins for carbon dioxide cycles.* Exp. Therm. Fluid Sci. 32, 560–570 (2007).

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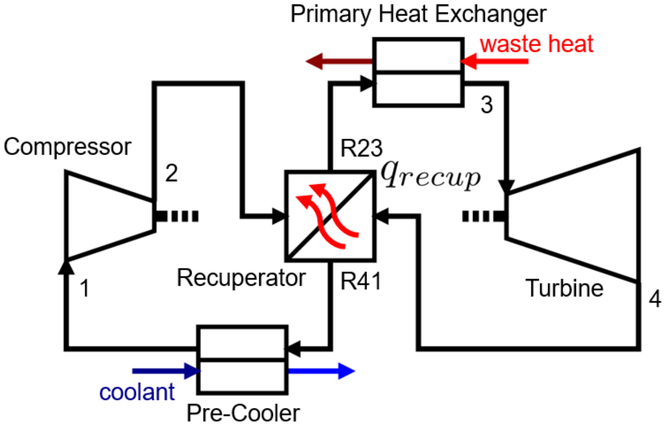
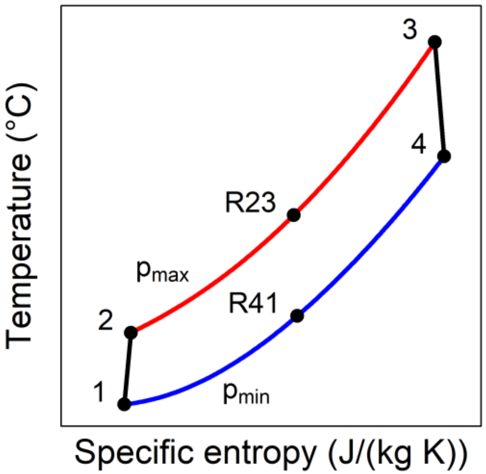
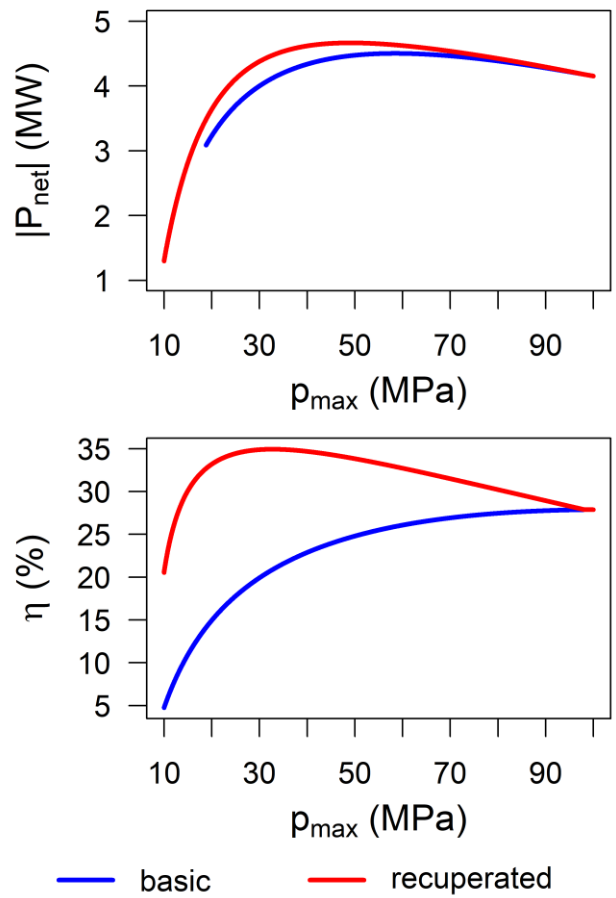
Power cycle

Industrial Waste Heat Utilization

Component	Turbine	Compressor	Recuperator
Efficiency	90 %	85 %	90 %







Method – Analytical Model

Background

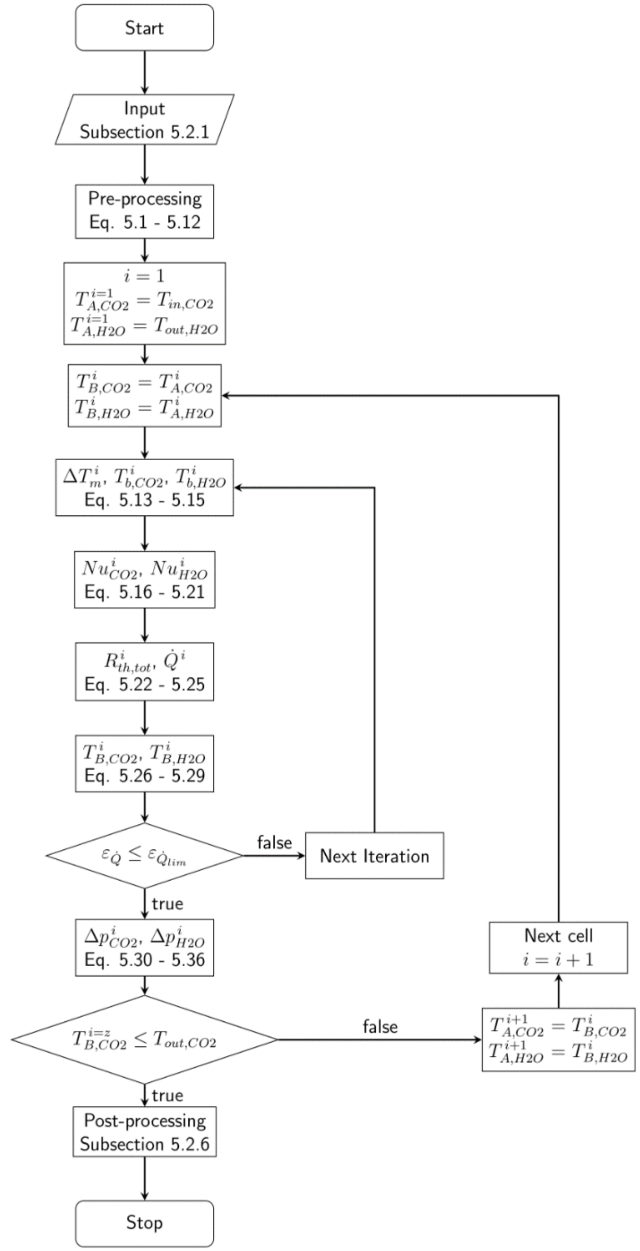
- → rapid assessment of different configurations
- → based on empirical heat transfer correlations
- → iterative process

Investigated parameter range

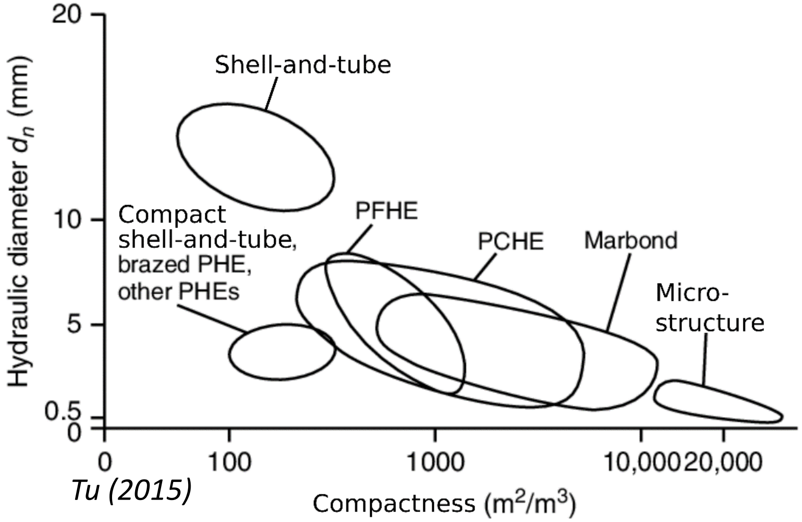
- → d_{CO2} = 0.5 – 3.0 mm
- → d_{H2O} = 0.5 – 3.0 mm
- → G_{CO2} = 100 – 900 kg/(m² s)

Results

- → required PCHE body length
- → volumetric heat flux
- → sCO2 and coolant pressure drop

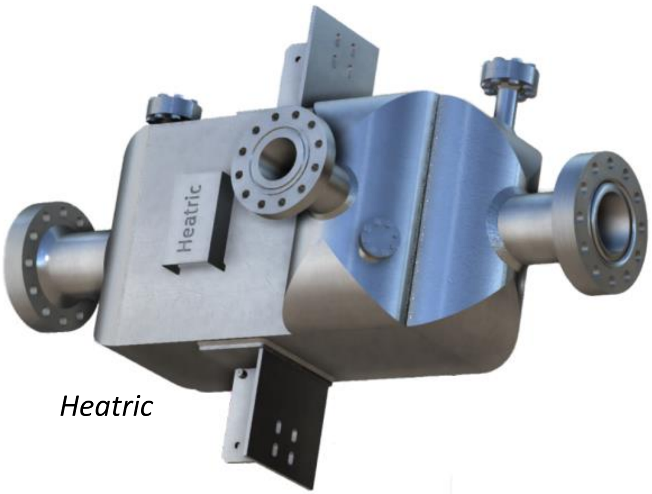


Heat Exchangers

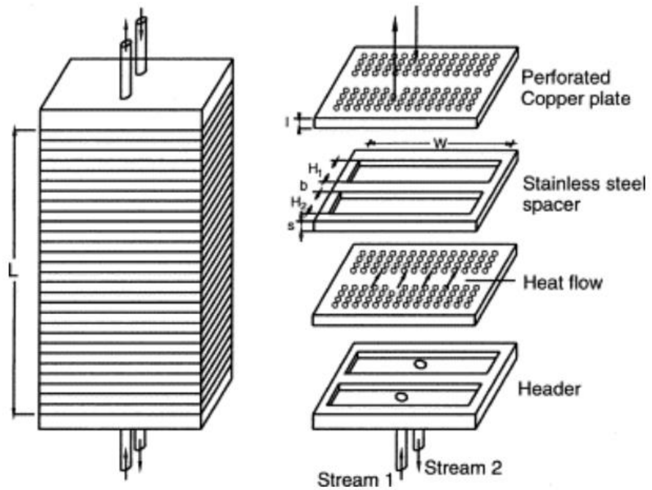


Tu (2015)

Heat exchanger design	p_{max} MPa	T_{max} °C	β m^2/m^3
Shell-and-tube	60	500	≈ 100
PHE (gaskets)	2	200	≈ 200
PHE (brazed)	3	220	≈ 200
PFHE	9	650	800-1500
TFHE	20	200	720-3300
Spiral	2	400	≈ 200
PCHE	>40	>800	200-5000
Perforated plate matrix	100	800	≈ 6000
Marbond	>40	900	≈ 10000
Microstructure	>50	>800	10000-30000



Heatric



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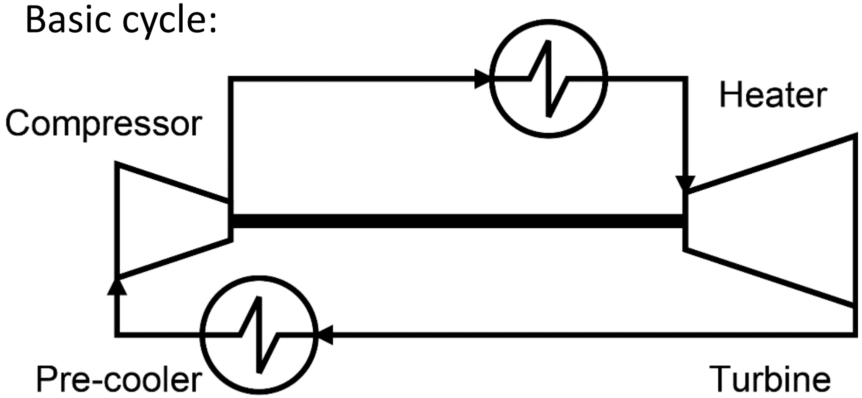
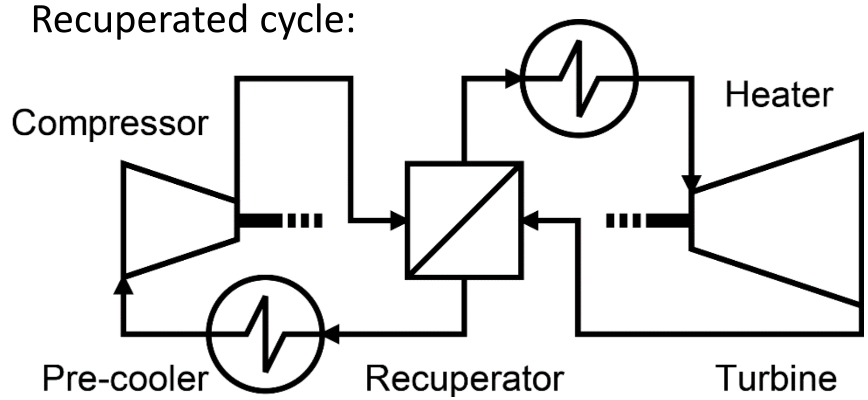
sCO2 Power Cycle

Cycle Layouts

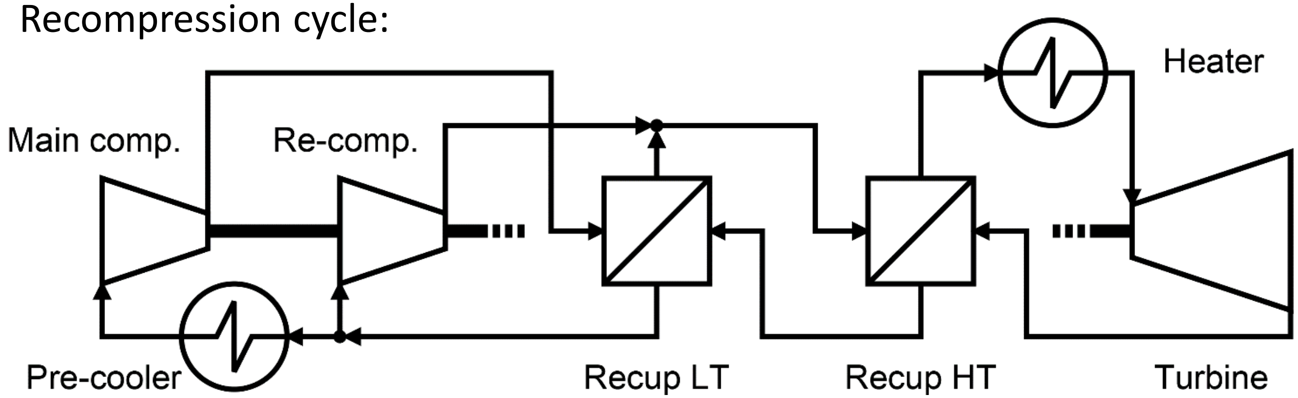
Basic cycle:



Recuperated cycle:



Recompression cycle:



Supercritical Carbon Dioxide (sCO2)

Heat Transfer Correlations

Rate of heat flow:

$$\dot{Q} = \alpha\, A_{ht}\,(T_w - T_b)$$

Nusselt number:

$$Nu = \frac{\alpha\, L_c}{\lambda}$$

Reynolds number:

$$Re = \frac{\rho\, u\, d_{hyd}}{\mu}$$

Prandtl number:

$$Pr = \frac{\nu}{a}$$

Relationship forced convection

$$Nu = \mathrm{f}\,(Re, Pr)$$

Gnielinski (1975):

$$Nu_b = \frac{\frac{f_b}{8}\, Re_b\, Pr_b}{1 + 12.7\sqrt{\frac{f_b}{8}}\left(Pr_b^{\frac{2}{3}} - 1\right)}$$

$$f_b = (1.8 \log_{10} Re_b - 1.5)^{-2}$$

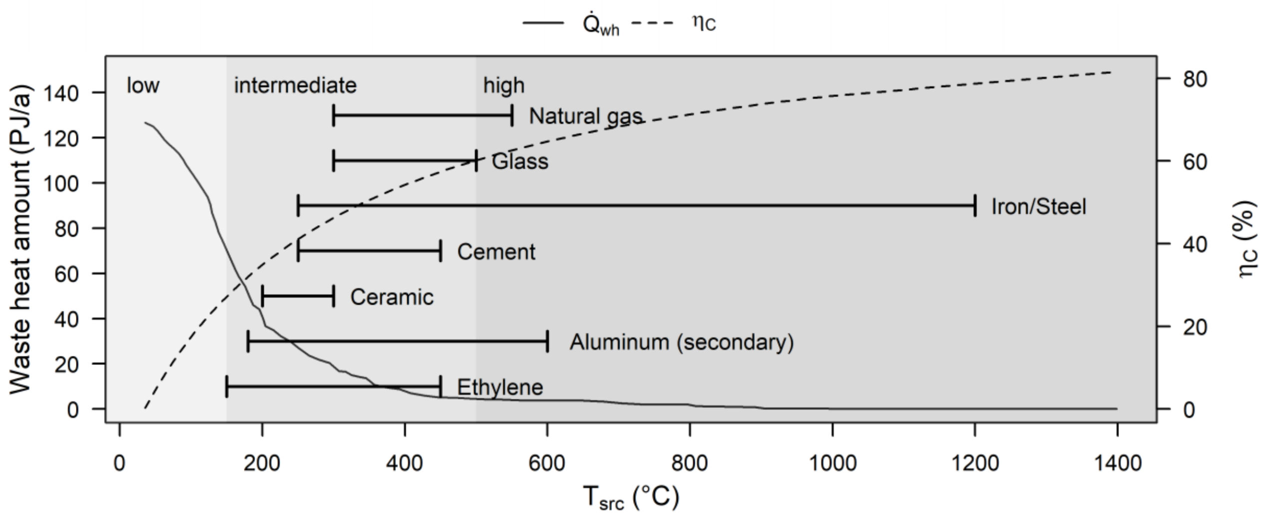
Jackson (2002):

$$Nu_b = 0.0183\, Re_b^{0.82}\, Pr_b^{0.5} \left(\frac{\rho_w}{\rho_b}\right)^{0.3} \left(\frac{\bar{c}_p}{c_{p,b}}\right)^{n}$$

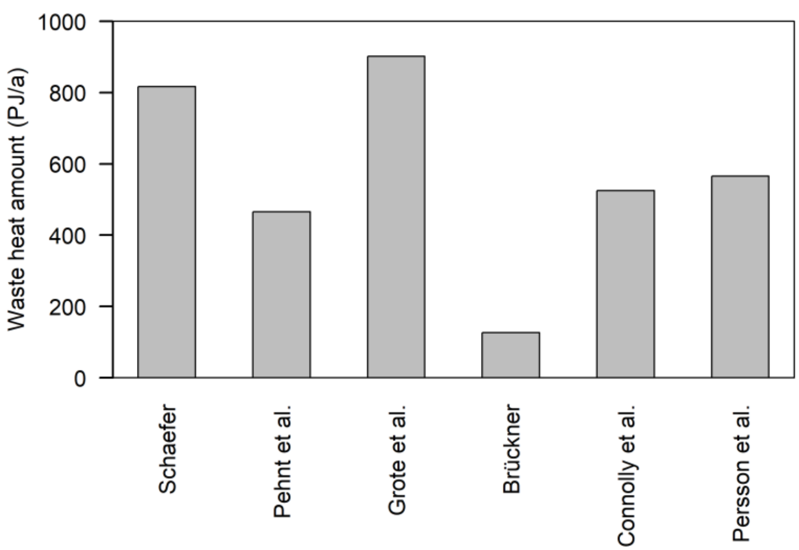
$$n = \begin{cases} 0.4 & \text{for } T_b < T_w < T_{pc} \text{ or } 1.2\,T_{pc} < T_b < T_w \\ 0.4 + 0.2\left(\frac{T_w}{T_{pc}} - 1\right) & \text{for } T_b < T_{pc} < T_w \\ 0.4 + 0.2\left(\frac{T_w}{T_{pc}} - 1\right)\left[1 - 5\left(\frac{T_b}{T_{pc}} - 1\right)\right] & \text{for } T_{pc} < T_b < 1.2\,T_{pc} \text{ or } T_b < T \end{cases}$$

Industrial Waste Heat

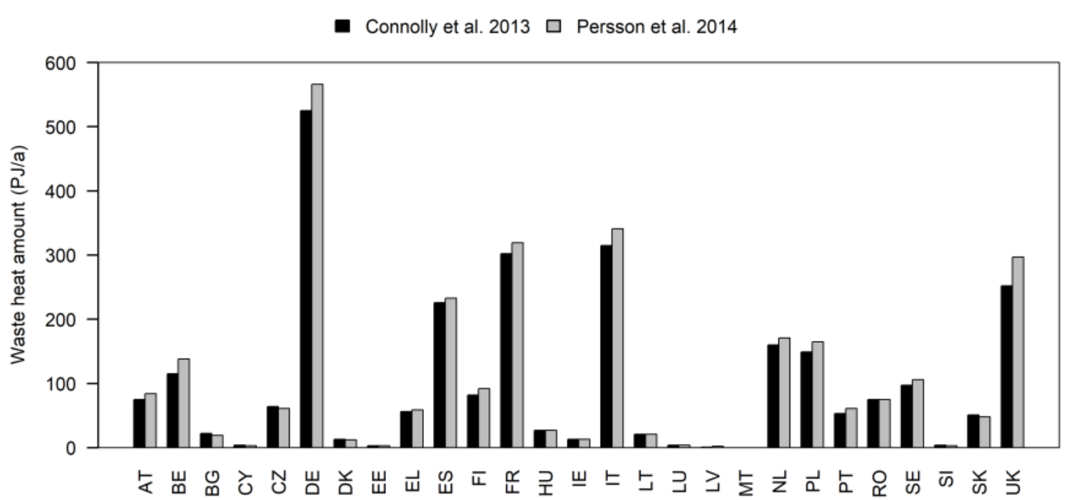
Situation



Germany:

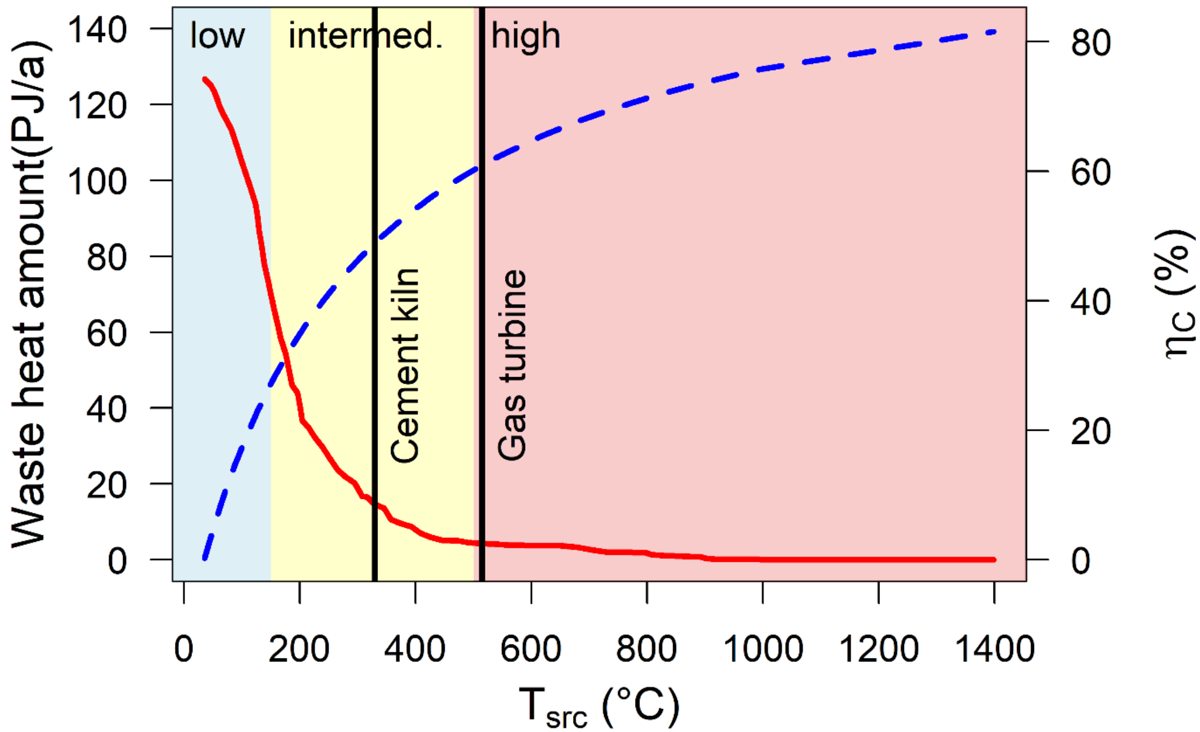


EU-27:



Industrial Waste Heat

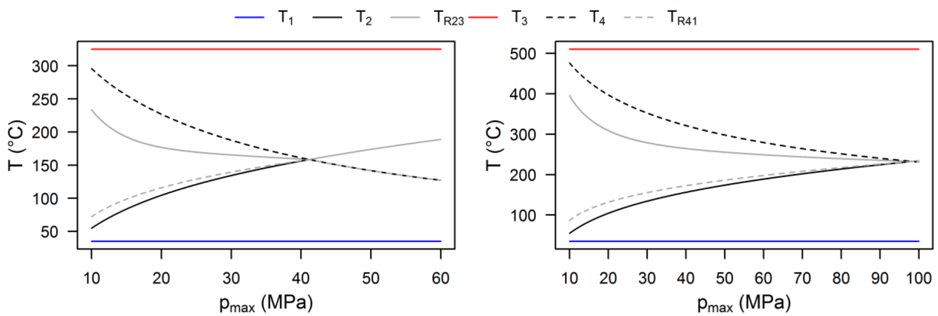
Waste Heat Sources



Power Cycle Model

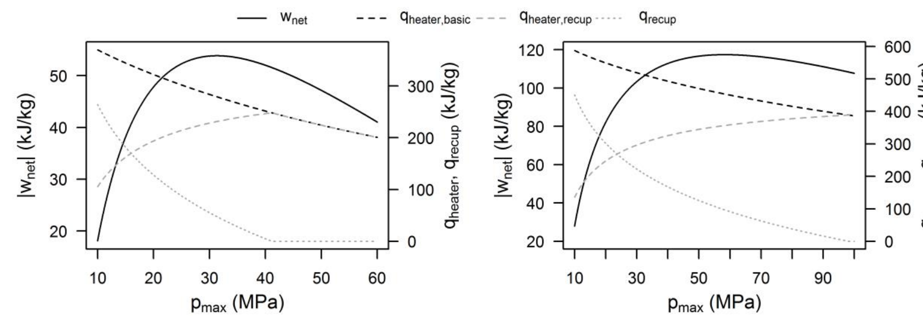
Results

Temperatures of thermodynamic states



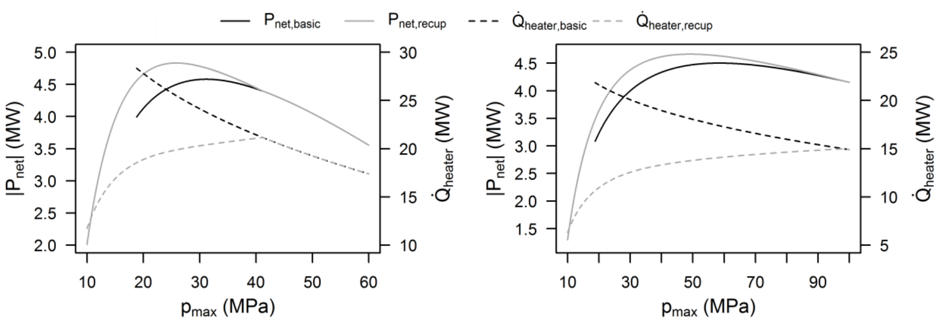
(a) Cement kiln WHR system (b) Gas turbine WHR system

Specific net work and specific heats



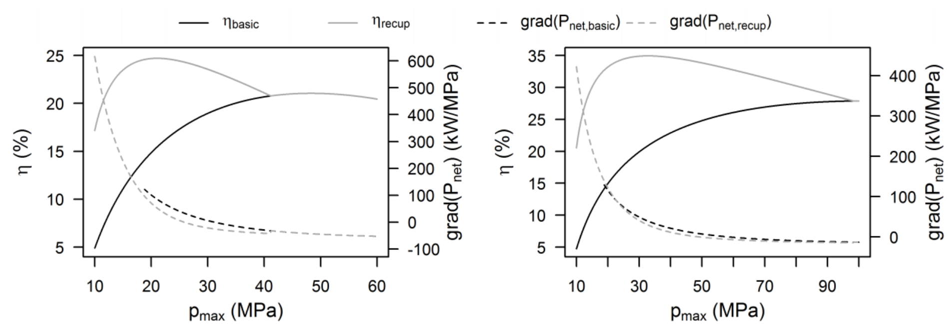
(a) Cement kiln WHR system (b) Gas turbine WHR system

Net power and heater heat duty



(a) Cement kiln WHR system (b) Gas turbine WHR system

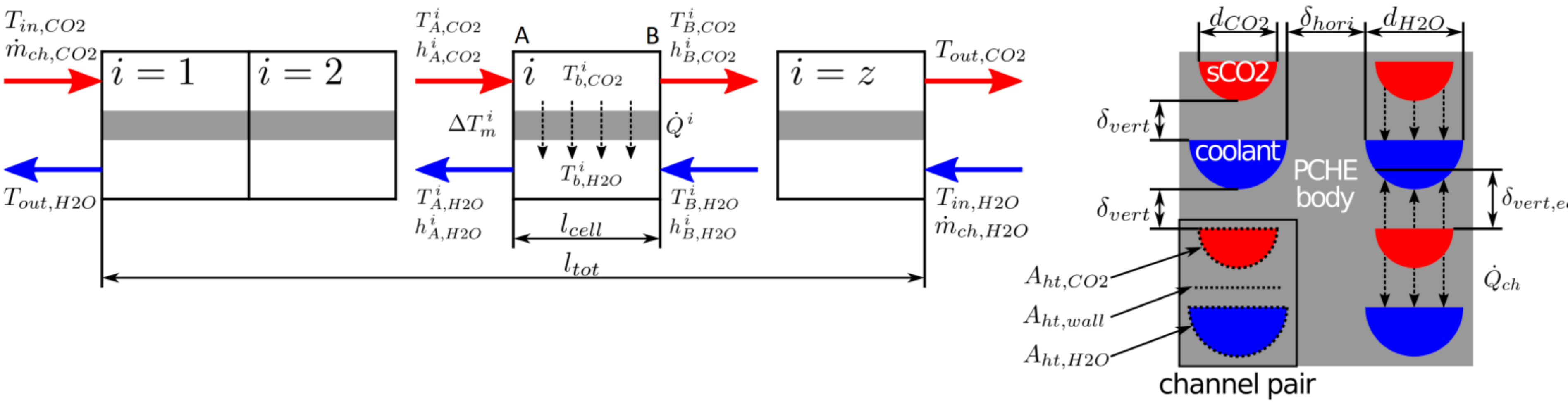
Efficiency and power gradient



(a) Cement kiln WHR system (b) Gas turbine WHR system

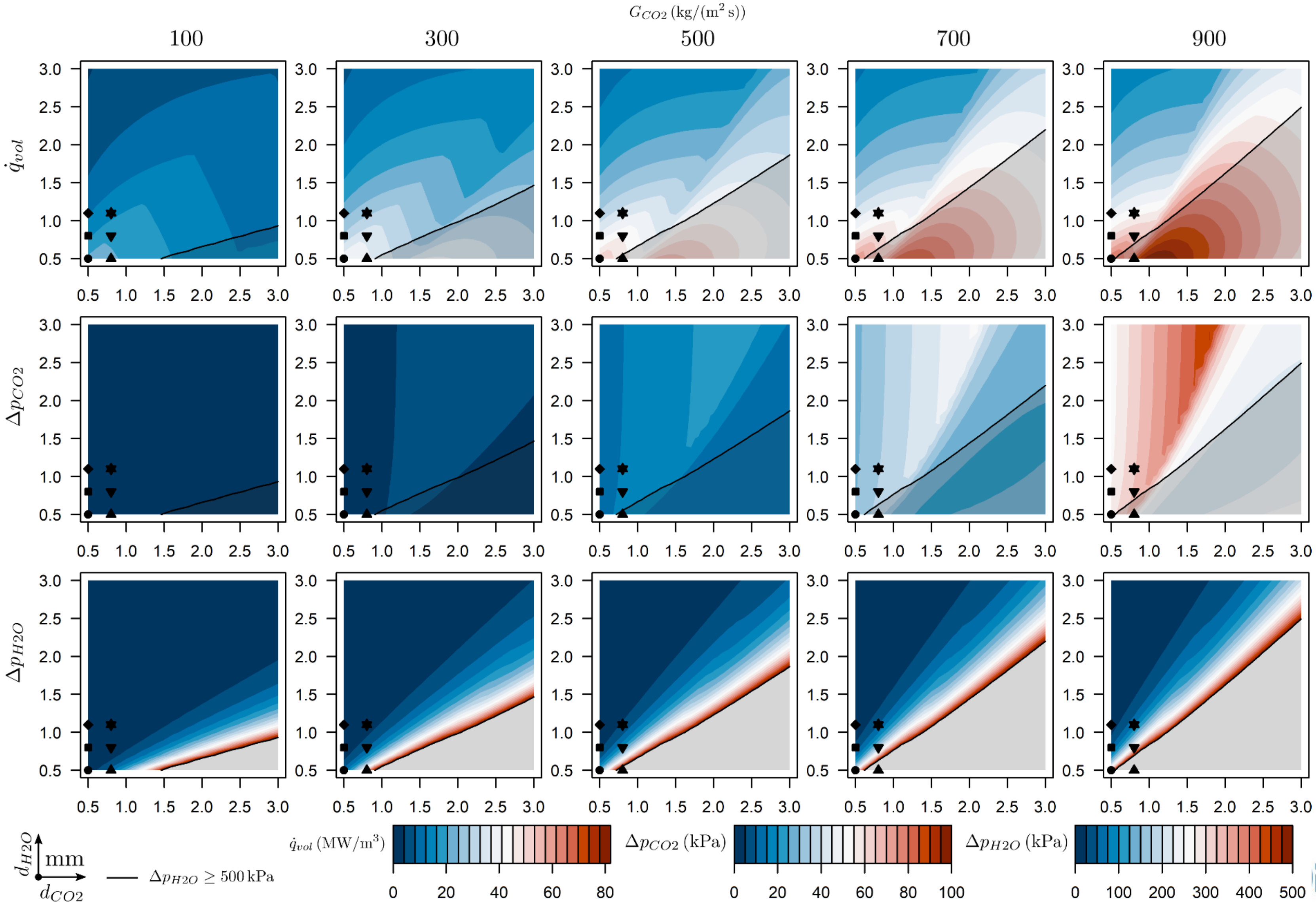
Analytical Model

Calculation Scheme



Analytical Model

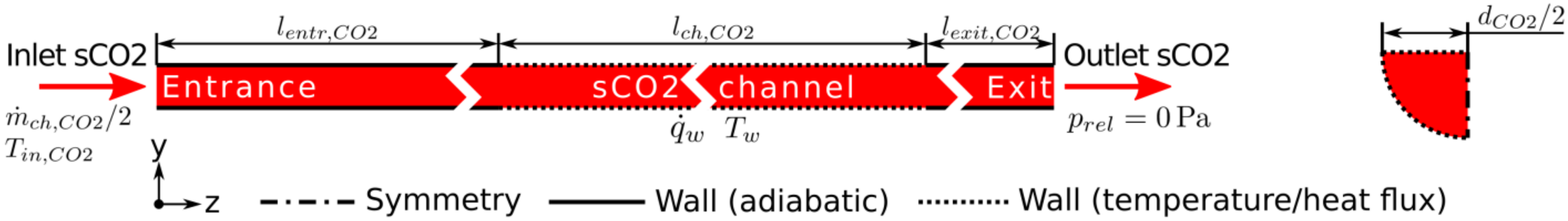
Results – Gas Turbine WHR



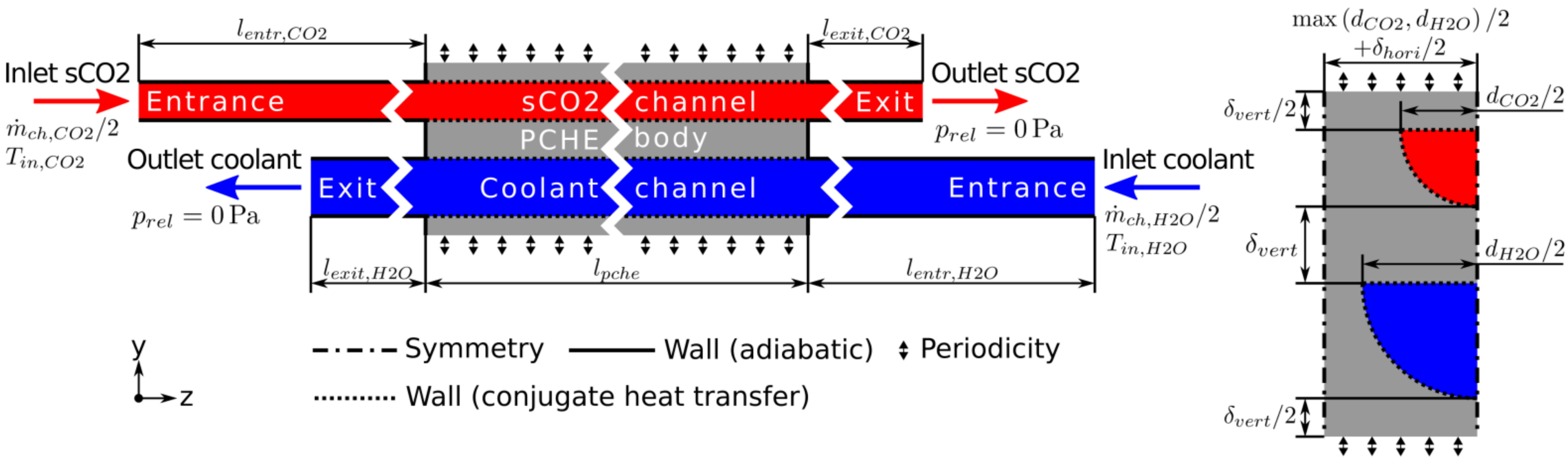
Numerical Model

Geometry

Single-channel model

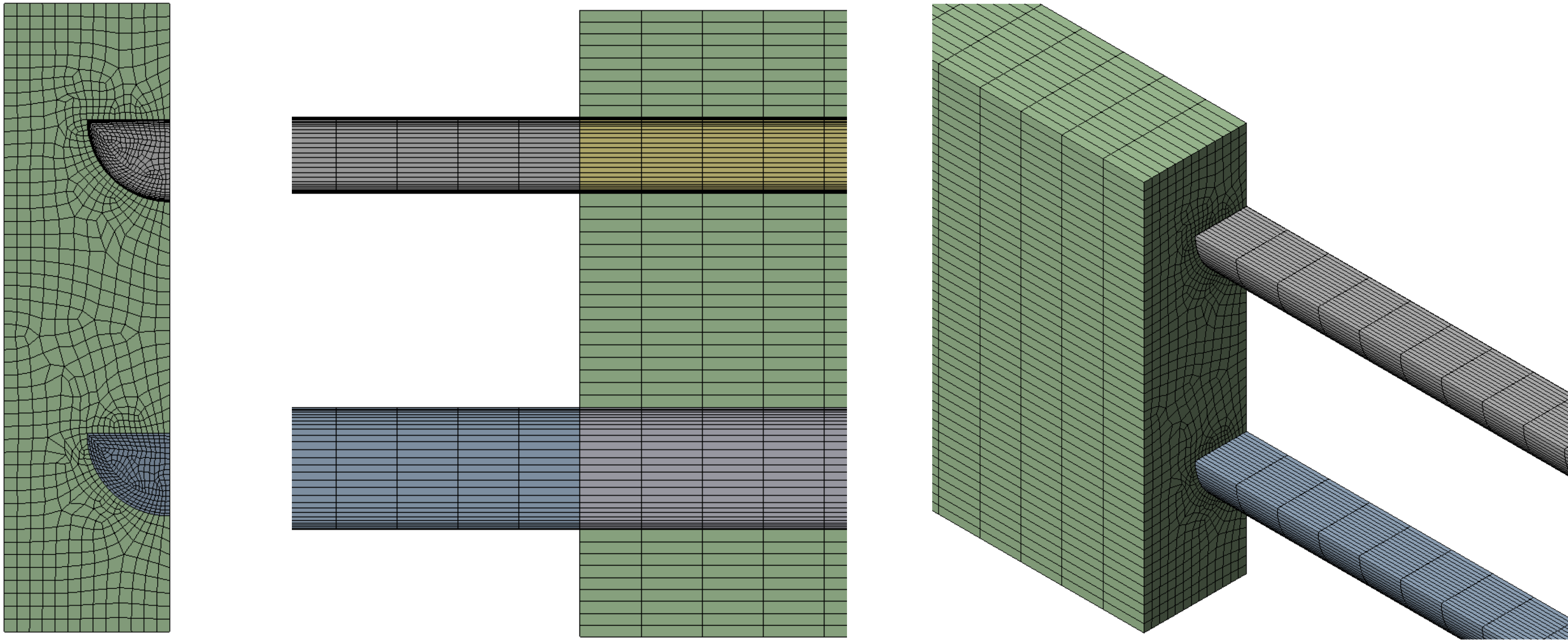


Pre-cooler model



Numerical Model

Mesh



→ 1.5 – 9.6 million elements

→ inflation layer $y^+ < 1$

Numerical Model

Turbulence Model

Reynolds stresses

$$-\rho\overline{u_i'u_j'} = \mu_t\left(\frac{\partial\overline{u}_i}{\partial x_j} + \frac{\partial\overline{u}_j}{\partial x_i}\right) - \frac{2}{3}\rho k\delta_{ij}$$

Turbulent kinetic energy

$$k = \frac{1}{2}\overline{u_i u_j}$$

Turbulent kinetic energy

$$\frac{\partial\left(\rho u_j k\right)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_{k3}}\right)\frac{\partial k}{\partial x_j}\right] + P_k - \beta'\rho k\omega$$

Turbulent frequency

$$\frac{\partial\left(\rho u_j \omega\right)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_{\omega 3}}\right)\frac{\partial \omega}{\partial x_j}\right] + \alpha_3\frac{\omega}{k}P_k - \beta_3\rho\omega^2 + (1 - F_1)\frac{2\rho}{\sigma_{\omega 2}\omega}\frac{\partial k}{\partial x_j}\frac{\partial \omega}{\partial x_j}$$

Blending function for wall distance

$$F_1 = \tanh\left[\left(\min\left[\max\left(\frac{\sqrt{k}}{\beta'\omega y}, \frac{500\nu}{\omega y^2}\right), \frac{4\rho k}{CD_{kw}\sigma_{\omega 2}y^2}\right]\right)^4\right]$$

Kinematic eddy-viscosity

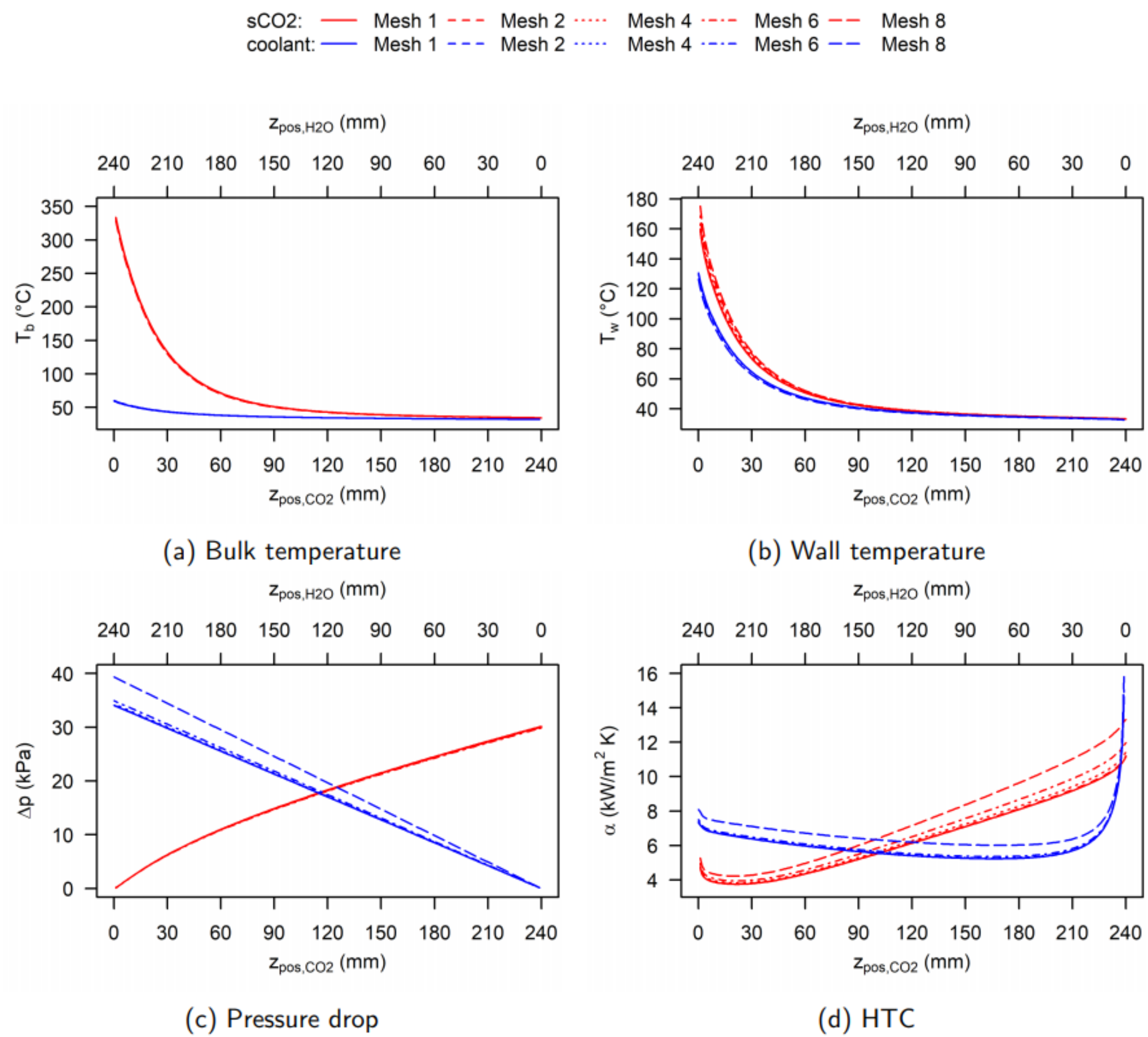
$$\nu_t = \frac{a_1 k}{\max\left(a_1\omega, SF_2\right)}$$

Blending function to restrict eddy-viscosity limiter

$$F_2 = \tanh\left(\left[\max\left(\frac{2\sqrt{k}}{\beta'\omega y}, \frac{500\nu}{\omega y^2}\right)\right]^2\right)$$

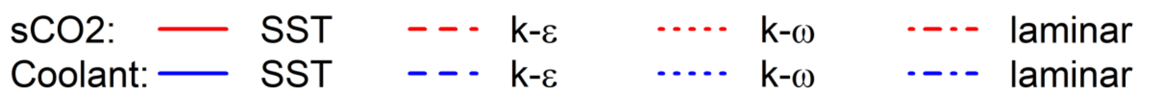
Numerical Model

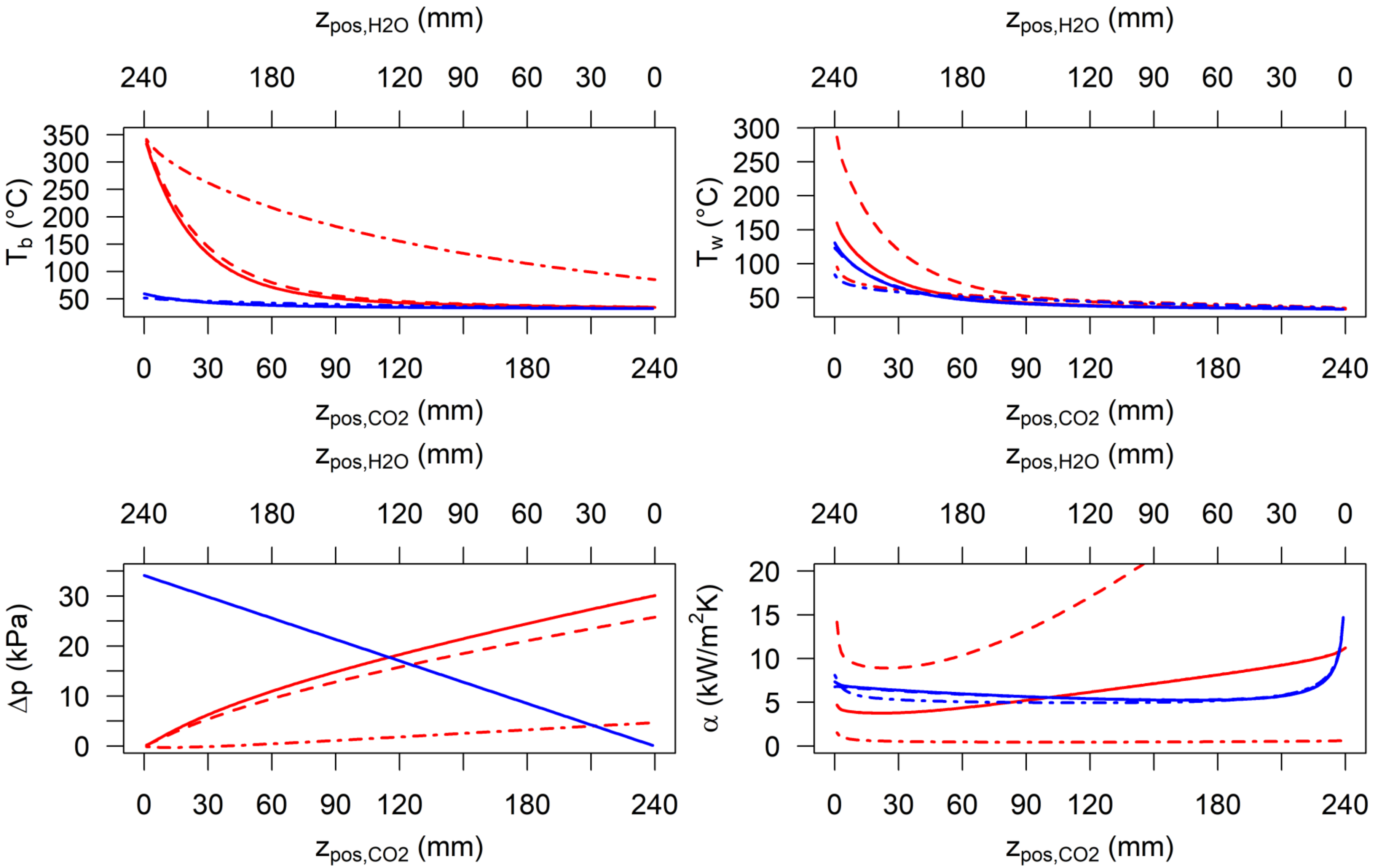
Mesh Independence Study



Numerical Model

Turbulence Models

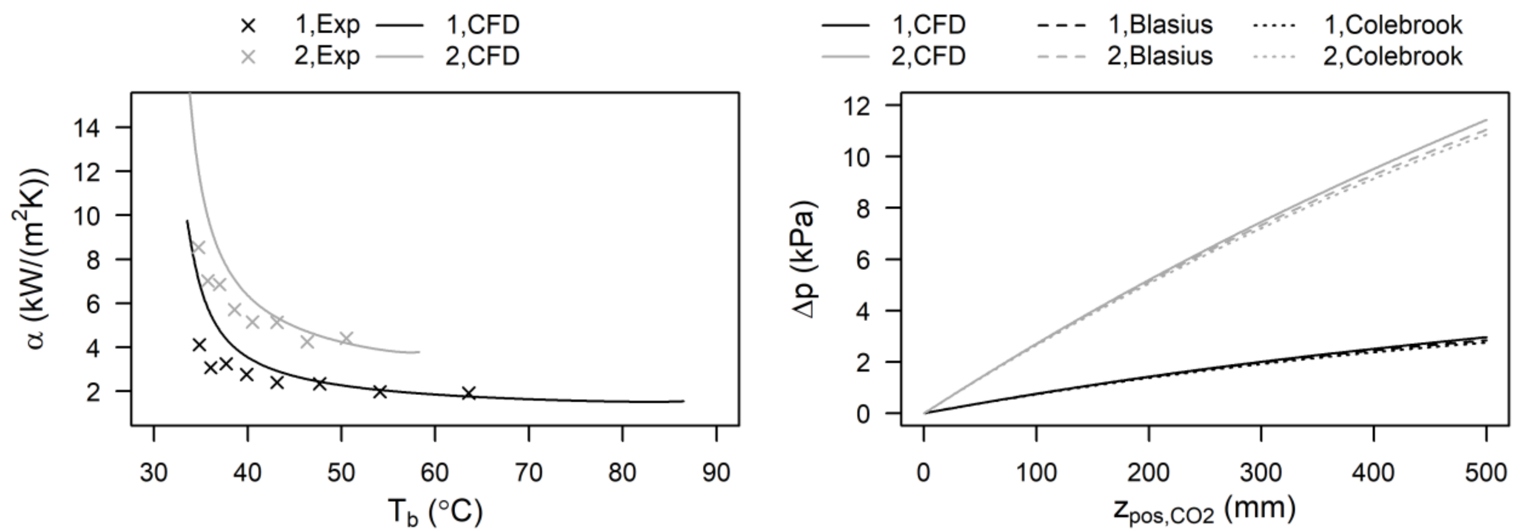




Numerical Model

Experimental Validation

#	d_{CO2} mm	$l_{entr,CO2}$ mm	$l_{ch,CO2}$ mm	$l_{exit,CO2}$ mm	G_{CO2} $kg/(m^2\,s)$	$\dot{q}_w$ kW/m^2	$T_{in,CO2}$ °C	ρ_{ref} kg/m^3
1	1.9	200	500	100	326	−23.2	90	217.0
2	1.9	200	500	100	762	−33.9	60	233.4



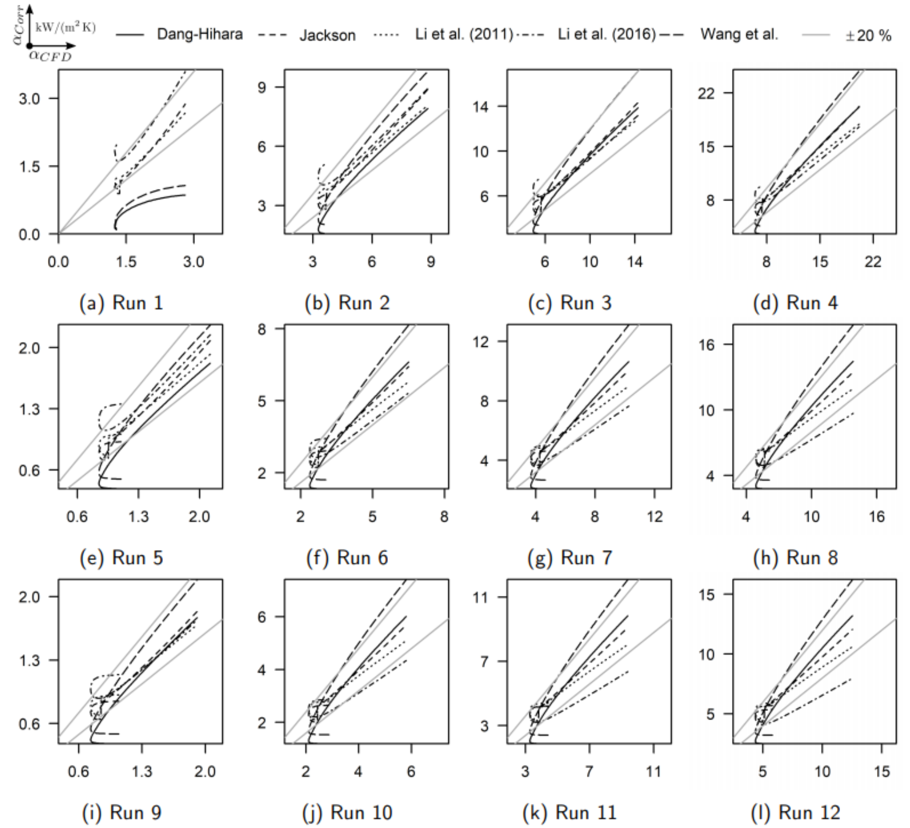
(a) HTC (b) Pressure drop

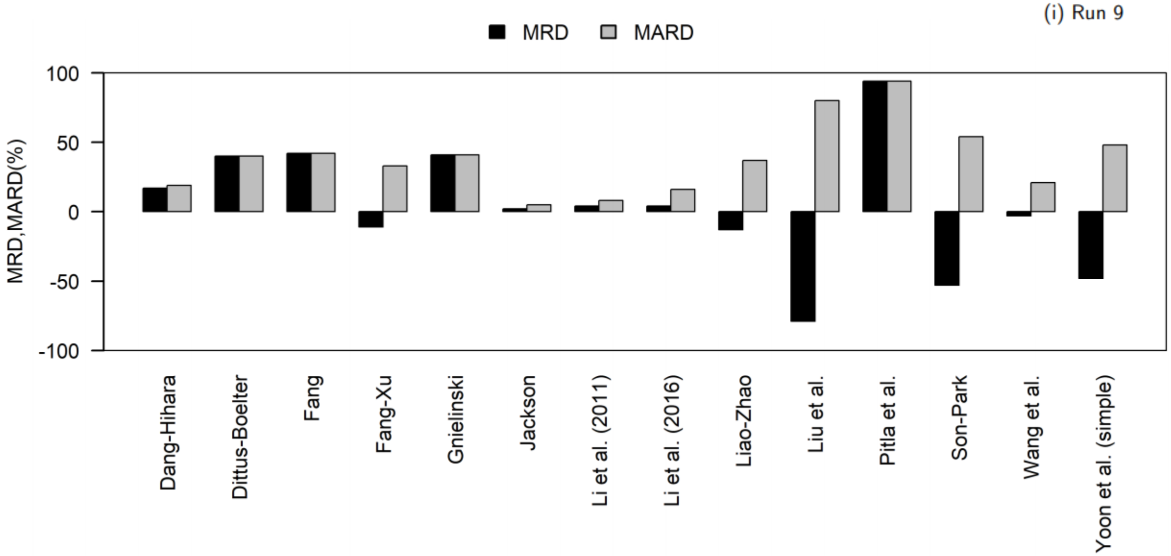
Kruizenga, A. et al. *Heat Transfer of Supercritical Carbon Dioxide in Printed Circuit Heat Exchanger Geometries.* J. Therm. Sci. Eng. Appl. 3, (2011).

Numerical Model

Correlation Comparison

#	d_{CO2} mm	$l_{ch,CO2}$ mm	G_{CO2} kg/(m^2 s)	$\dot{m}_{CO2}$ g/s	u m/s	Re $\times 10^3$
Run 1	0.50	60	100	0.010	0.4 to 1.6	1 to 2
Run 2	0.50	80	400	0.039	1.5 to 6.2	4 to 6
Run 3	0.50	90	700	0.069	2.6 to 10.9	7 to 11
Run 4	0.50	100	1000	0.098	3.7 to 15.5	10 to 16
Run 5	1.75	240	100	0.120	0.4 to 1.6	4 to 5
Run 6	1.75	320	400	0.481	1.5 to 6.2	15 to 22
Run 7	1.75	350	700	0.842	2.6 to 10.9	25 to 38
Run 8	1.75	370	1000	1.203	3.7 to 15.5	36 to 55
Run 9	3.00	440	100	0.353	0.4 to 1.6	6 to 9
Run 10	3.00	590	400	1.414	1.5 to 6.2	25 to 38
Run 11	3.00	660	700	2.474	2.6 to 10.9	44 to 66
Run 12	3.00	690	1000	3.534	3.7 to 15.5	62 to 94





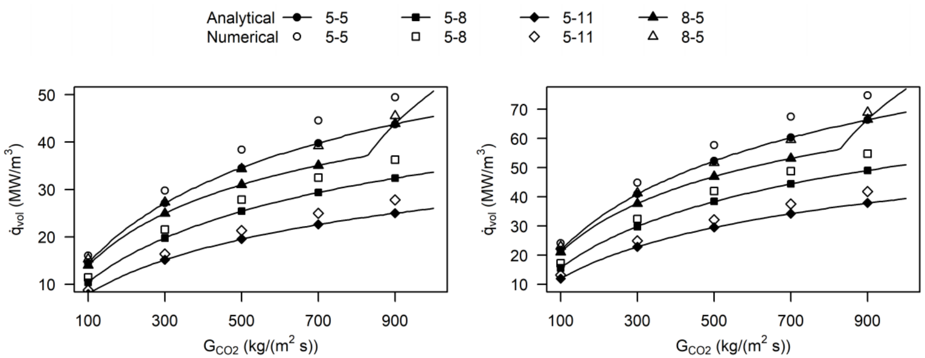
$$MRD = \frac{1}{N}\sum_{i=1}^{N}\frac{\alpha_{i,corr} - \alpha_{i,CFD}}{\alpha_{i,CFD}}$$

$$MARD = \frac{1}{N}\sum_{i=1}^{N}\left|\frac{\alpha_{i,corr} - \alpha_{i,CFD}}{\alpha_{i,CFD}}\right|$$

Numerical Model

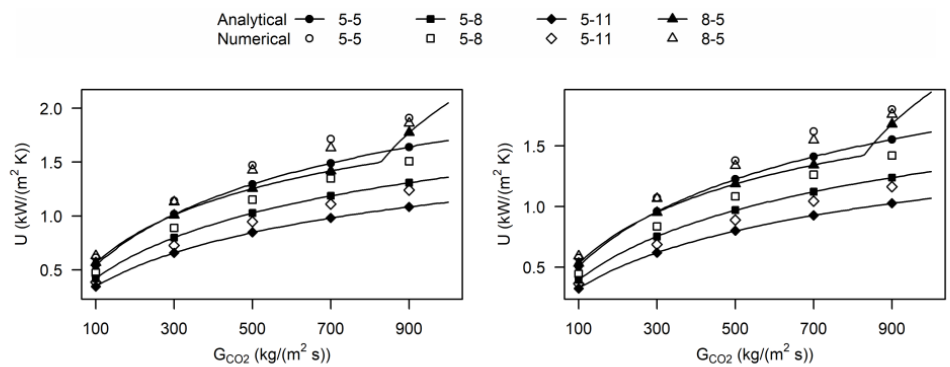
Results

Volumetric heat flux



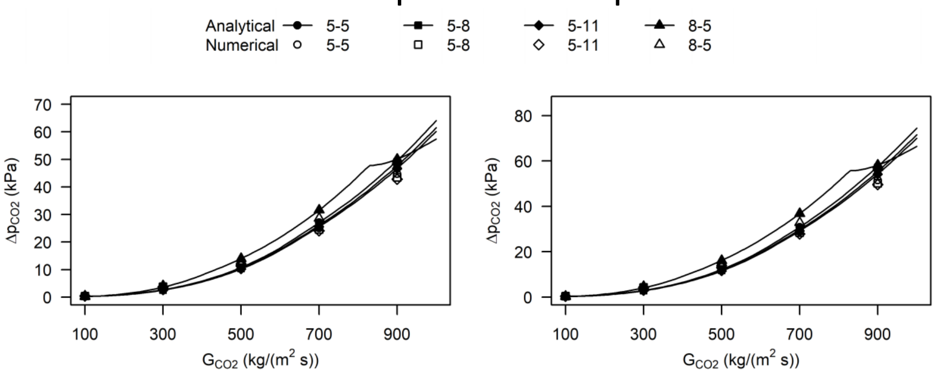
(a) Cement kiln WHR system (b) Gas turbine WHR system

Overall heat transfer coefficient



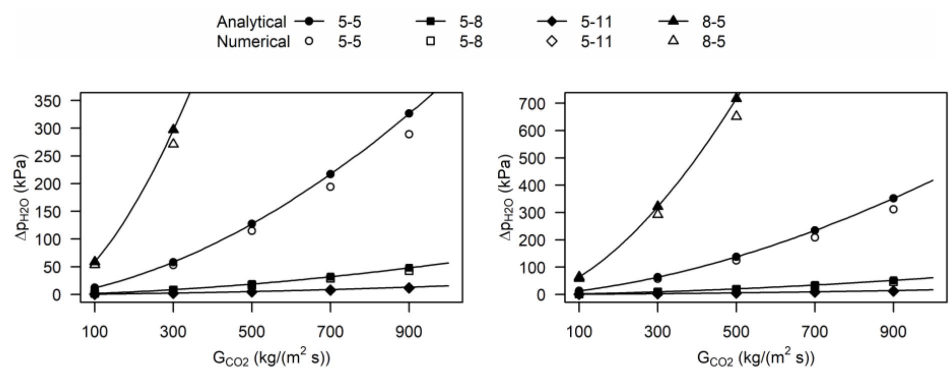
(a) Cement kiln WHR system (b) Gas turbine WHR system

sCO2 pressure drop



(a) Cement kiln WHR system (b) Gas turbine WHR system

Coolant pressure drop

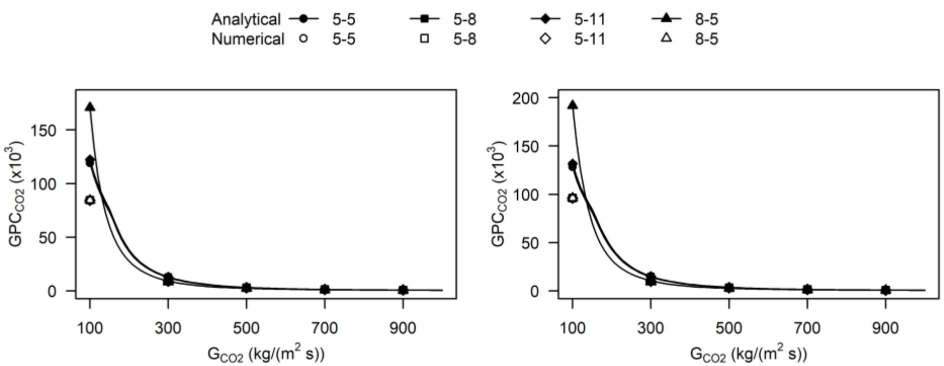


(a) Cement kiln WHR system (b) Gas turbine WHR system

Numerical Model

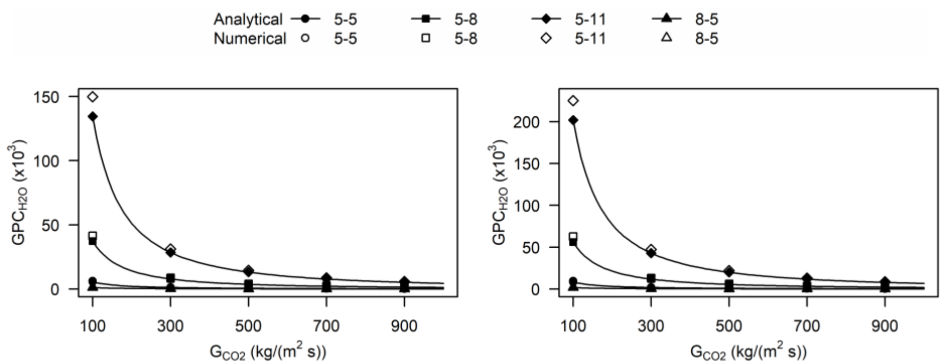
Results

Global performance sCO2 channel



(a) Cement kiln WHR system (b) Gas turbine WHR system

Global performance coolant channel



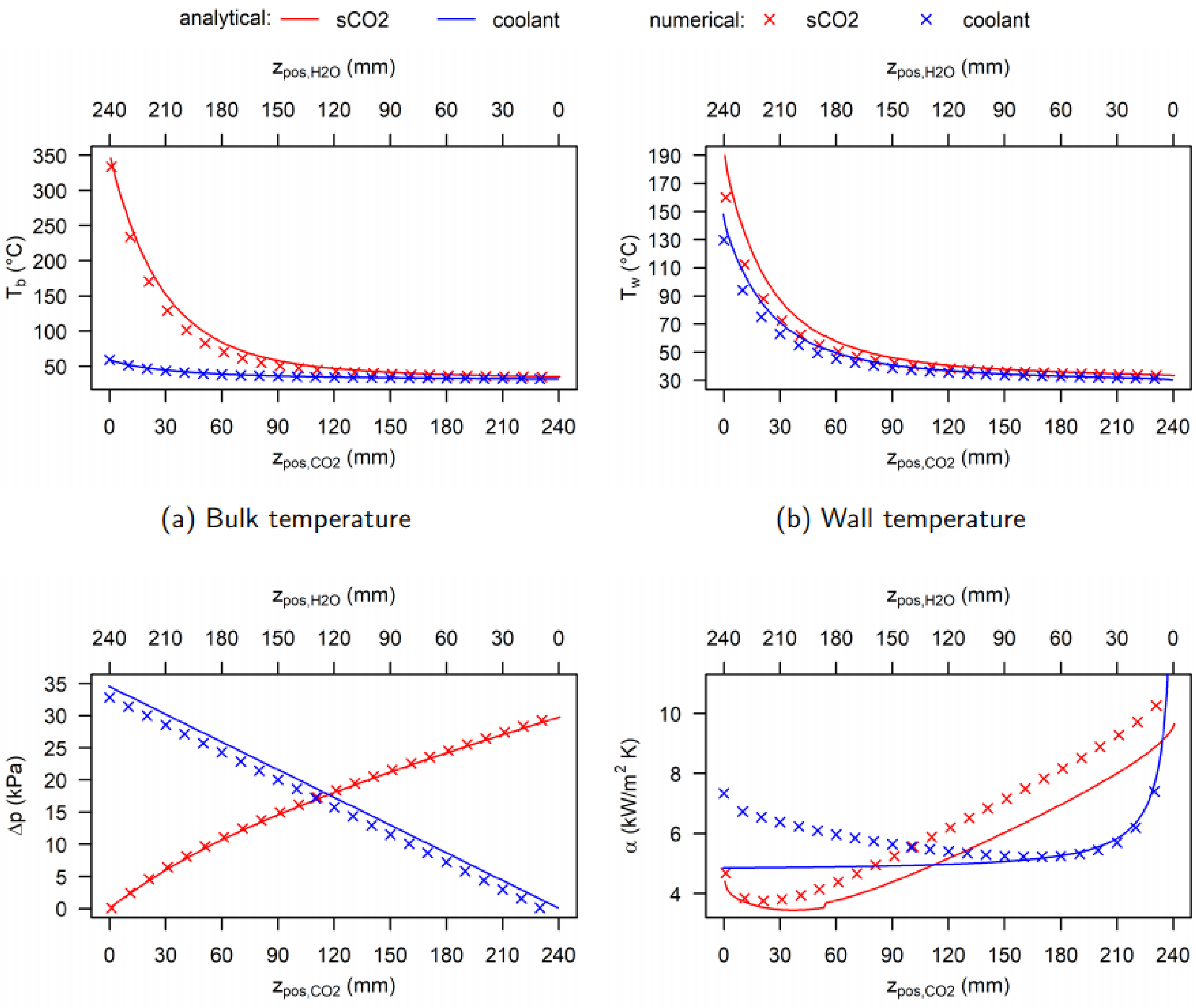
(a) Cement kiln WHR system (b) Gas turbine WHR system

$$GPC = \frac{\dot{Q}_{tot}}{\Delta p_{tot}\,\overline{\dot{V}}}$$

Numerical Model

Comparison Analytical and Numerical Model

Configuration:
GT-5-8-700



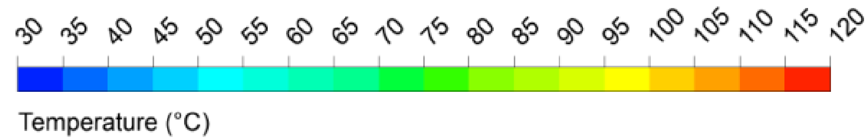
(c) Pressure Drop (d) HTC

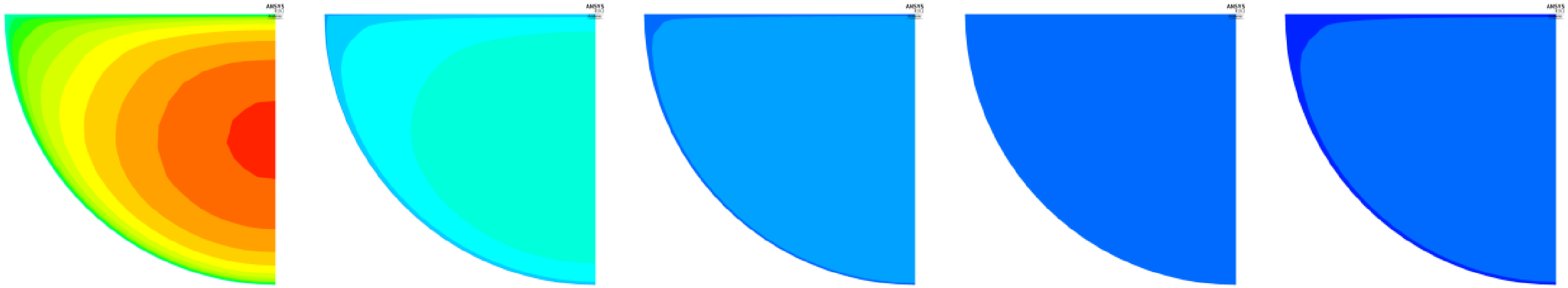
Numerical Model

Visualization

Configuration:
GT-5-8-700

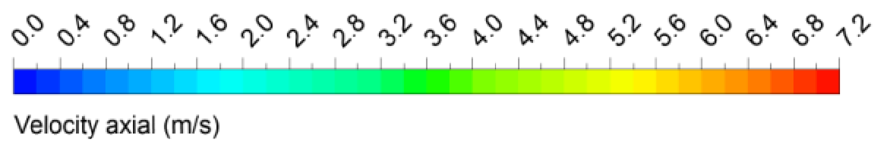
sCO2 bulk temperature

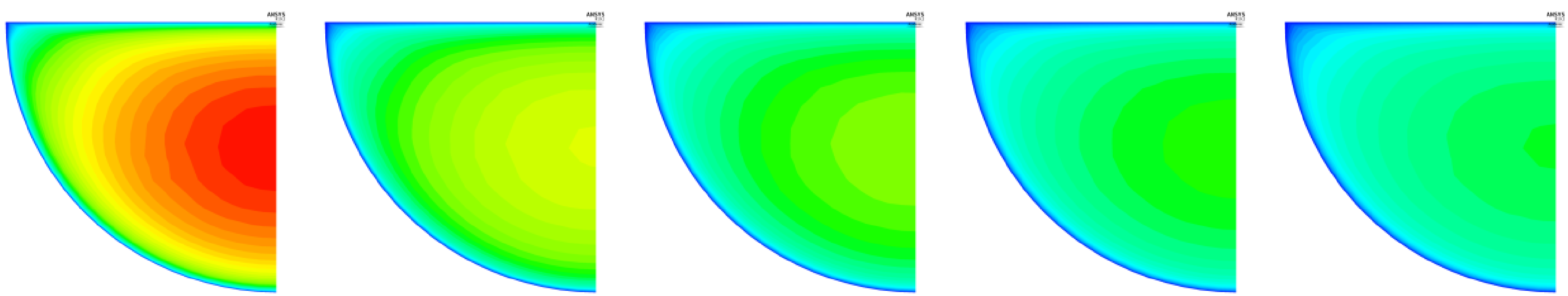




(a) 40 mm (b) 80 mm (c) 120 mm (d) 160 mm (e) 200 mm

sCO2 velocity axial





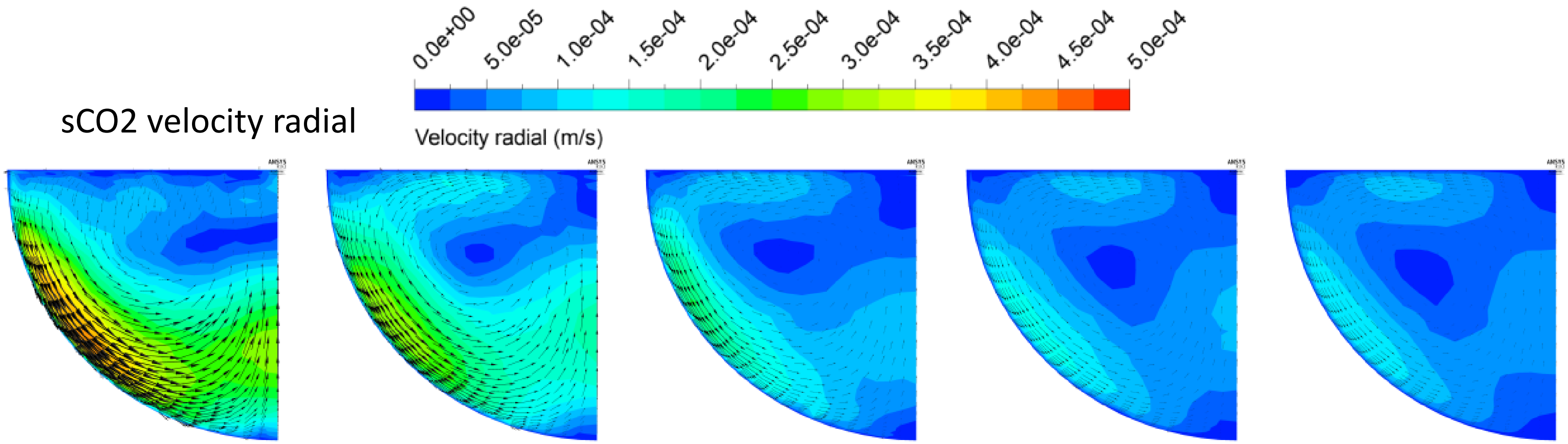
(a) 40 mm (b) 80 mm (c) 120 mm (d) 160 mm (e) 200 mm

Numerical Model

Visualization

Configuration:
GT-5-8-700

sCO2 velocity radial



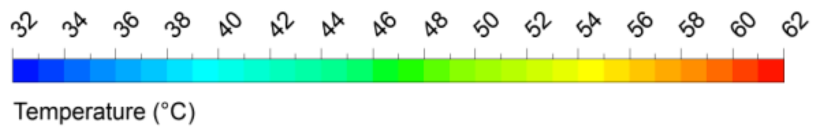
(a) 40 mm (b) 80 mm (c) 120 mm (d) 160 mm (e) 200 mm

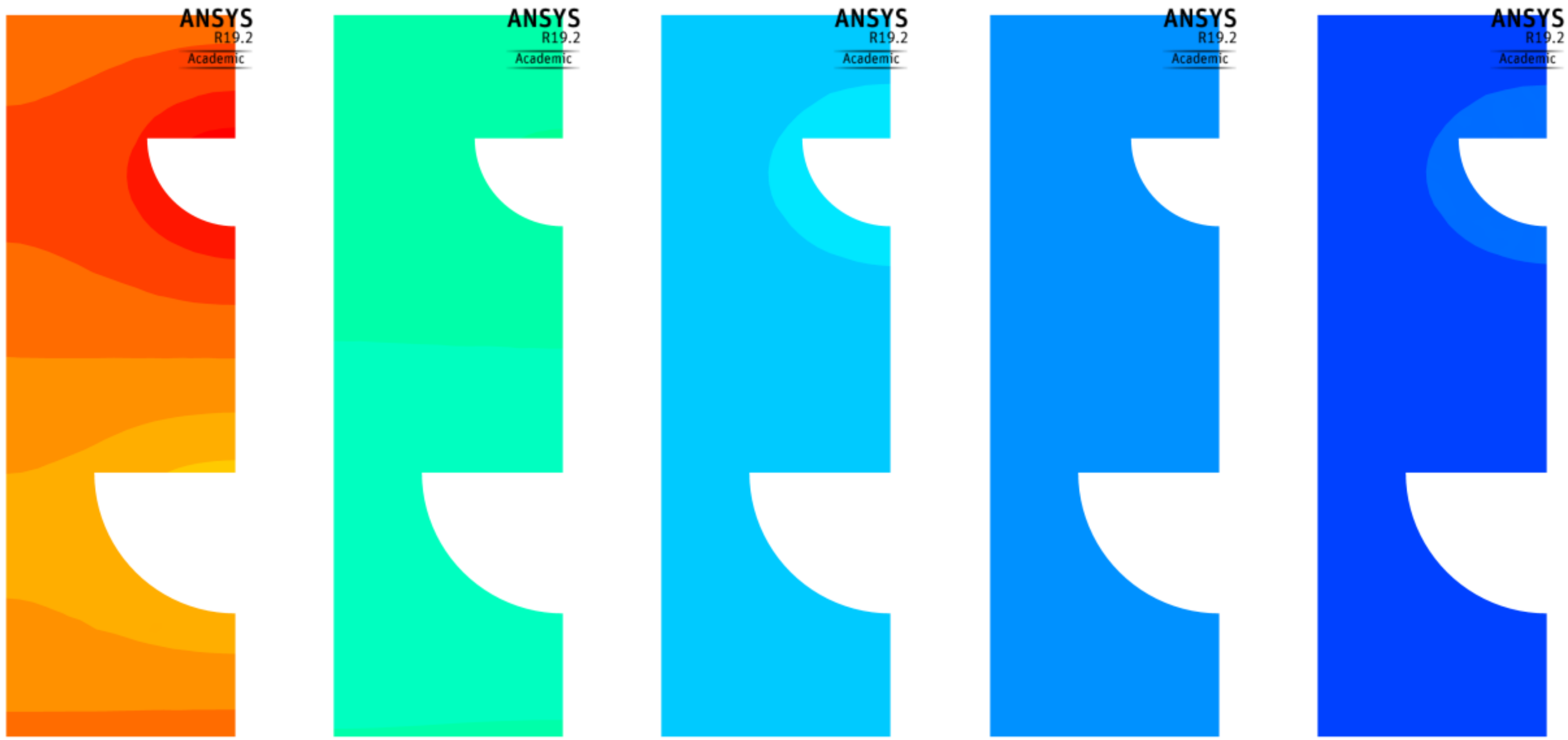
Numerical Model

Visualization

Configuration:
GT-5-8-700

PCHE body temperature





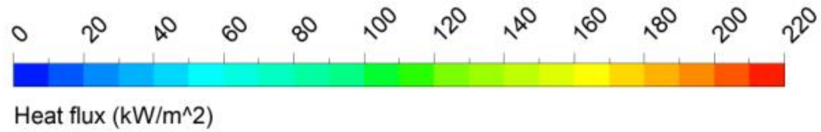
(a) 40 mm (b) 80 mm (c) 120 mm (d) 160 mm (e) 200 mm

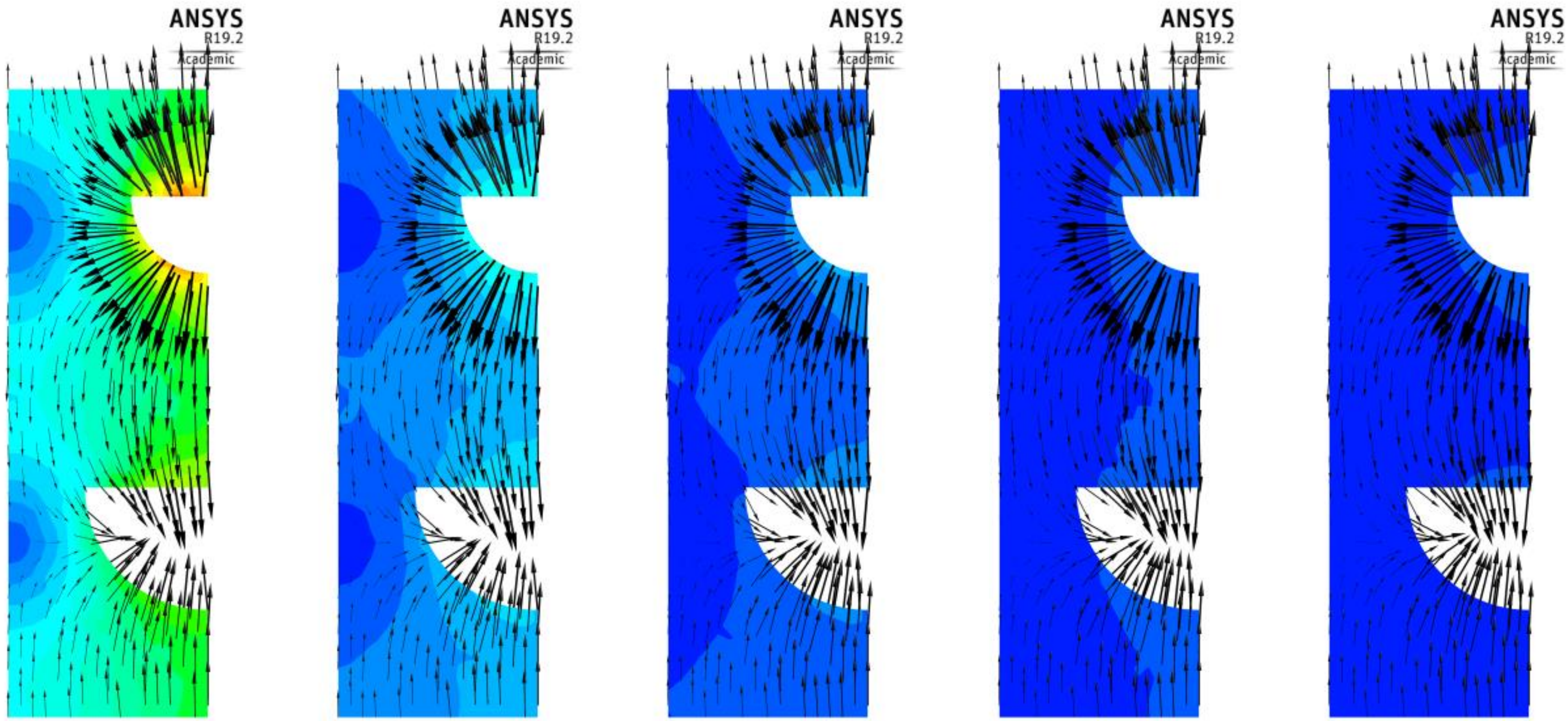
Numerical Model

Visualization

Configuration:
GT-5-8-700

PCHE body heat flux





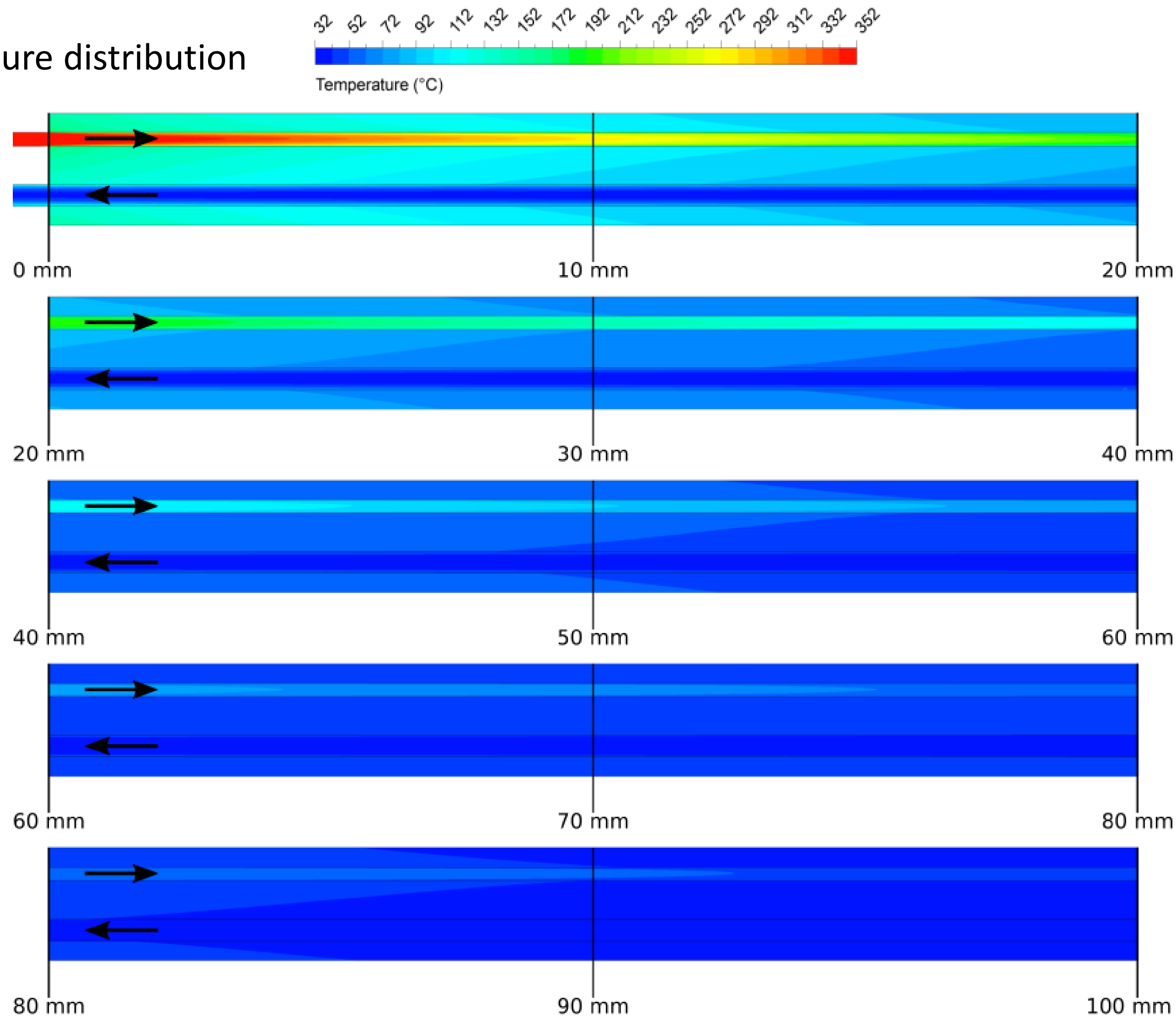
(a) 40 mm (b) 80 mm (c) 120 mm (d) 160 mm (e) 200 mm

Numerical Model

Visualization

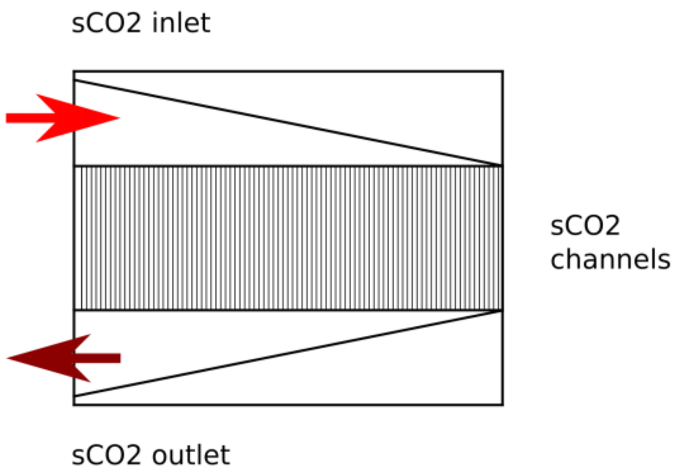
Configuration:
GT-5-8-700

PCHE temperature distribution

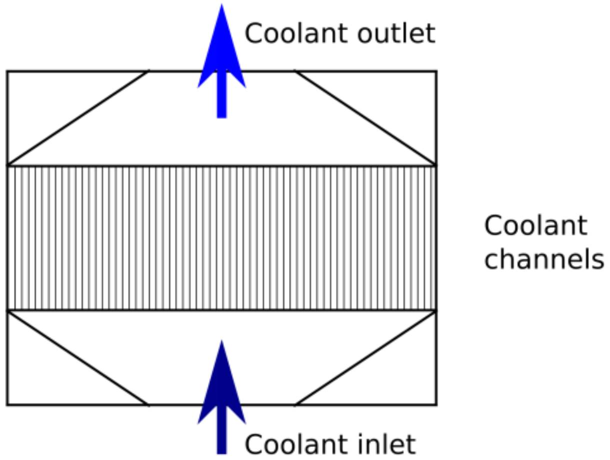


Design Proposal

Flow Distribution



(a) sCO2 plate



(b) Coolant plate