Supercritical CO_2 Power Cycles Symposium

Tutorial:

Heat Exchangers for Supercritical CO2 Power Cycle Applications

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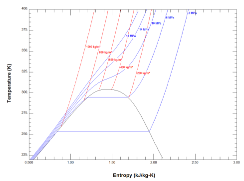
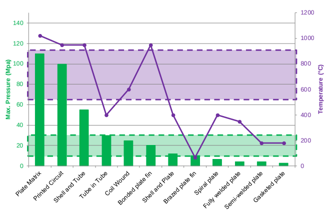
Ed Green — Thar Energy

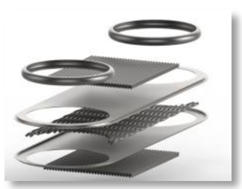
The following slides present an overview of heat exchangers in supercritical CO_2 applications



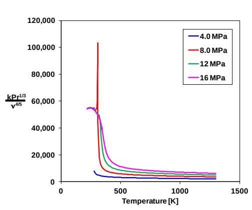
Introduction to sCO_2



Heat exchangers in sCO2 cycle applications



Heat exchanger mechanical design for sCO_2



Hydraulic design and heat transfer in supercritical fluids

Brief Introduction to S-CO2

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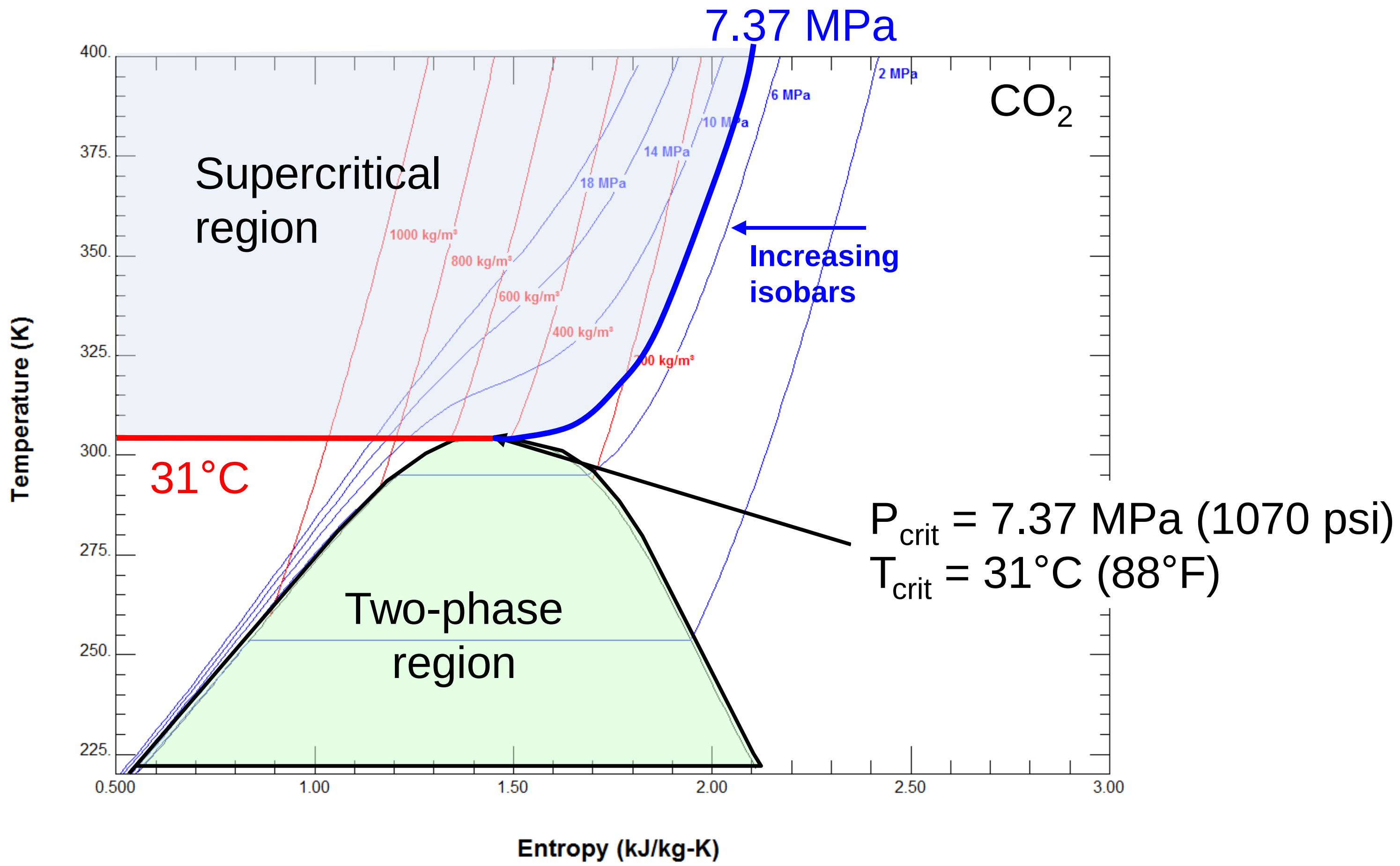
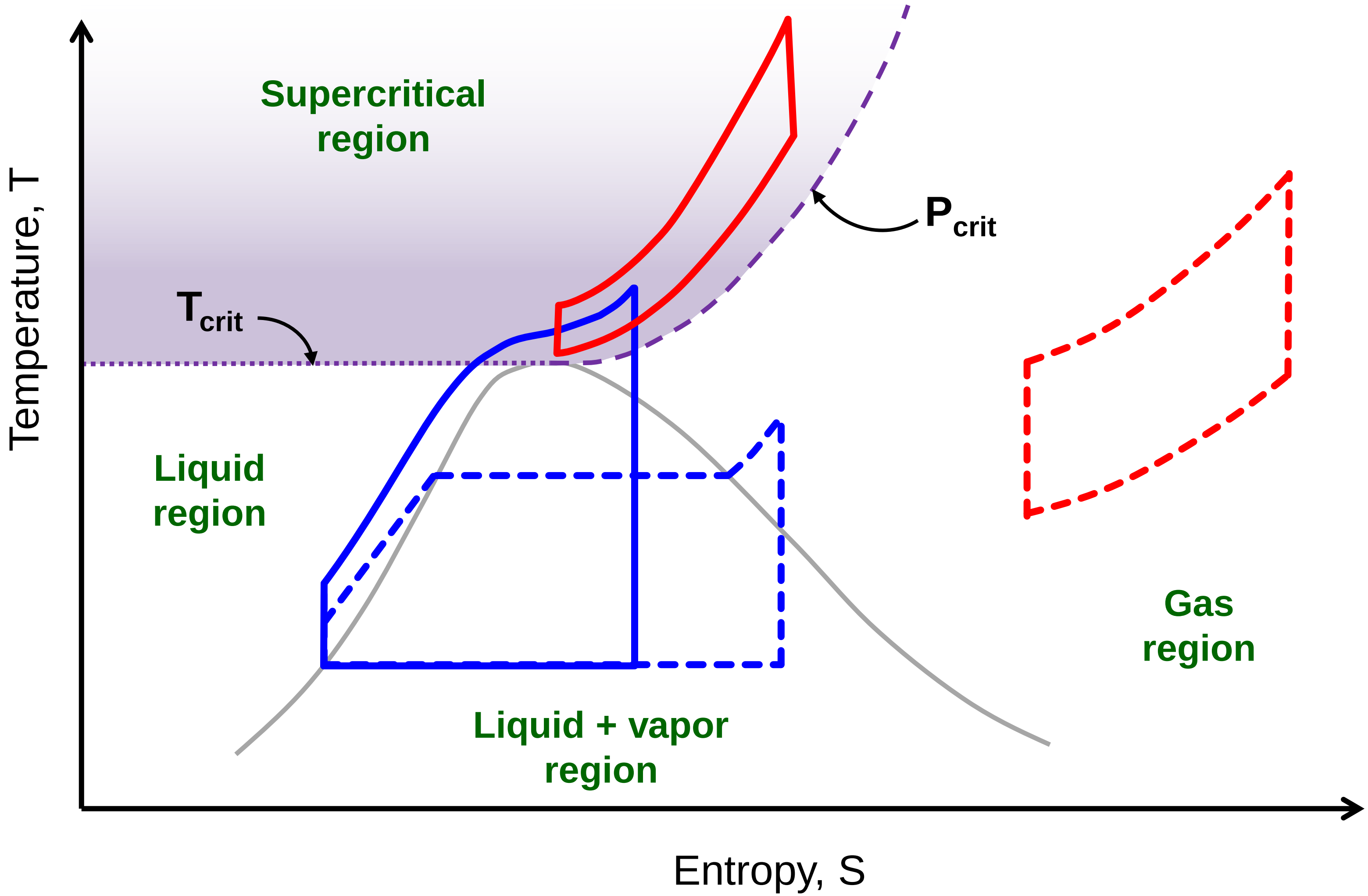
A fluid is supercritical if the pressure and temperature are greater than the critical values



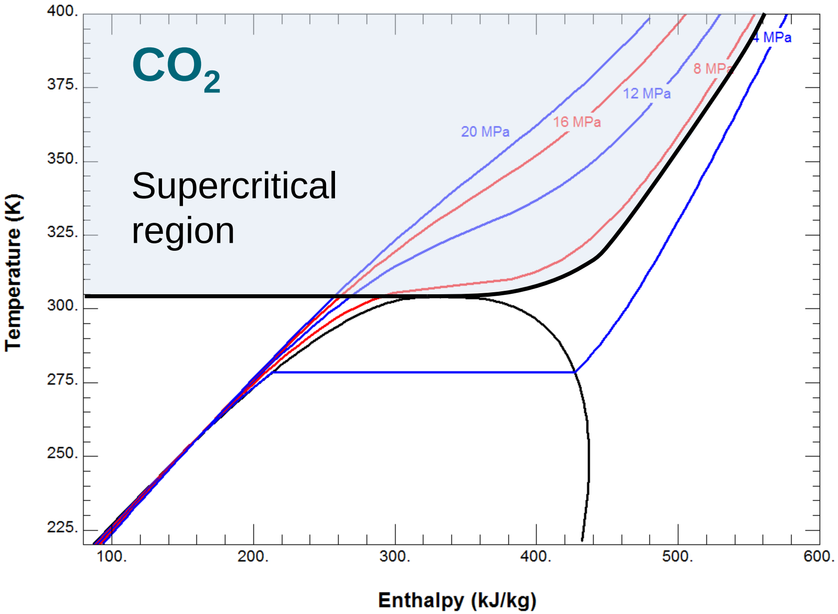
Source: Musgrove et al. GT2012-70181

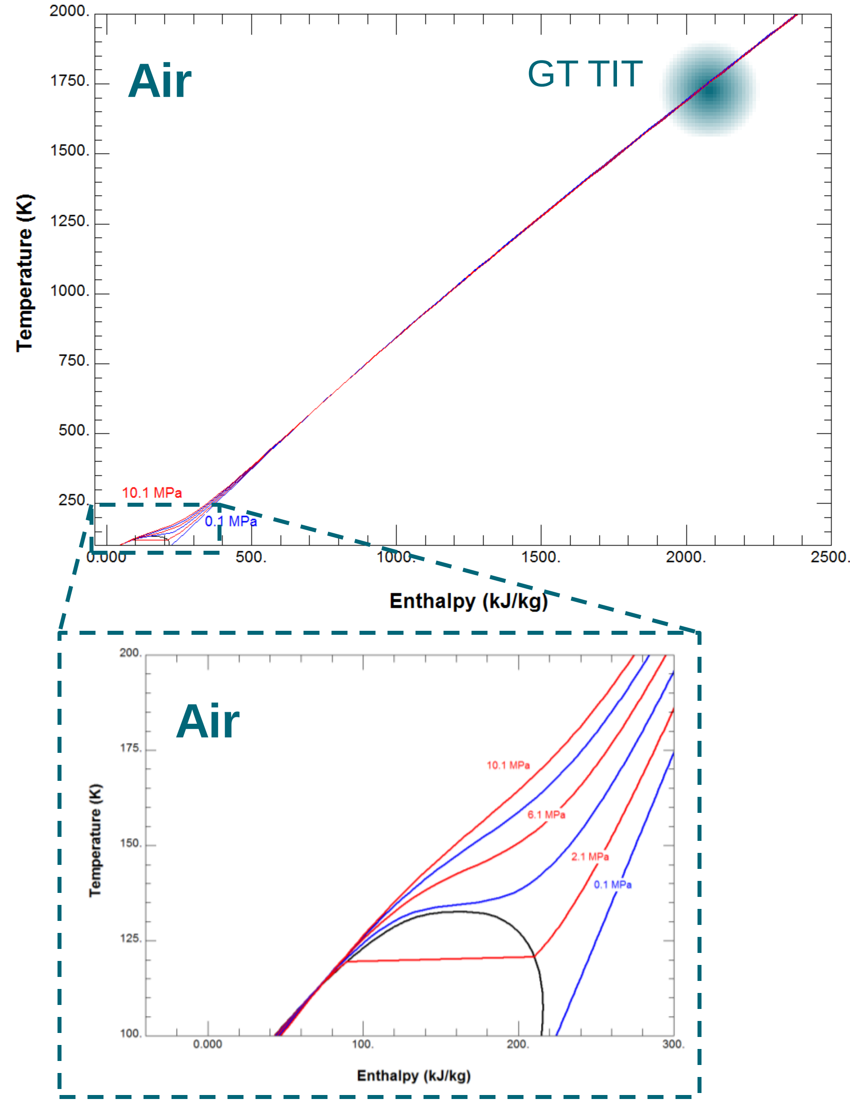
A power cycle is supercritical if part of the cycle takes place in the supercritical phase region



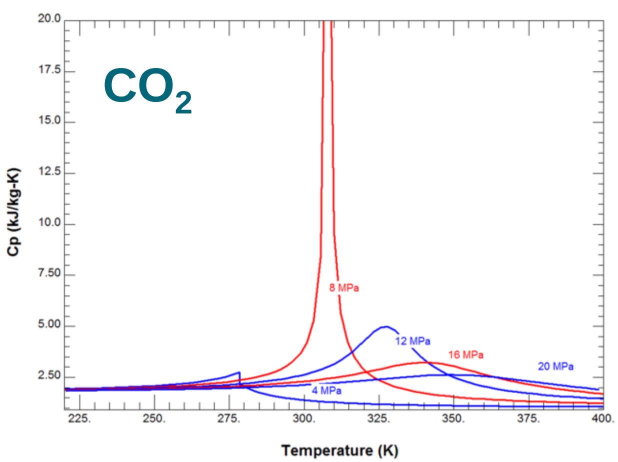
Source: Musgrove et al. GT2012-70181

Fluids operating near their critical point have dramatic changes in enthalpy



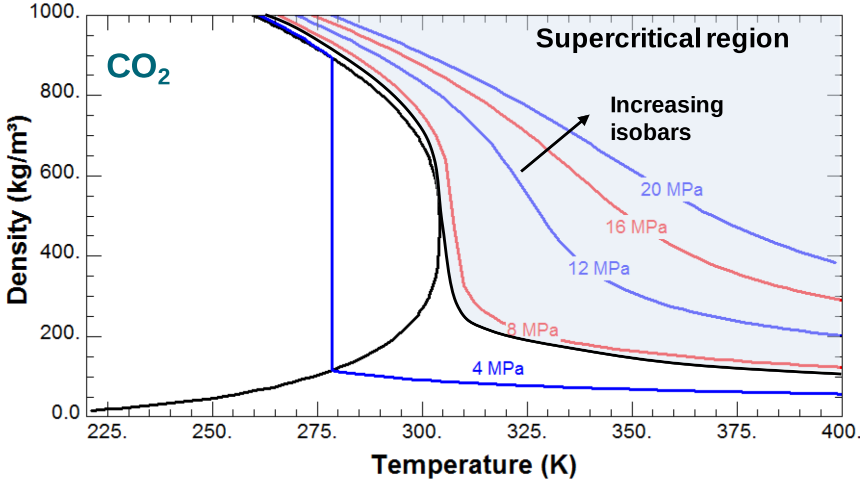


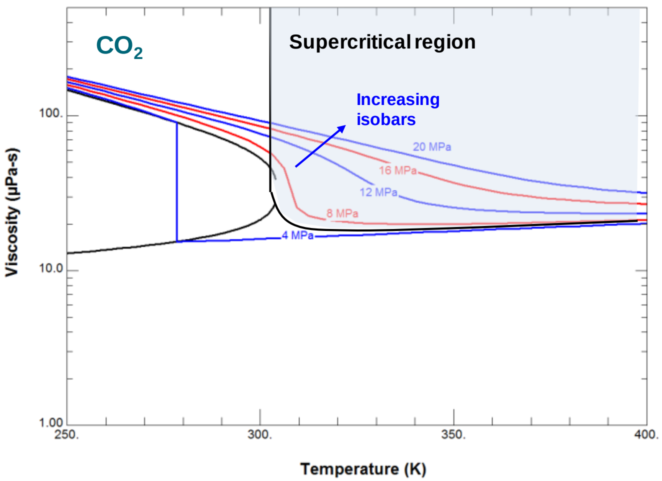
$$C_p = \left.\frac{\partial h}{\partial T}\right)_p$$

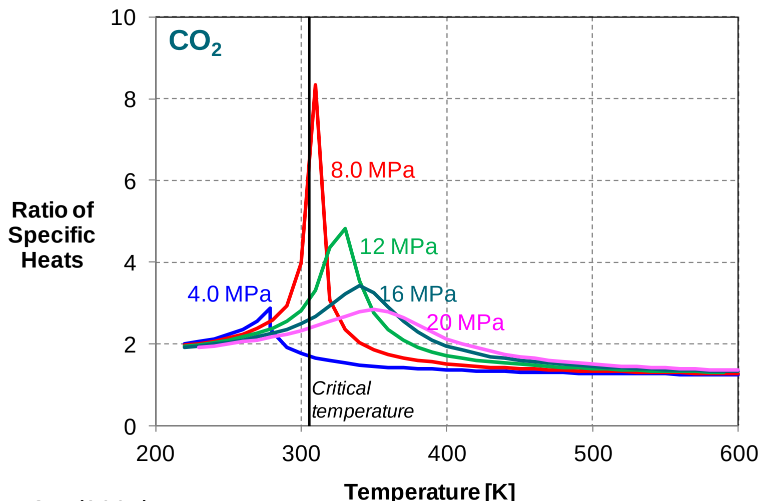


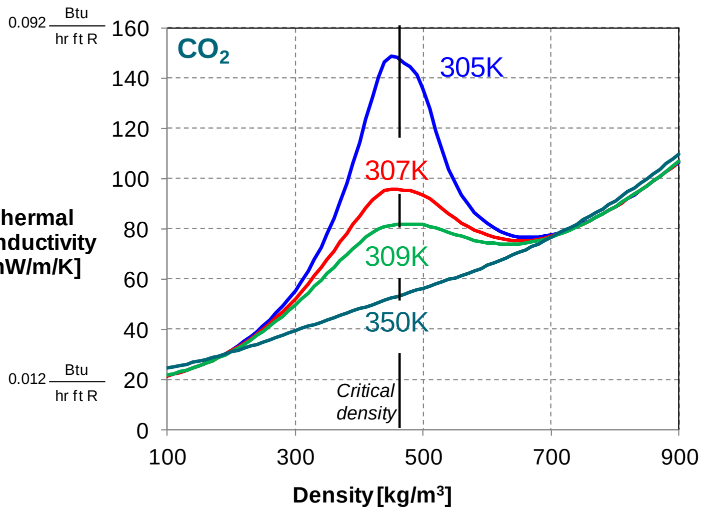
Source: Musgrove et al. GT2012-70181

Operating near the critical point allows dramatic changes in fluid properties to be exploited



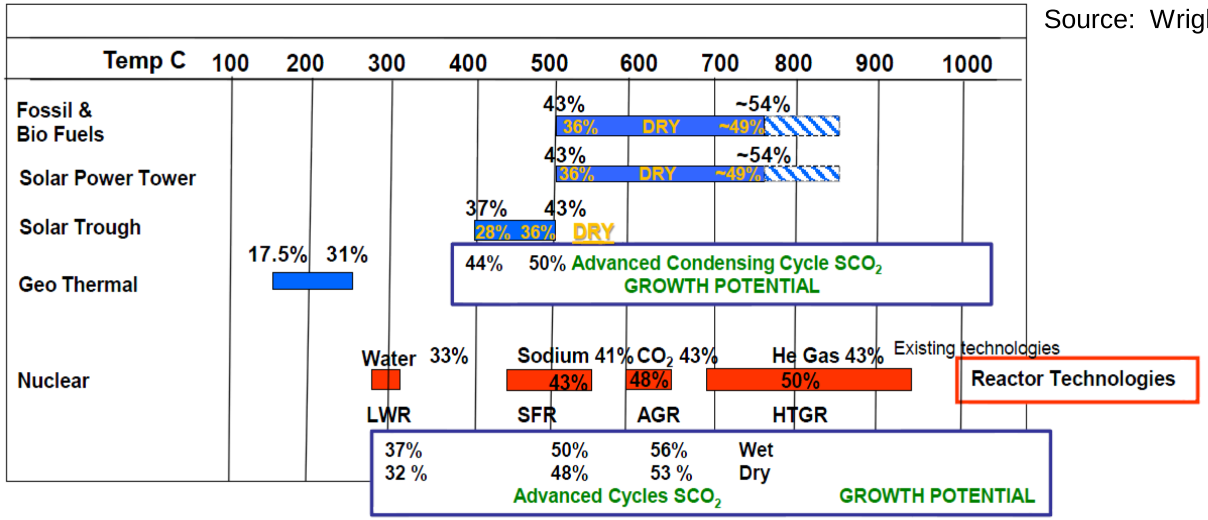






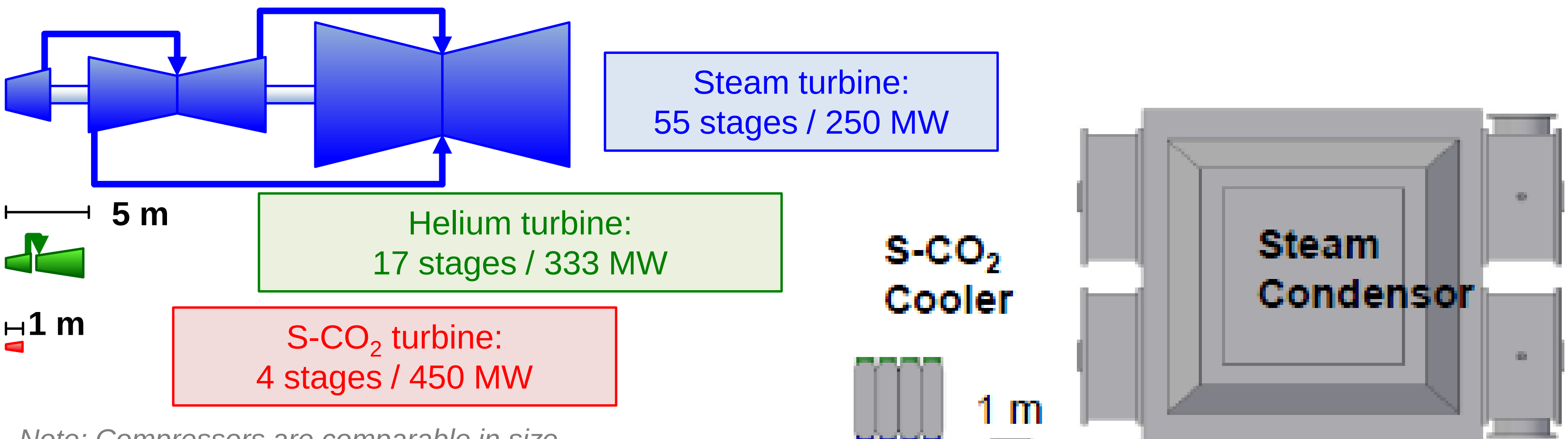
REFPROP (2007)

S-CO2 power cycles allow a range of thermal sources and small machinery



Source: Wright (2011)

Assumptions (Turbomachinery Eff (85%/87%/90% : MC/RC/T), 5 K Approach T, 5% dp/p losses, Hotel Losses Not In Included, Dry Cooling at 120 F)

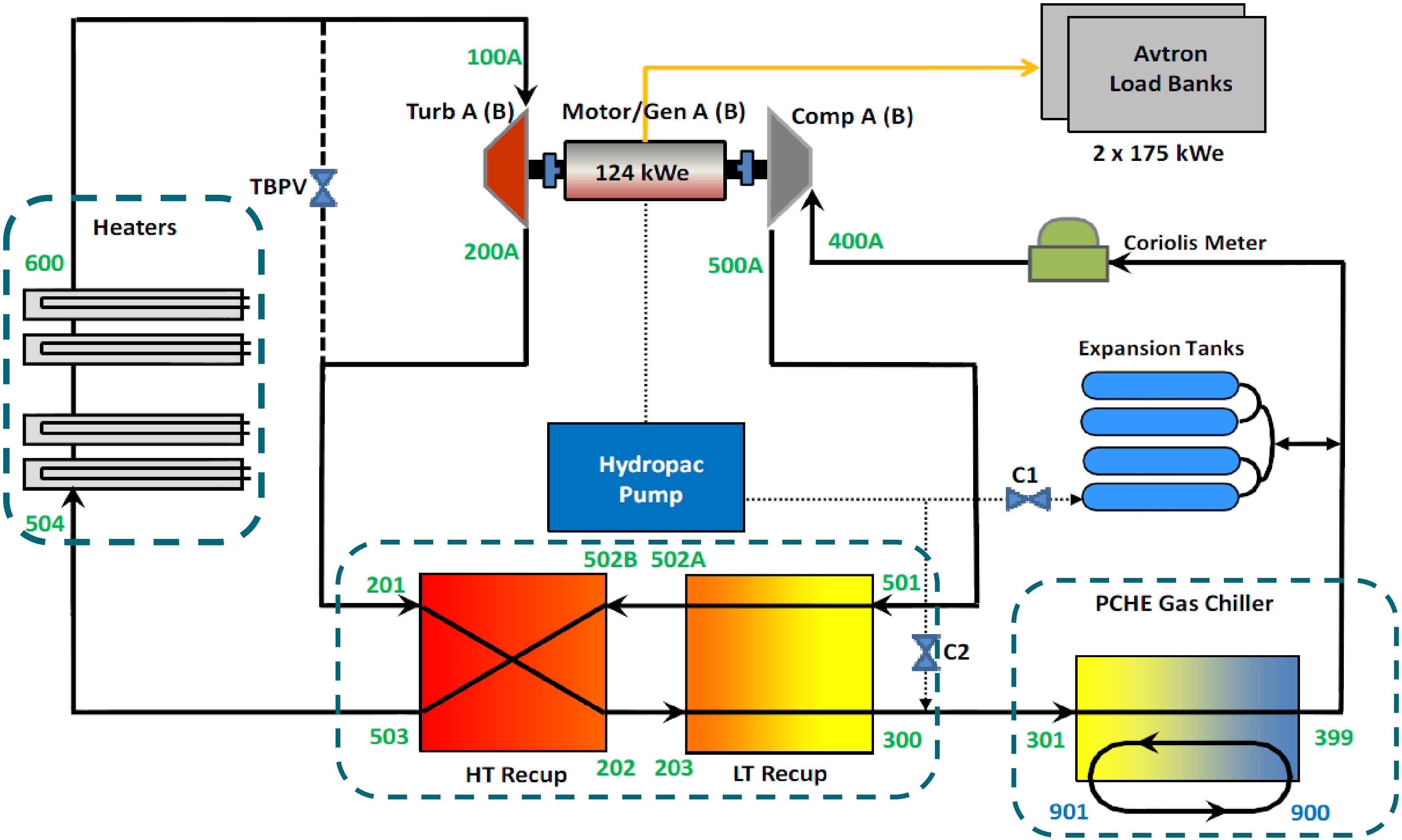


Note: Compressors are comparable in size

Adapted from Dostal (2004)

Source: Wright (2011)

Heat exchangers are typically used for heat addition and as recuperators



[Conboy et al. 2012]

Heat Exchangers in SCO2 power conversion cycles

Jorge Montero Carrero

Heatric

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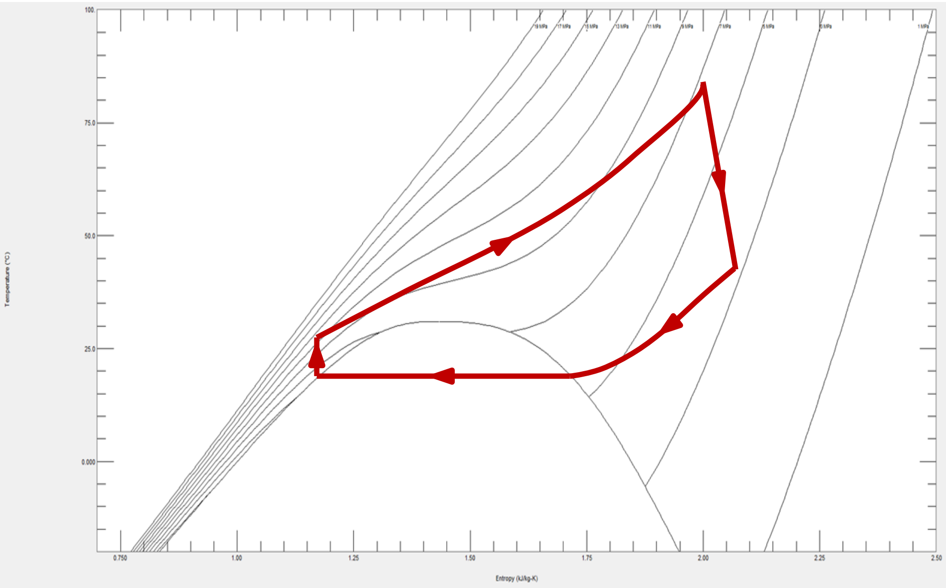
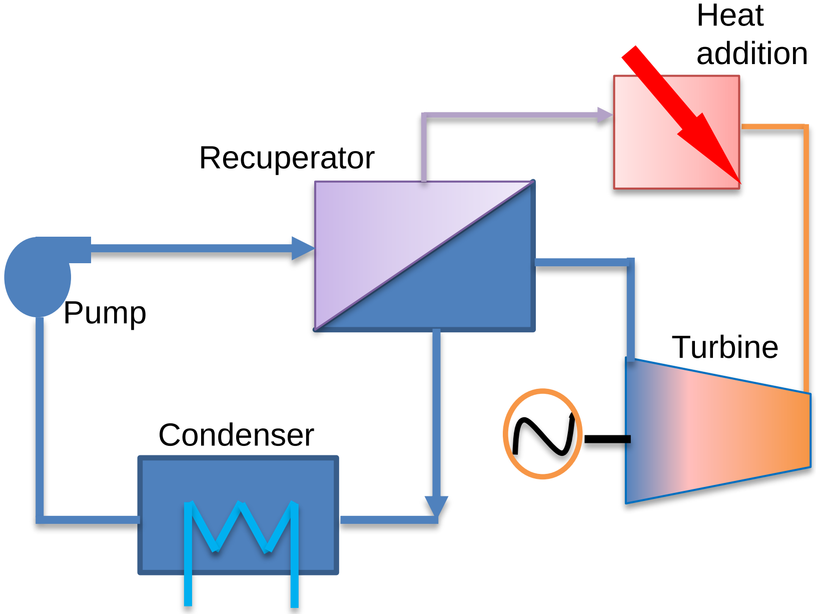
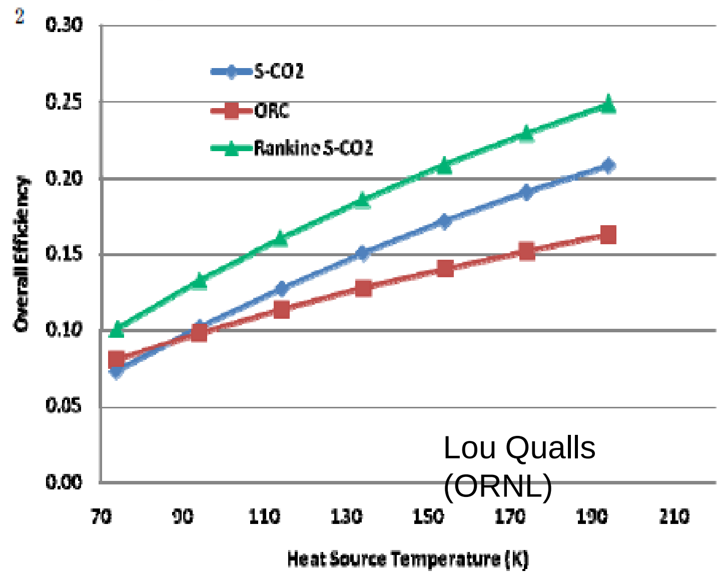
SCO_2 Rankine Cycles

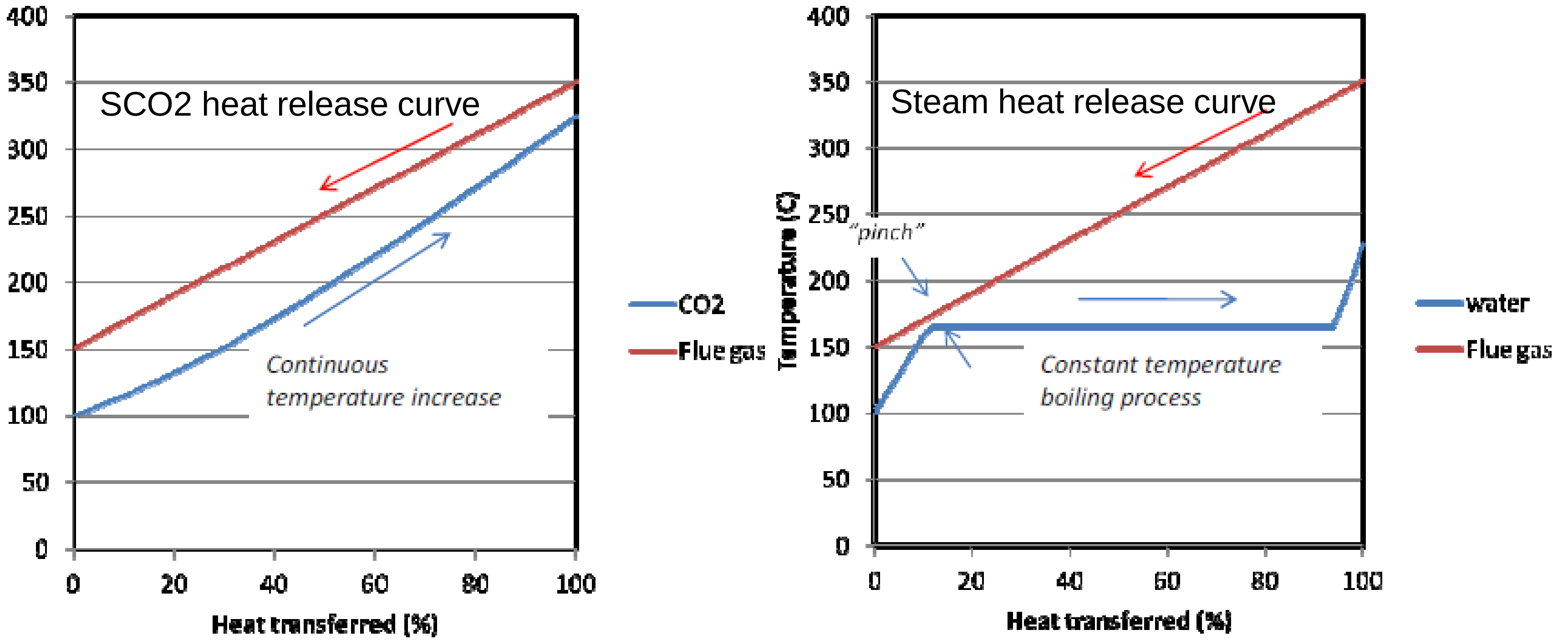




- 20 – 25 % first law efficiency
- Up to 10 % more efficient than ORC
- Heat Sources include Geothermal, exhaust gasses, industrial waste, solar, etc

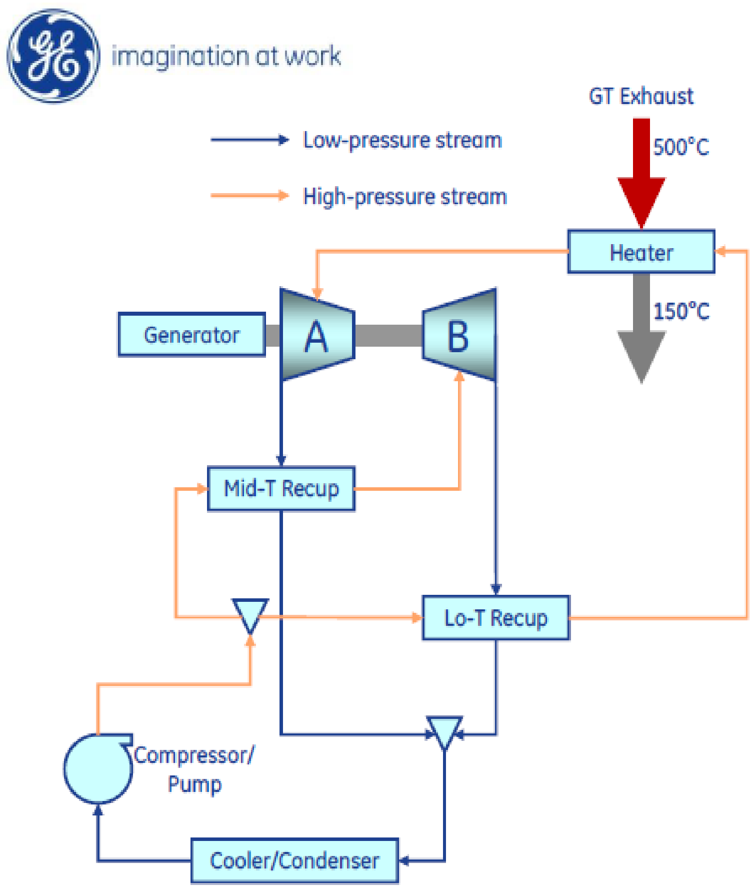


Exchanger application in SCO_2 Cycles

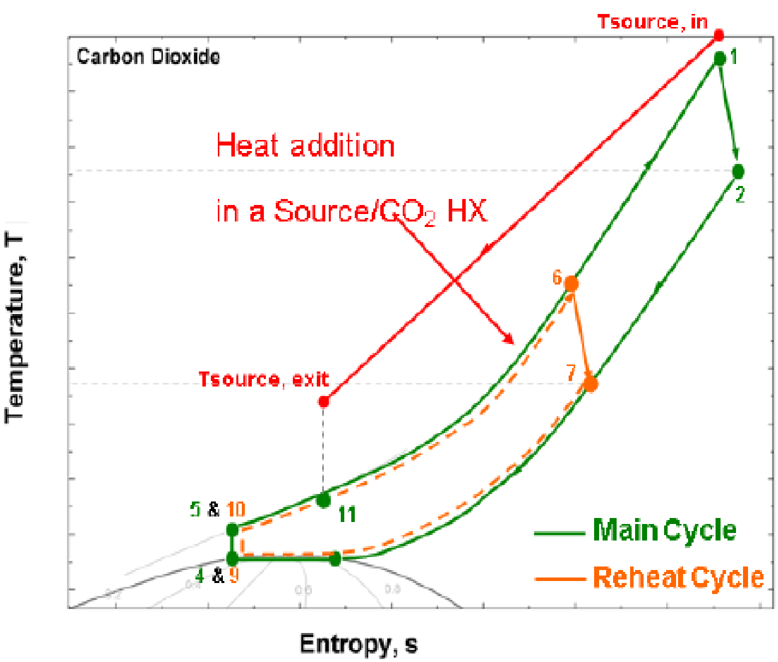


- Better heat recovery possible in SCO2 cycles with single phase exchangers
- Two phase boiling at constant temperature (steam cycles) limits close temperature approach (pinching)

Applications using SCO_2 Rankine Cycles

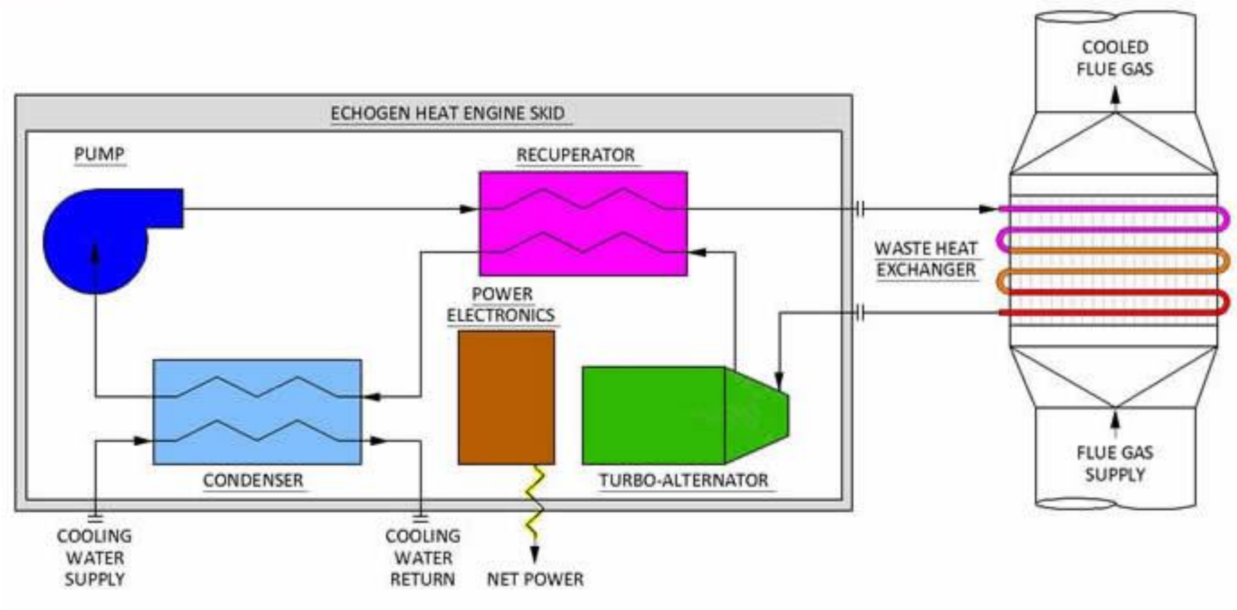


Courtesy of GE GRC (patent pending)



- 30% first-law efficiency
- Better utilization of exhaust energy
- 10% more power output compared to ORC
- Compact turbo-machinery with low footprint

Echogen EPS systems



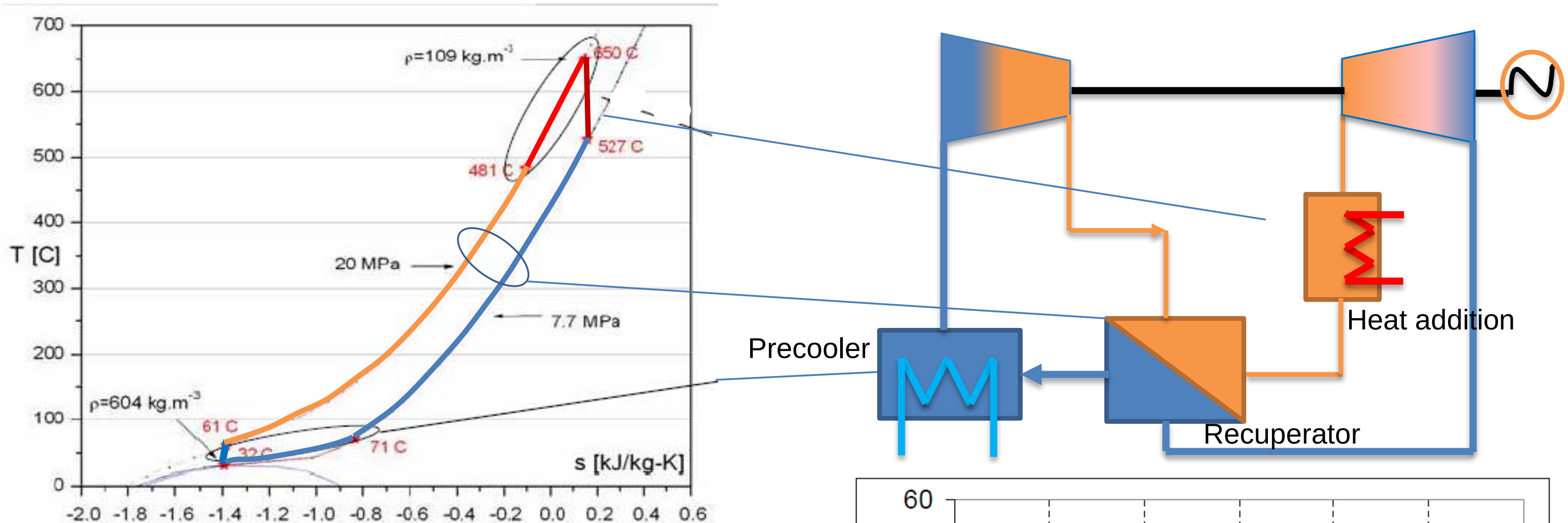
Echogen Commercialisation

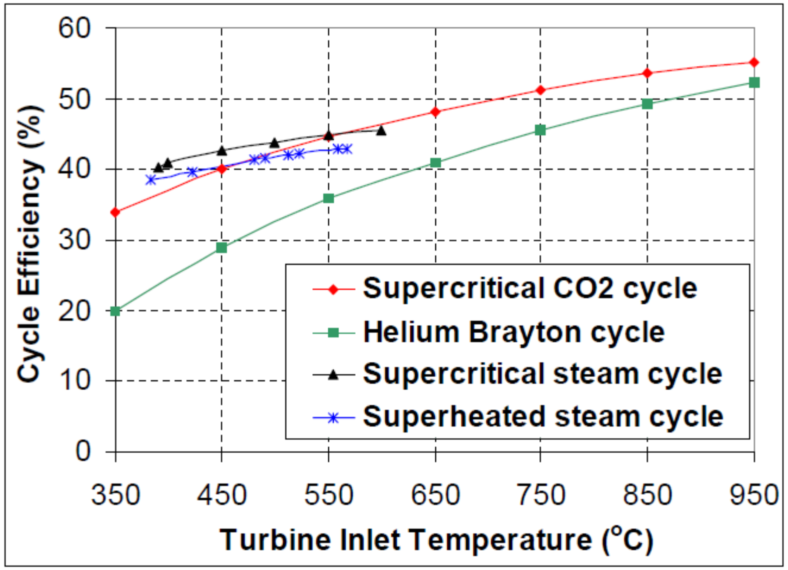
- Built and tested demonstration unit
- Since designed and built commercial scale system, EPS100 (6-8 MW)
  - Tested at Dresser Rand's facility at Olean in New York
- Similar system, EPS 30 (1.5 MW), currently in design for commercial introduction in 2016

Echogen used compact exchangers
>300m² heat transfer area
~13000kg
Core ~ 1.5 x 1.5 x 0.5 m

Comparable S&T:
>850m²
~50000kg
Shell ~ 1.2m diameter x 12m length

Exchangers in SCO_2 Brayton Cycles





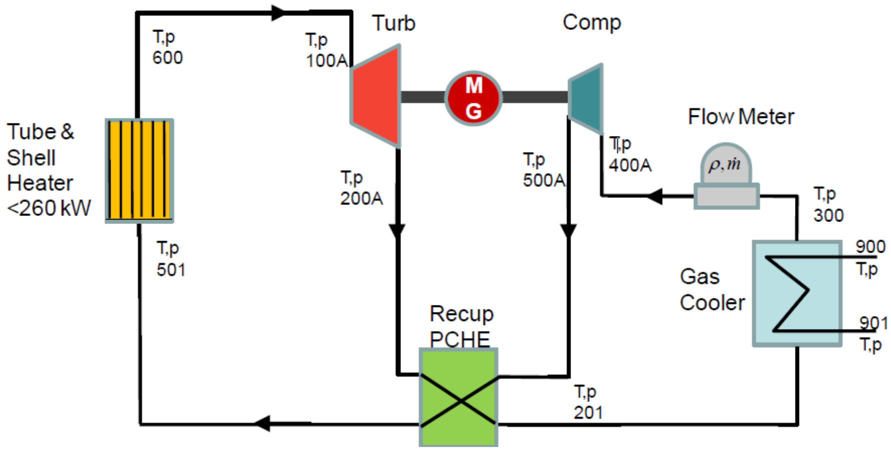
- Better fuel-power conversion efficiency
- Require high turbine inlet temperatures for efficient operation
- Simple cycles are highly recuperative
- Compressive work takes significant portion of developed power

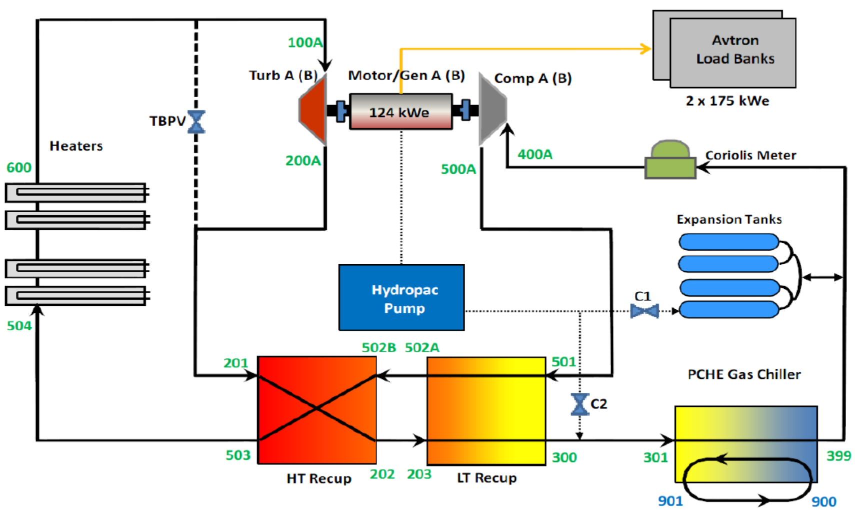
Exchangers that can be used in Brayton cycle include

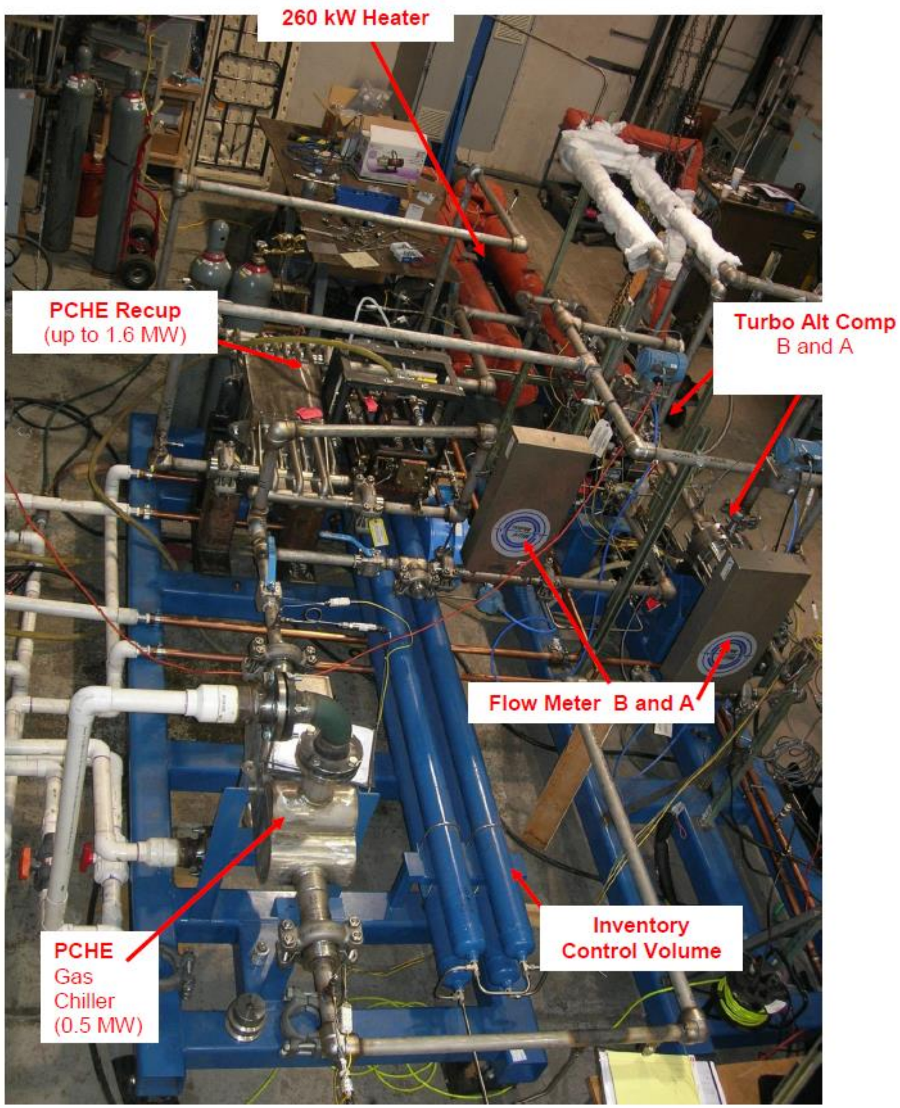
- Spiral wound exchanger
- Shell and tube
- Diffusion Bonded exchangers (plate fin and etched channels)
- Hybrid exchangers
- Finned tube and shell
- Plate and shell
- Porous media (metallic foam) exchangers

Sandia / Barber Nichols Inc.

Sandia has built and tested simple and recompression SCO2 test loops







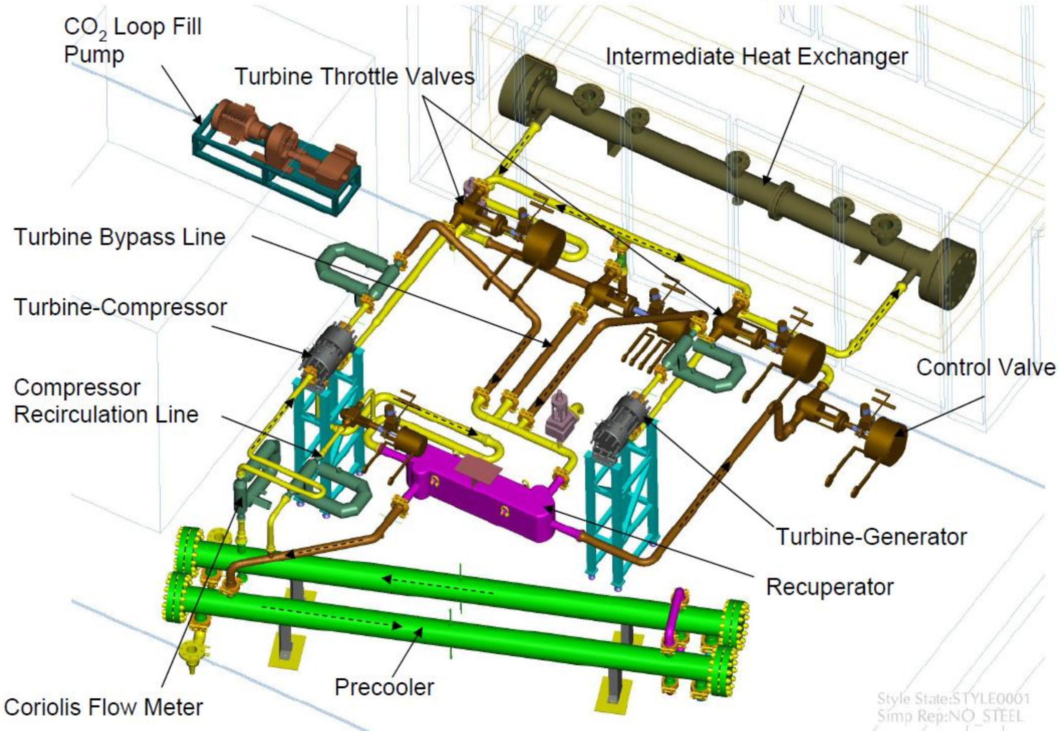
Wright; Conboy

Sandia Heat Exchangers used

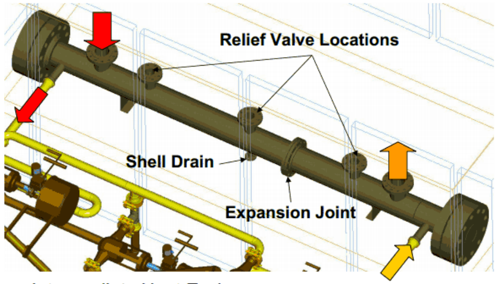
- HT Recuperator
  - 2.27 MW
  - 482°C (900°F)
  - 17.24 MPa (2500 psig)
- LT Recuperator
  - 1.6 MW
  - 454°C (849°F)
  - 17.24 MPa (2500 psig)
- Gas Chiller
  - 0.53 MW
  - 149°C (300°F)
  - 19.31 MPa (2800 psig)
- 6 ‘Shell and Tube’ heaters
  - U tubes contained resistance wire heaters

Bechtel – Integrated Test System

IST Physical Layout



IST Components: Intermediate Heat Exchanger



- Intermediate Heat Exchanger
  - Shell: MultiTherm PG-1
  - Tubes: CO_2
  - 1 shell, 230 tubes, 3/8" OD
  - 10" NPS OD x ~17' long
  - Expansion joint to accommodate differential thermal growth

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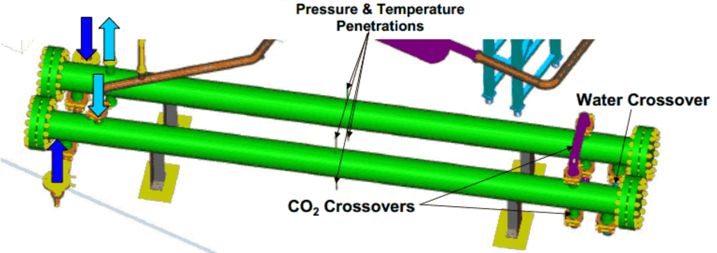
IST Components: Precooler



- Precooler
  - Shell: CO_2
  - Tubes: H_2O
  - 2 shells, 77 tubes each, 5/8" OD
  - 10" NPS OD x ~19' long
  - Locations for temperature and pressure measurements at the center of each shell

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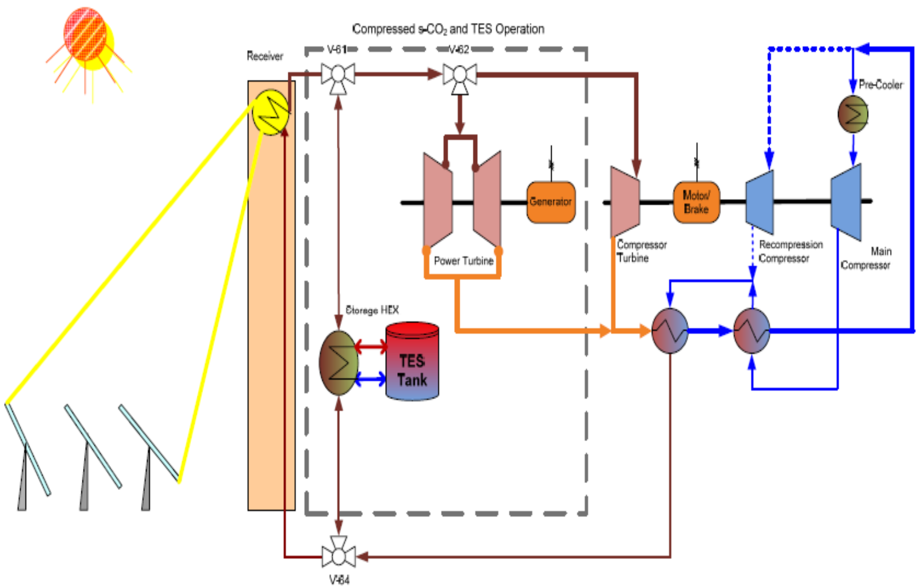
HOFFMAN Process, Inc.

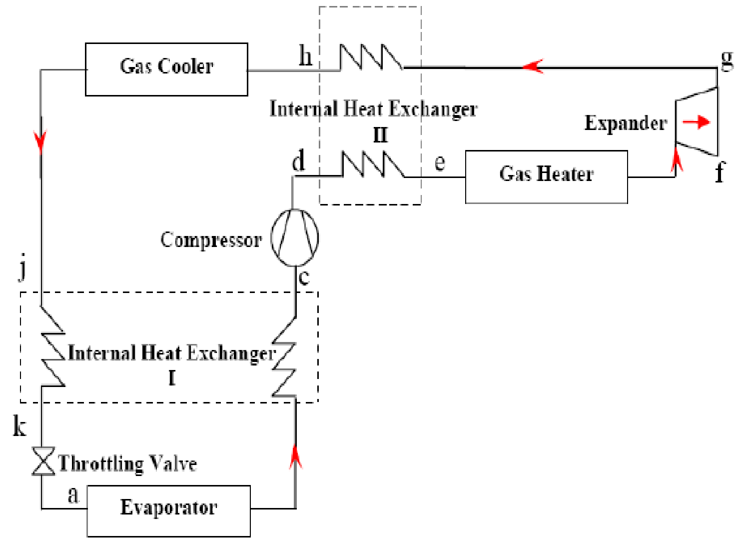
Other Advanced SCO_2 power cycles include

CSP closed-loop recompression Brayton cycle with thermal storage



Modular power tower design

Cooling and power Combined cycles



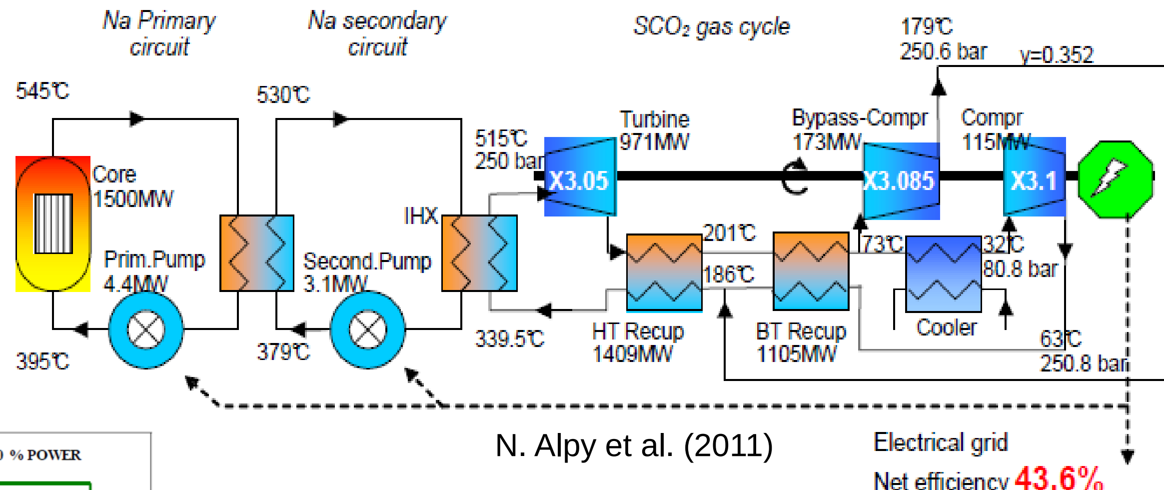
Tri-generation if the gas cooler provides heating service

The lower thermal mass makes startup and load change faster for frequent start up/shut down operations and load adaption than a HTF/steam based system

Ma and Turchi, (2011)

SCO_2 Brayton Power conversion for SFRs

CEA Astrid test program- research shows significant efficiency increase using SCO2 (43.6%) compared to existing (180 bar) N2 cycle (37.8%)



N. Alpy et al. (2011)

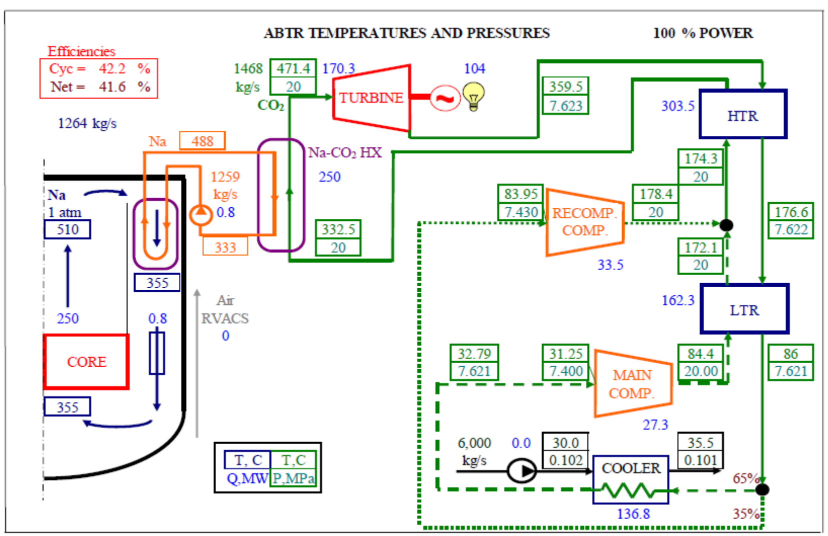


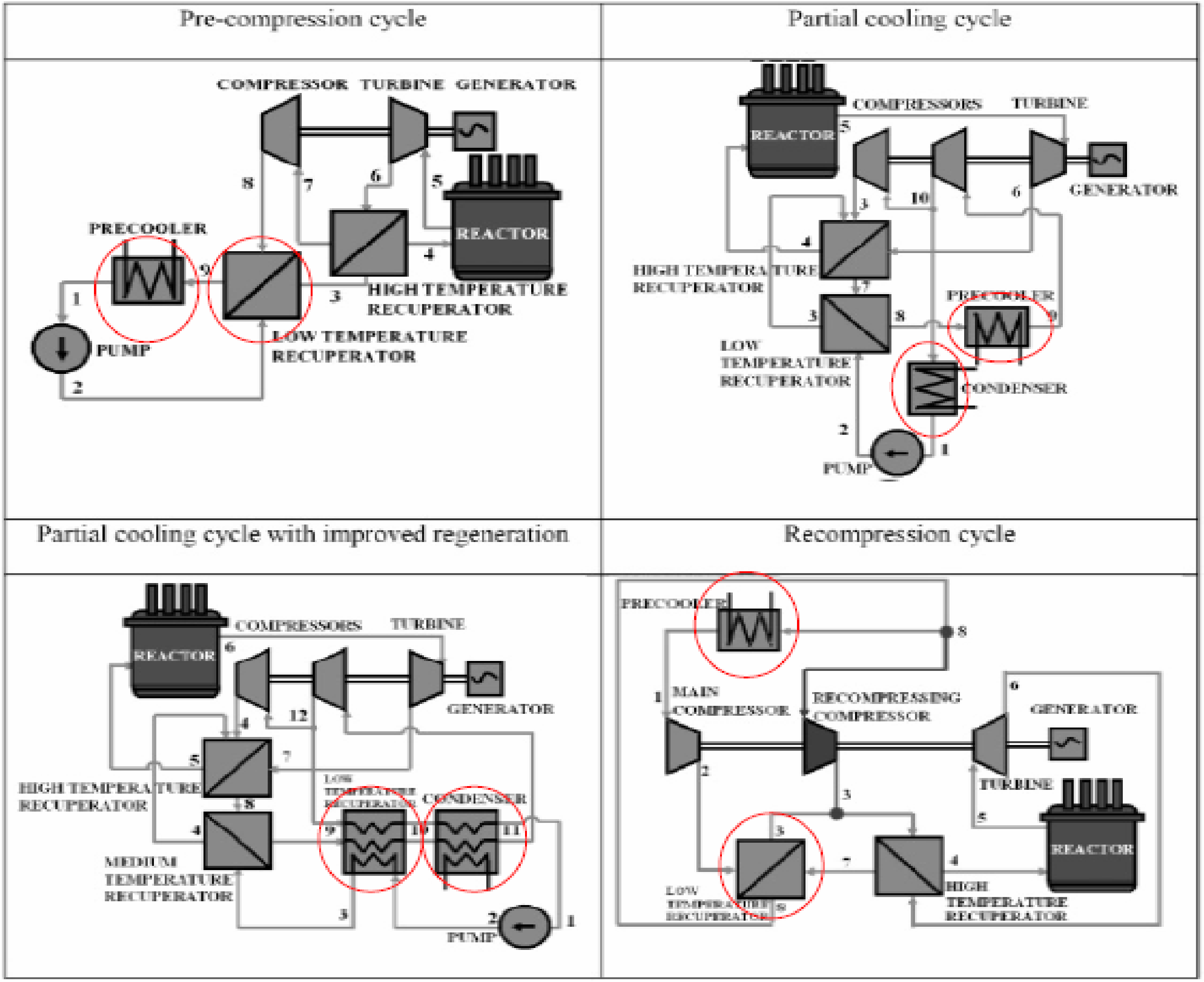
Figure 21. S-CO_2 Cycle Performance with "Ideal" Heat Exchangers.

ANL-GenIV-103 report

Advanced Burner Test Reactor (ABTR) concept design study by ANL.

Potential efficiency increase to 45%

Future modifications to advanced cycles will require more heat exchanger applications



(Dostal et al. 2006)

System Optimisation for Heat Exchangers

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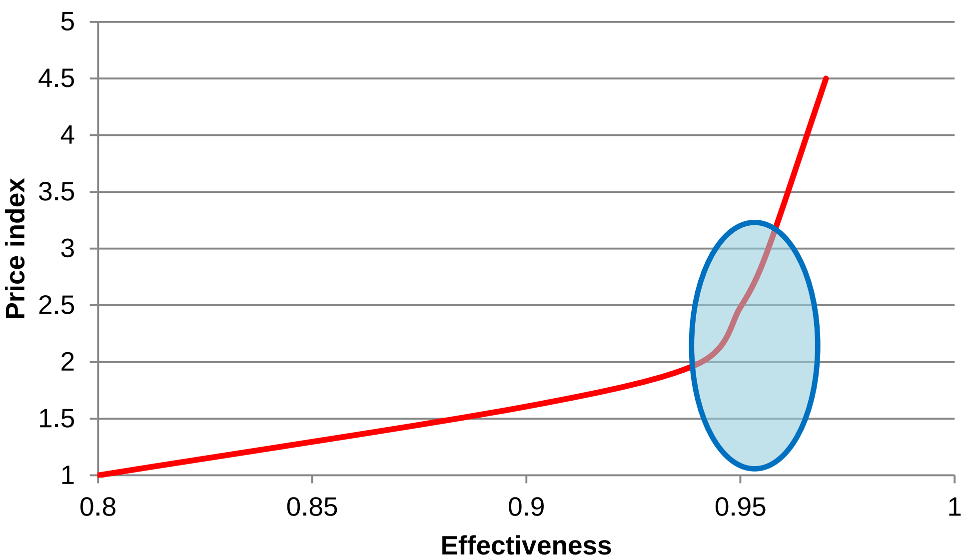
Heat exchanger design considerations

- Plant efficiency vs CAPEX
    - Close temperature approach requires high effectiveness recuperators
    - Higher design temp requires high nickel alloy
- Large property changes require sensitivity checks
    - Operating conditions
    - Pressure levels
- Off design points including turn-down conditions need to be analysed for avoiding pinch point and reversal

Heat exchangers currently form a large part of the overall system cost

CAPEX vs OPEX studies are required to find optimum operating point of the system

- Temperature approach and pressure drop both greatly affect price

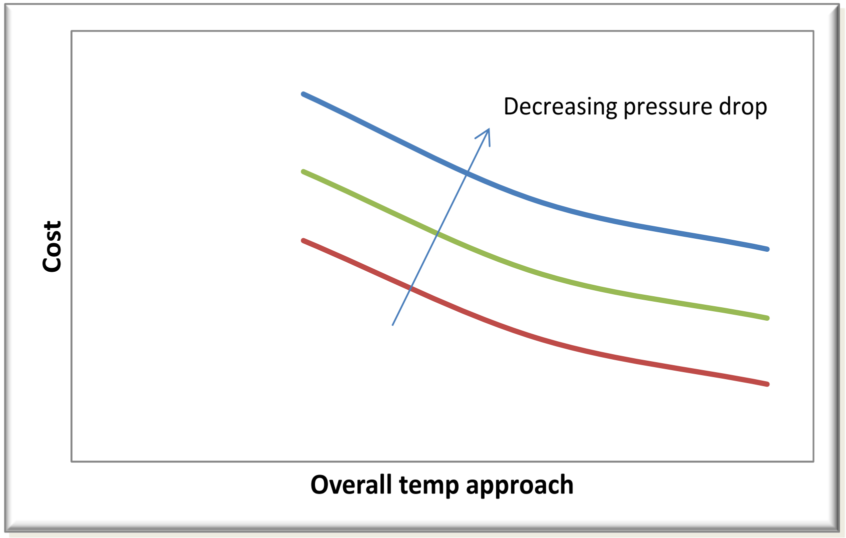


$$Eff = 1 - \frac{\Delta T}{T_{hi} - T_{ci}}$$

Where ΔT = minimum temperature approach

Pinch point varies per technology type. Graph shown for PCHE.

Optimum point varies depending on process conditions and technology type used



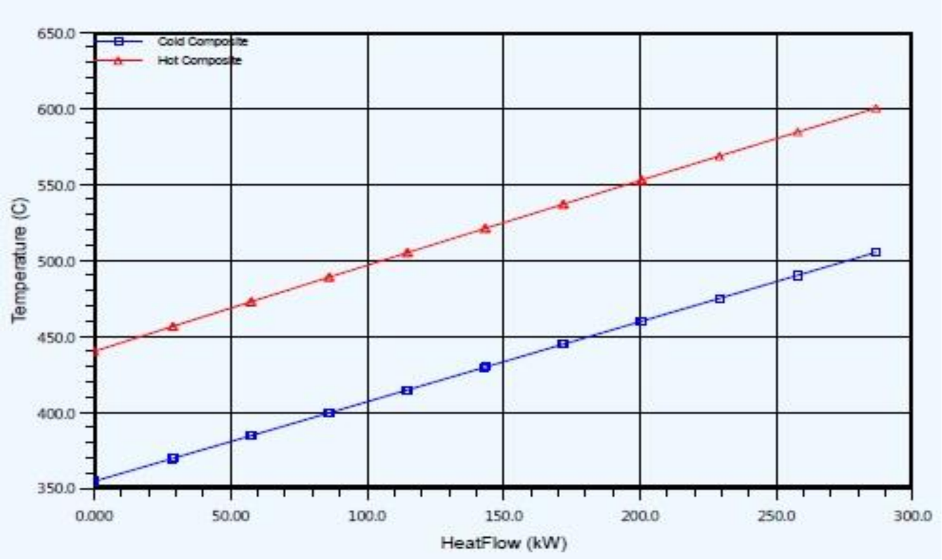
Split into HT and LT Recuperators

The Recuperator is often split in two sections: Hot and Cold Recuperators

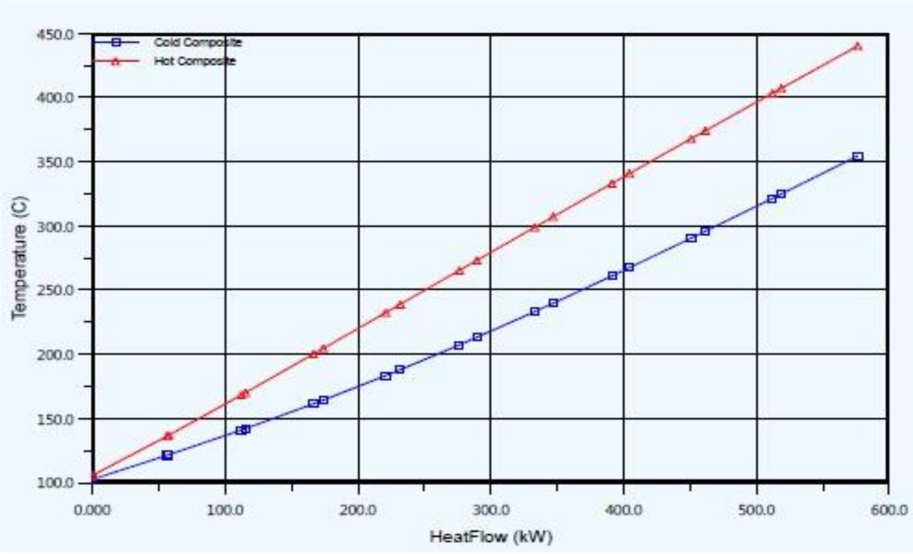
Selection of the middle point needs a detailed study

The HT Recuperator is mechanically driven due to the relative low material strength at high design temperatures and may require high nickel alloy

HT Recuperator

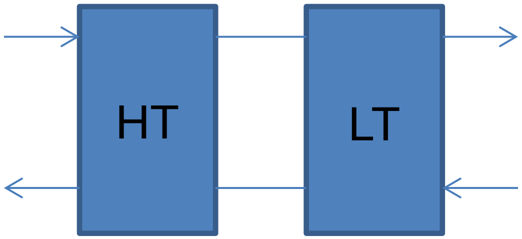


LT Recuperator



Breaking the recuperator in two sections also reduces the thermal gradient per unit

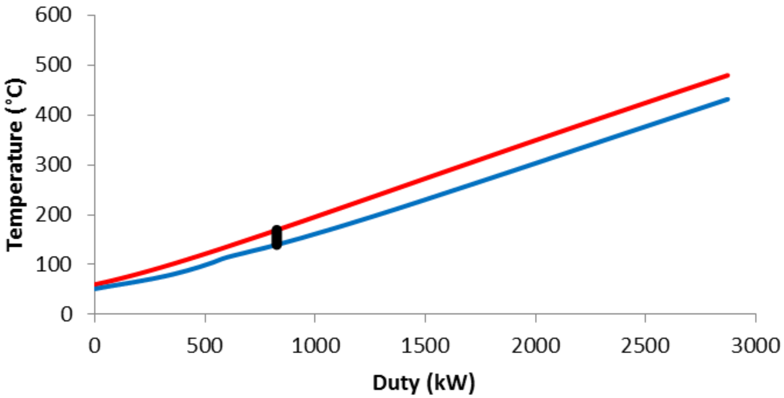
The LT Recuperator, less constrained mechanically, will typically have a larger duty and the pinching point



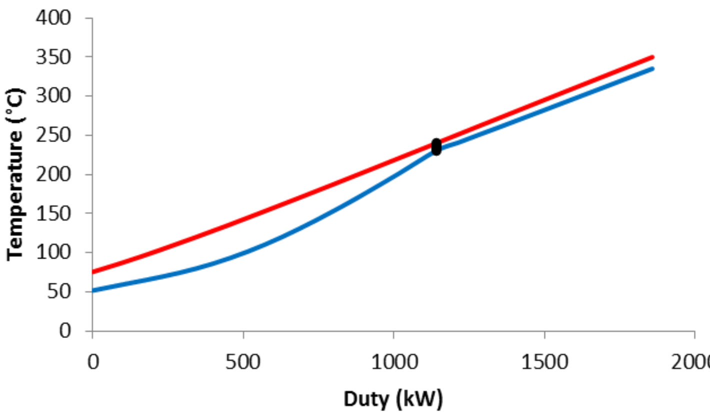
Design Cases need careful consideration

Reducing the inlet temperature away from the designed operating temperature can drastically change heat curve.

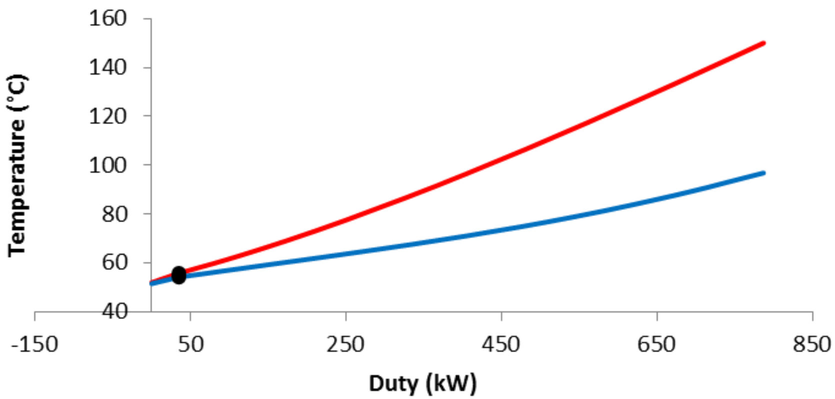
If lowered to much will cause pinch point in HT exchanger. Leaving LT exchanger redundant.



Design conditions: 480°C Inlet



350°C Inlet



150°C Inlet

Economy of Scale

Another important factor to take into account when commercializing these cycles is the economy of scale.

- The relation between the cost of the heat exchangers and the duty of the plant is not linear
- As the size of the plant increases, the price per kW of the exchangers decreases logarithmically

HEXs suited for SCO2 applications

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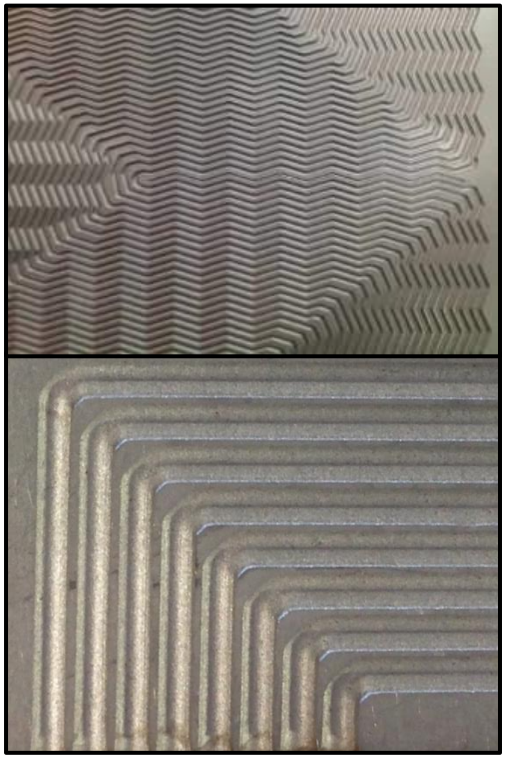
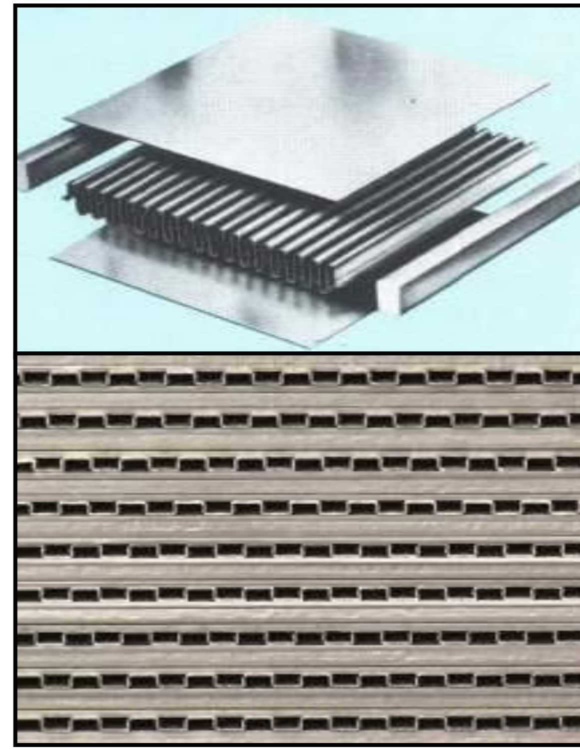
General Overview

Exchanger type	Advantages	Disadvantages
Shell & Tube	- Most commonly available - Wide range of design conditions - Versatile in service	- Lower thermal efficiency - Subject to vibration issues - Large overall footprint
Compact	- Multiple configurations available - High thermal efficiency - Small overall footprint - Low initial purchase cost - Thermo-mechanical strain tolerance	- *Small flow channels** - Limited inspection access for the core - Not well understood by operators

*Also an advantage

Heatric PCHE



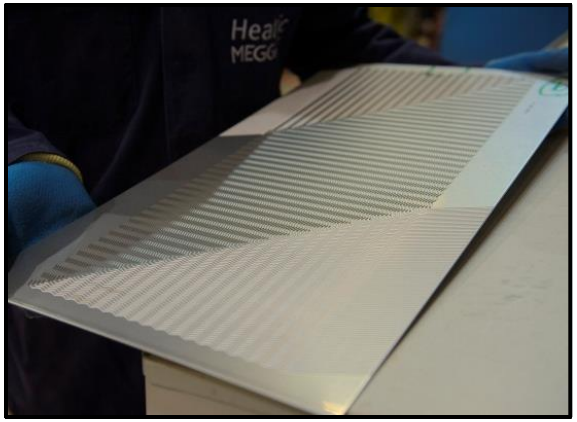


PCHE
Printed Circuit Heat Exchanger

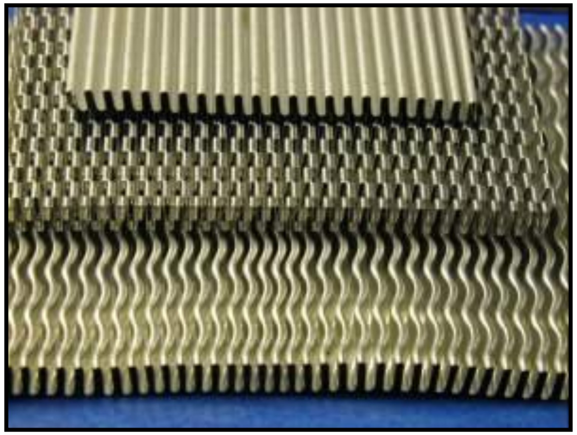
H^2X
Hybrid

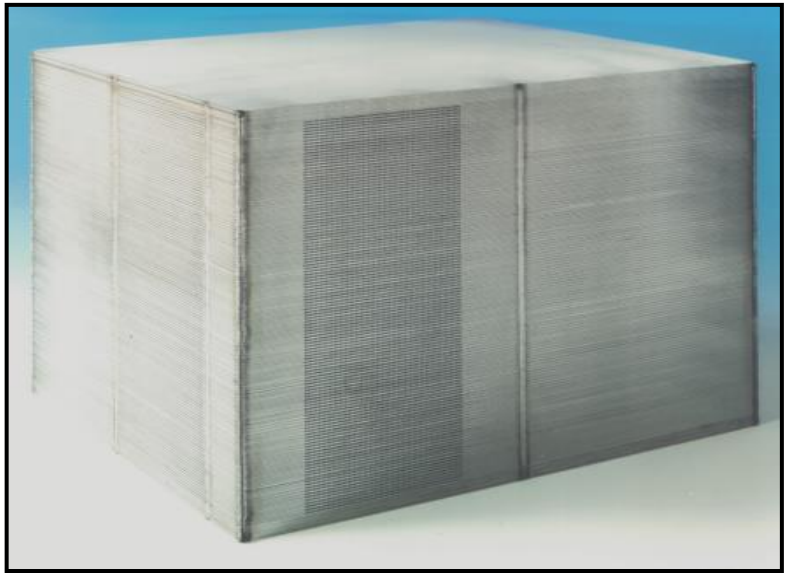
FPHE
Formed Plate Heat Exchanger

Main Components



Etched plates
Or
Formed plates

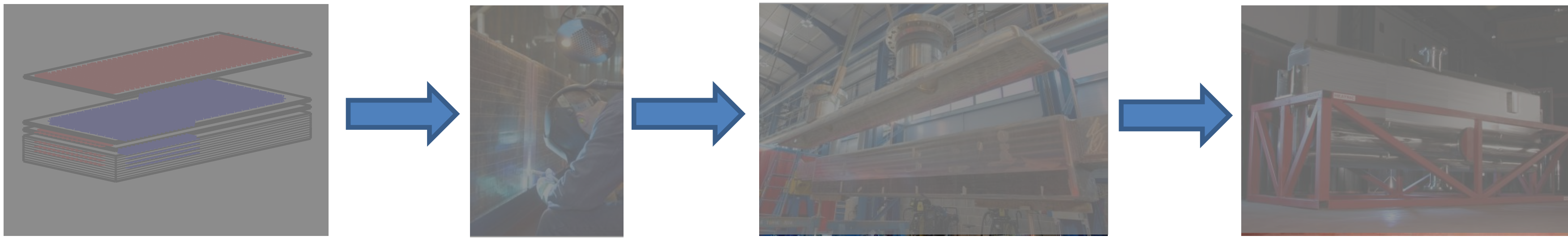


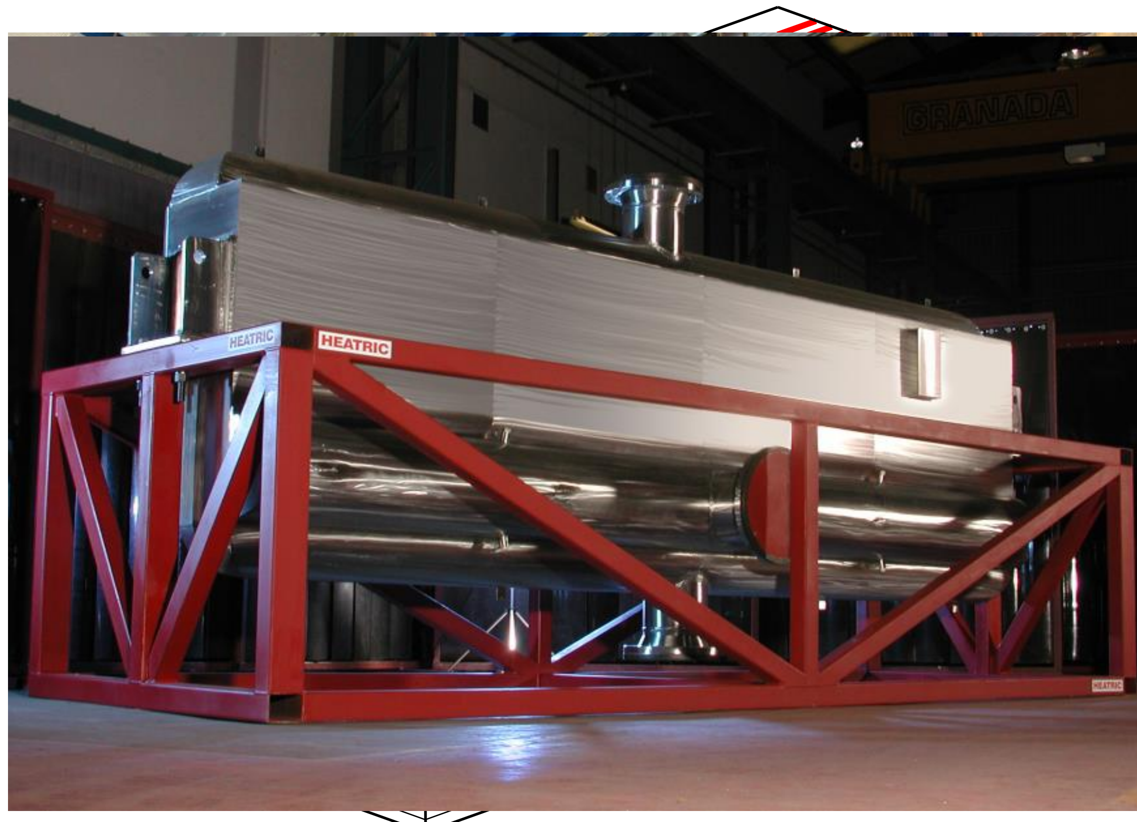


Diffusion
bonded core

Headers,
nozzles,
flanges

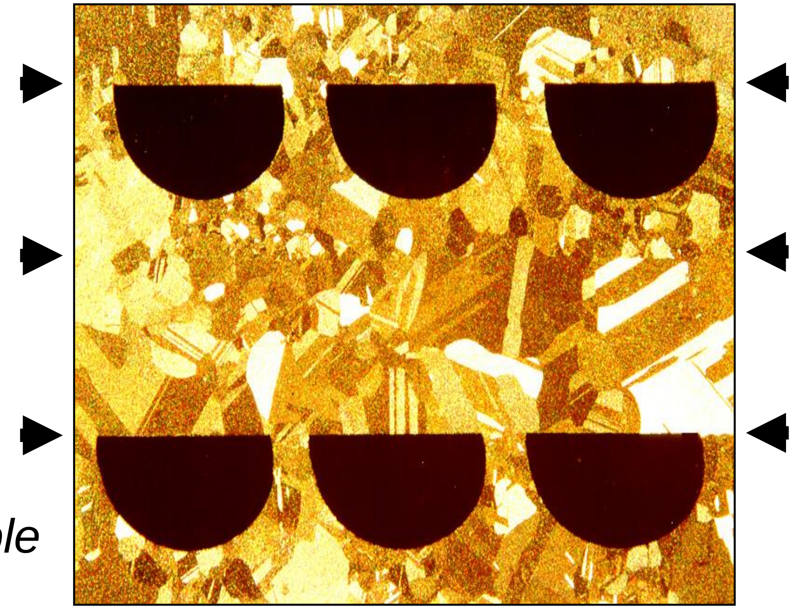
Construction





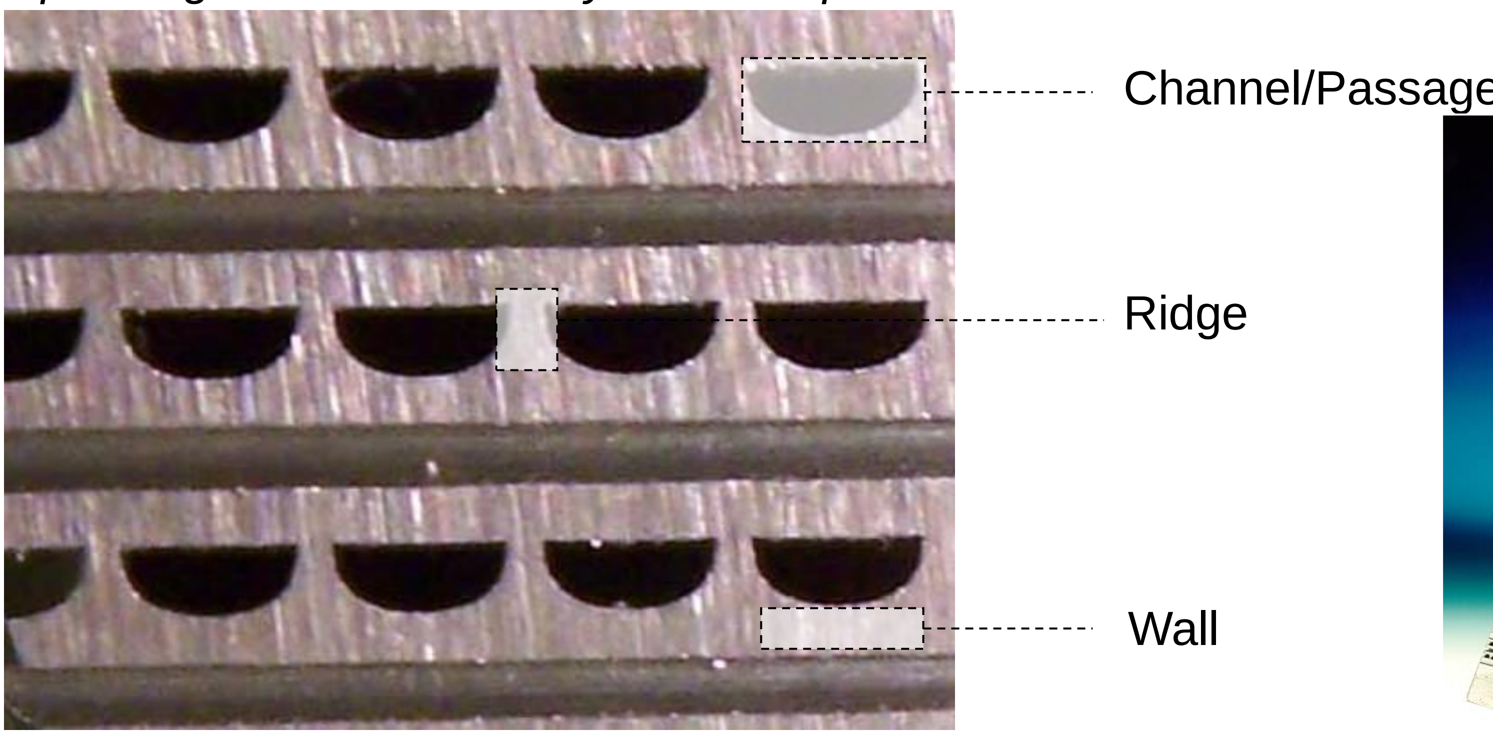
1. Stack and Diffusion Bond Core
2. Block to block joints
3. Assemble headers, nozzles and flanges
4. Weld headers, nozzles and flanges to core

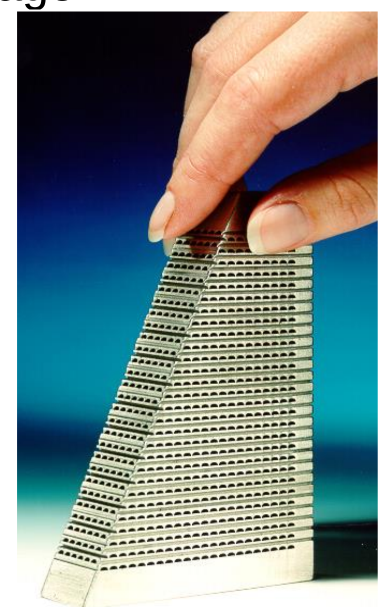
Core Details



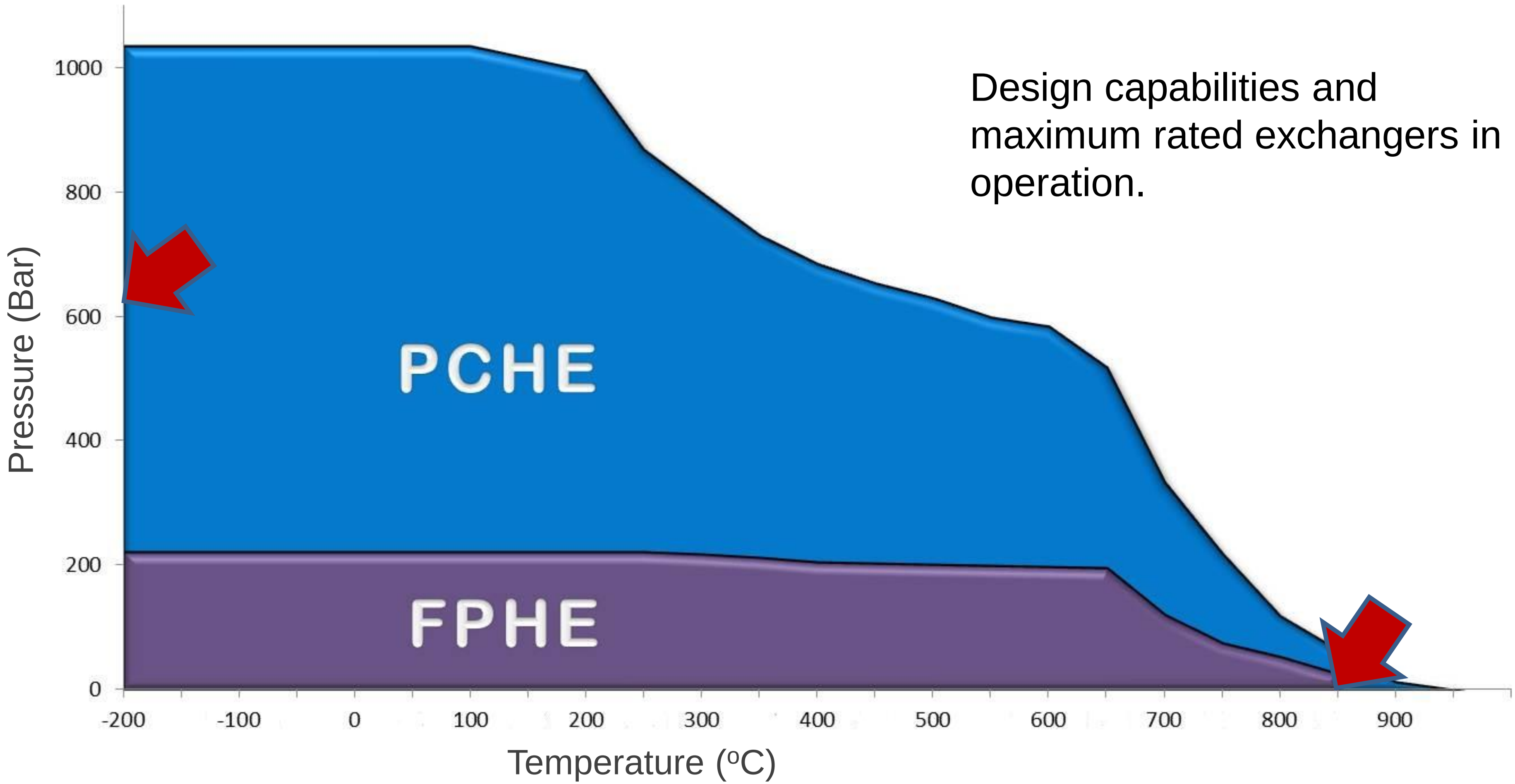
<u>Current Typical Dimensions</u>
Channel Depth – 1.1 mm
Plate Thickness – 1.69 mm
Individual core block – 600 x 600 x 1500 mm
Total unit length – 8500 mm
Hydraulic Diameter – 1.5 mm

Cores are bespoke designed and values are variable depending on thermal and hydraulic requirements





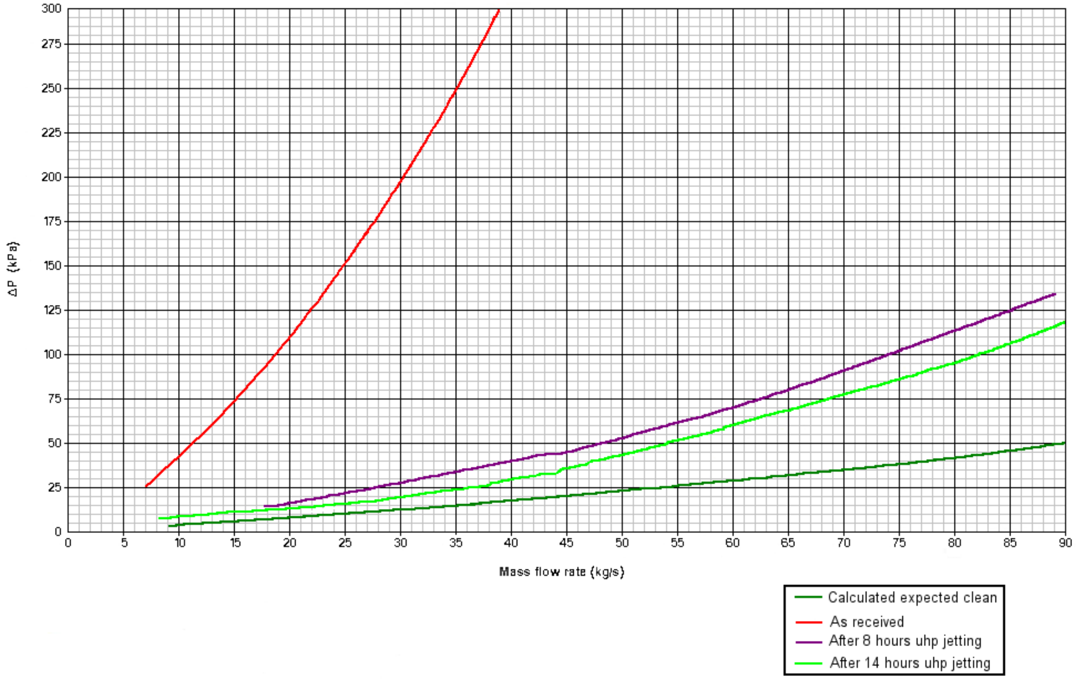
Operating Conditions



Design capabilities and maximum rated exchangers in operation.

Maintenance

- Mechanical
  - Ultra High Pressure (UHP) water jetting
  - Successfully used to clean core and headers
- Chemical
  - Can be used with UHP or standalone



before UHP …

… and after

Design Considerations for Heat Exchangers in the Brayton sCO2 Cycle

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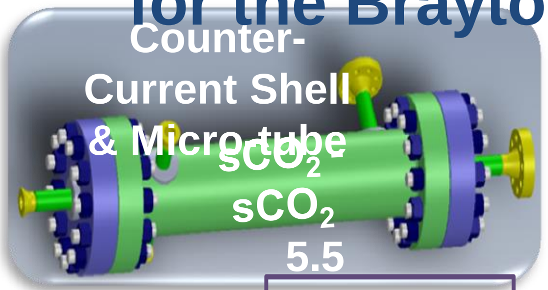
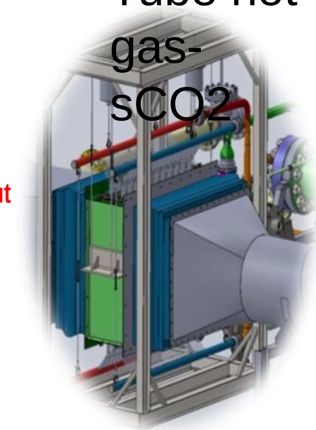
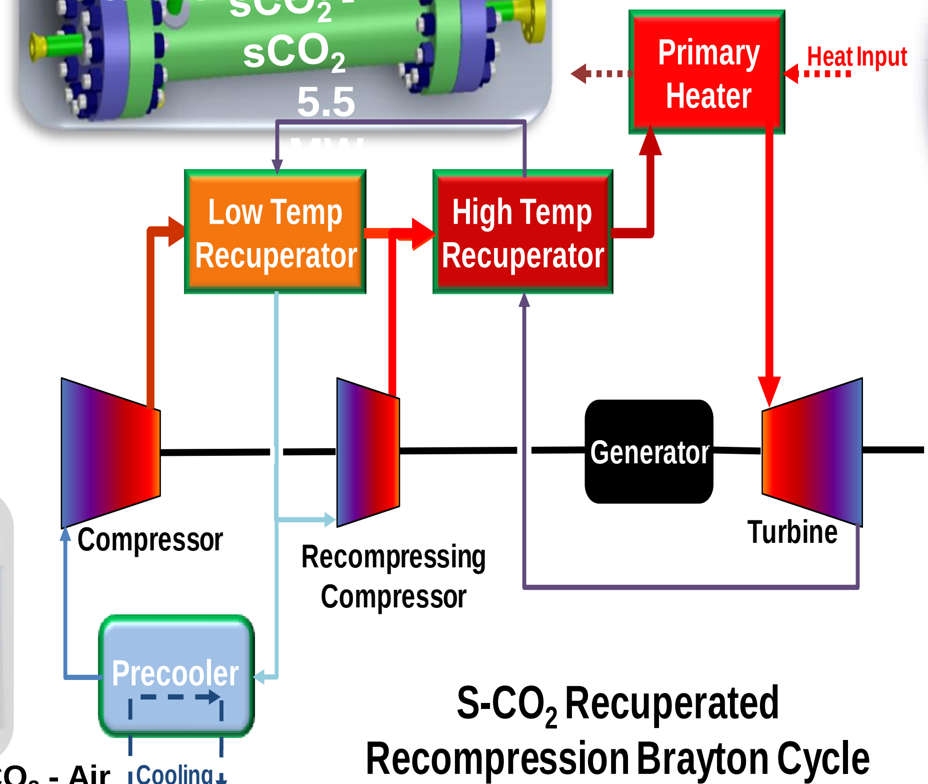
Thar Energy sCO_2 Recuperators, Heater HXs & Pre-cooler HXs for the Brayton Cycle



Bare Tube hot gas-sCO2





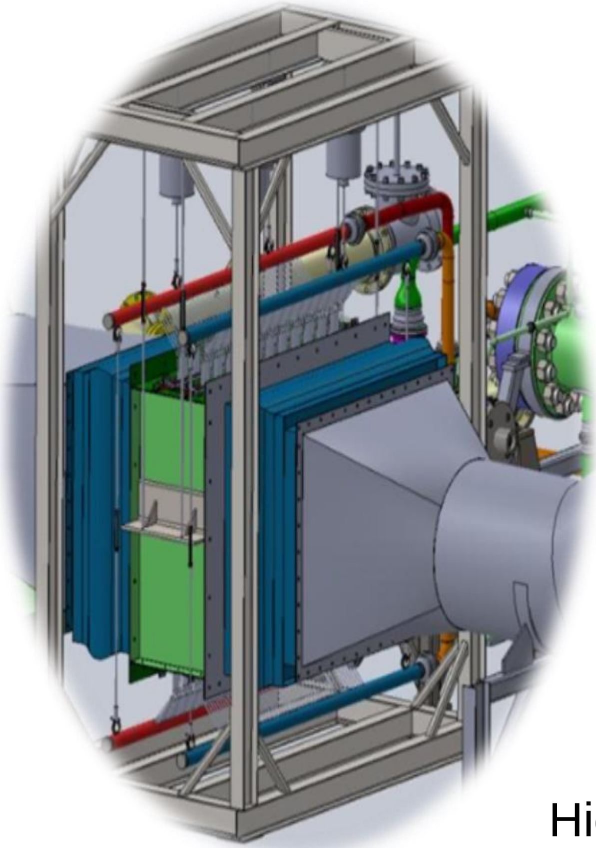
or water

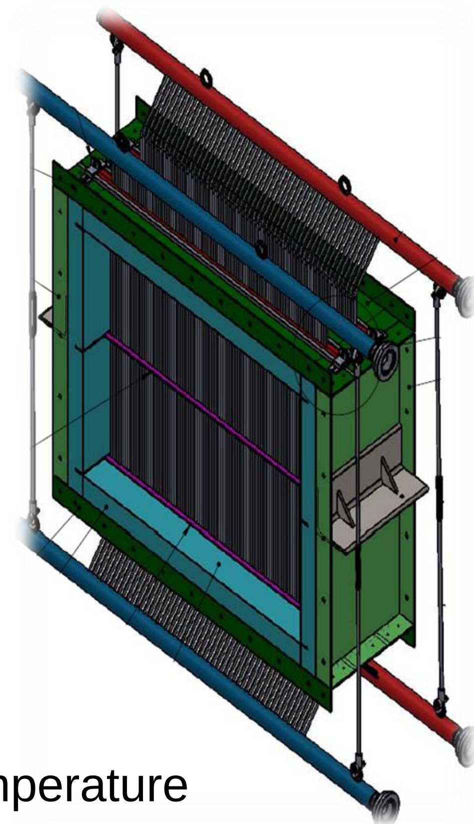
Micro-channel sCO_2 - Air HX
Or
Counter-Current Shell & Tube
sCO2 - Water HX

Primary Heater

Bare Tube Hot Gas - sCO_2 HX

Inconel 740H Construction





High temperature design considerations already discussed by others in this

Heater Design considerations

- Super-alloys for high temperature corrosion
- Design to creep/stress-rupture rather than yield strength

Design Conditions:

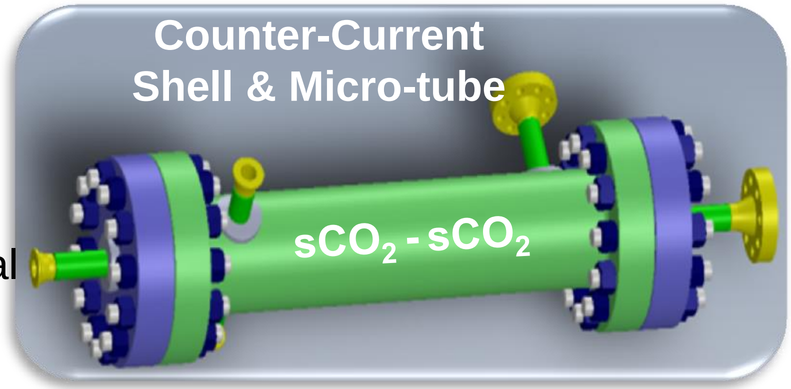
Gas Fired Burner/Blower Outlet Temperature: 870°C

sCO_2 Outlet Temperature: 715°C

High Temperature Recuperator and Low Temperature Recuperator

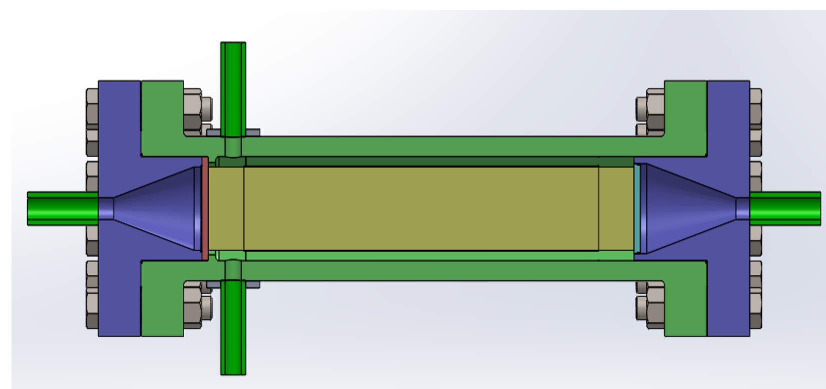
Counter-Current Shell & Microtube Heat Exchanger

- ASME Sec VIII, Div I Stamped Pressure Vessel
- High thermal efficiency
- Floating Head Design to Reduce Thermal Stresses
- Easy Serviceability and Maintenance
- Replaceable Tube Bundle
- Low Cost <$100 per kWt
- Scalable 1 - 1,000 MWe facilities



HTR Design Considerations

- Requires nickel alloys for high temperature
- Depending on the selected material, may be designing to allowable yield strength or to creep /stress rupture



LTR Design Considerations

- Lower temperatures can use stainless steels

Many Recuperator design considerations already discussed by others in this tutorial

Pre-Cooler Design Consideration
Air-Cooled or Water-Cooled

Advantages of Air-Cooled

- In areas that have scarcity of water - doesn't require a water source
- Lower Initial Cost
- Less Maintenance

Advantages of Water-Cooled

- More Compact
- Higher efficiency
- Larger heat removal capabilities
- Longer lifespan

Disadvantages of Air-Cooled

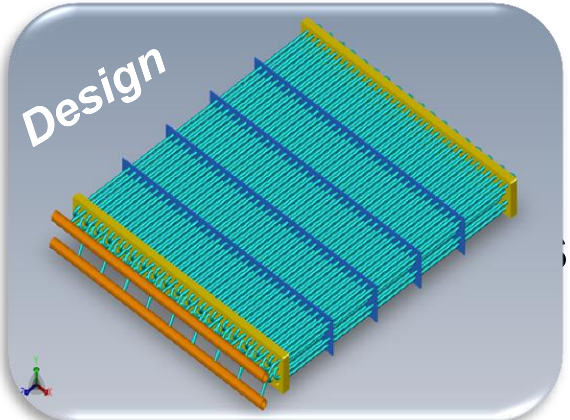
- Lower efficiency
- Shorter lifespan
- Lower heat removal capabilities
- Noisier
- Peak output is limited on hot days

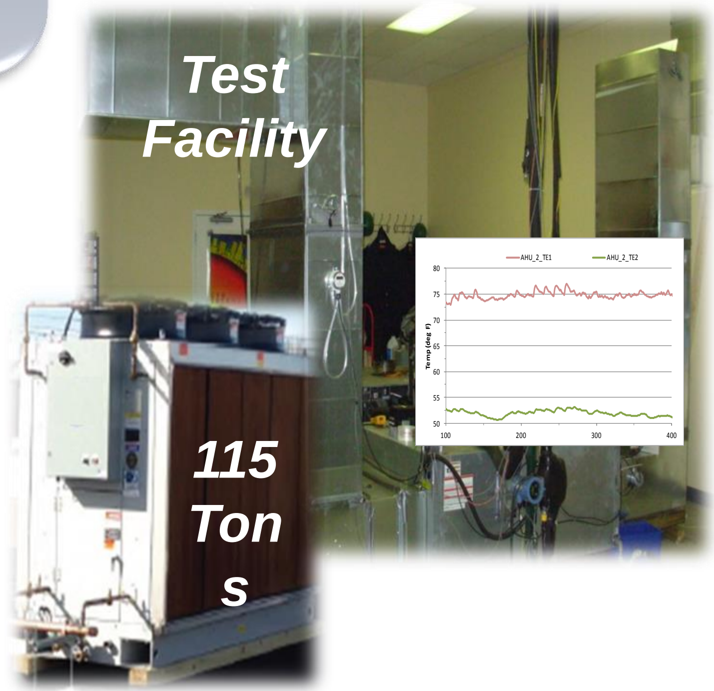
Disadvantages of Water-Cooled

- Requires a water source
- Higher initial cost
- More maintenance
- Water treatment costs

Pre-Cooler Design Consideration Air-Cooled

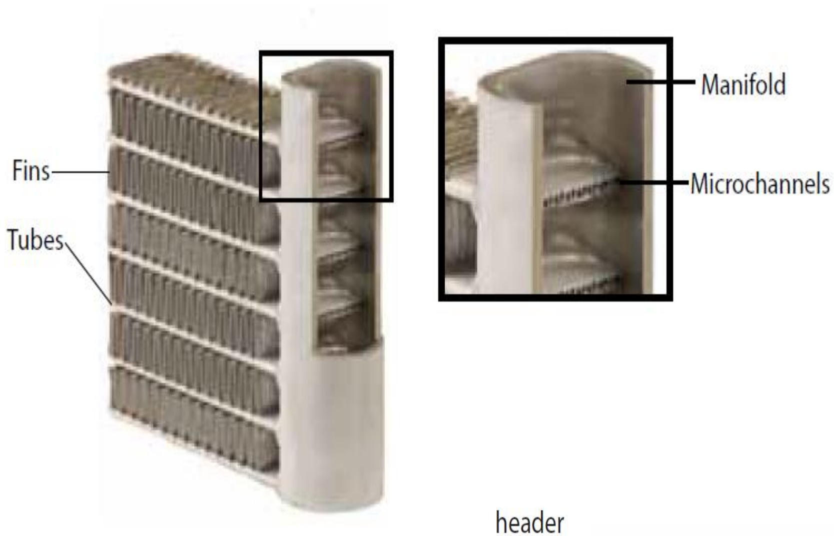
Micro-channel sCO_2 - Gas Cooler HXs

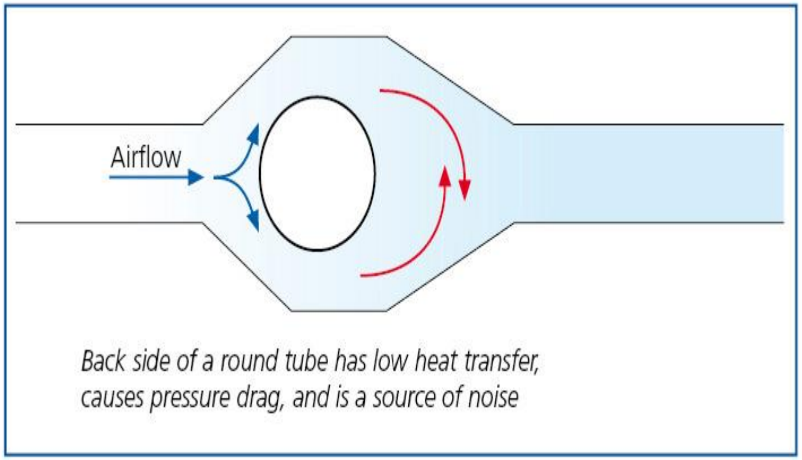


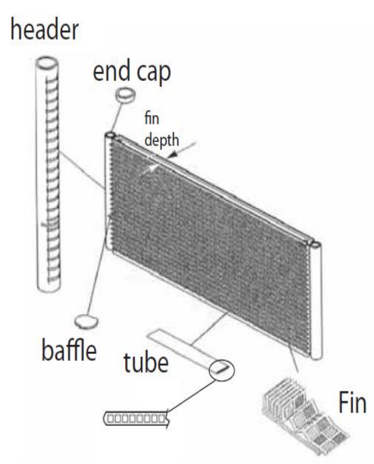


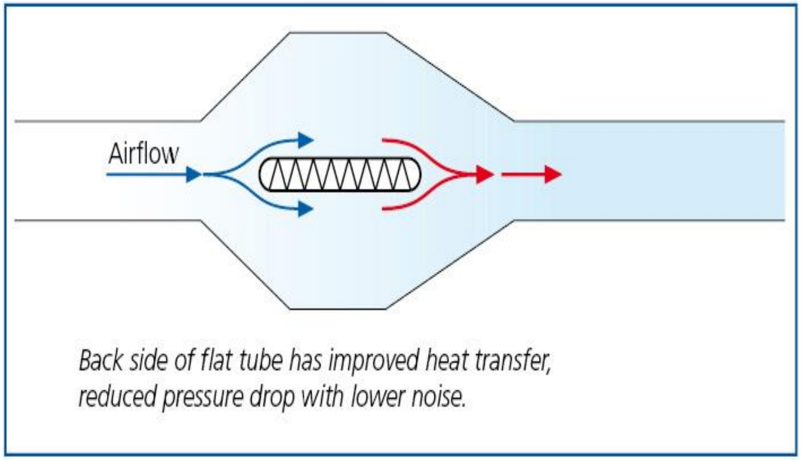
CO_2-Air Approach Temperature as Low as 2°C

Pre-Cooler Design Consideration Air-Cooled

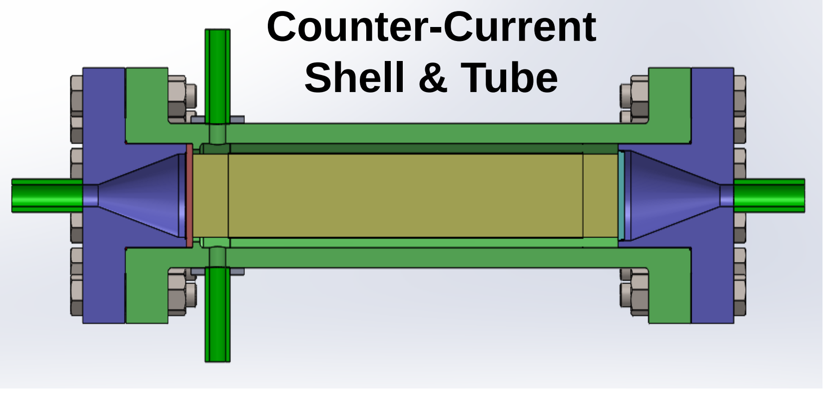








Pre-Cooler Design Consideration Water-Cooled



Configuration Options to Consider when using water

- Open Loop, Untreated Water – Shell and Standard Tube Diameters (5/8” to 1.0” diameter)
- Open or Closed Loop, Filtered, Soft Water – Shell and Small Tube Diameters (1/8” to 3/8” diameter)
- Closed Loop, Filtered, Demineralized Water – Shell and Micro-Tube or micro-channel

Pre-Cooler Design Consideration Water-Cooled

Cooling water generally on the tube side

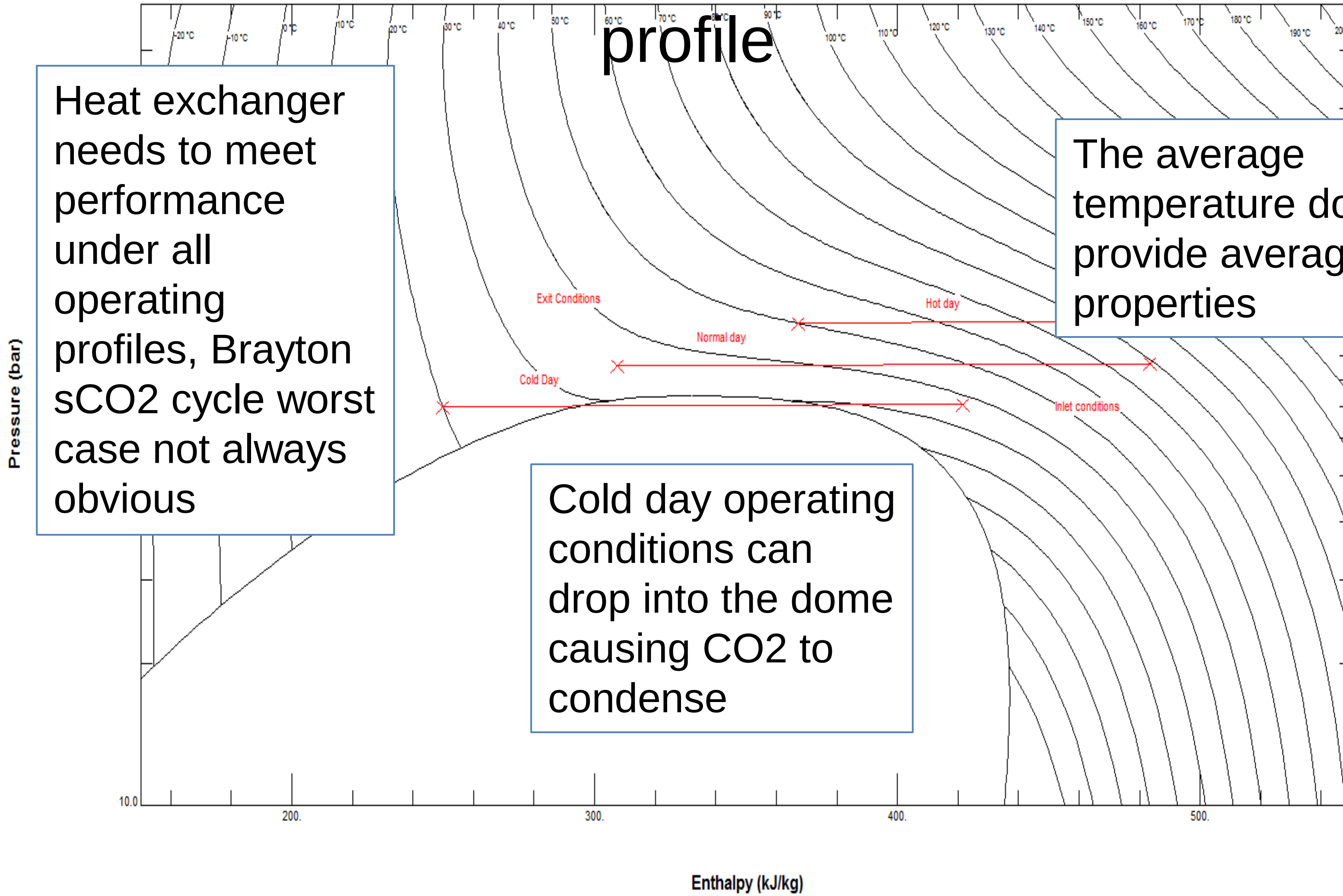
- Facilitates cleaning the tubes either mechanically or by water jet
- Possible to inspect individual tubes for pitting corrosion
- Fewer sedimentary problems occur due to the simpler flow path
- Easier to maintain a minimum velocity to reduce fouling

Pre-Cooler Design Consideration Water-Cooled General Design Guidelines

- Pressure Vessel Design requirements per ASME Section VIII
- Design guidelines per TEMA Standards of the Tubular Exchanger Manufacturers Association
- Water velocities are typically designed between 1.5 and 2.5 m/s
- Bulk water temperature should not exceed 50°C
- To avoid severe mal-distribution between tubes or passages, the total pressure drop across the tubes or passages should be at least 5 times the inlet nozzle pressure drop

Pre-Cooler Design Considerations

Ambient conditions affect the heat exchanger profile



Heat exchanger needs to meet performance under all operating profiles, Brayton sCO2 cycle worst case not always obvious

The average temperature does not provide average properties

Cold day operating conditions can drop into the dome causing CO2 to condense

Pre-Cooler Design Consideration Condensing

- Cannot design using LMTD method, use segmented model
- The S-CO_2 power system may take advantage of lower heat rejection temperatures by allowing the pre-cooler to condense the CO_2.
- Condensation lowers the compressor inlet pressure, increases the fluid density and increases the compression ratio
- The liquid to vapor density ratio is roughly a factor of 2:1. Because of this small density ratio, a radial compressor may be able to “pump” liquid CO_2.

Pre-Cooler Design Consideration Condensing

- Heat transfer resistance on the condensing side of an exchanger is made up of two parts
  1. Resistance of the condensate film
  2. Resistance of the vapor film between the vapor and condensate interface
- The transfer through the film is conductive and generally reduces the heat transfer
- The value of the condensate film resistance depends on the geometry of the surface, vapor shear stress, turbulent/laminar, external/internal to the tube, horizontal/vertical etc
- When condensation is expected the heat exchanger should be sloped to direct the flow towards the outlet and to prevent flooding lower tubes or passages

Pre-Cooler Design Consideration Typical Tubing Materials

- Copper Alloys – CuNi 90/10, CuNi 70/30, Admiralty, Al Brass
- Titanium Alloys – Ti Grade 2
- Ferritic Stainless Steel – TP439, Sea-Cure, Al29-4C
- Duplex Stainless Steel – Al2003, 2205, 2507
- Austenitic Stainless Steel – TP304, TP316, TP317, 254SMO, AL6XN

No one material is perfect for all applications. Tradeoffs in cost vs. reliability depends on water quality

Pre-Cooler Design Consideration Corrosion Susceptibility

- Galvanic Corrosion
- Pitting corrosion
- Intergranular Corrosion
- Chloride Stress Corrosion Cracking
- Erosion Corrosion
- Fretting
- Crevice Corrosion
- Selective Leaching
- MIC microbe influenced corrosion
- Hydrogen embrittlement
- Corrosion Fatigue

Pre-Cooler Design Consideration Guidelines for Corrosion Avoidance

- Avoid water velocities below 1 m/s to prevent excess deposits which can lead to fouling and local corrosion
- Maximum water velocity of 2.5 m/s to prevent erosion
- Maintain water temperatures below 50C, above that temperature fouling increases significantly due to inverse solubility
- Avoid designing crevices
- Selection of metals should be with similar galvanic potential
- Use cathodic protection when metals have different galvanic potential
- Control water chemistry when water contains halides to minimize pitting
- Minimize particles or droplets in fluid to prevent erosion
- Avoid vibratory or cyclic loading of close contact parts to avoid fretting
- Drain and dry after hydro-test or run in and avoid long term wet layup - MIC

Heat Exchanger Types
Continued

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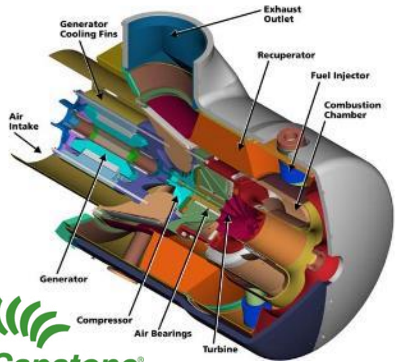
Plate-Matrix Heat Exchangers – An Overview

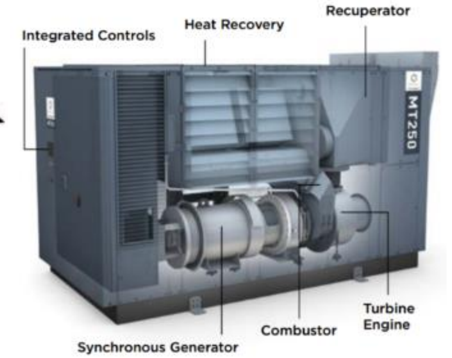


Capstone

(30, 65, and 200 kW)

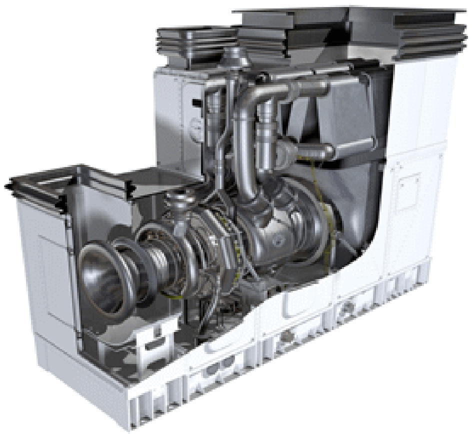
Ingersoll Rand

(70 and 250 kW)



FLEXENERGY

(250 and 333 kW)



ROLLS ROYCE

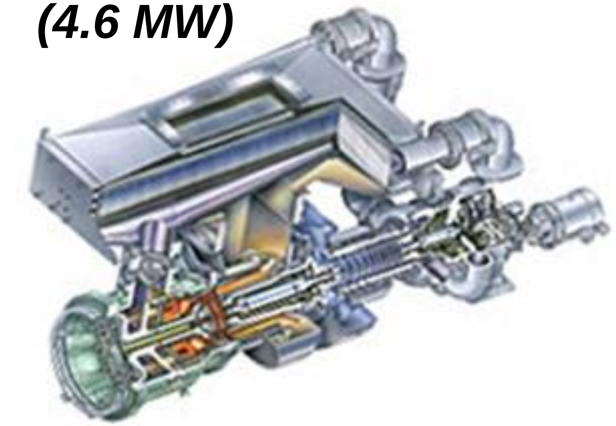
Rolls-Royce®

WR-21 (25.2 MW)

Solar® Turbines

A Caterpillar Company

Mercury-50
(4.6 MW)



The Plate-Matrix Unit Cell

External low-pressure matrices

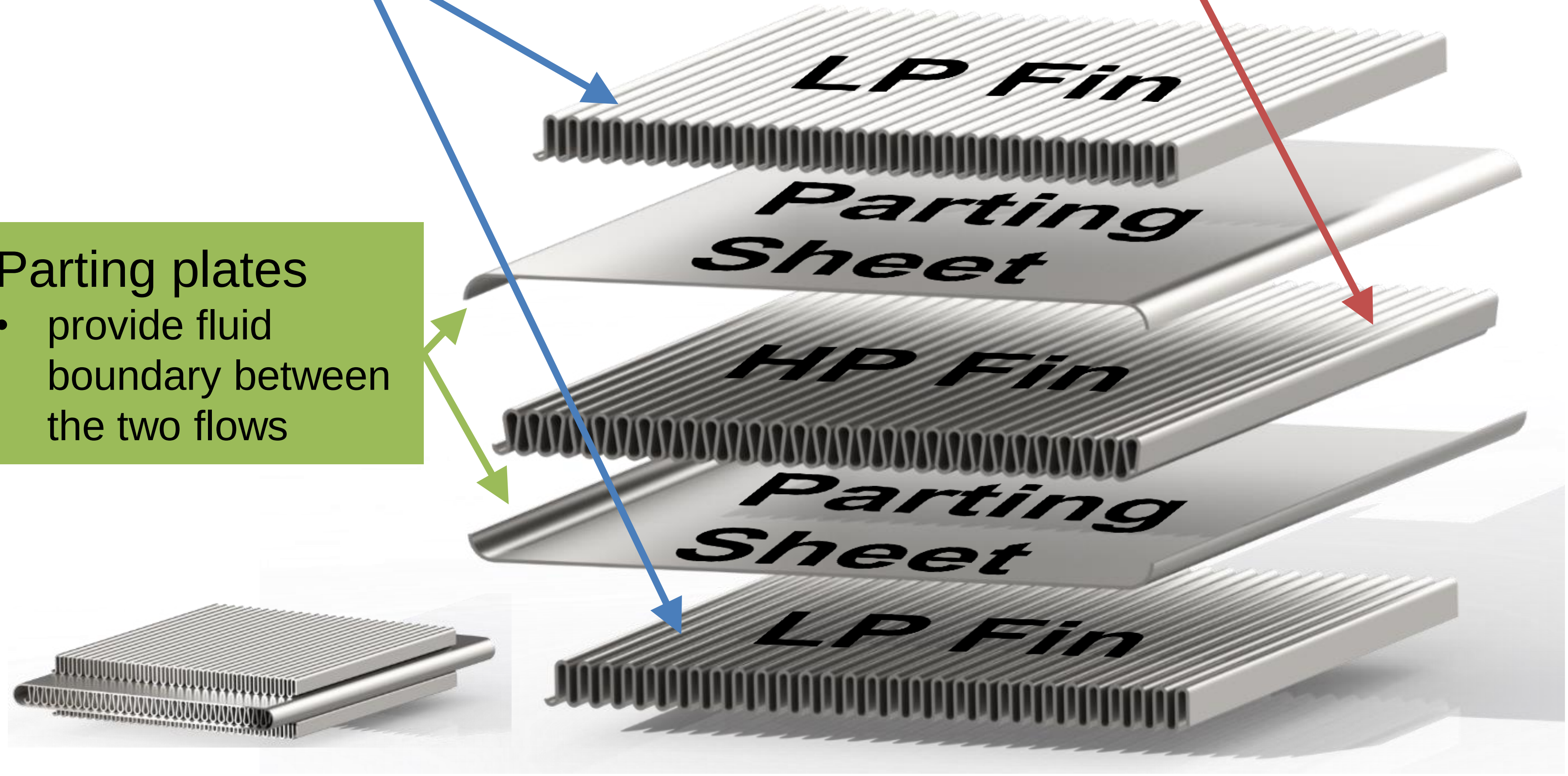
- Enhances the heat transfer of the low-pressure fluid as it flows between adjacent unit cells

Internal high-pressure matrix

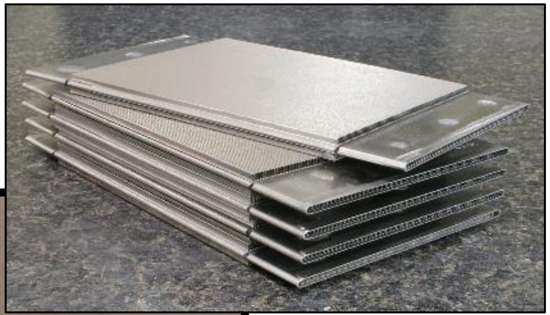
- Enhances the heat transfer of the high pressure fluid as it flows between the two parting plates
- Can serves as structural features for high-pressure (sCO_2) applications

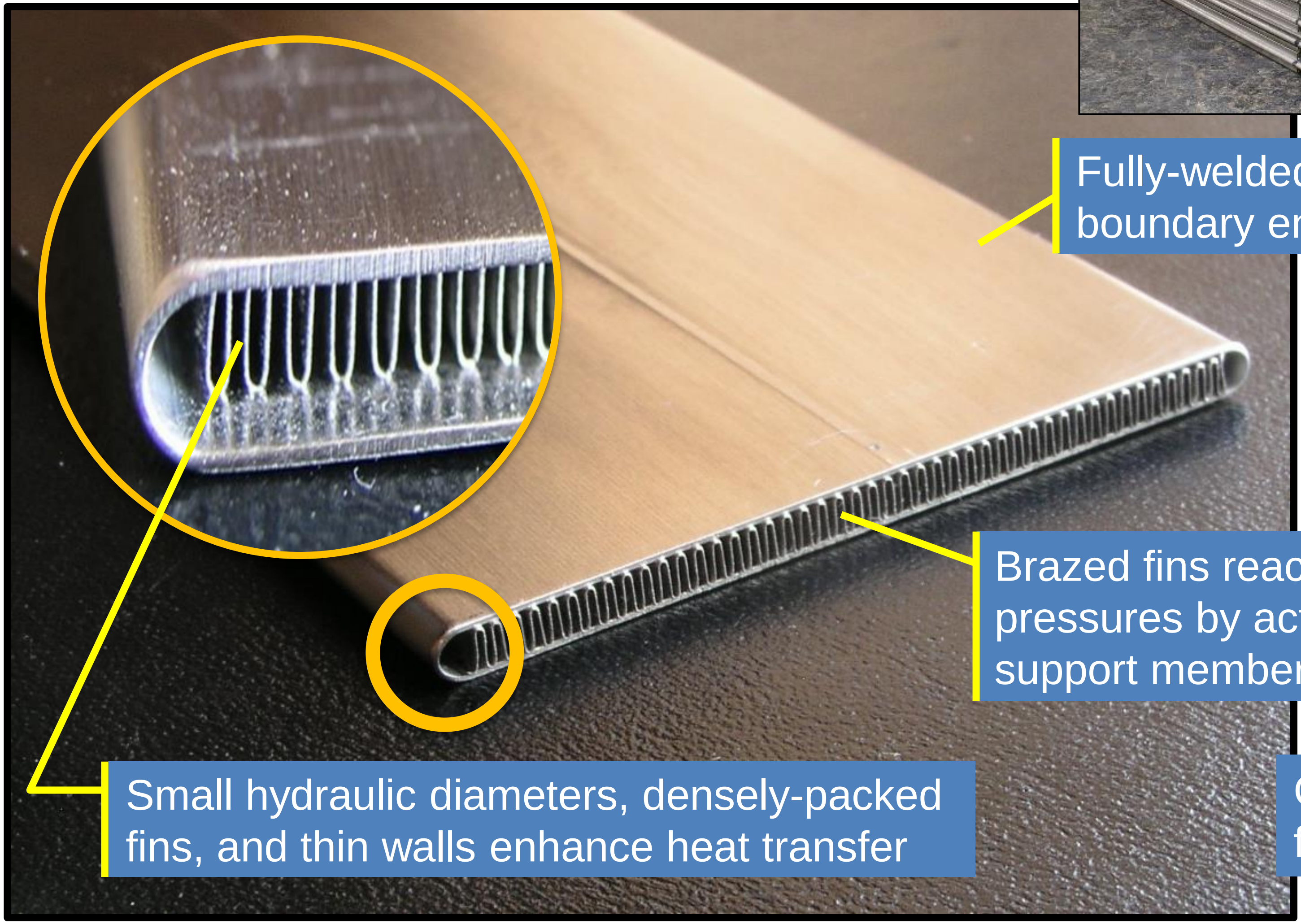
Parting plates

- provide fluid boundary between the two flows



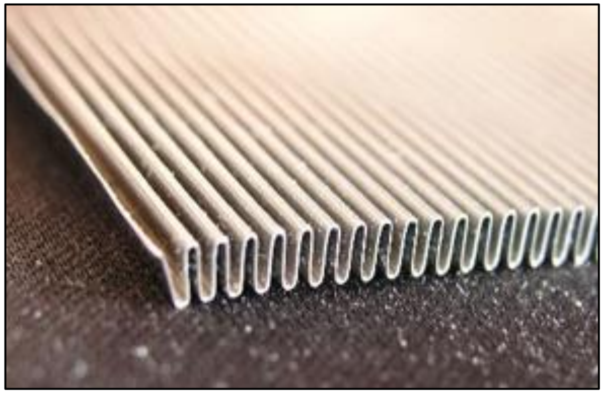
Unit Cell Design

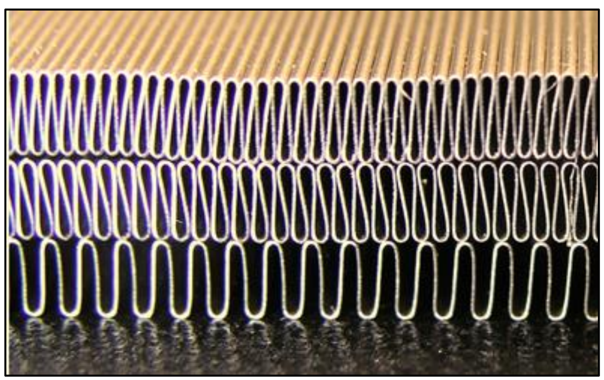




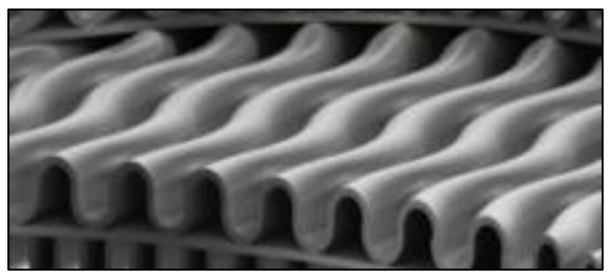
Heat Transfer Matrices

Straight Fin





WavyFin



Wire Mesh

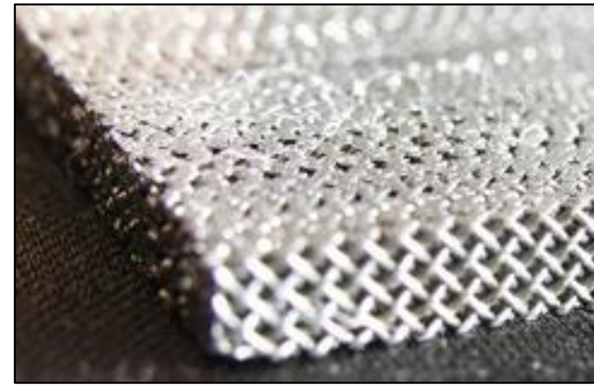
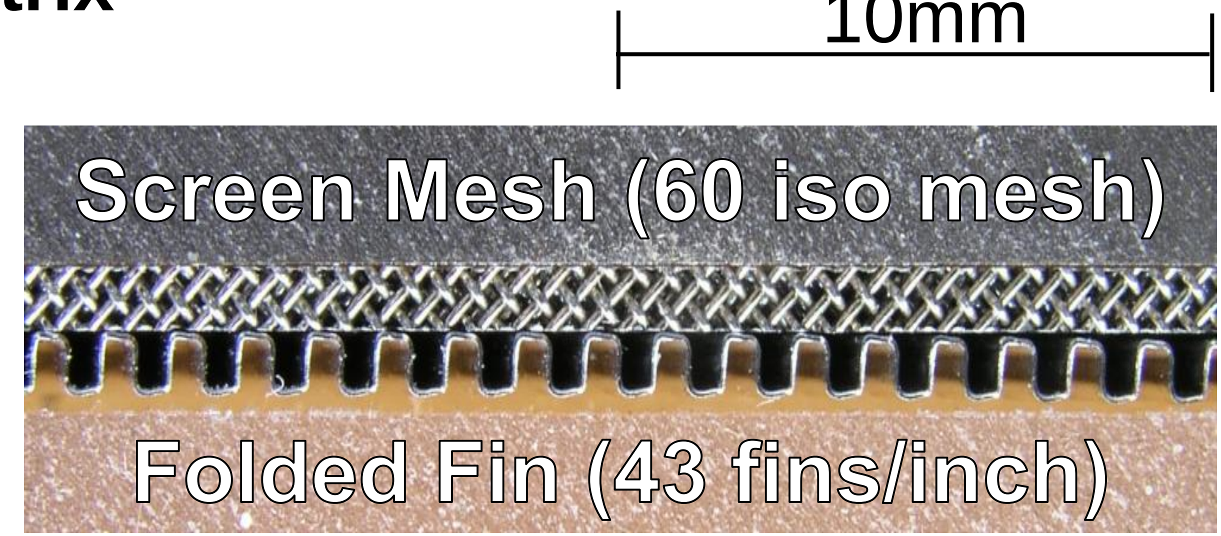


Plate-Matrix Heat Exchangers

Choosing a Matrix

- Cost
- Mass
- Footprint
- Size (Volume)



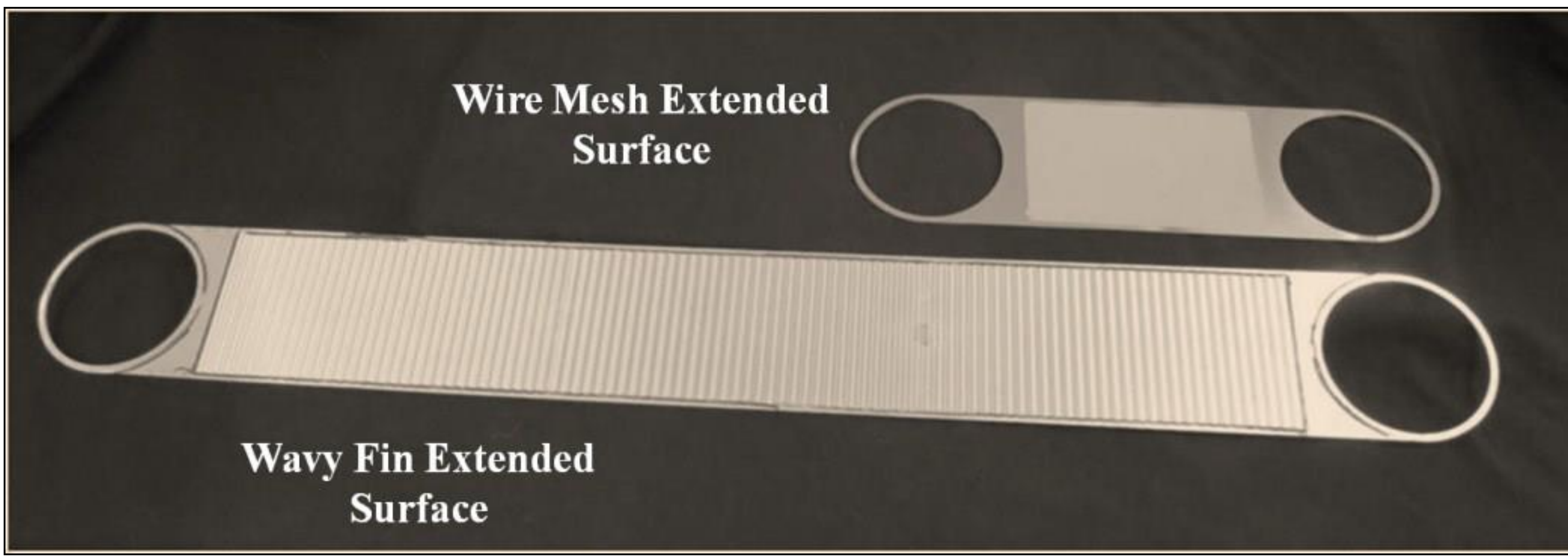
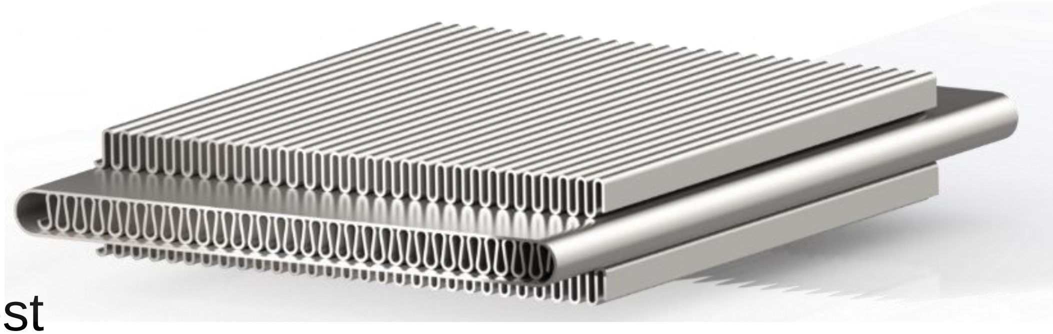


Plate-Matrix Heat Exchangers

The Unit Cell - Characteristics



- Inspectable at the unit-cell level
  - Identifies issues (leaks, poor bonds) at the earliest possible processing point
  - Avoids expensive scrap/repair for local defects
- Enables the independent specification of extended surfaces for each flow
- Manifolds and headers may be integrated directly cell
- Easily configurable flow orientations:
  - Counterflow for maximum heat exchanger potential
  - Crossflow for mismatched flows (e.g. radiator-type applications)

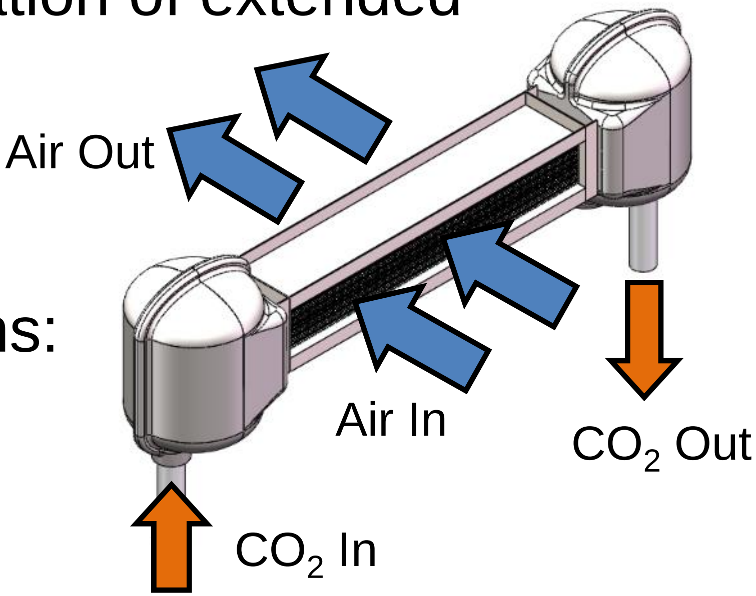


Plate-Matrix Heat Exchangers

Plate-Matrix Heat Exchanger Cell Counter Flows

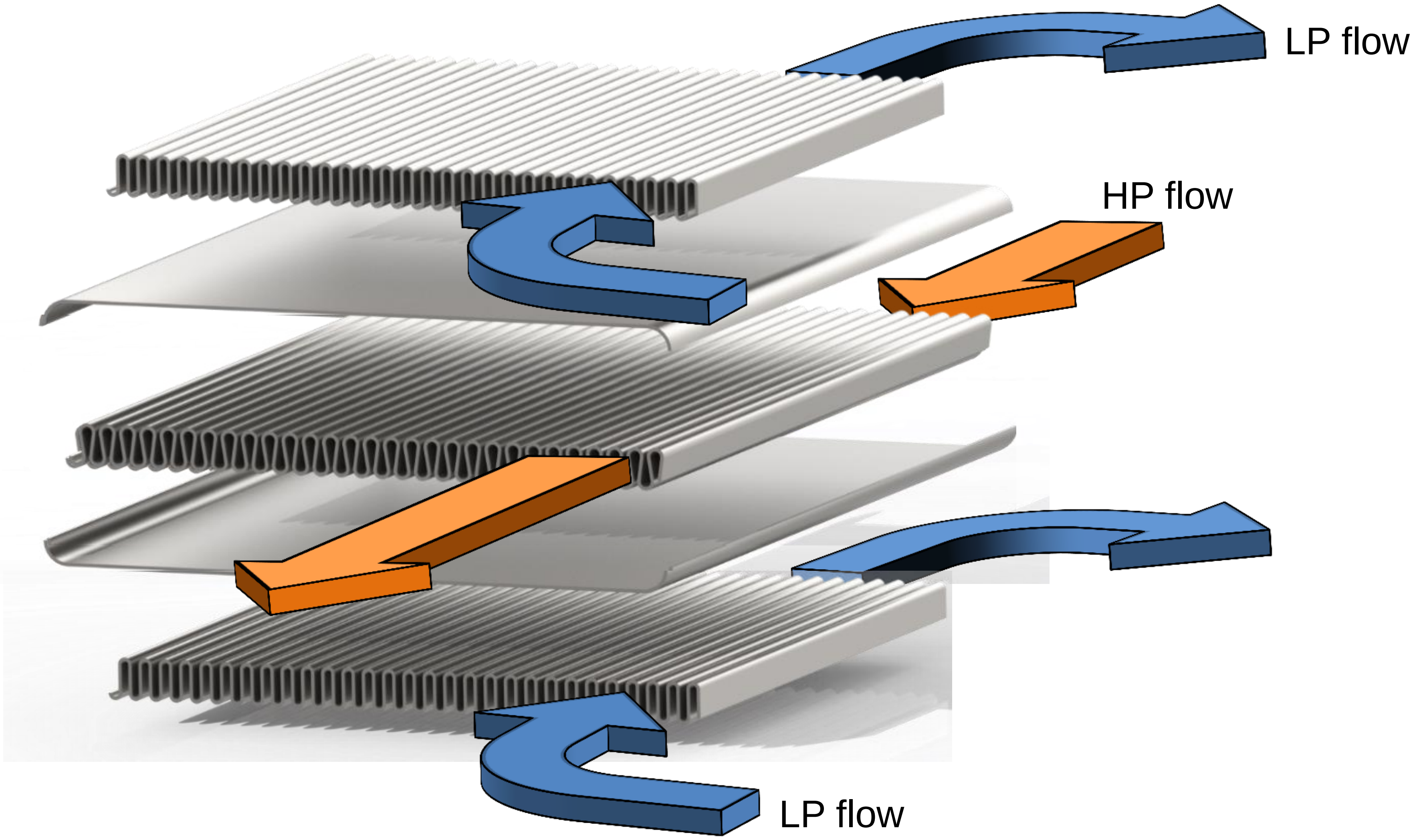
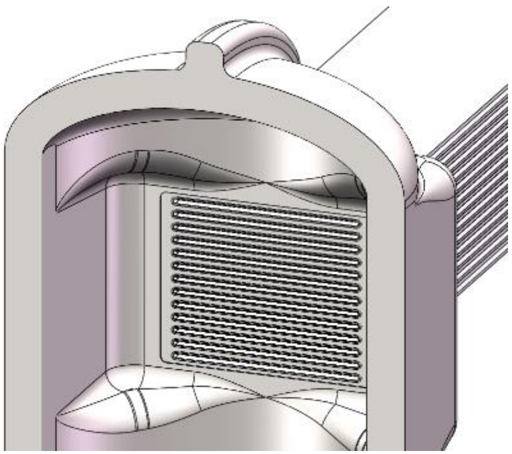


Plate-Matrix Heat Exchanger Manifolds

- Multiple unit-cells are attached to each other at the high-pressure manifolds



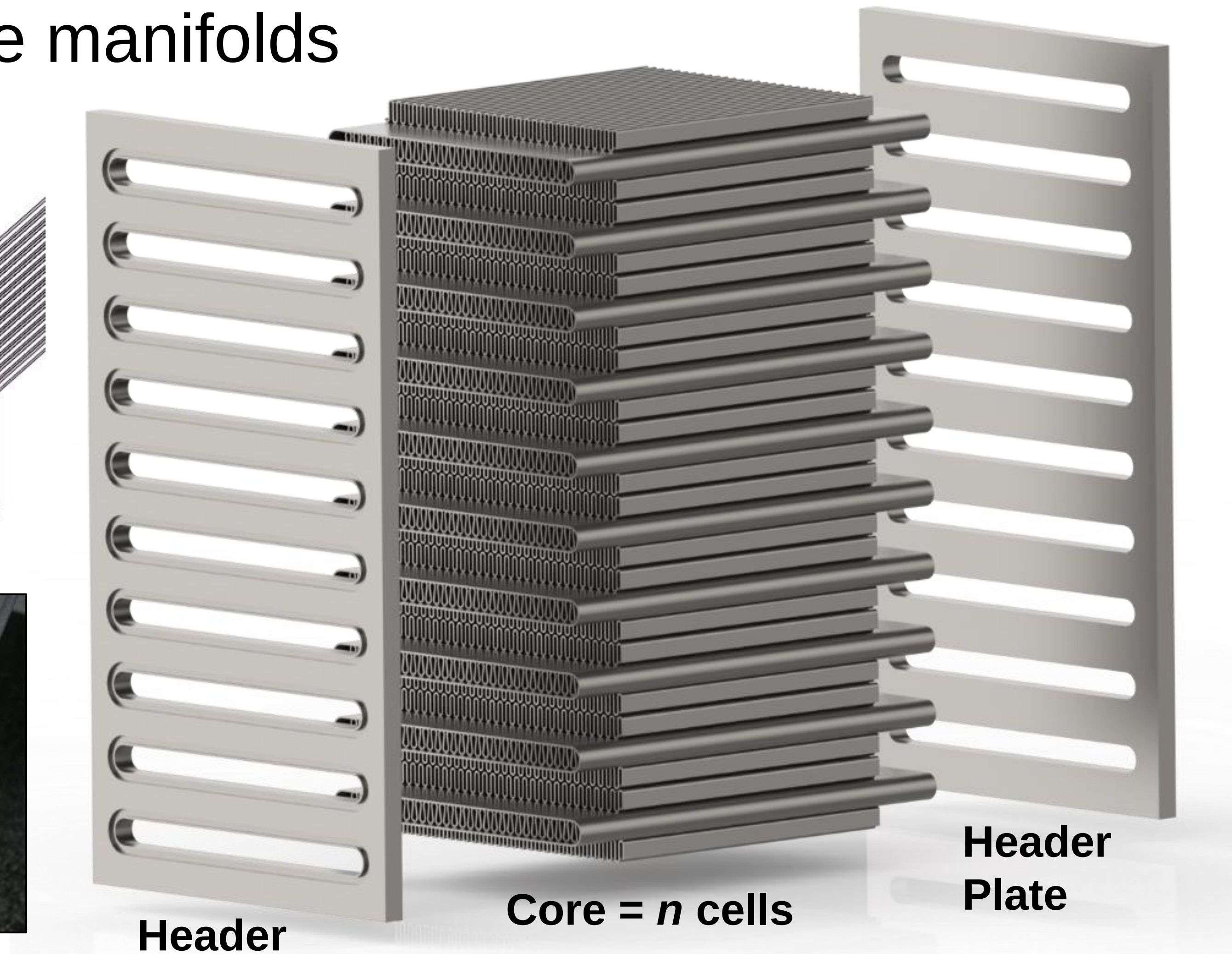
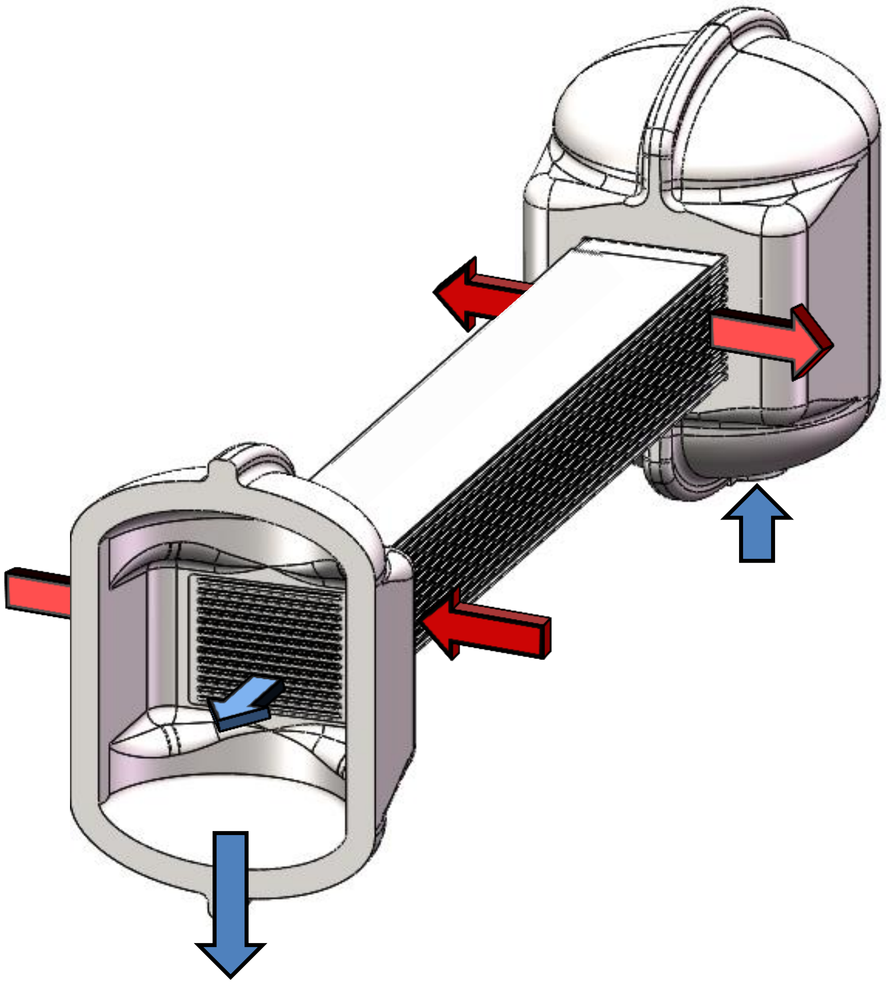


Plate-Matrix Heat Exchangers

Plate-Matrix Heat Exchanger Cores

- Multiple unit-cells are attached to each other at the high-pressure manifolds



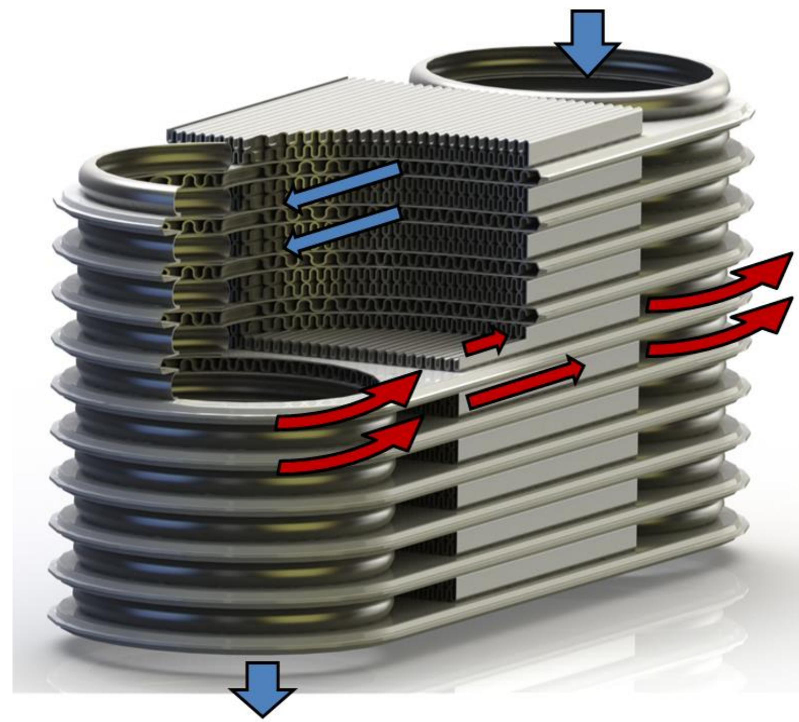
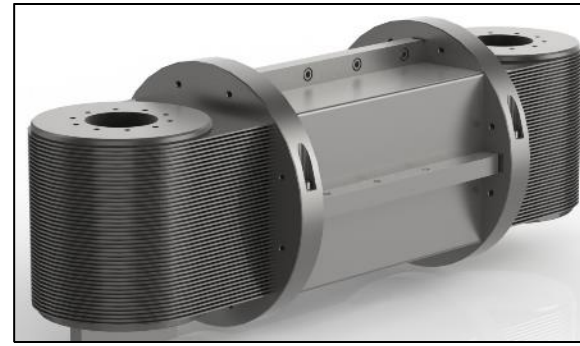


Plate-Matrix Heat Exchangers

Pressure Vessel Packaging

- Standard configurations mount modular cores in standard ASME-stamped pressure vessels and/or pipes
  - Compact high-performance surfaces enable minimal volume solutions
- Alternative high-pressure packaging designs may require ASME qualification



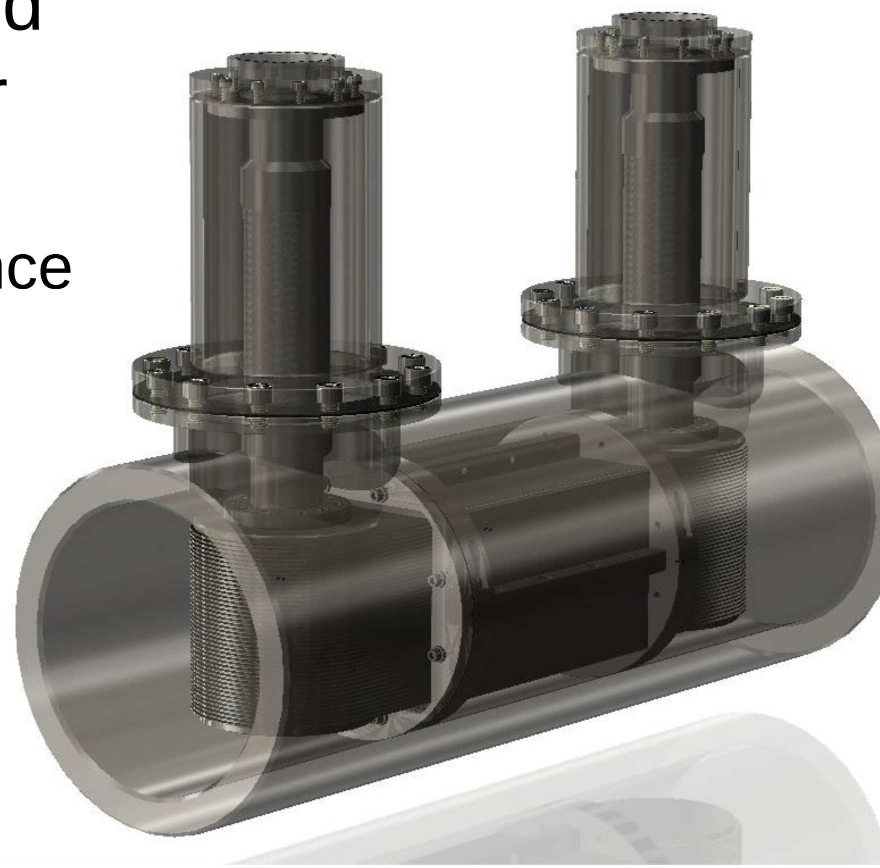
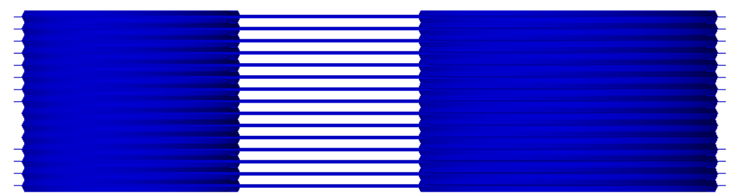


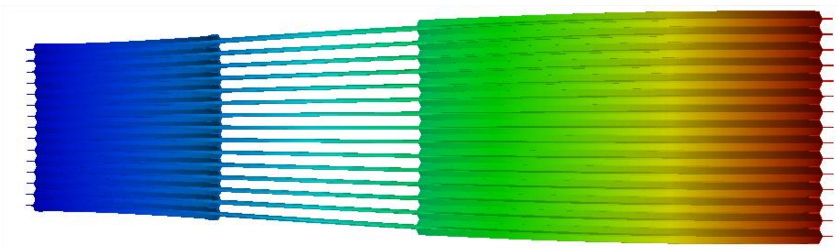
Plate-Matrix Heat Exchangers

Thermo-Mechanical Strain Tolerance

- Non-monolithic construction provides thermo-mechanical strain tolerance
  - Each unit cell represents a unique slip plane within the assembly
  - The associated low mechanical stiffness can accommodate temperature differences without inducing stresses on the assembly



Cold (Isothermal)



Hot

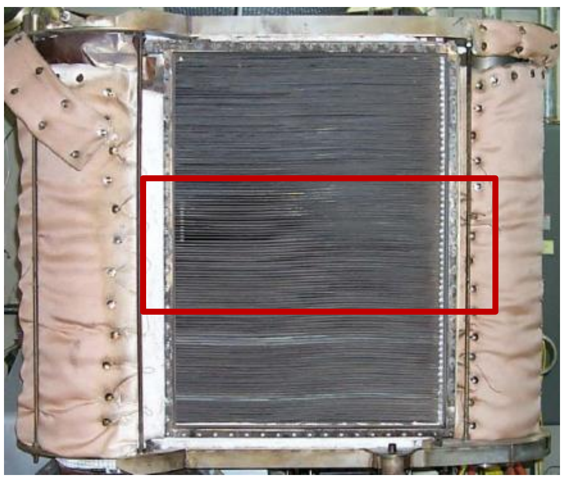


Plate-Matrix Heat Exchangers

Heat Exchanger Mechanical Design and Validation for S-CO_2 Environments

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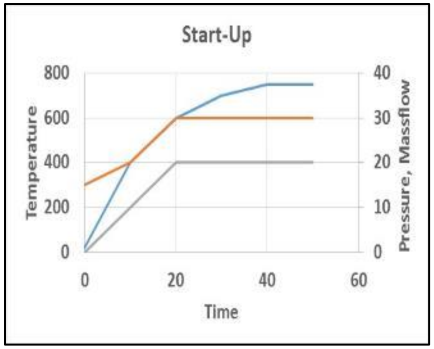
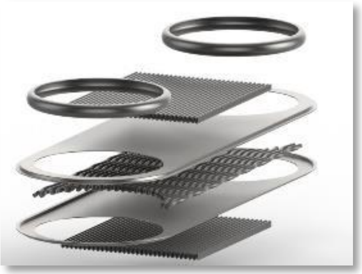
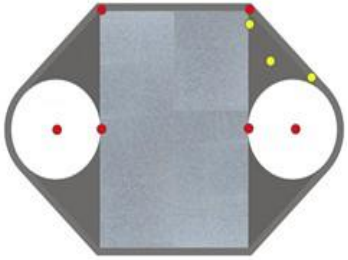
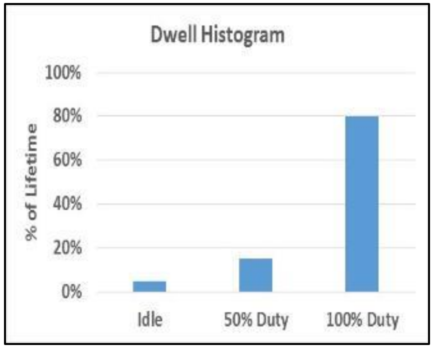
Design Methodology

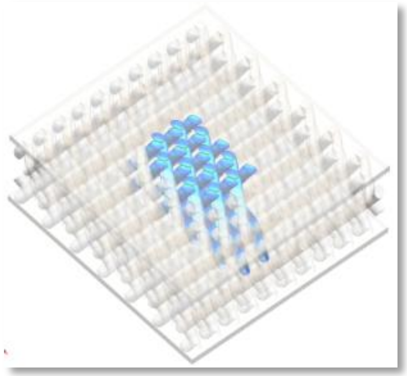
Mission Definition → **Mechanical Design and Simulations** → **Configured and Processed Materials Characterization** → **Thermal and Strain Validation & Endurance**







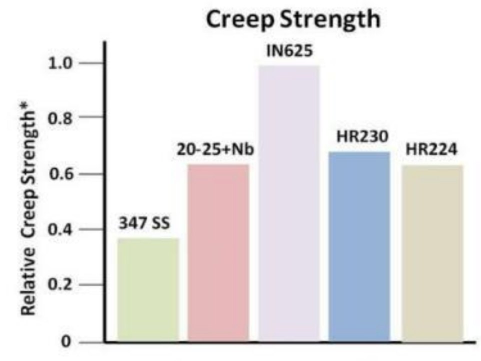




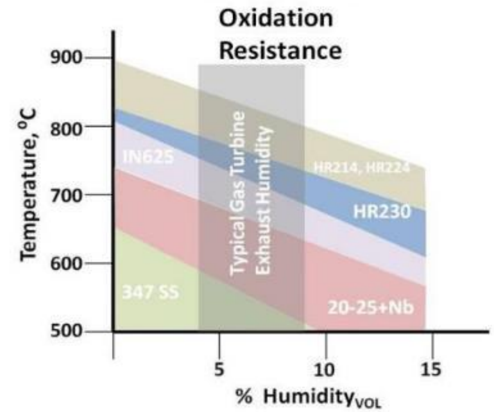
Requirements-to-Design Validation Method

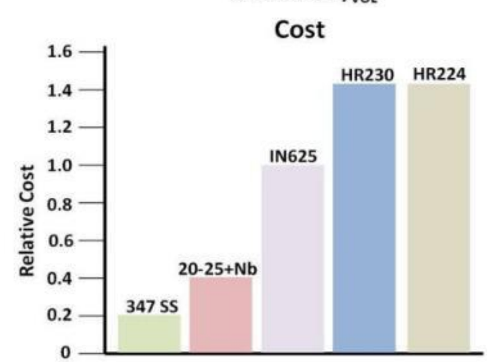
- Specify Requirements in terms of mission profiles
  - Including dwells and transient maneuvers
- Render thermal hydraulic design into mechanical design

- Initial analyses with substrate material properties:
  - temperature
  - stress/strain
  - durability
- Characterize as configured/processed materials as loaded in operation
  - creep
  - fatigue
- Validate/calibrate temperature and strain with actual heat exchanger cells
- Validate design with accelerated endurance testing
  - greater ΔT
  - greater pressure
  - design temperatures at control points.

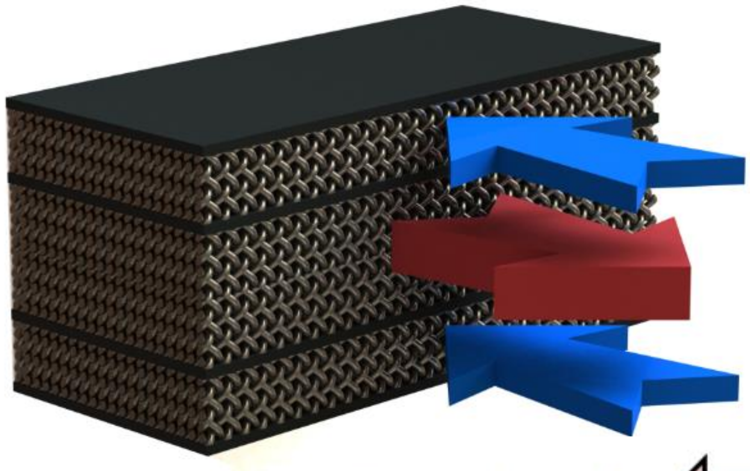


*10,000 Rupture at 650°C



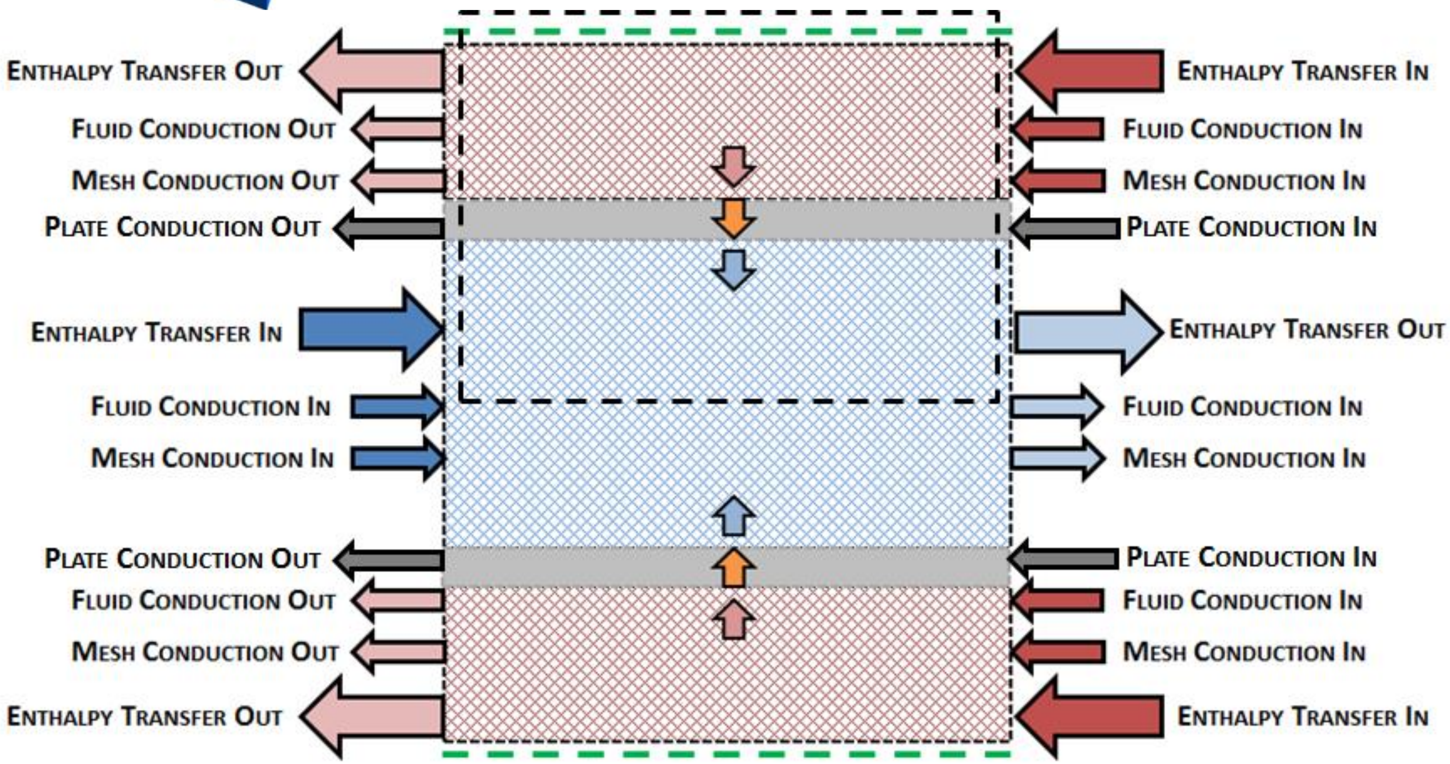


Heat Transfer Modeling



- Finite Difference modeling captures the non-intuitive nonlinear physical properties of supercritical fluids within heat exchangers (particularly in vicinity of critical point)

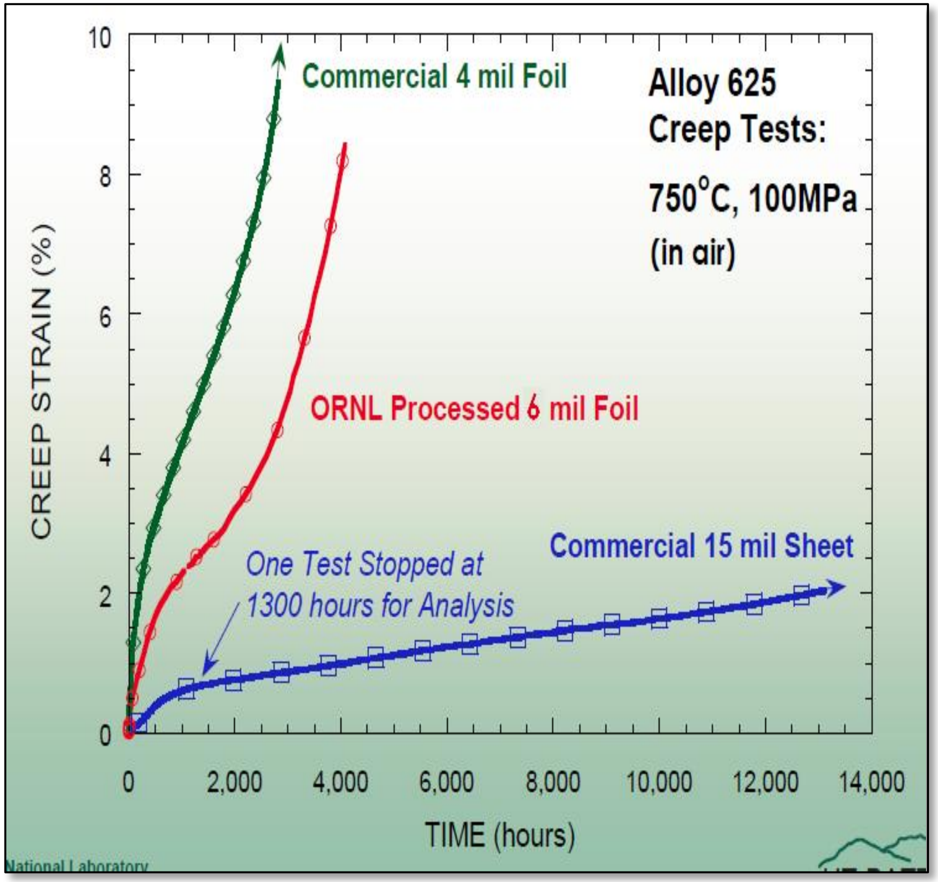
- Enthalpy change is used to calculate the heat gain (or loss) so as to capture the significant pressure dependence of the internal energy of the fluid
  - Δh(T,P) used instead of $\dot{m}c_p$(T)

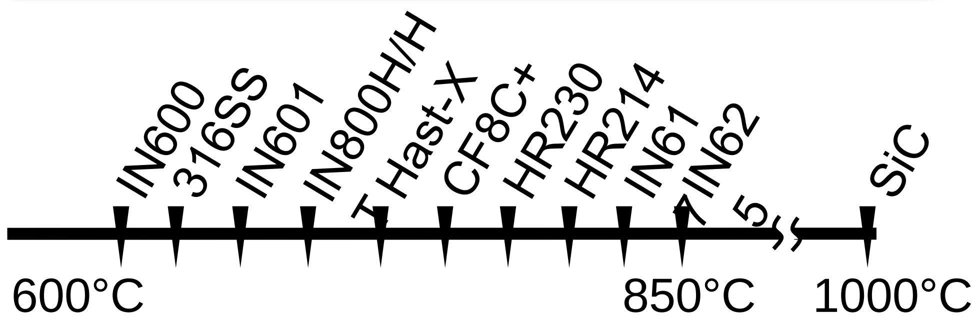


- Axial conduction losses – which may be significant in high-ε designs – are captured for both the parent material and the heat transfer enhancing structures

Creep Considerations

- High solidity structures – thick-walled tubes, dense extended surfaces.
- Ni-Cr alloys with precipitates in grain boundaries
- Choices: Alloy 625, Alloy 617, Alloy 718, Alloy 230, HR214™, HR224™
- Be careful of thickness. Sheet properties may not represent foil. (Grain size vs. thickness?)

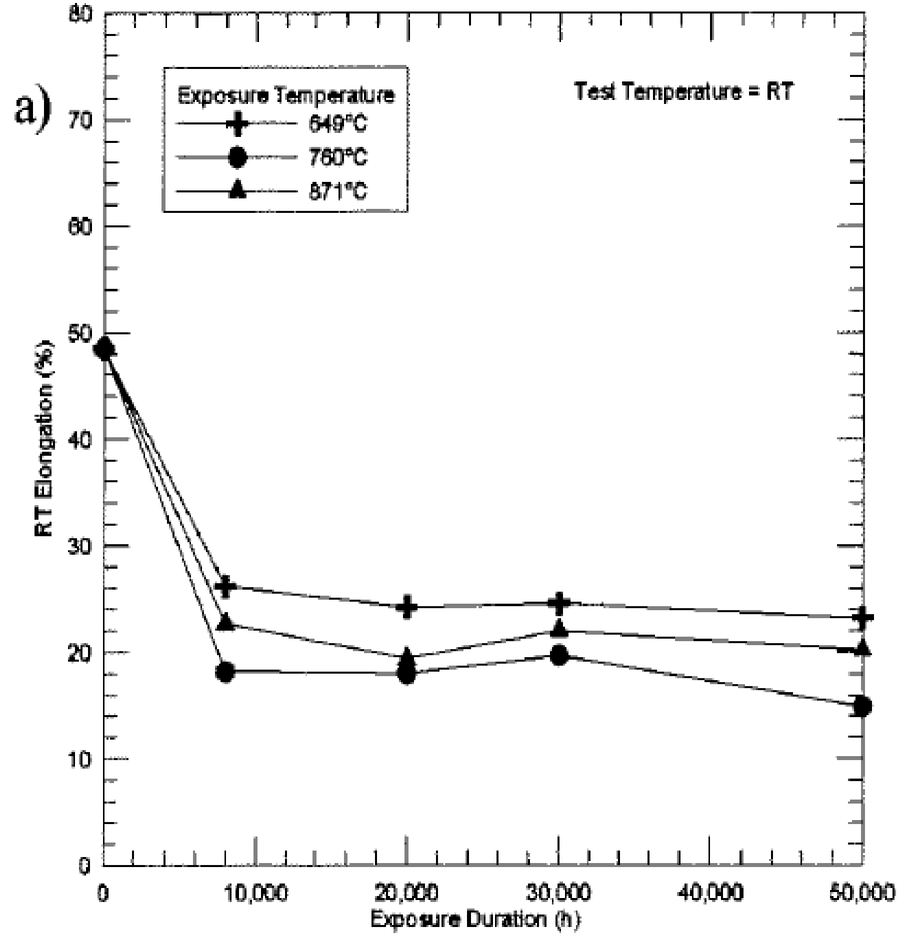




For 10,000 hour creep life at 70 MPa

Fatigue Considerations

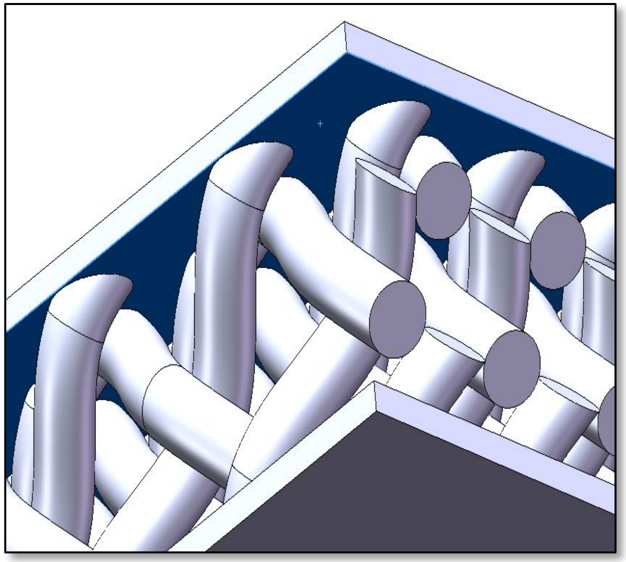
- Highly design dependent gradient selection for ΔT
- Structural compliance
  - Bigger is NOT stronger!
- Thick-thin avoidance
- Stress in weld-heat affected zones.
- Ductility – as processed, after aging

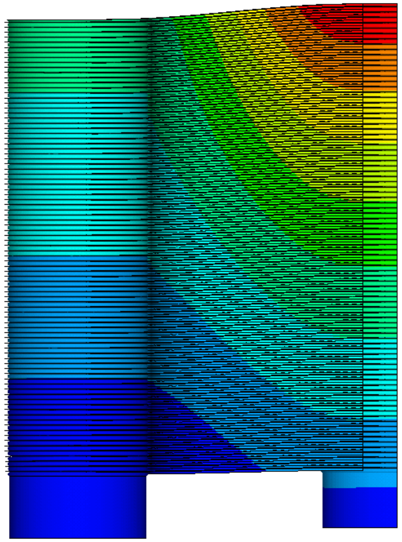


HR120 elongation with exposure at 649, 760 and 871ºC. Source: Pike & Srivastava Haynes Int’l

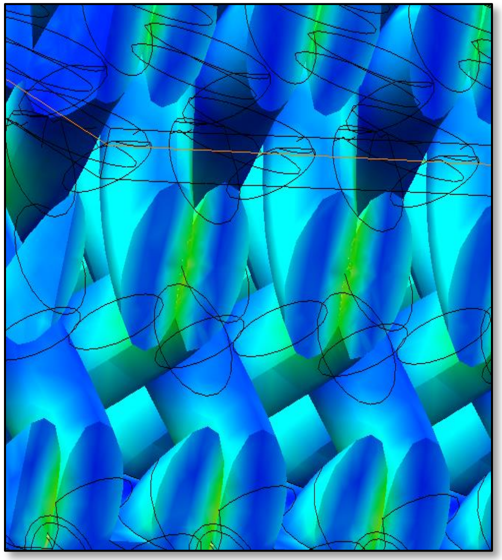
Simulations

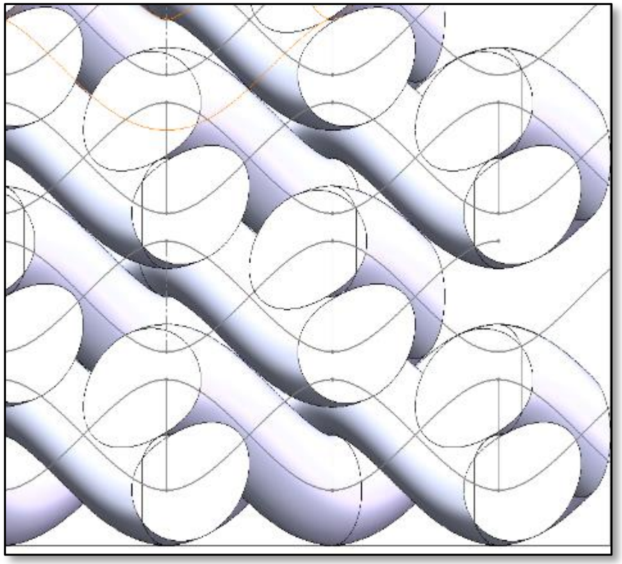
- Conduct thermal and structural FEA to determine temperature, stress, and strain
- Identify ‘control points; - details where damage may accumulate
- Perform initial life analyses to quantify creep, and fatigue





Core strain analysis

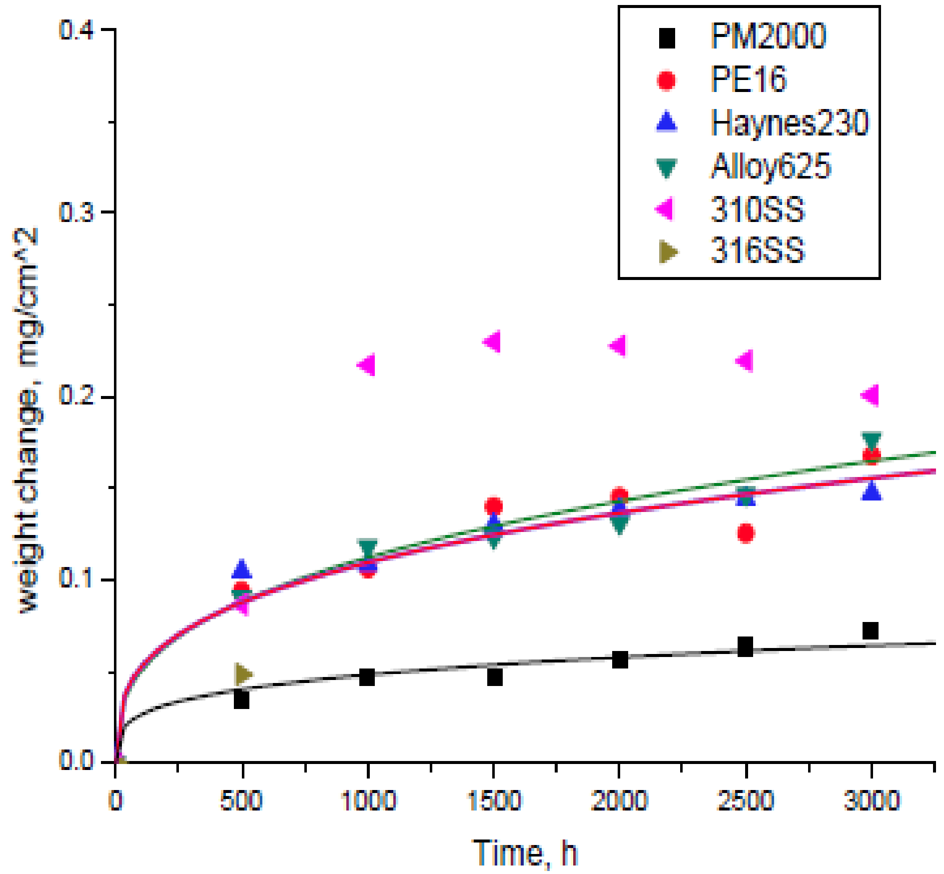




Wire-mesh analysis for creep and pressure-fatigue simulation.

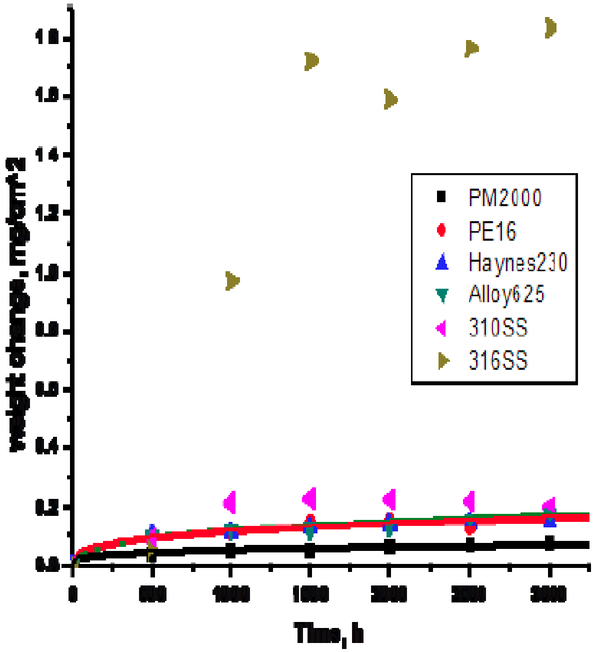
Corrosion Considerations

- Oxidation
- Scale evaporation with high temperature and/or humidity addition
- Ni and Cr basic protection
- Rare-earth additions to stabilize scale
- Aluminum addition for very low volatile Al_2O_3 scale over chromia
- >20% Cr is key to oxidation resistance at 650^0C according to Sridharan et al.

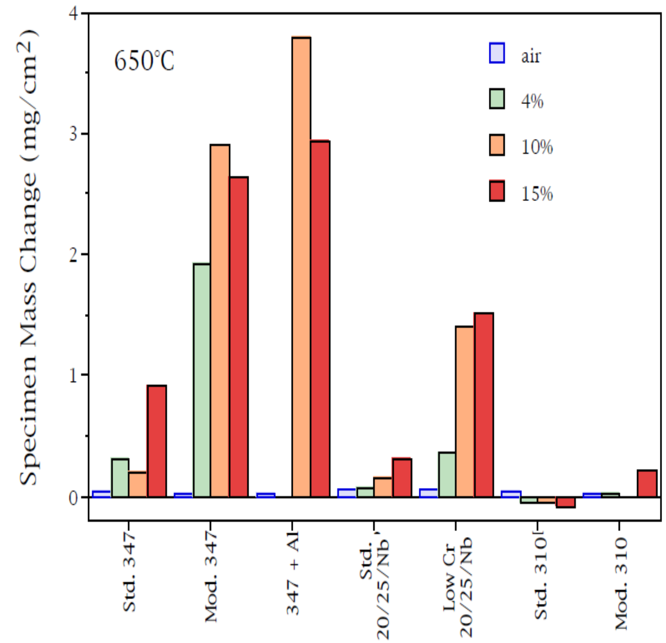


Source: Sridharan, Anderson, et al - University of Wisconsin, sCO_2 Power Cycle Symposium, Boulder, CO 2011

Type 310SS 650°C Oxidation sCO_2 vs. Air



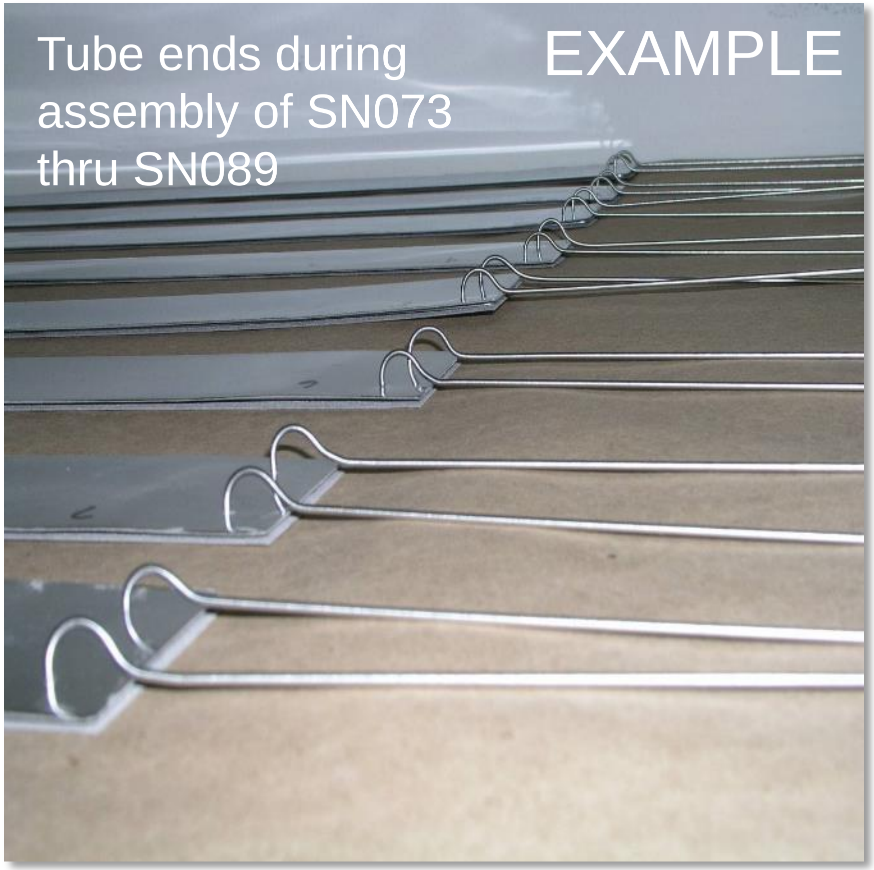
Sridharan, Anderson, University of Wisconsin, et al, sCO_2 Power Cycle Symposium, Boulder, CO 2011



Pint (ORNL) and Rakowski (Allegheny Ludlum), Effect of Water Vapor on the Oxidation Resistance of Stainless Steel

1. 0.25 mg/cm^2 gain in sCO_2 vs. 0.045 in laboratory air after 1,000 hours
2. Aluminum addition with addition of humidity?

Testing As Configured/Processed Material



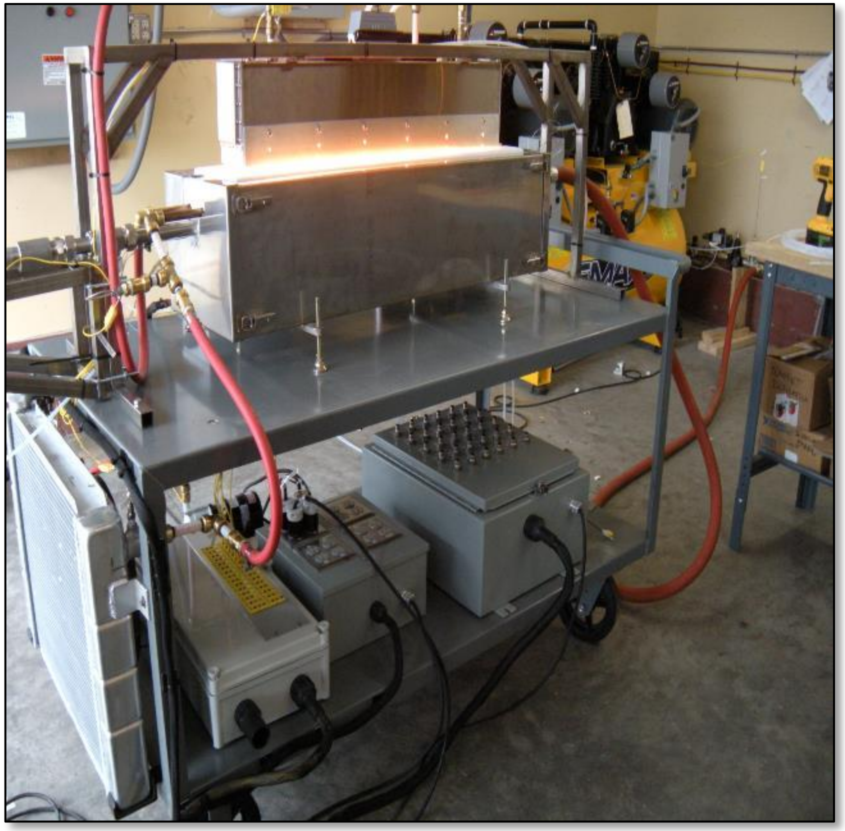
This final batch of heat exchanger cells were of high quality, leak tight and suitable for creep tests

- Example: If pressure is the steady load dominating creep or fatigue, pressure is used in characterization
  - Includes all configuration and processing effects
  - Avoids interpretation of ‘like’ data and loading.

- sCO_2 pressurization for possible corrosion interaction

Thermo-Mechanical Fatigue Testing

- If high radiant flux loads produce damage, material is characterized accordingly
- Burner rig or furnace is appropriate for characterization under cyclic convective loading



Radiant (High Flux) Test Rig

High Temperature Furnace

Hydraulic Design with Supercritical Fluids

Shaun Sullivan

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Hydraulic Design – Supercritical Fluids

$$\Delta P_{total} = \Delta P_{inlet\,manifold} + \Delta P_{entrance} + \Delta P_{internal\,flow} + \Delta P_{exit} + \Delta P_{outlet\,manifold}$$

$$\Delta P_{internal\,flow} = f\,\frac{L}{D_h}\,\frac{1}{2}\,\rho\,V^2$$

$$f = f\,(e, D_h, V, \rho, \mu)$$

$$V = \frac{\dot{m}}{\rho\,A_f}$$

Geometric parameters

Fluid properties and mass flow

Hydraulic Design – Modeling Considerations

- The non-linear behavior of supercritical fluids – particularly near the critical point – makes endpoint calculations risky
  - Finite difference or integrated methods necessary to capture non-intuitive property behavior

- The strong property dependence on pressure makes sensible heat calculations risky
  - Use enthalpy change Δh(T,P) to calculate energy gain or loss, instead of $\dot{m}c_p$

Hydraulic Design – Correlations and Calculations

- Internal Flow $\Delta P = f \frac{L}{D_h} \frac{1}{2} \rho V^2$
  - *f* may be derived from:
    - Moody Chart
    - Kays and London (NB: friction factor *f* = 4*Fanning Friction Factor)
    - empirical correlation

- Porous Media $\Delta P = \frac{Q\mu L}{kA_f}$

Q = volumetric flow rate
κ = permeability

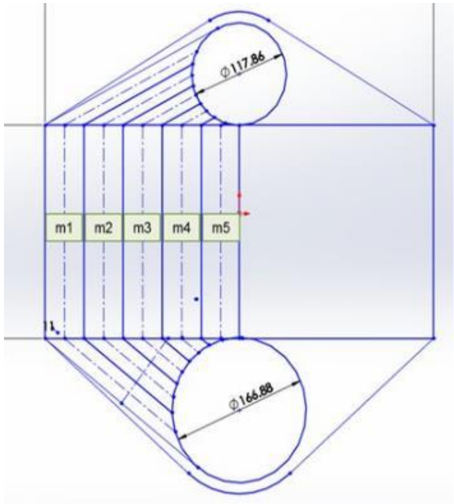
- Wire-Mesh $f = \frac{2\rho \Delta P}{G^2 \beta t}\left(\frac{1-\varepsilon}{\varepsilon}\right)^{0.4}$

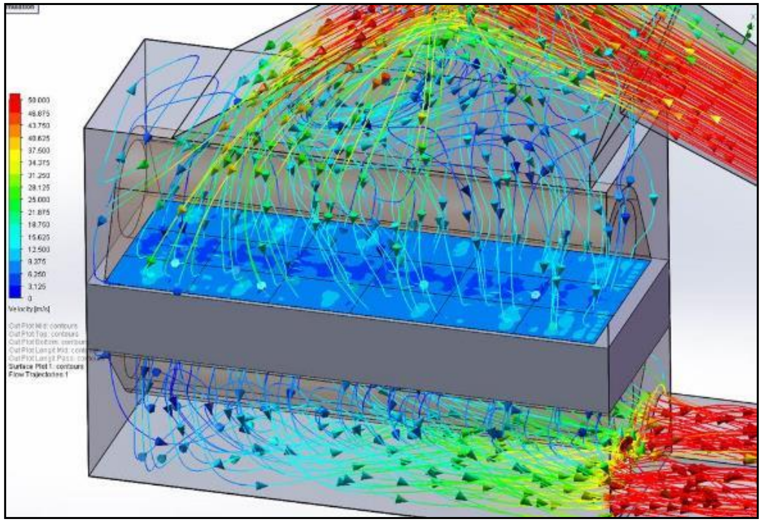
G = internal mass velocity
β = surface area/volume
ε = porosity

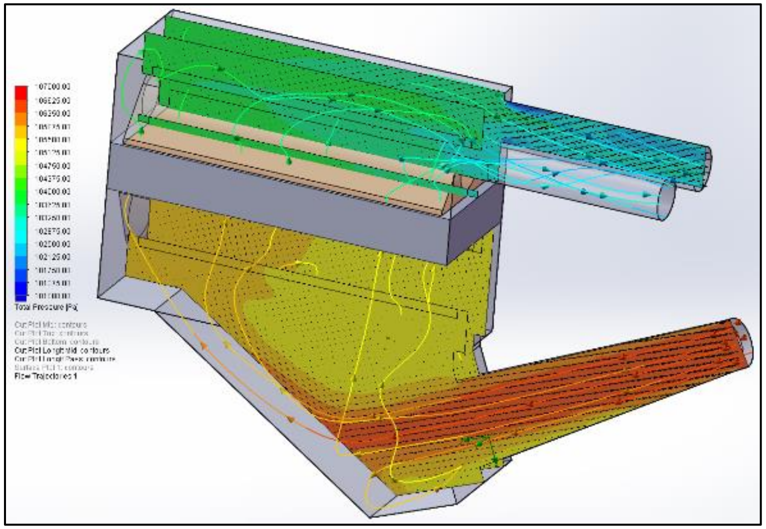
- CFD

Hydraulic Design – Flow Distribution

- Headered or unheadered, the net pressure loss along any given flowpath will be the same
  - Uniform flow may be imposed by tailoring the area ratio to account for differences in density and velocity profile
  - Headered channels may impose unequal flow resistances, resulting in unequal passage flows
    - Performance must be assessed on a mass-averaged basis







sCO2 Heat Transfer

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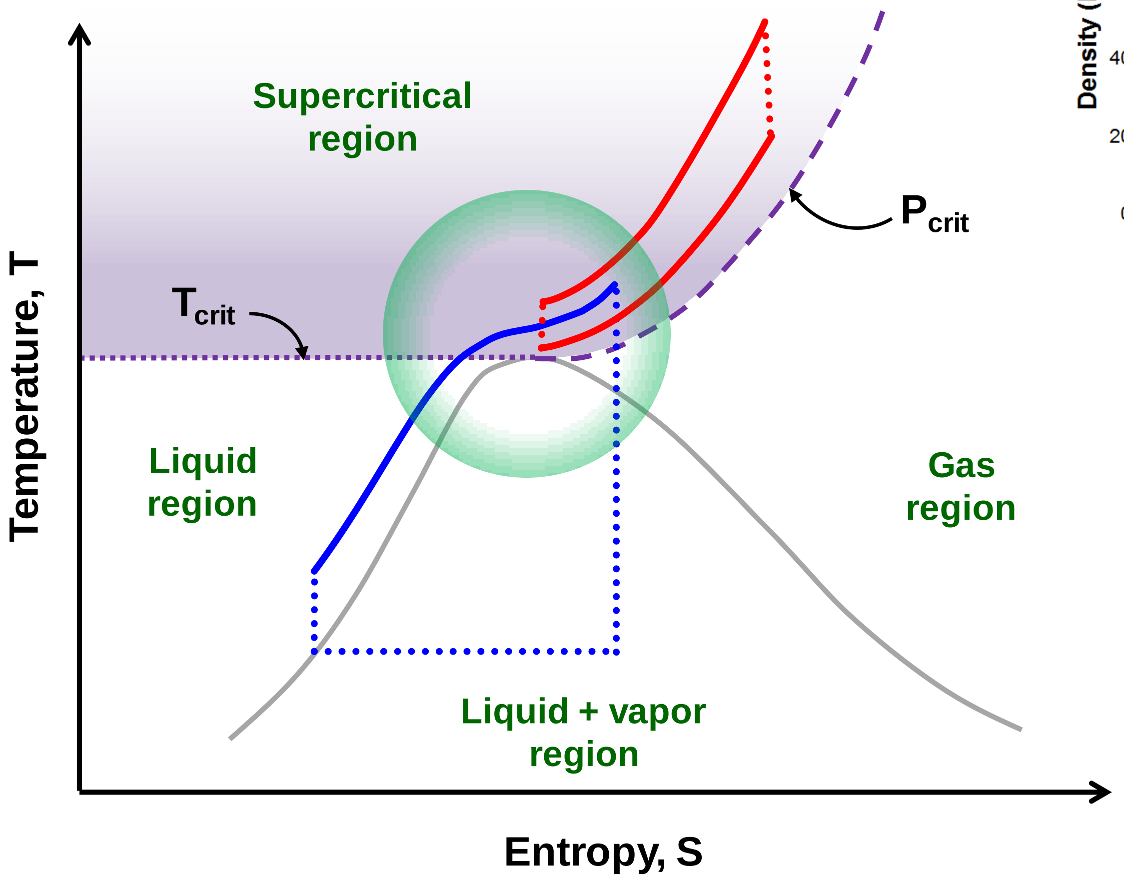
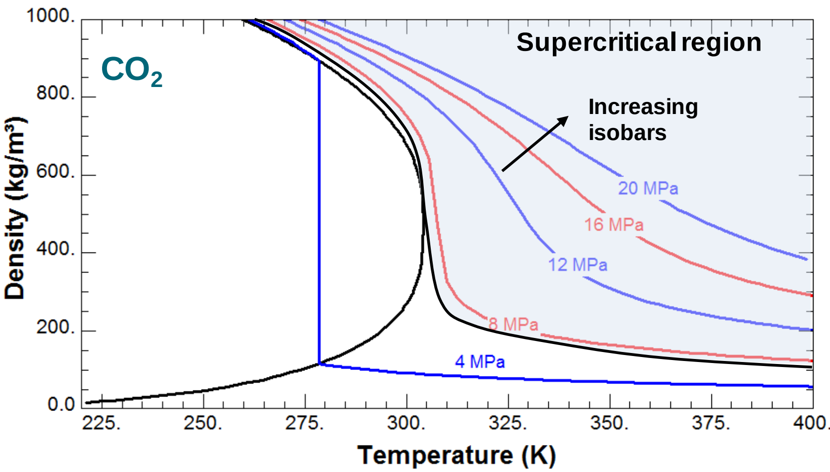
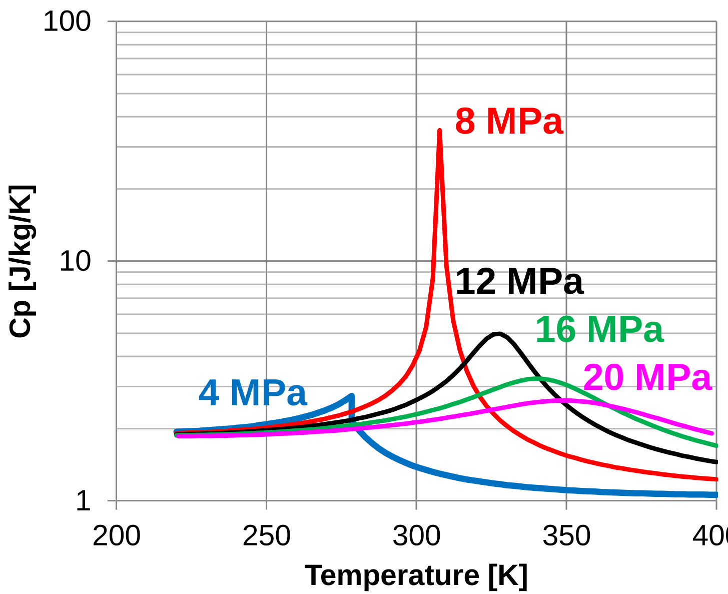
Fluid property effects near the critical point allow for less approximations in heat exchanger sizing







Typical approximations for heat exchanger sizing are not valid for near-critical sCO2

General equation | **Typical approximation**

Heat transfer

$$Q = w\left(i_{c,o} - i_{c,i}\right)$$

$$Q = wC_p(T_o - T_i)$$

$$Q = UA\Delta T_{LM}$$

$$Q = \varepsilon C_{min}\left(T_{h,i} - T_{c,i}\right)$$

$$\varepsilon = f(NTU, C_{min})$$

$$C_{min} = min\left[\left(wC_p\right)_c, \left(wC_p\right)_h\right]$$

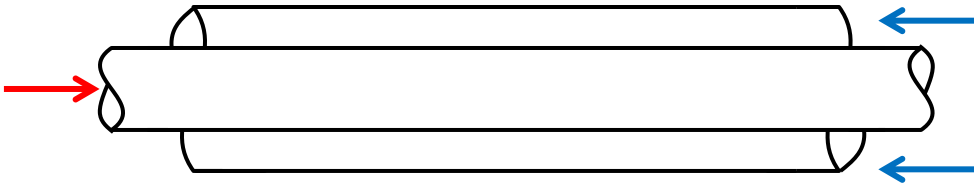
Overall heat transfer coefficient

$$\frac{1}{UA} = \frac{1}{(hA)_i} + \frac{ln\left({}^{D_o}/_{D_i}\right)}{2\pi kL} + \frac{1}{(hA)_o}$$

$$h = f(Nu) = CRe^x Pr^y$$

Nomenclature
i = enthalpy
h = heat transfer coefficient
w = mass flow rate

Subscripts
c = cold stream
h = hot stream
i = inlet
o = outlet

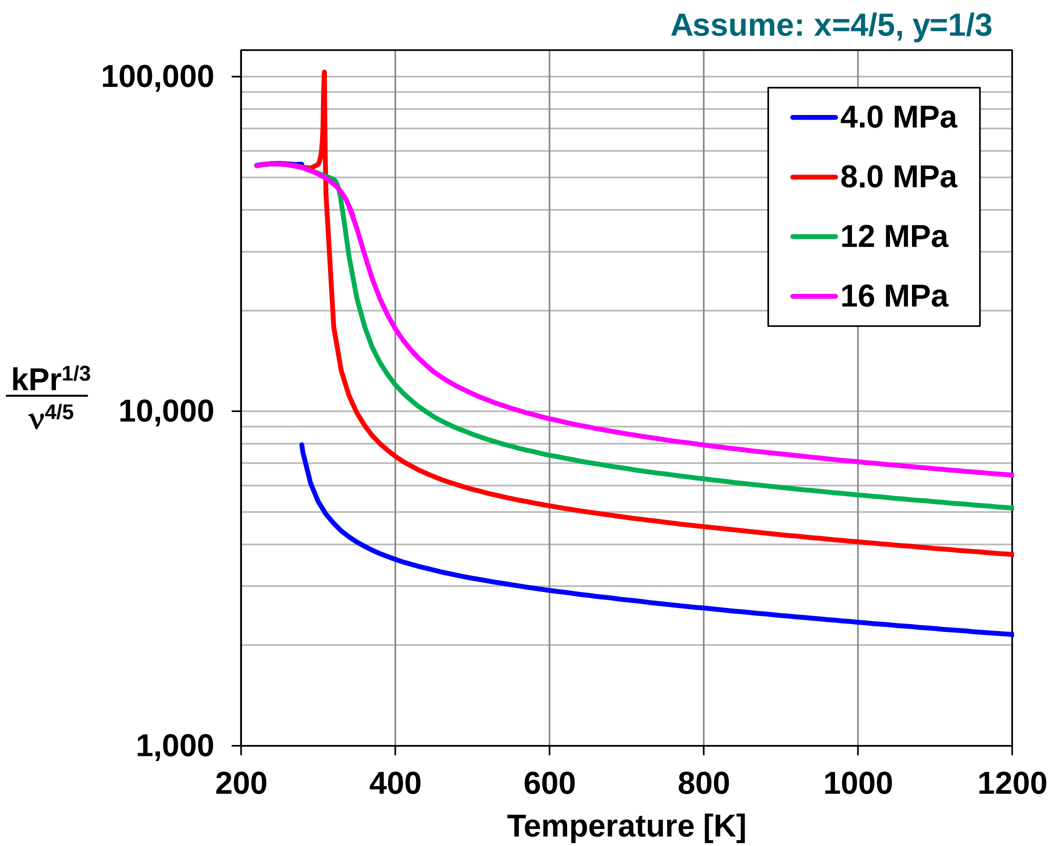


Typical correlations based on average fluid properties are not applicable near the critical point

$$Q = hA\Delta T$$

$$h = f\left\{\frac{k}{L}Re^{x}Pr^{y}\right\}$$

$$h = f\left\{\frac{(VD)^{x}}{L}\left(\frac{k}{\nu^{x}}(Pr)^{y}\right)\right\}$$

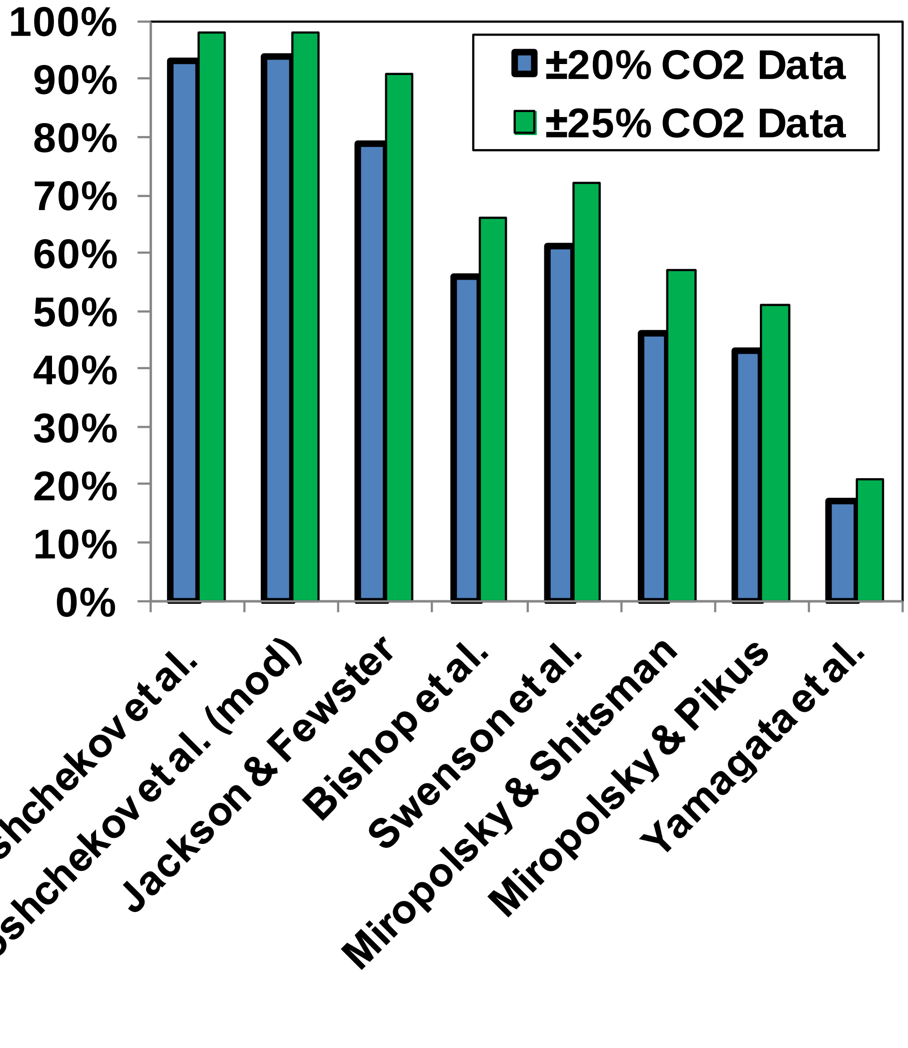


Dittus-Boelter type correlations with property variation are valid when buoyancy is negligible

Test data screened for buoyancy

$$Nu_b = CRe_b^{m1} Pr_b^{m2} \left(\frac{\rho_w}{\rho_b}\right)^{m3} \left(\frac{\overline{C}_p}{C_{p_b}}\right)^{m4}$$

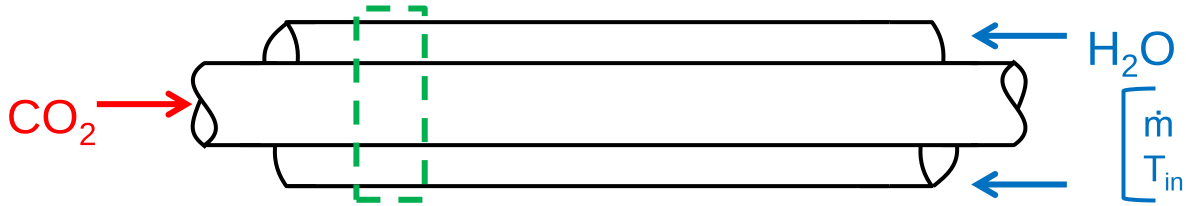
b = bulk
w = wall



[Values from Jackson 2013]

An example counter-flow heat exchanger is used to illustrate calculation methods

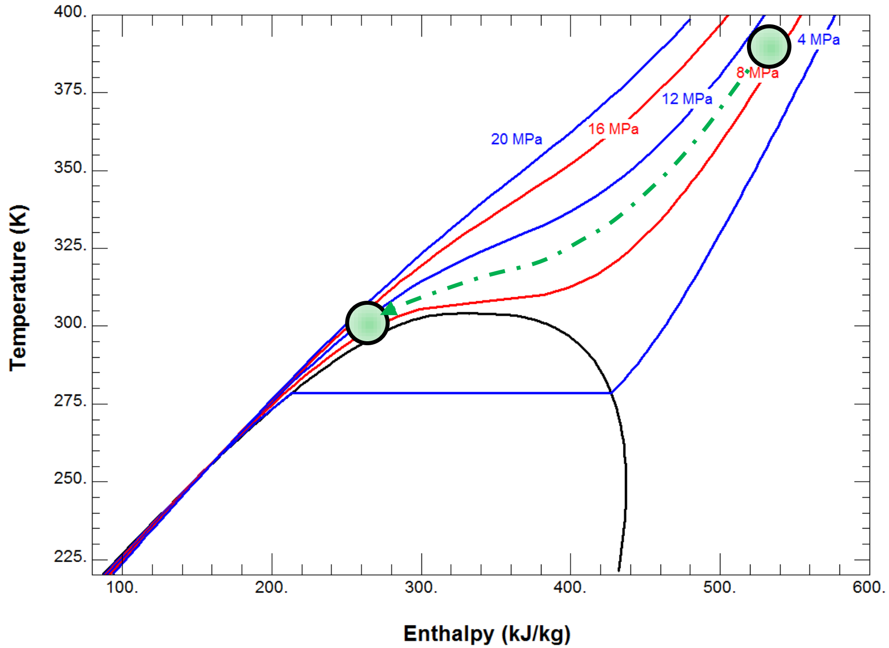
$\dot{m} = 0.01963$ kg/s
$T_{in} = 394.2$ K

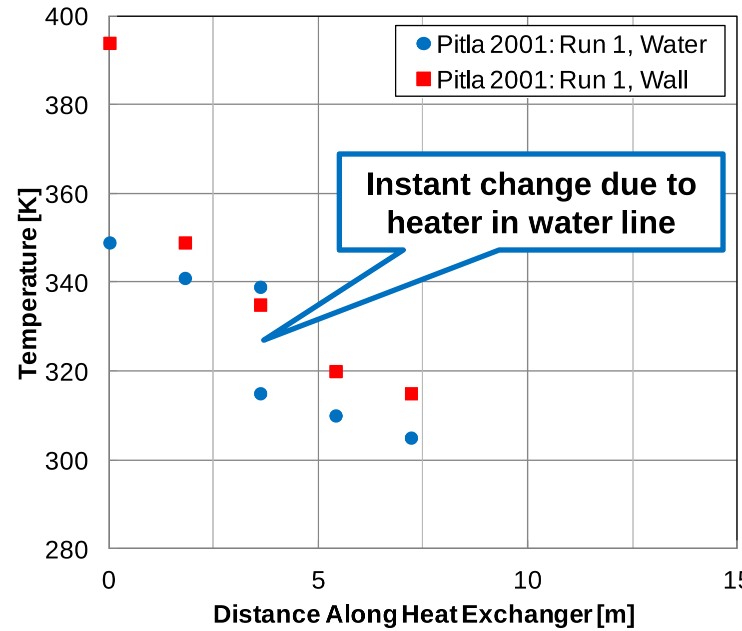


$\dot{m} = 0.04011$ kg/s
$T_{in} = 293.8$ K

Assumptions:
- one-dimensional
- steady-state
- frictionless flow

Validation of the method is based on test data from [Pitla 2001]





Conventional heat exchanger calculation methods can be compared to a discretized enthalpy method

ε-NTU Method (average fluid properties):

$$NTU = \frac{UA}{C_{min}} \qquad \varepsilon = \frac{1 - \exp\left[-NTU(1 - C_r)\right]}{1 - C_r \exp\left[-NTU(1 - C_r)\right]}$$

$$C = \dot{m} C_p$$

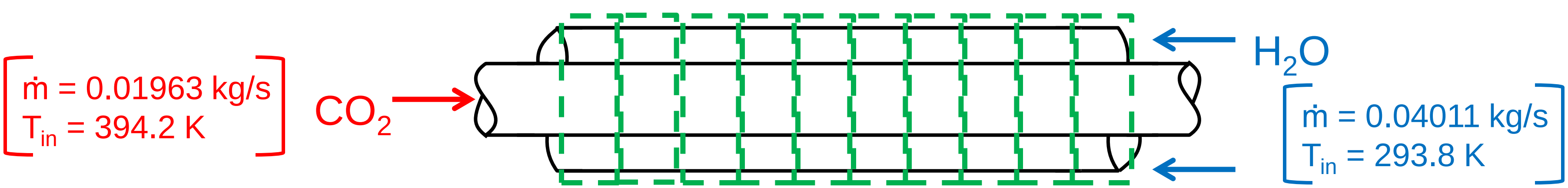
ε = Q/Qmax = 98.6%

A 1st order, backward difference discretization of the energy equation:

$$\left(h_{c,n}\right)^{i} = \left(h_{c,n-1}\right)^{i-1} + \frac{UA}{\dot{m}_c}\left(T_{h,n} - T_{c,n}\right)^{i-1}$$

n = node
i = iteration
h = hot stream
c = cold stream

ε = Q/Qmax = 32.5%



The heat exchanger should be discretized to accurately account for fluid property variations

Discretized energy equaiton:

$$\left(h_{c,n}\right)^{i} = \left(h_{c,n-1}\right)^{i-1} + \frac{UA}{\dot{m}_c}\left(T_{h,n} - T_{c,n}\right)^{i-1}$$

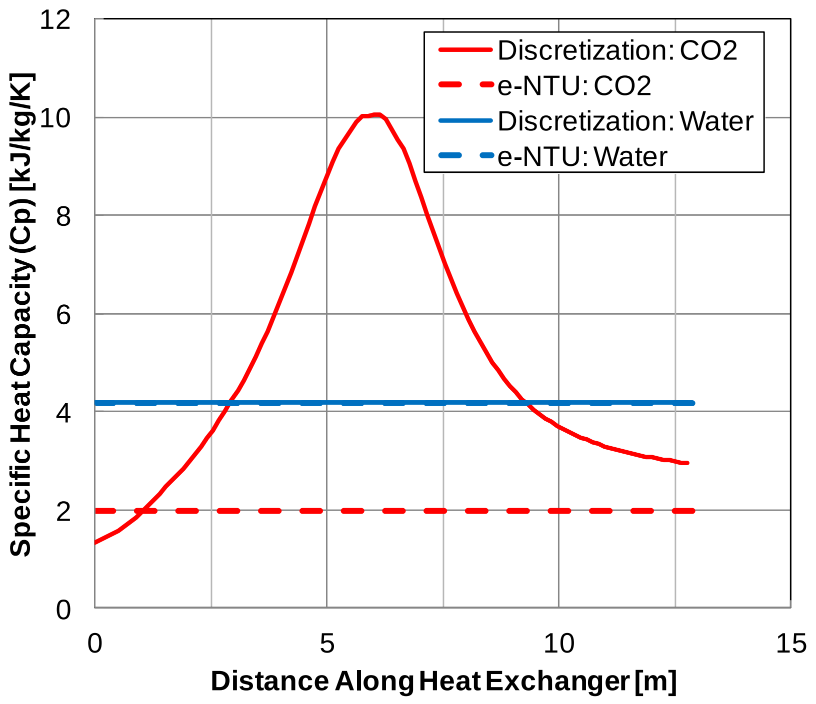
n = node
i = iteration
h = hot stream
c = cold stream

ε-NTU Method (average fluid properties):

$$NTU = \frac{UA}{C_{min}}$$

$$\varepsilon = \frac{1 - \exp\left[-NTU\left(1 - C_r\right)\right]}{1 - C_r \exp\left[-NTU\left(1 - C_r\right)\right]}$$

$$C = \dot{m} C_p$$



Note: The heater is removed from the calculation

Discretizing the heat exchanger accounts for property differences that affect fluid temperature

Discretized energy equaiton:

$$\left(h_{c,n}\right)^{i} = \left(h_{c,n-1}\right)^{i-1} + \frac{UA}{\dot{m}_c}\left(T_{h,n} - T_{c,n}\right)^{i-1}$$

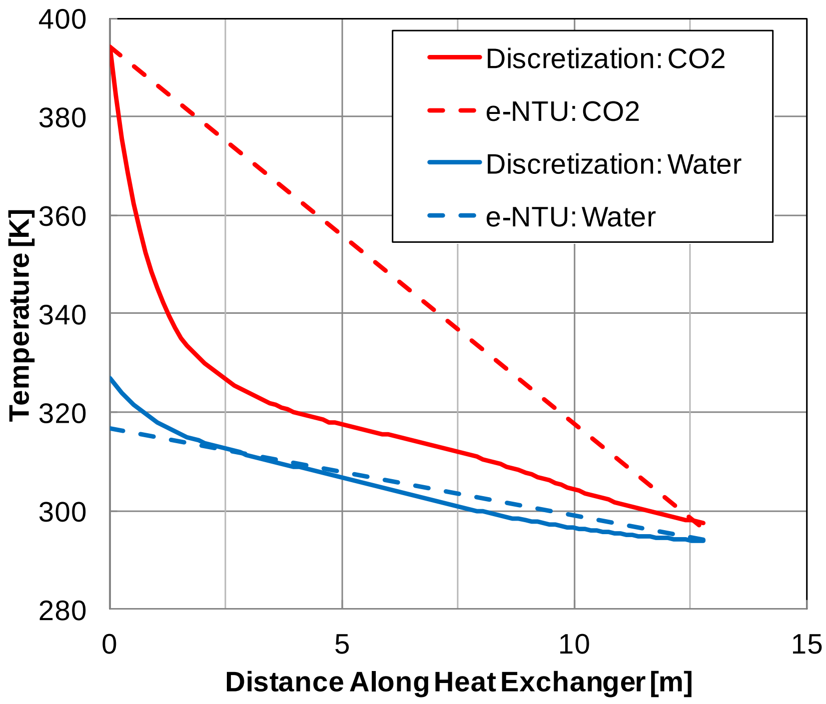
n = node
i = iteration
h = hot stream
c = cold stream

ε-NTU Method (average fluid properties):

$$NTU = \frac{UA}{C_{min}}$$

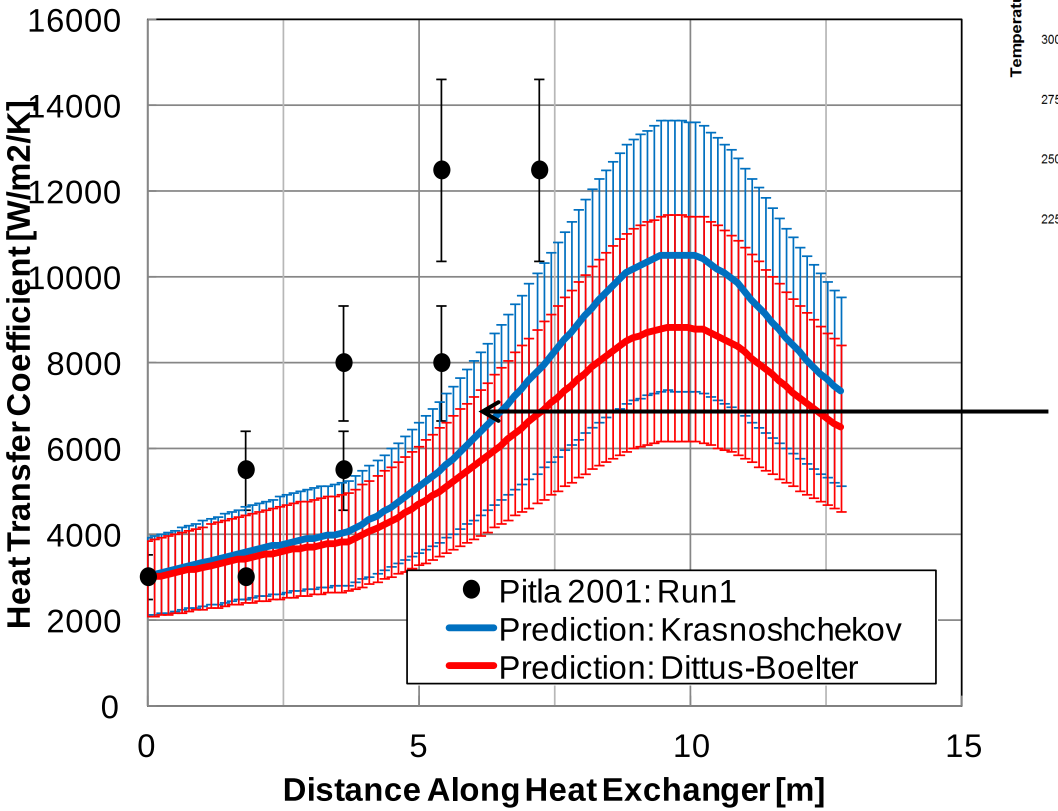
$$C = \dot{m}C_p$$

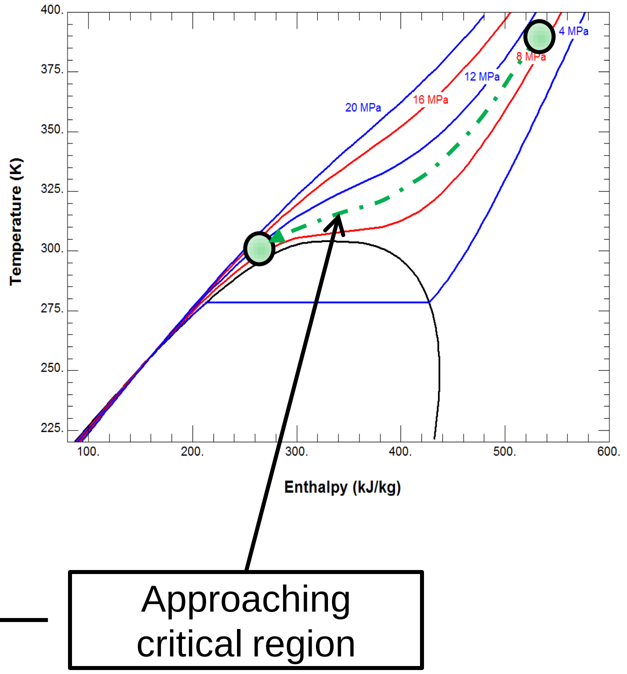
$$\varepsilon = \frac{1 - \exp\left[-NTU(1 - C_r)\right]}{1 - C_r \exp\left[-NTU(1 - C_r)\right]}$$



Note: The heater is removed from the calculation

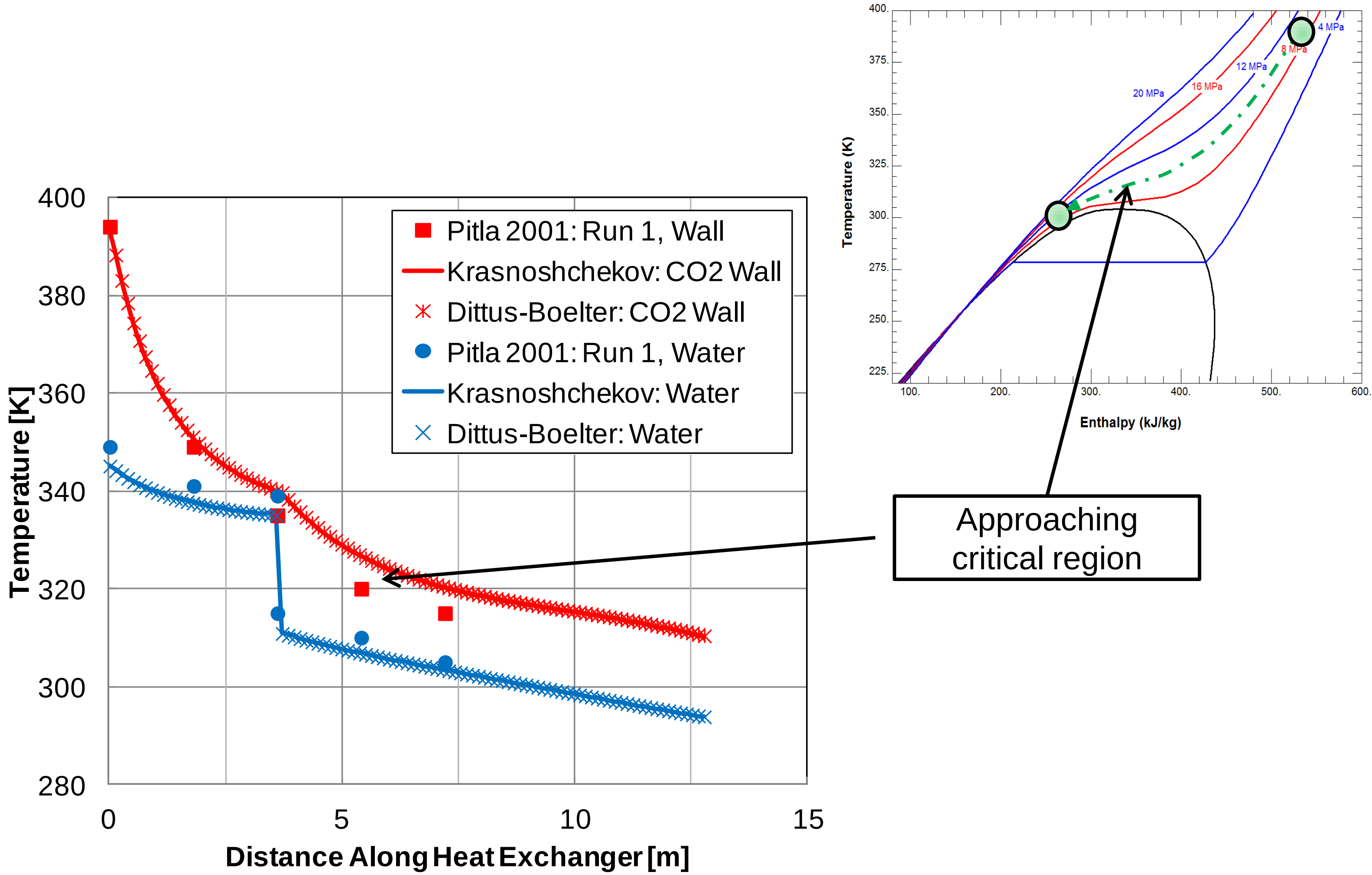
Property changes in the critical region cause heat transfer variations between correlations





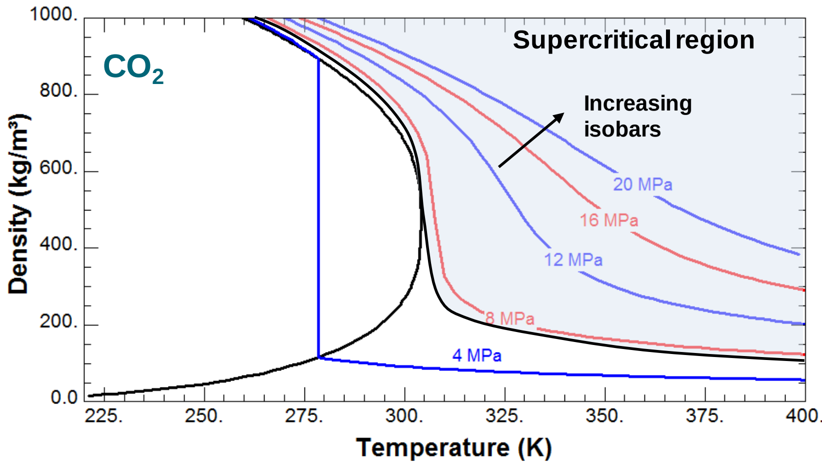
Note: 30% uncertainty bars applied to correlations

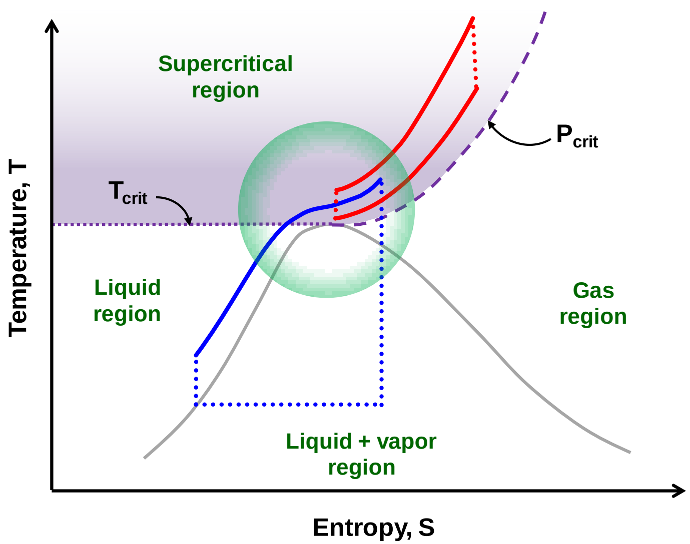
Heat transfer variations from correlations can be negligible on temperature prediction



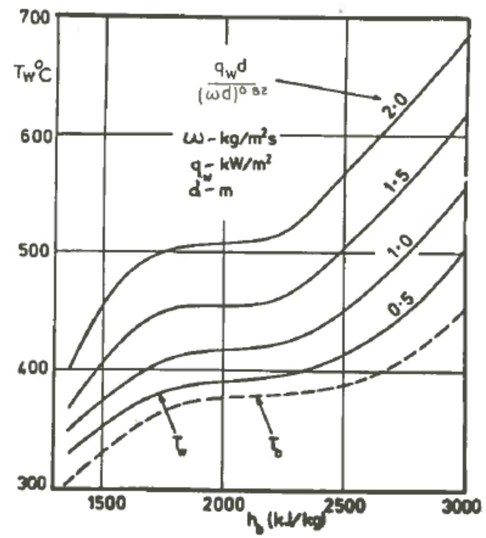
Note: 30% uncertainty bars applied to correlations

Depending on flow conditions, buoyancy effects can influence heat transfer coefficients



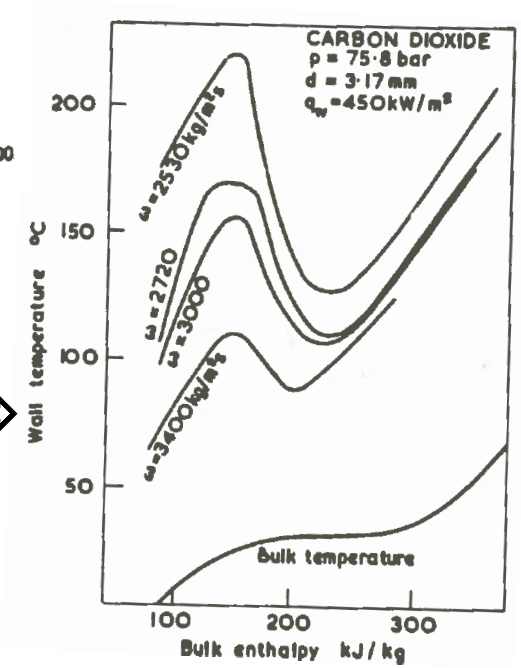


Correlations do not predict deterioration due to buoyancy



Krasnoshchekov correlation

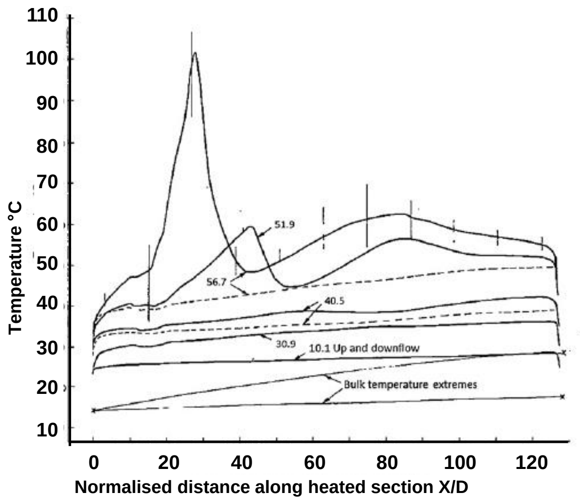
Experimental data (Shiralkar & Griffith 1970)



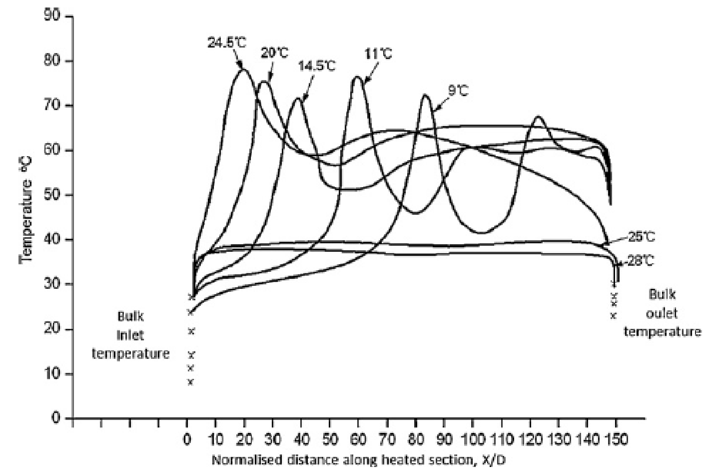
[Figures from Jackson 1979b]

S-CO2 flow in vertical tubes indicates local heat transfer is a strong function of fluid properties

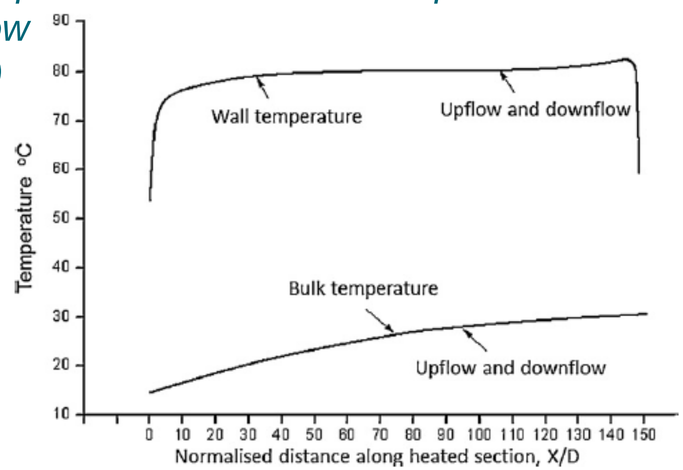
Flow direction and heat flux affect wall temperature distribution



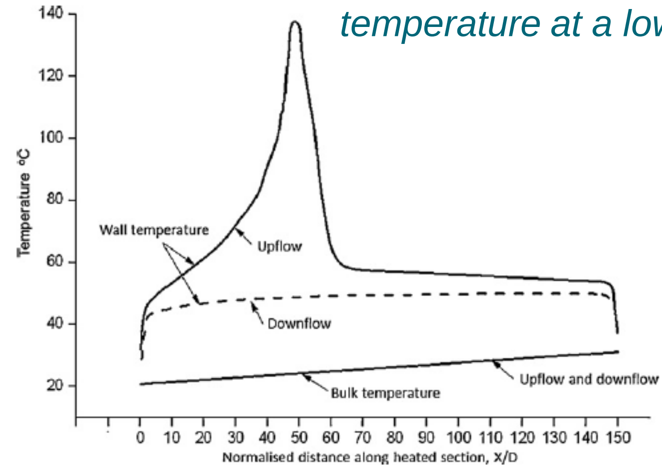
Inlet fluid temperature affects the axial location of the wall temperature peak



Up/down flow produces similar wall temperatures at high mass flow ($Re \sim 2.5x10^5$)

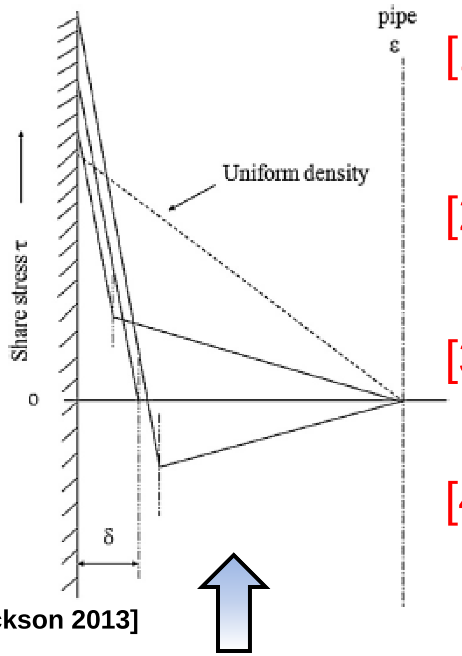


The upward flow direction produces a peak wall temperature at a low mass flow ($Re \sim 4x10^4$)



Figures from [Jackson 2013]

Heat transfer deteriorates and recovers due to buoyancy effects near the wall



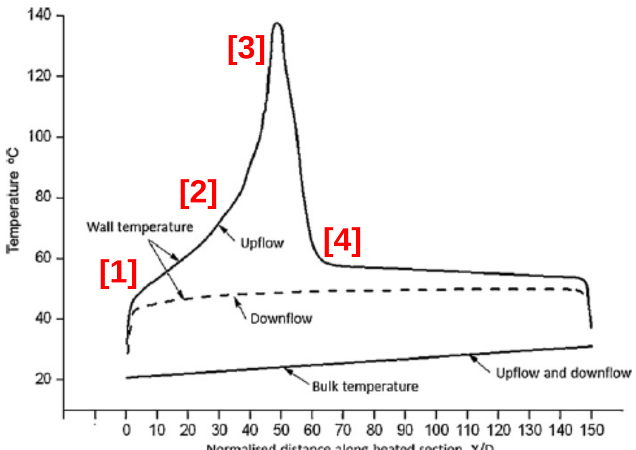
[Jackson 2013]

[1] Wall heating reduces the fluid density near the wall to cause buoyant flow near the wall

[2] Growth of the buoyant wall layer causes the wall shear stress to decrease

[3] Turbulence production reduces as the shear stress decreases – causing a 'laminarization' of the flow

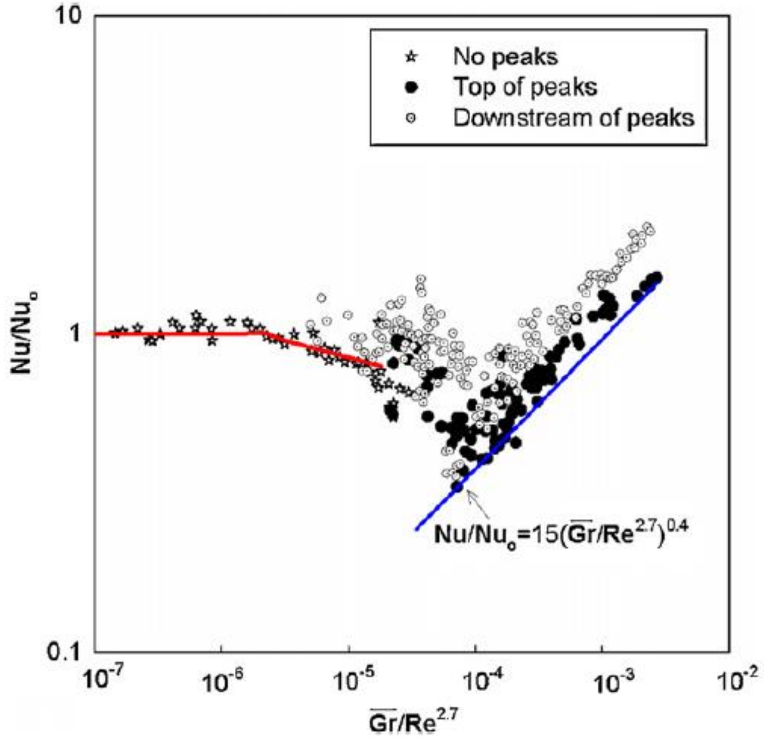
[4] Turbulence production is restored when the buoyant layer is thick enough to exert an upward force on the core flow



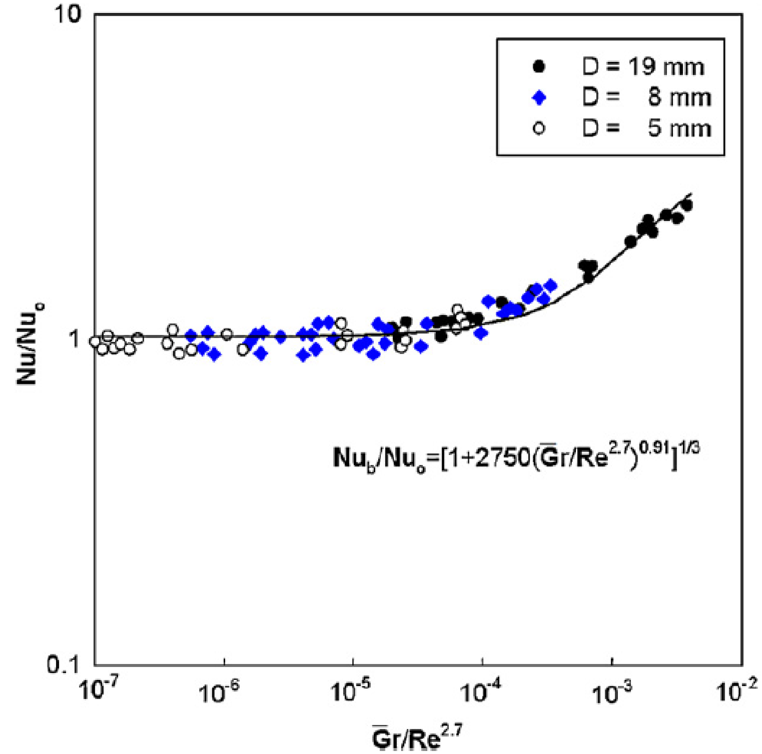
[Jackson 2013]

Buoyancy significantly affects vertical tube heat transfer by reducing or promoting turbulence

Upward flow



Downward flow

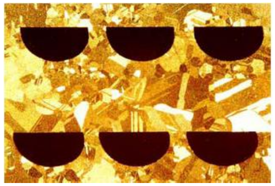


The onset of buoyant effects in upward flow: $$\left(\frac{Gr_b}{Re_b^{\;2.7}}\right)\left(\frac{\mu_w}{\mu_b}\right)\left(\frac{\rho_b}{\rho_w}\right)^{1/2} > 10^{-5}$$

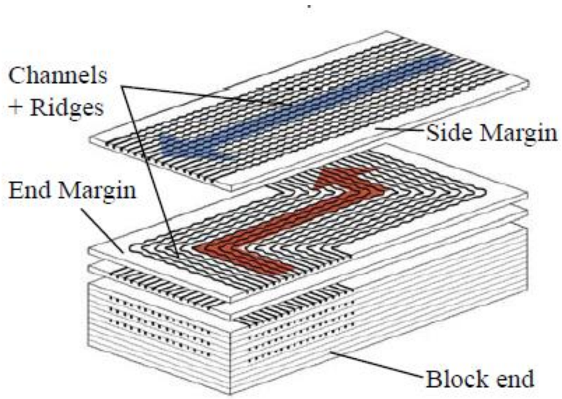
[Jackson 1979a]

Nu_0 = Nusselt number for forced convection

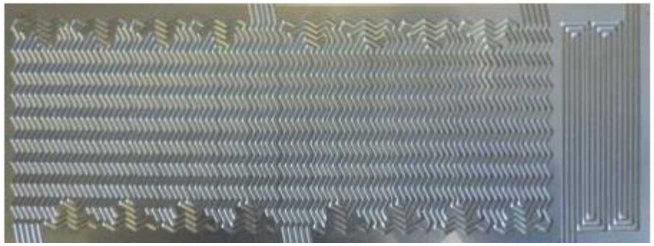
Real sCO2 applications have the difficult task of testing/correlating for complex HX geometries



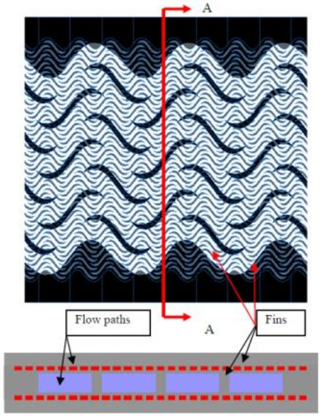
[Le Pierres 2011 et al.]



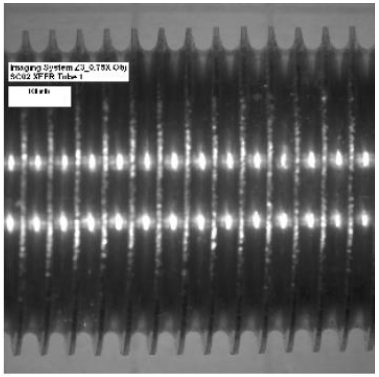
[Le Pierres 2011 at al.]



[Le Pierres 2011 at al.]



[Utamura 2007]



[Nehrbauer 2011]

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Backup Slides

S-CO2 flow in vertical tubes indicates local heat transfer is a strong function of fluid properties

Flow direction and heat flux affect wall temperature distribution

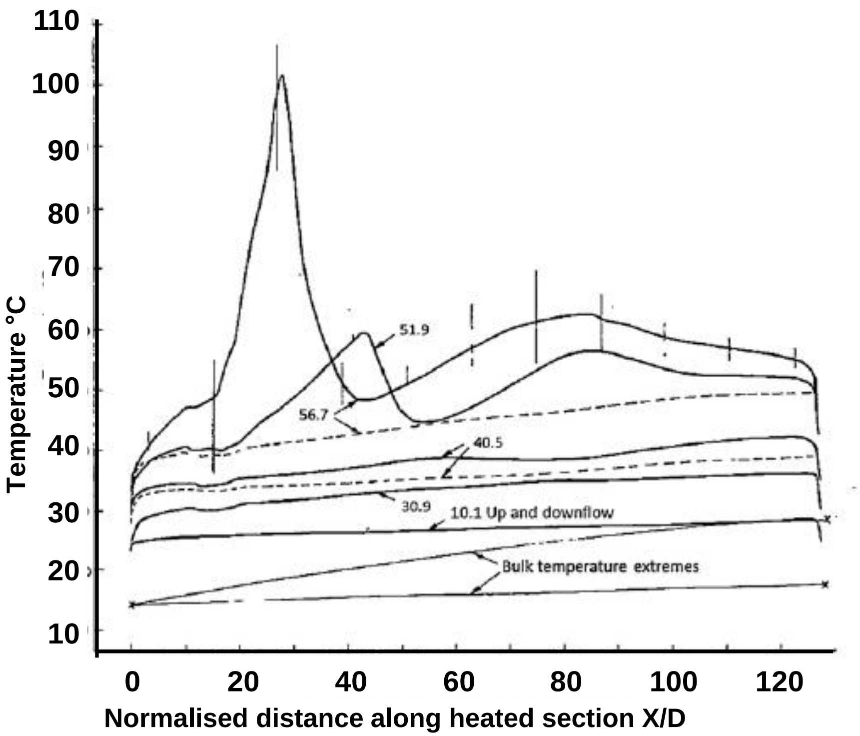

Fig. 4. Localized deterioration of heat transfer with upward flow; 19 mm diameter tube. Upflow is denoted by solid lines; downflow by broken lines, mass flow rate 0.160 kg/s; bulk inlet temperature 14 °C; wall heat flux as indicated, 30.9, 40.5, 51.9, 56.7 kW/m^2.

[Jackson 2013]

Inlet fluid temperature affects the axial location of the wall temperature peak

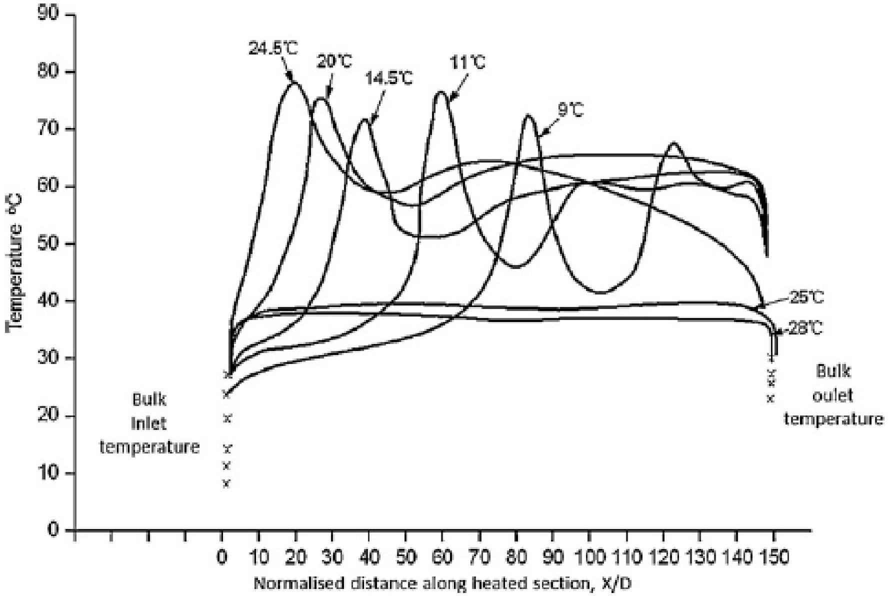

Fig. 8. Effect of reducing inlet fluid temperature, 8 mm diameter tube, upflow only. Pressure 7.58 MPa; inlet temperatures, 9 °C, 11 °C, 14.5 °C, 20 °C, 24.5 °C; mass flowrate 0.02 kg/s; wall heat flux 33.6 kW/m^2; Re ~ 4×10^4.

[Jackson 2013]

S-CO2 flow conditions can reduce the effect of fluid property changes on local heat transfer

Upward and downward flow directions produce similar wall temperatures at high mass flow ($Re \sim 2.5 \times 10^5$)

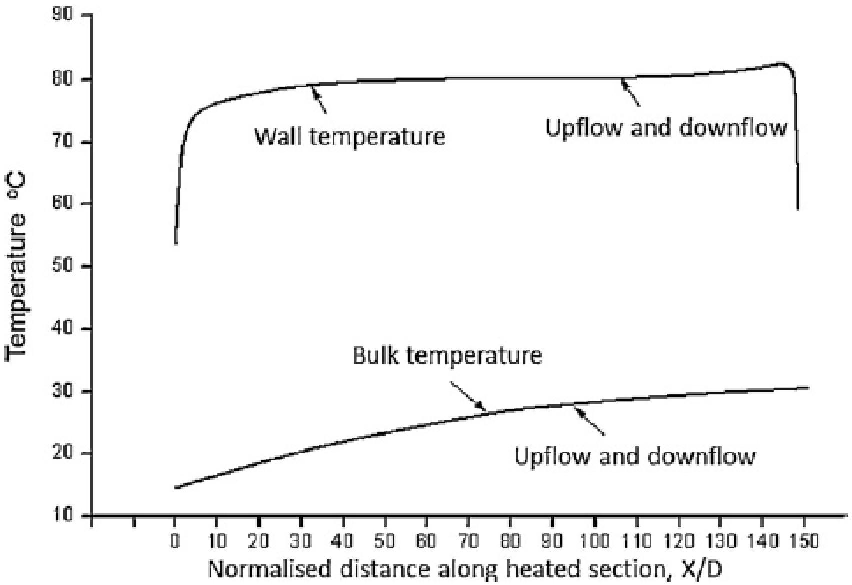

Fig. 9. Highest mass flow rate, 5mm diameter tube, pressure 7.58 MPar, upflow only. Mass flowrate 0.0645 kg/s; wall heat flux 455 kW/m^2.

[Jackson 2013]

The upward flow direction produces a peak wall temperature at a low mass flow ($Re \sim 4 \times 10^4$)

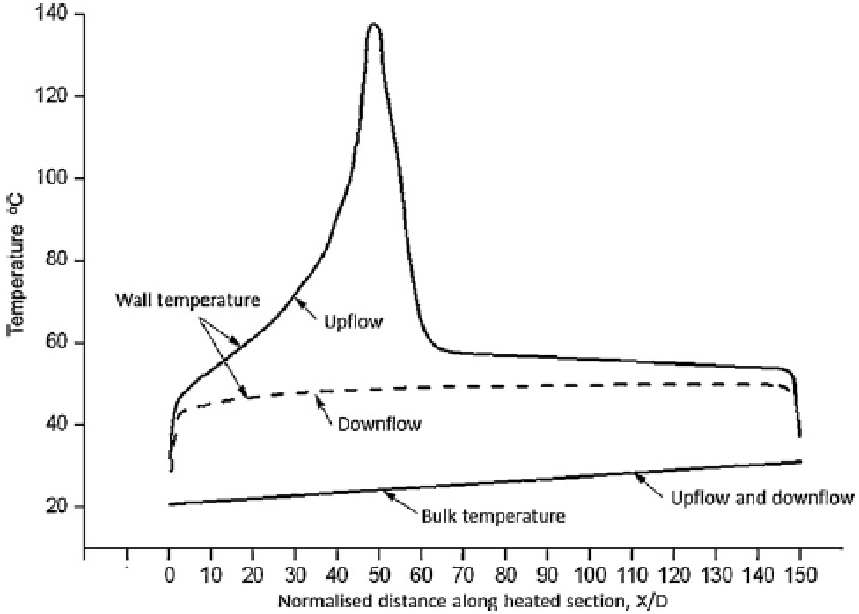

Fig. 10. Further reduction of flow rate, 5 mm diameter tube, upflow and downflow. Pressure 7.58 MPa; mass flow rate 0.0129 kg/s; wall heat flux 68 kW/m^2; $Re \sim 4 \times 10^4$.

[Jackson 2013]