The 6th International Supercritical CO2 Power Cycles Symposium
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Tutorial:

Heat Exchangers for Supercritical CO2 Power Cycle Applications

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Heat Exchangers for Supercritical CO2 Power Cycle Applications

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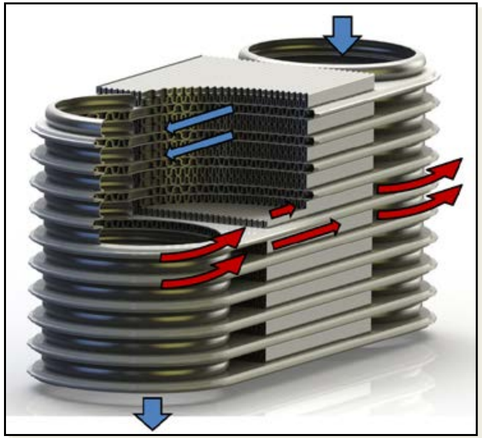
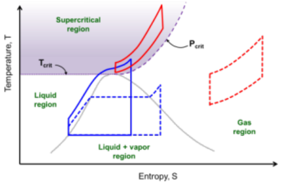
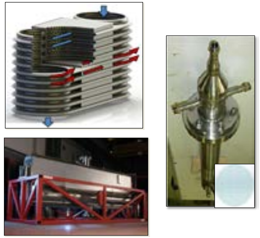


The following slides present an overview of heat exchangers in supercritical CO_2 applications

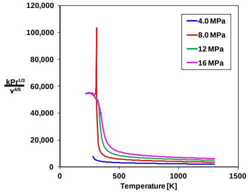


General heat exchanger overview and design trades



Specific heat exchangers for sCO_2

Heat exchanger mechanical design for sCO_2



Hydraulic design and heat transfer in supercritical fluids

Heat Exchangers in sCO_2 power conversion cycles

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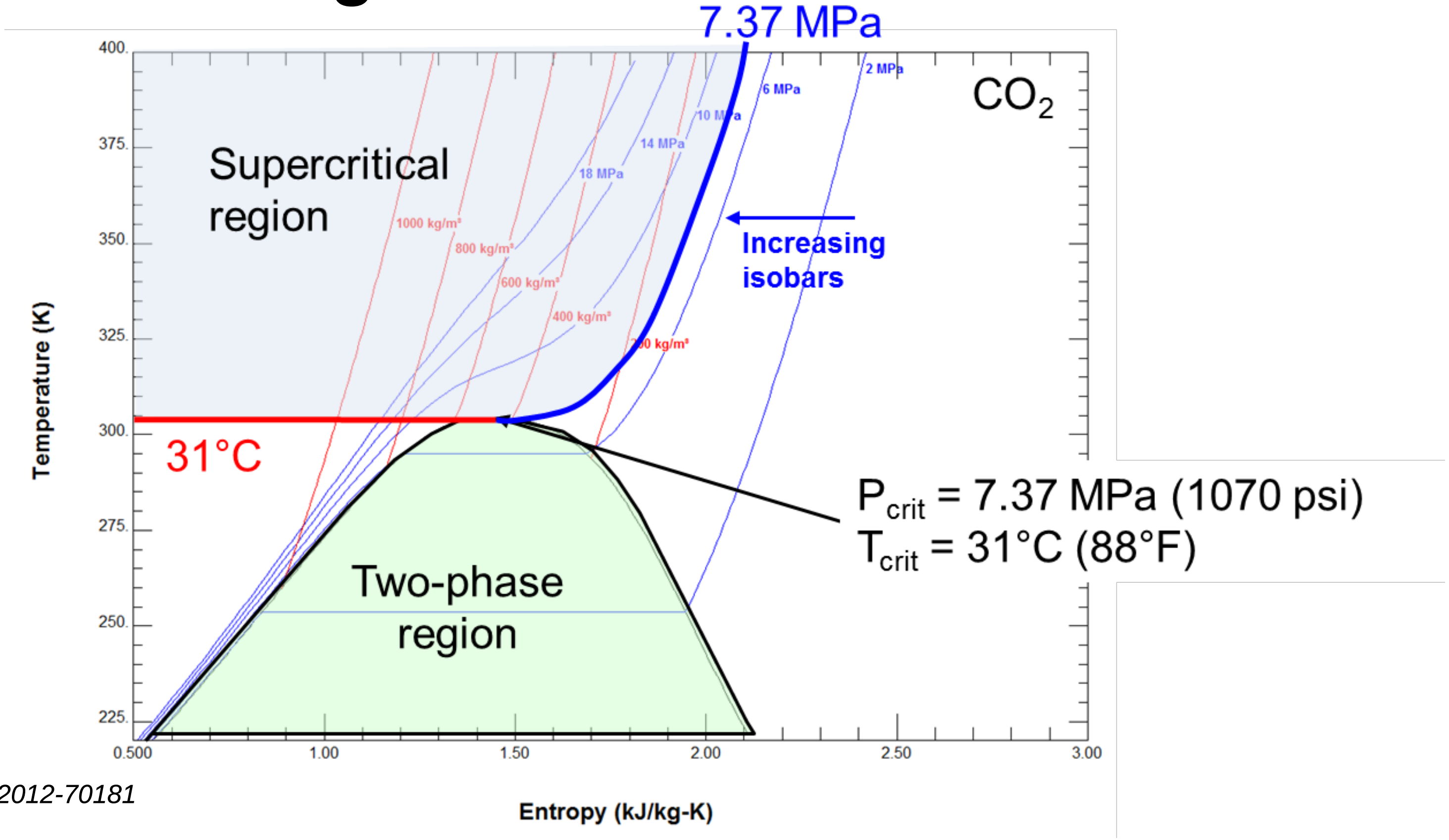
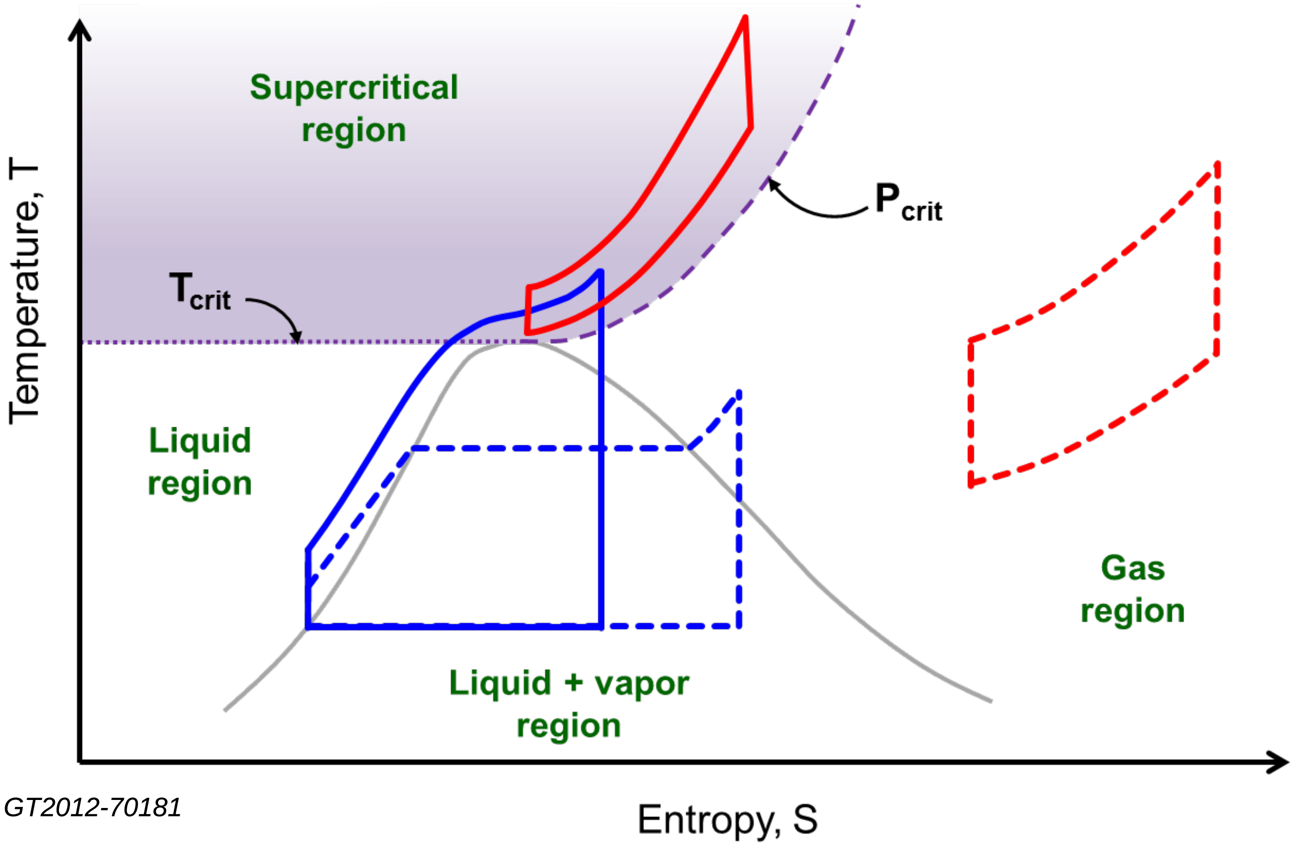
A fluid is supercritical if the pressure and temperature are greater than the critical values



Source: Musgrove et al. GT2012-70181

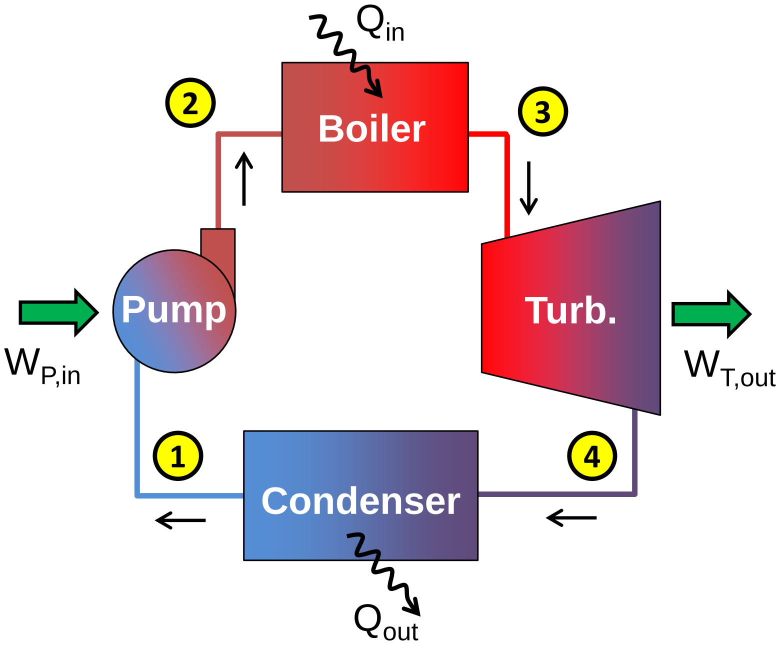
A power cycle is supercritical if part of the cycle takes place in the supercritical phase region



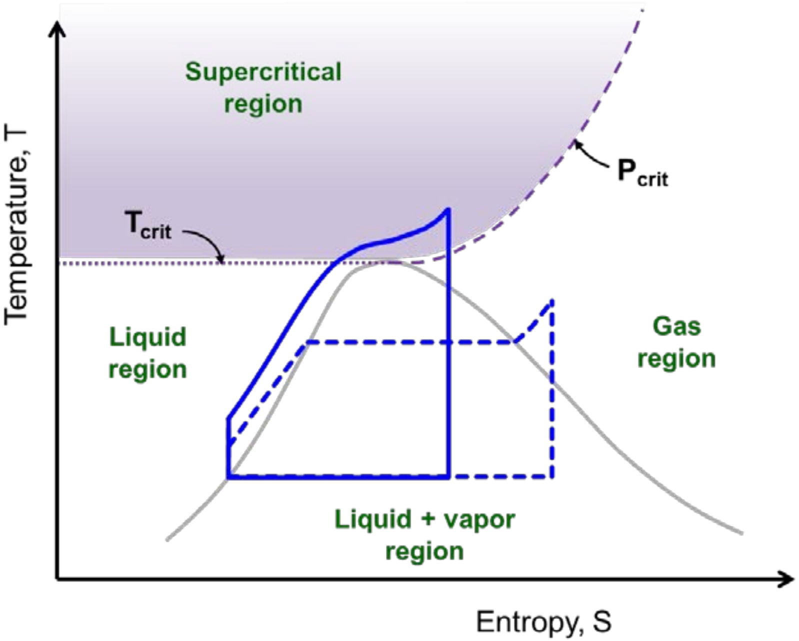
Source: Musgrove et al. GT2012-70181

A Rankine cycle requires heat exchangers for phase change

Heat Input:
- Typically indirect-fired like a boiler or steam generator



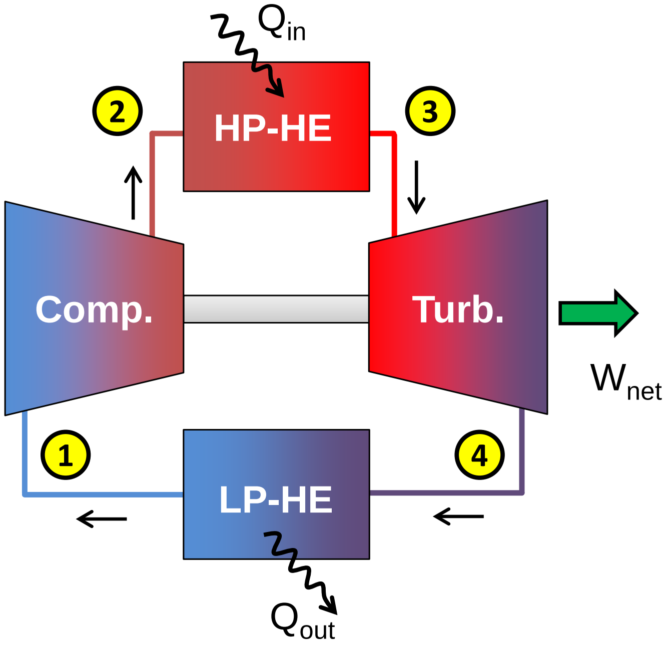
Heat Rejection:
- Cooling by air or condensing towers

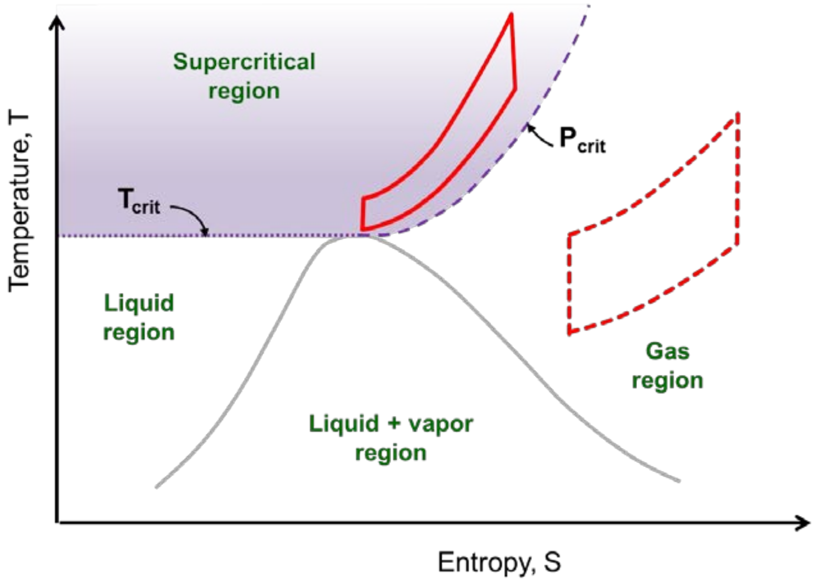


A Brayton Cycle requires heat exchangers for single-phase heat transfer

Heat Input:
- Direct-fired (oxy-combustion)
- Indirect-fired like a boiler

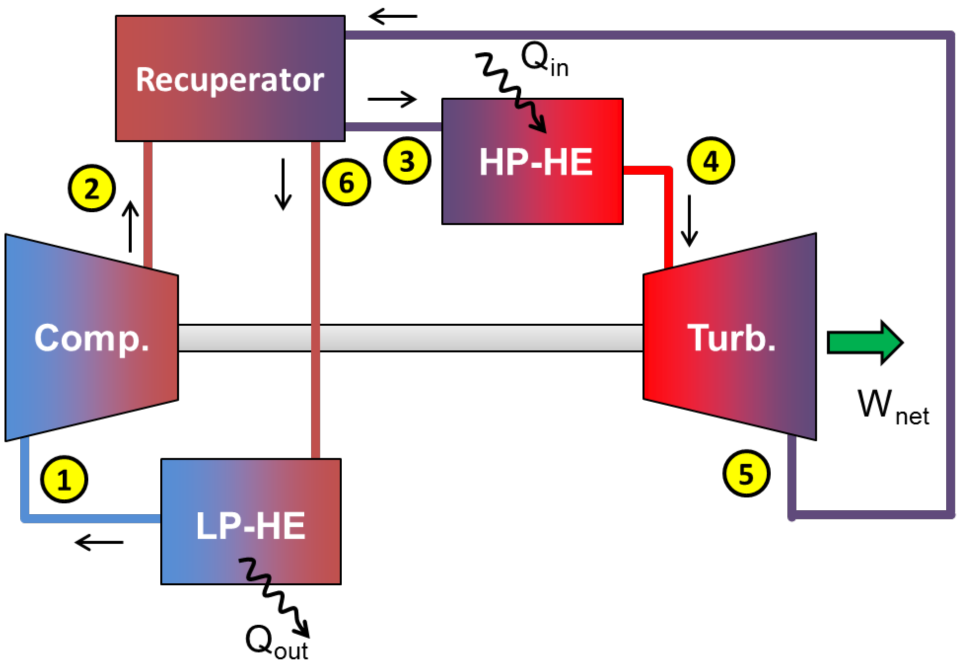
Heat Rejection:
- Closed-loop cycle – uses cooling water or cooling air

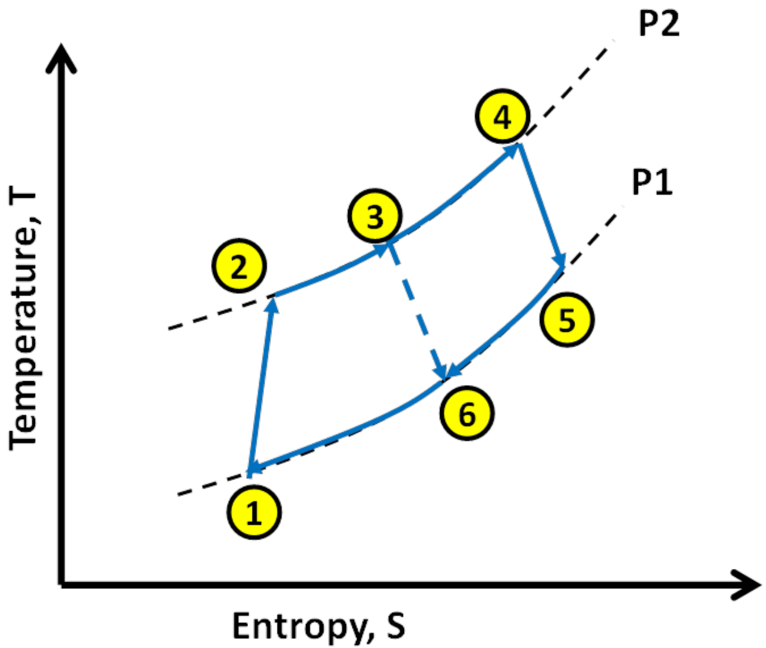




A recuperator exchanges heat within the cycle to improve overall cycle thermal efficiency

Recuperators generally transfer heat between separated flow streams





The number and types of heat exchangers depend on the cycle design

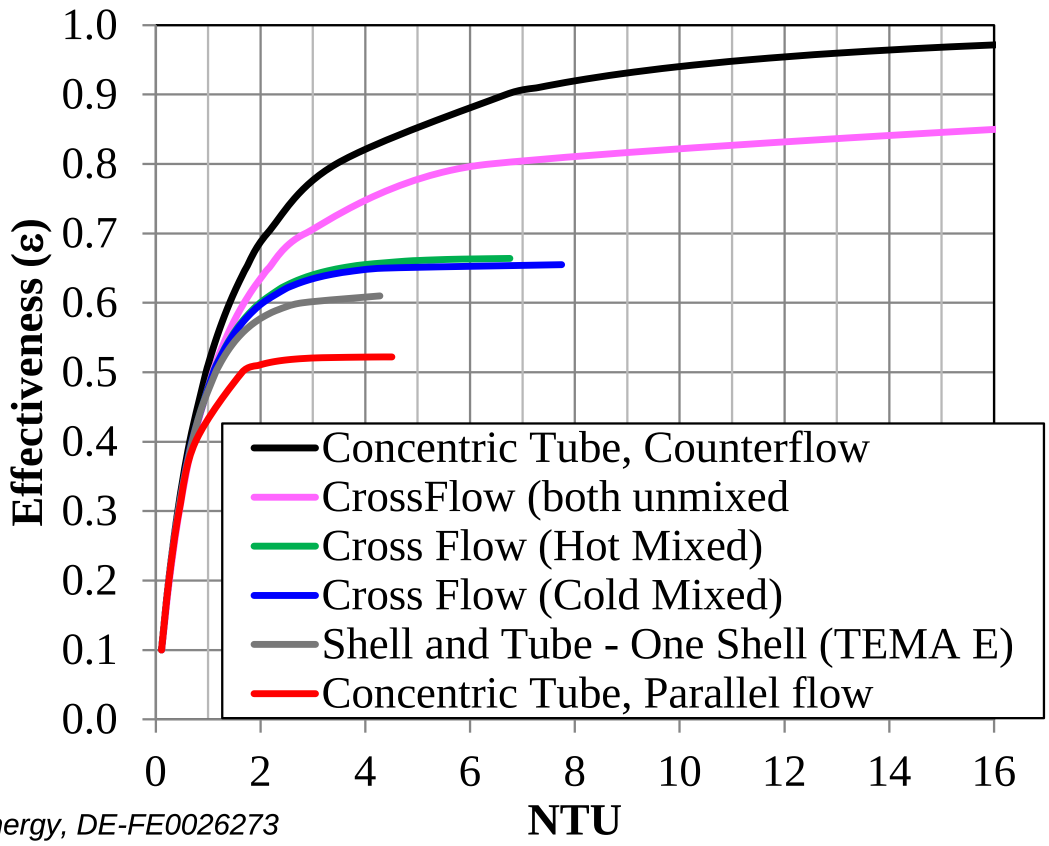
Super-critical Brayton cycle:

- Heater
- Cooler
- Single-phase heat transfer
- Optional: High temperature recuperator for the cycle
- Optional: Low temperature recuperator for the cycle
- Optional: Recuperator for waste heat recovery

Super/trans-critical Rankine cycle:

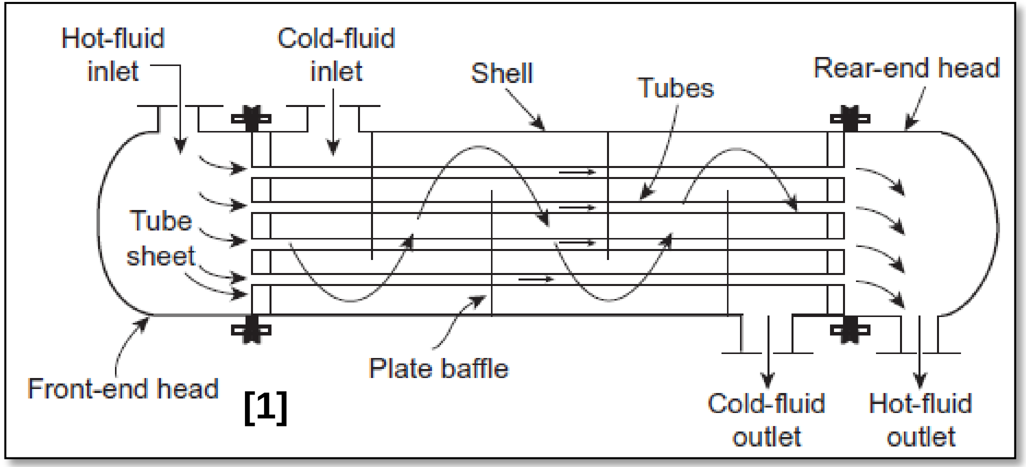
- Heater
- Cooler
- Multi-phase heat transfer (separator?)
- Optional: High temperature recuperator
- Optional: Low temperature recuperator

Most heat exchangers for sCO_2 are a counter-flow configuration because of the high effectiveness



Courtesy Thar Energy, DE-FE0026273

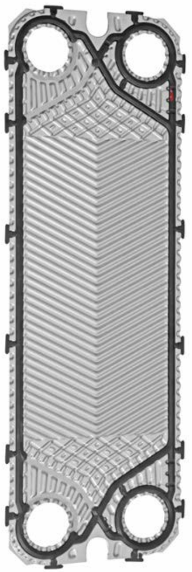
Some Conventional Heat Exchanger Layouts



[1] Shah, R. K., and Sekulic, D. P., 2003, Fundamentals of Heat Exchanger Design, John Wiley & Sons, New Jersey.
[2] Alfa Laval [4] Shah, R. K., and Sekulic, D. P., 2003, Fundamentals of Heat Exchanger Design, John Wiley & Sons, New Jersey.

Plate-type Configuration

- Corrugated plates are stacked to create flow passages
- Layers and corrugations provide rigidity and structural support
- Plates are sealed by a gasket, weld, or braze – depending on operating conditions



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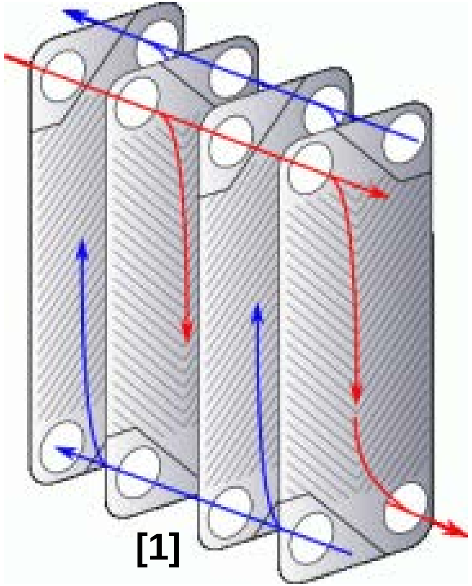
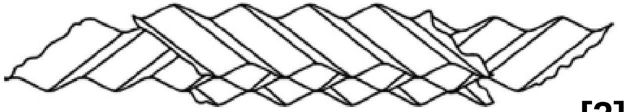


[1]



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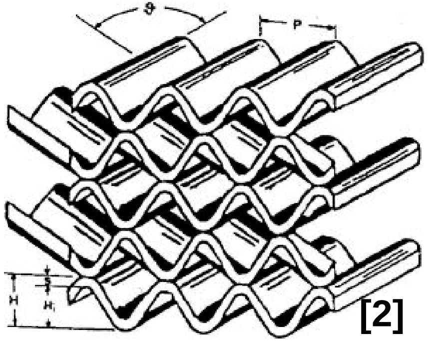

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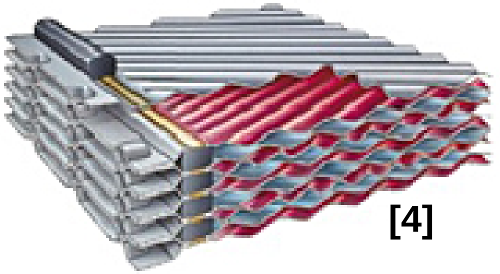
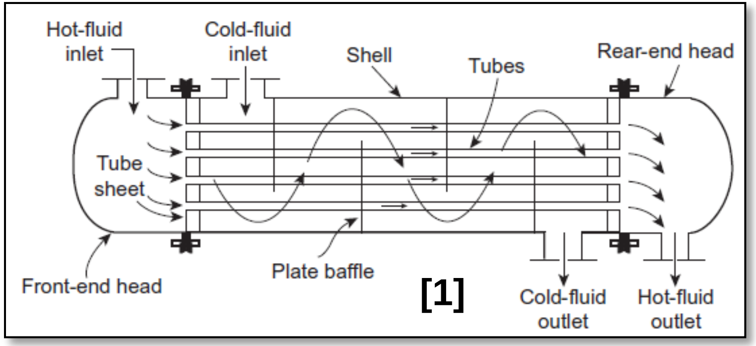


[4]

[1] Thomas Wicht, 2011, "Phase change behavior of ammonia-water mixtures in corrugated plate heat exchangers."
[2] L. Wang, B. Sunden, and R.M. Manglik, 2007, Plate Heat Exchangers: Design, Applications and Performance, WIT Press.
[3] Stuhrlingenterprise
[4] Alfa Laval, "Alfa Laval Launches WideGap Heat Exchanger,"Ethanol Produce Magazine.

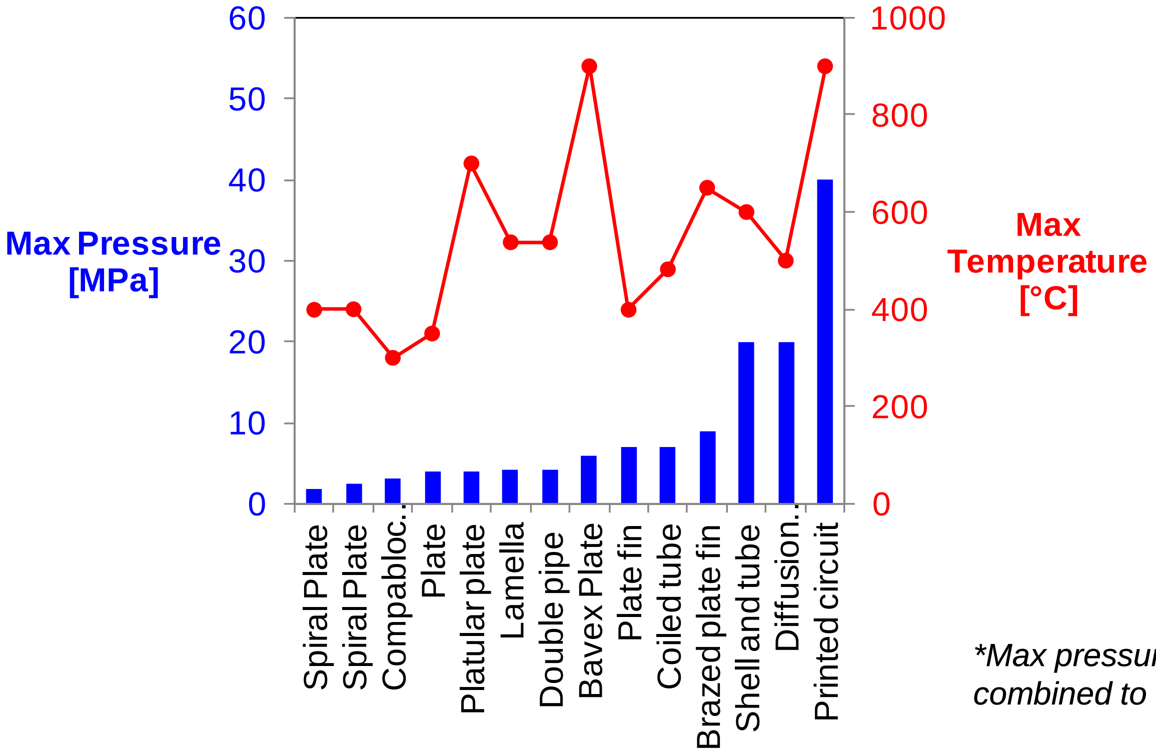
Shell and Tube Heat Exchangers

- Mechanical layout and design are detailed in ASME Boiler and Pressure Vessel Code and TEMA
- **The conceptual layout is simple:**
  - Casing
  - Tube bundle
  - Tube sheets
  - High pressure fluid usually in the tube



[1] Shah, R. K., and Sekulic, D. P., 2003, Fundamentals of Heat Exchanger Design, John Wiley & Sons, New Jersey.
[2] PRE-heat INC. [3] Southwest Thermal Technology, Inc

Heat exchanger type is dependent on the expected conditions



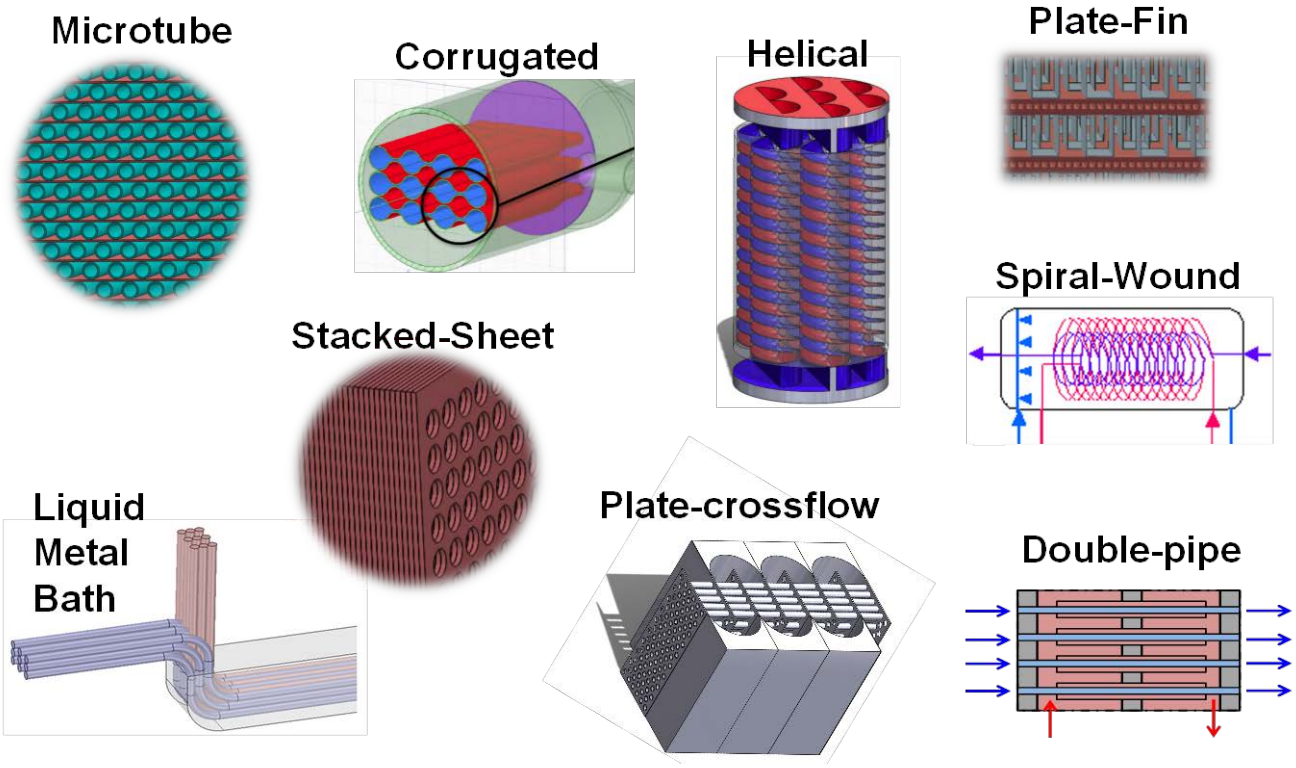
**Max pressure and Max Temperature should not be combined to select a heat exchanger from this chart!*

Data from:

[1] Shah, R. K., and Sekulic, D. P., 2003, Fundamentals of Heat Exchanger Design, John Wiley & Sons, New Jersey.

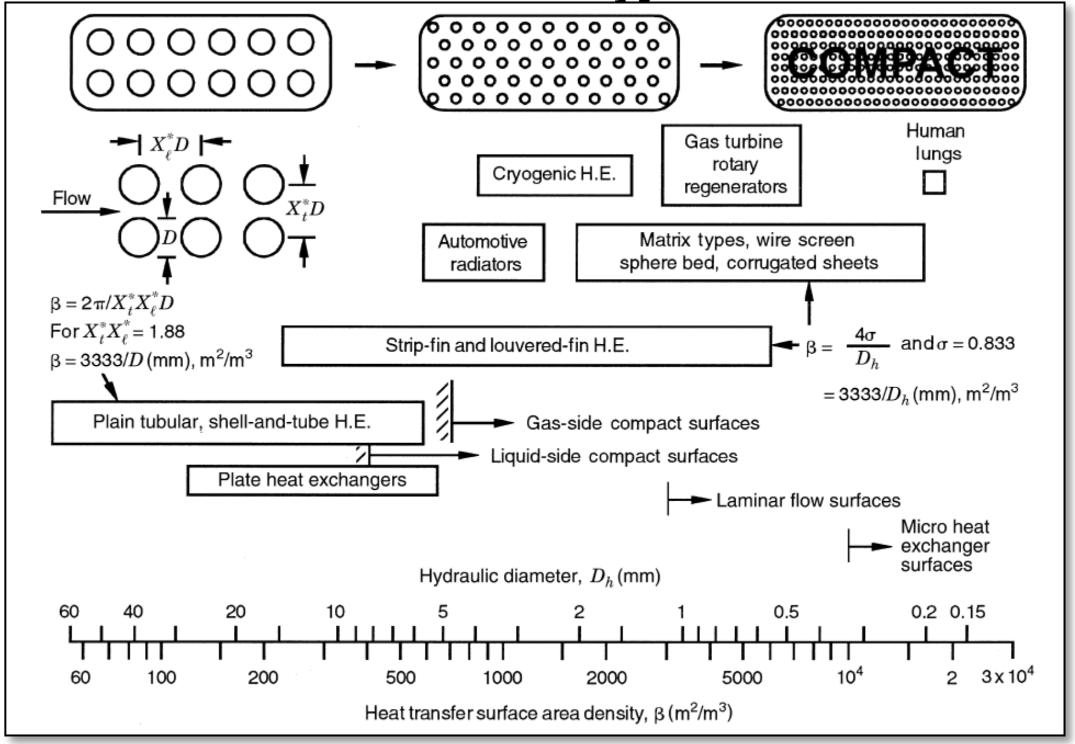
[2] Kuppan, T., 2000, Heat Exchanger Design Handbook, Taylor & Francis, New York.

The sky is the limit for heat exchanger concepts



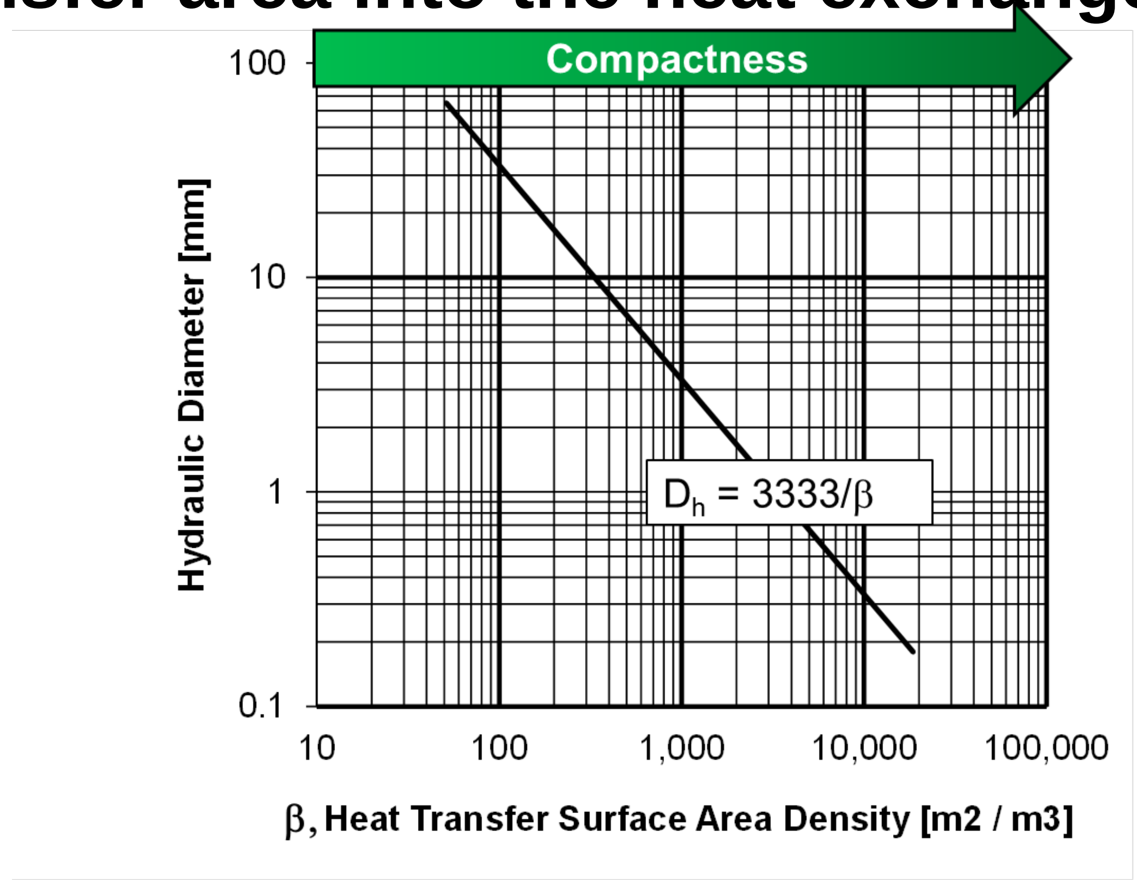
Courtesy Thar Energy, DE-FE0026273

How Much Heat Transfer Area in a Heat Exchanger?



Kakac, S., Bergles, A., and Mayinger, F., eds., 1981, Heat Exchangers: Thermal-Hydraulic Fundamentals and Design, Hemisphere Publishing Corporation, Washingtion.

The flow passage must decrease to pack more heat transfer area into the heat exchange volume



Design Trades for Heat Exchangers

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Heat exchanger design considerations

sCO_2 physical property variations require sensitivity checks

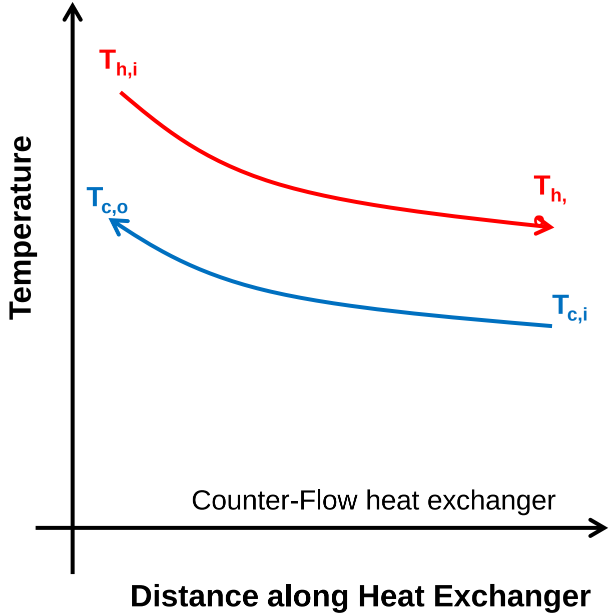
- Operating conditions
- Pressure levels
- Off-design points including turn-down conditions need to be analyzed for avoiding pinch point and reversal

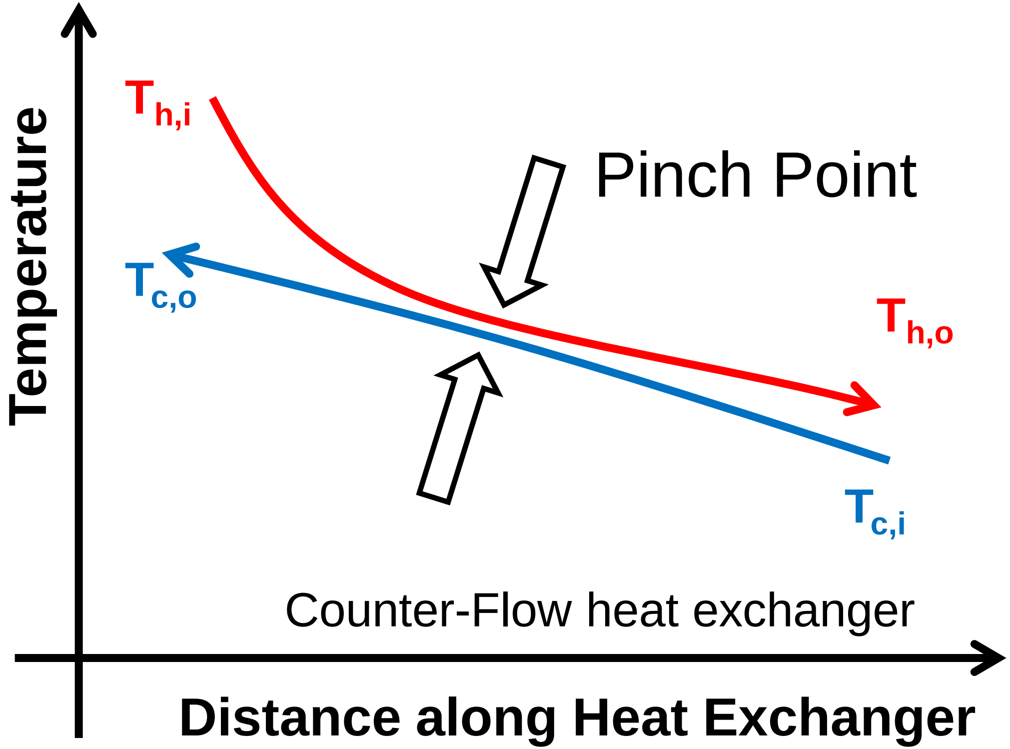
Plant efficiency vs HX CAPEX

- Close temperature approach requires high effectiveness recuperators
- High design temperature requires high nickel alloy

Real gas properties or phase change can create 'pinch' points in the temperature profile

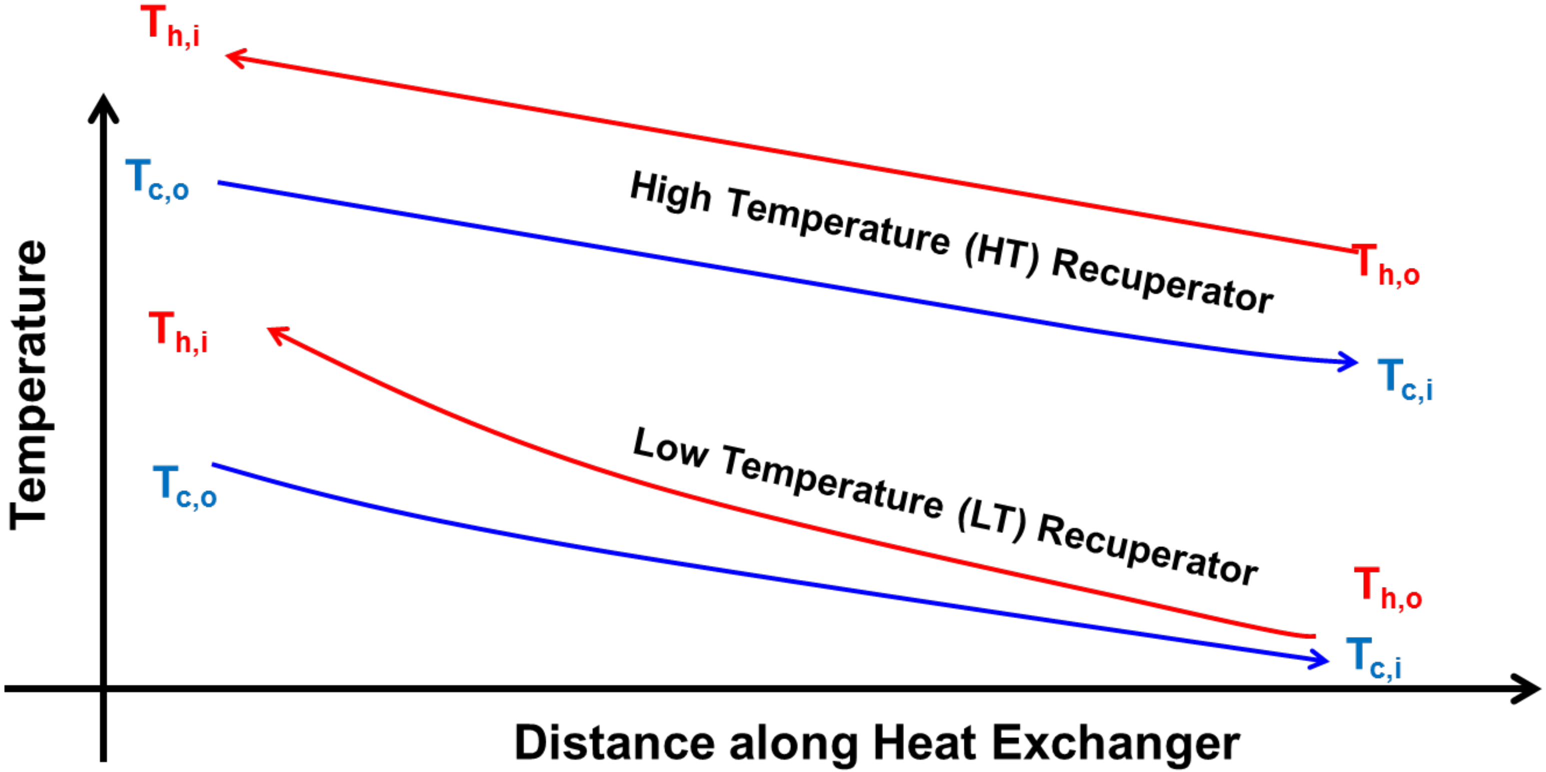
Pinch results in a poor design because the little-no heat is transfer when ΔT becomes very small



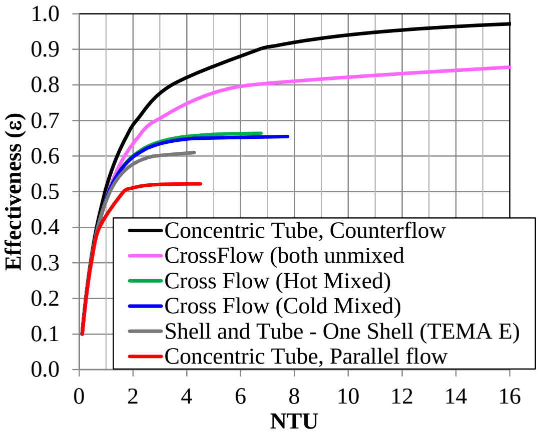


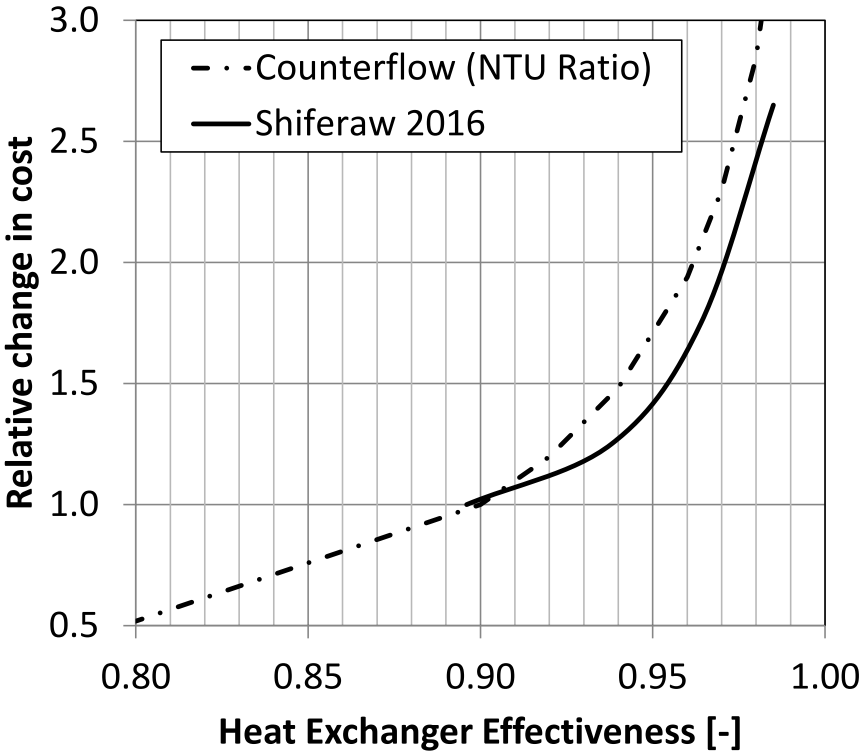
Recuperation can be split into high- and low-temperature units

Selecting the split point between recuperators is part of the cycle design



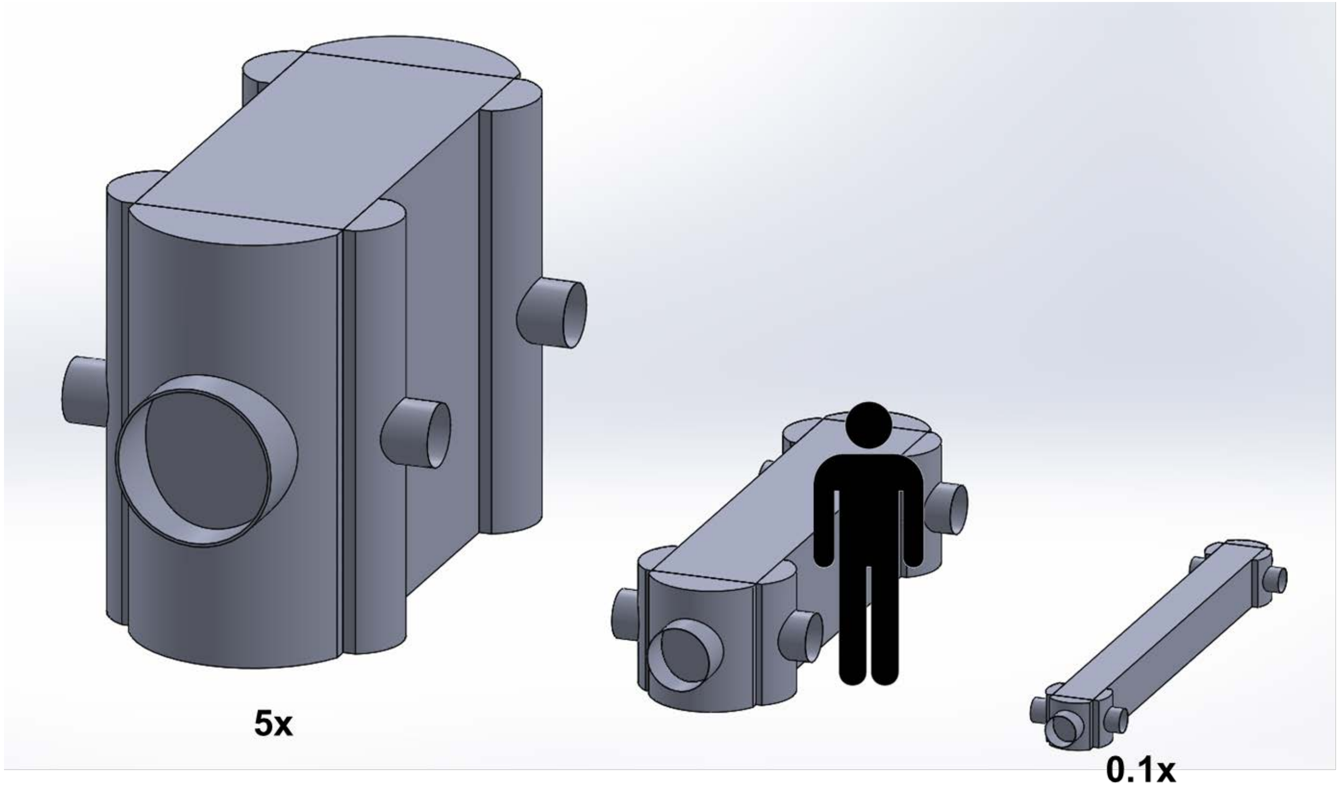
The required effectiveness can have a dramatic impact on heat exchanger size and cost



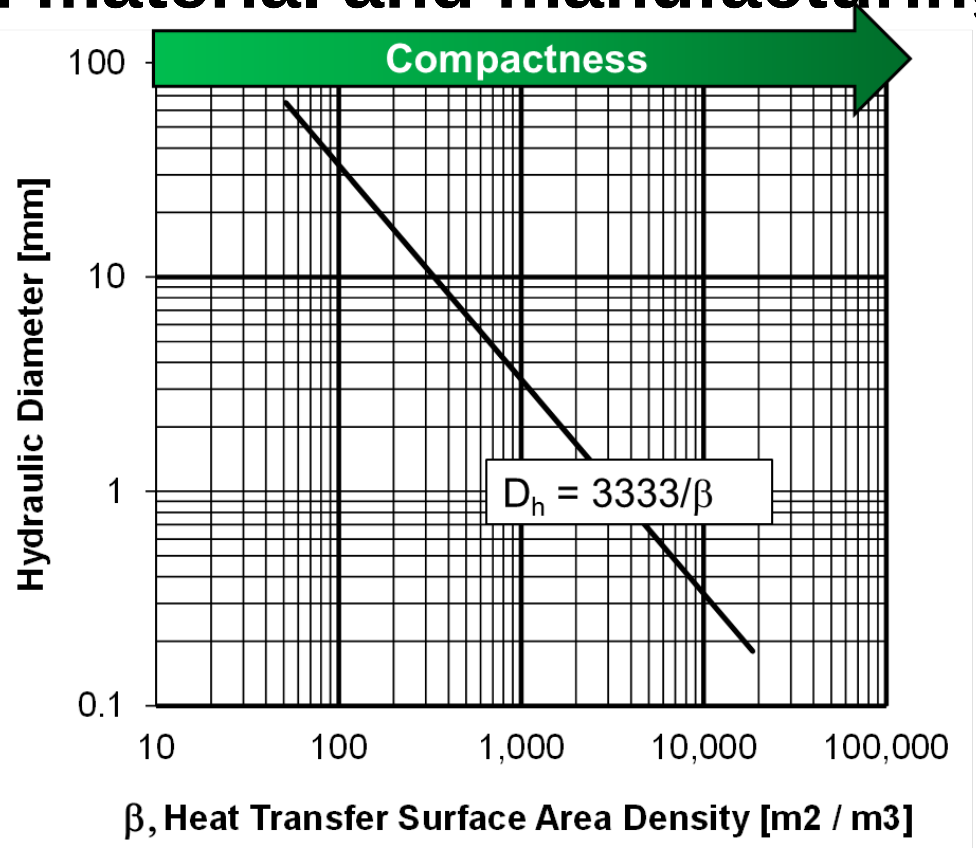


Shiferaw, D., 2016, "Economic Analysis of SCO2 Cycles with PCHE Recuperator Design Optimisation."

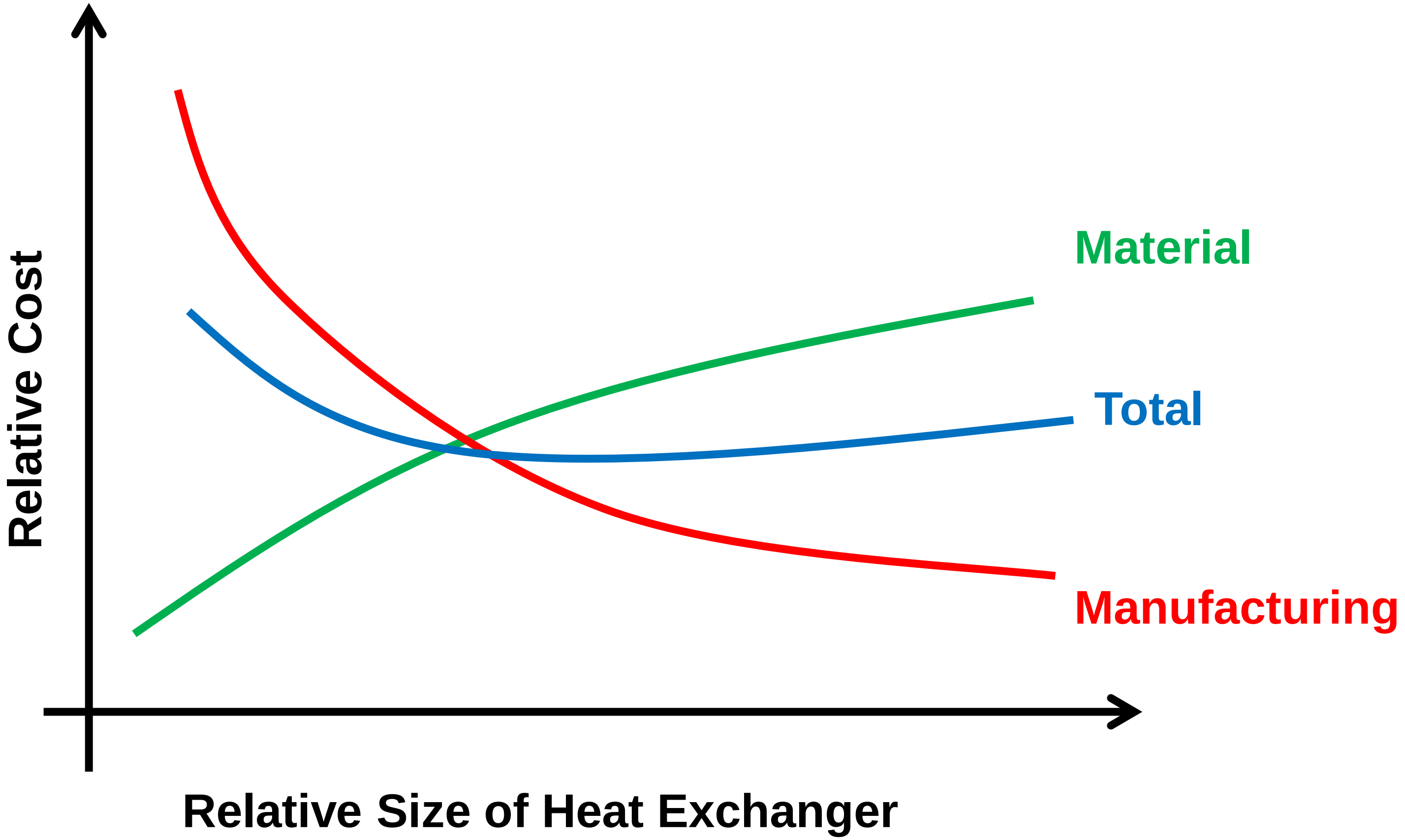
Economy of scale must also consider manufacturing limits as HXs are scaled to large thermal duties



As passage size and overall volume require a trade between material and manufacturing costs

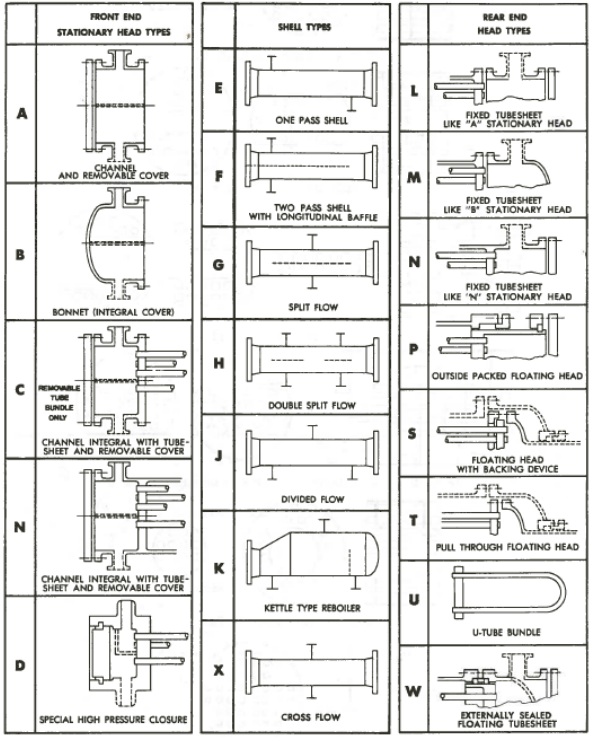


Trade studies in material and manufacturing selection are important to minimize cost

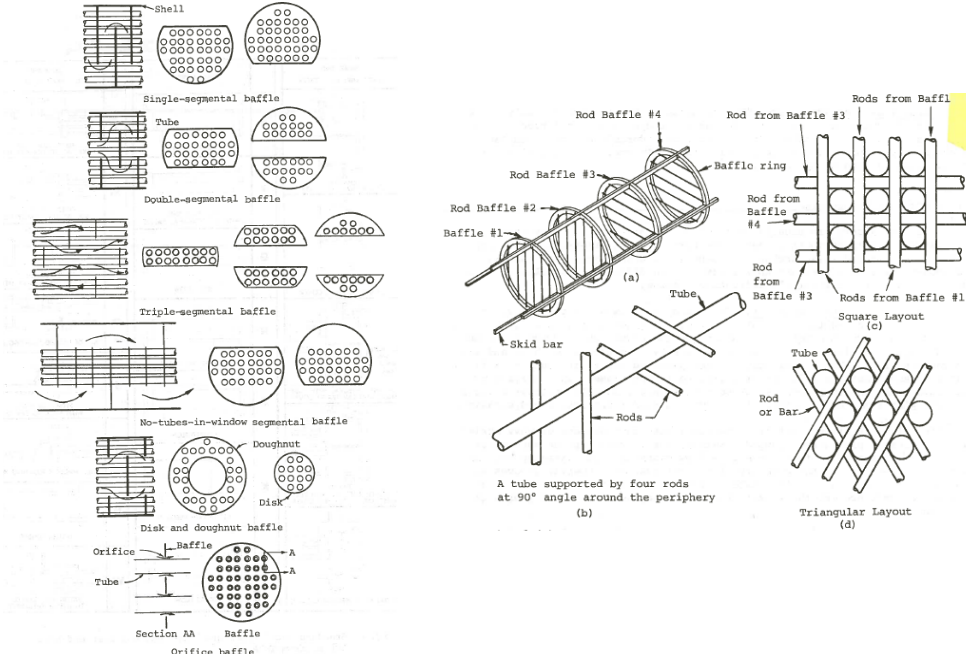


Many possible detail design options and trades for example, shell and tube

Shell and Tube Casing and Head

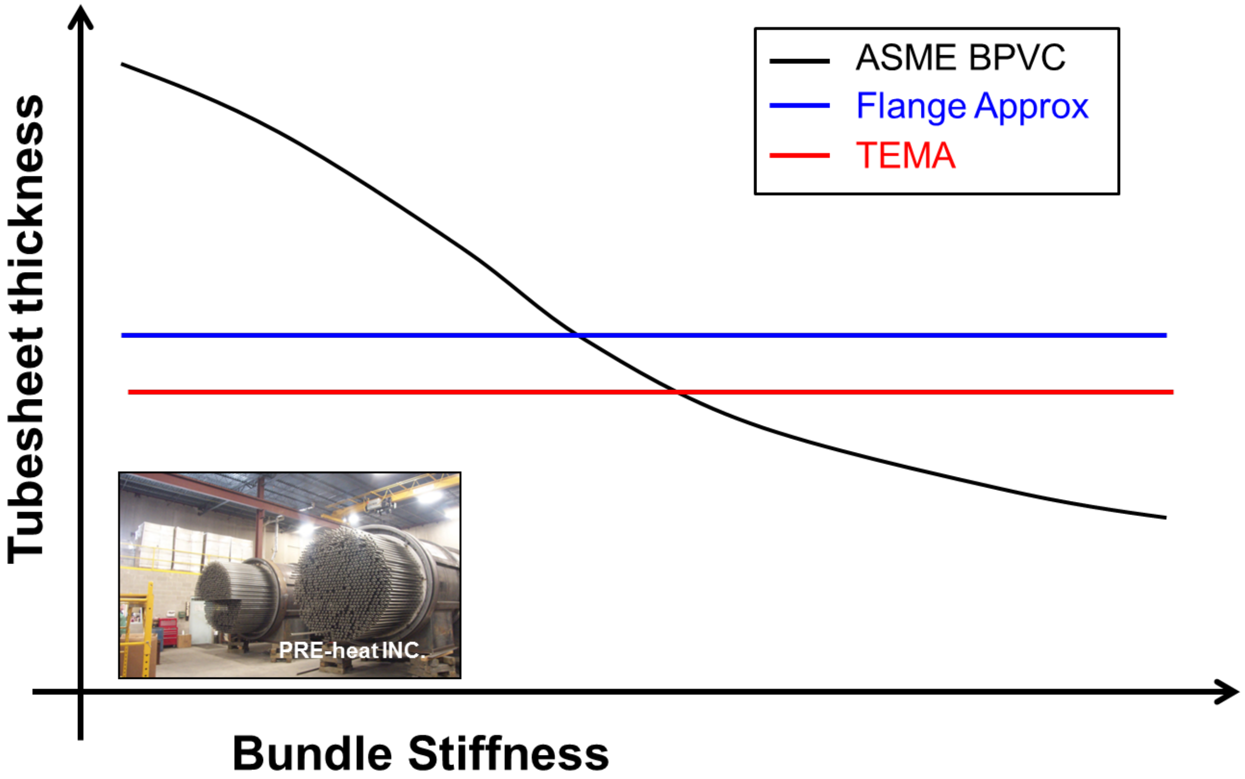


Tube bundle design



[1] Kakac, S., Bergles, A., and Mayinger, F., eds., 1981, Heat Exchangers: Thermal-Hydraulic Fundamentals and Design, Hemisphere Publishing Corporation, Washingtion.

Complex design codes can be used to optimize a heat exchanger design



Performance, cost, and ASME code calculations can be combined for optimization

- **Monte Carlo**
- **Genetic algorithm**
- **Response surface**
- **Neural network**

Genetic Algorithm
Variables (Dia, # Tubes , Length)
Objectives (Minimize Cost, Increase ε)
Constraints (ΔP, Thermal Duty)

Heat Exchanger Specific
Geometry
Initial Conditions
ASME code calculations
Cost estimates

mHX - source
Solution

Heat Exchangers Applied to sCO_2 Power Systems

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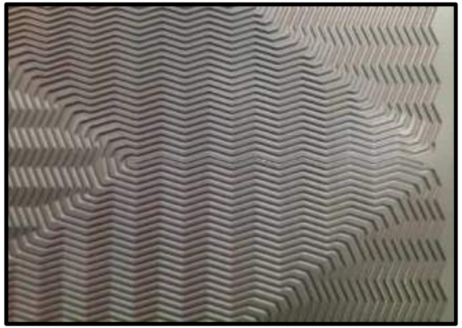
Heatric PCHE

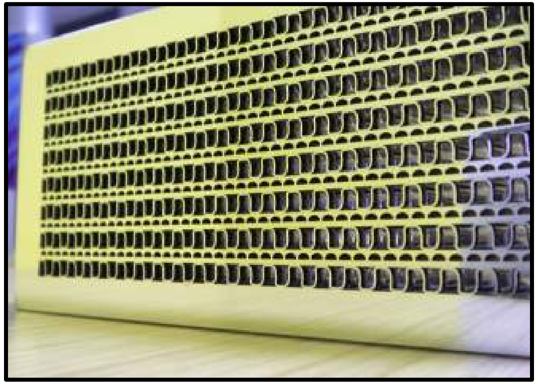
Heatric

MEGGITT

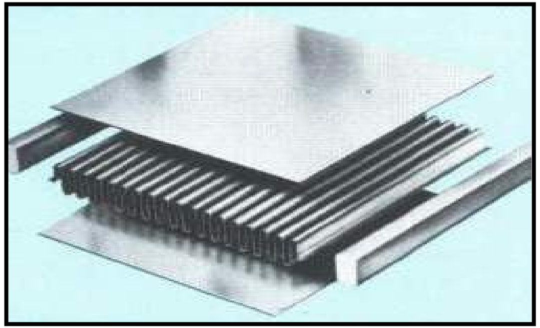
PCHE
Printed Circuit Heat Exchanger

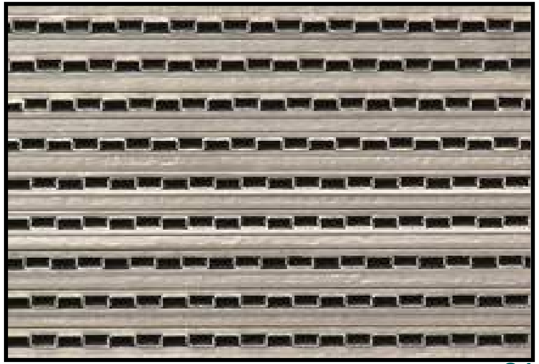


H^2X
Hybrid

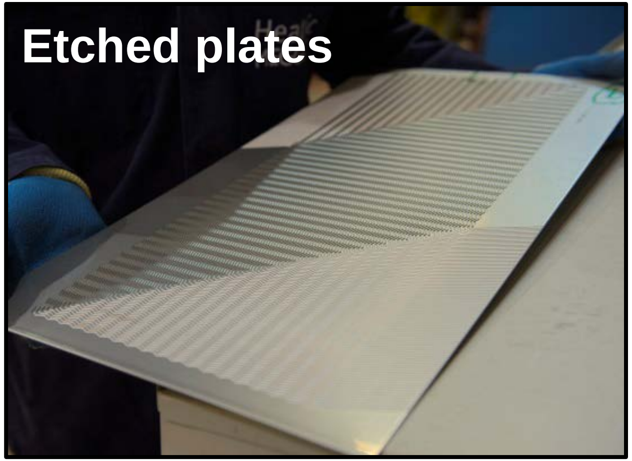


FPHE
Formed Plate Heat Exchanger



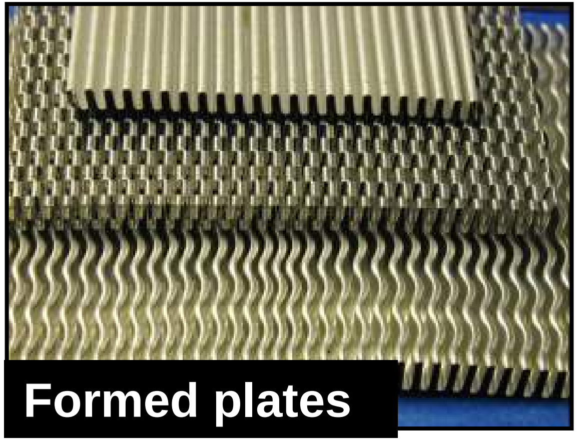


Main Components



Etched plates

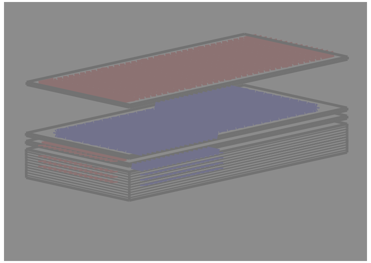
Diffusion bonded core

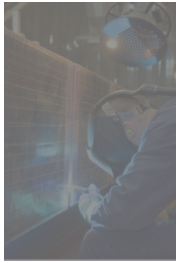


Formed plates

Headers, nozzles, flanges

Construction



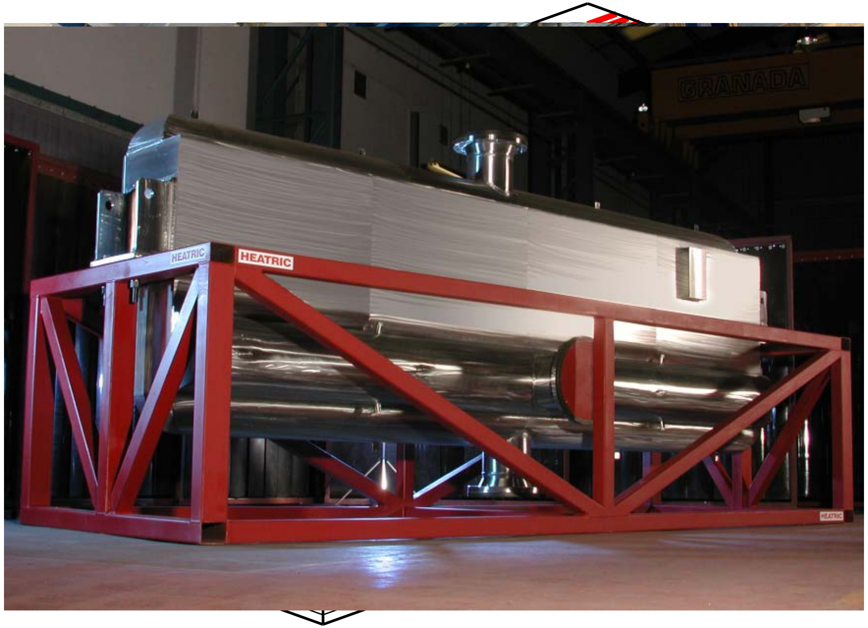


1. Stack and Diffusion Bond Core

2. Block to block joints

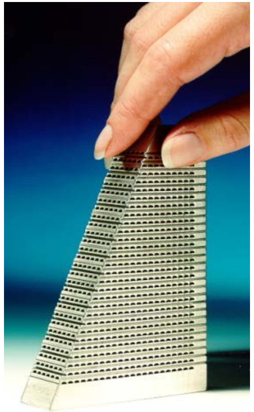
3. Assemble headers, nozzles and flanges

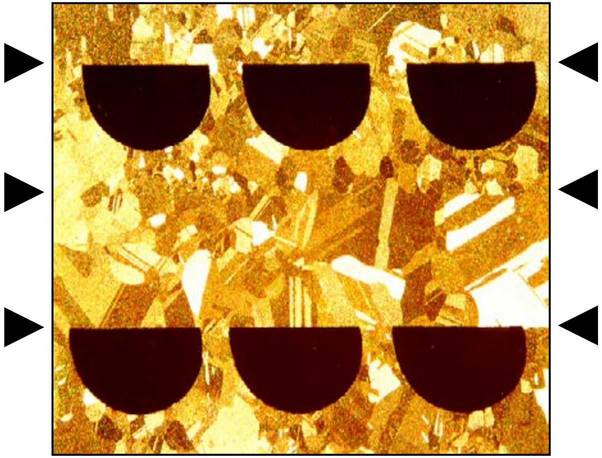
4. Weld headers, nozzles and flanges to core

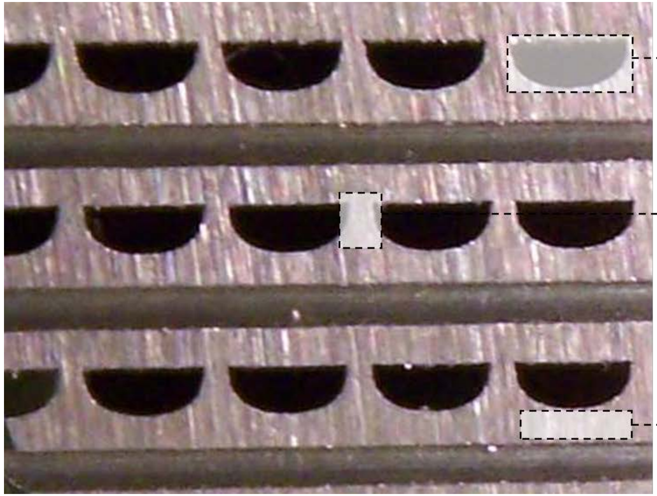


Core Details

<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

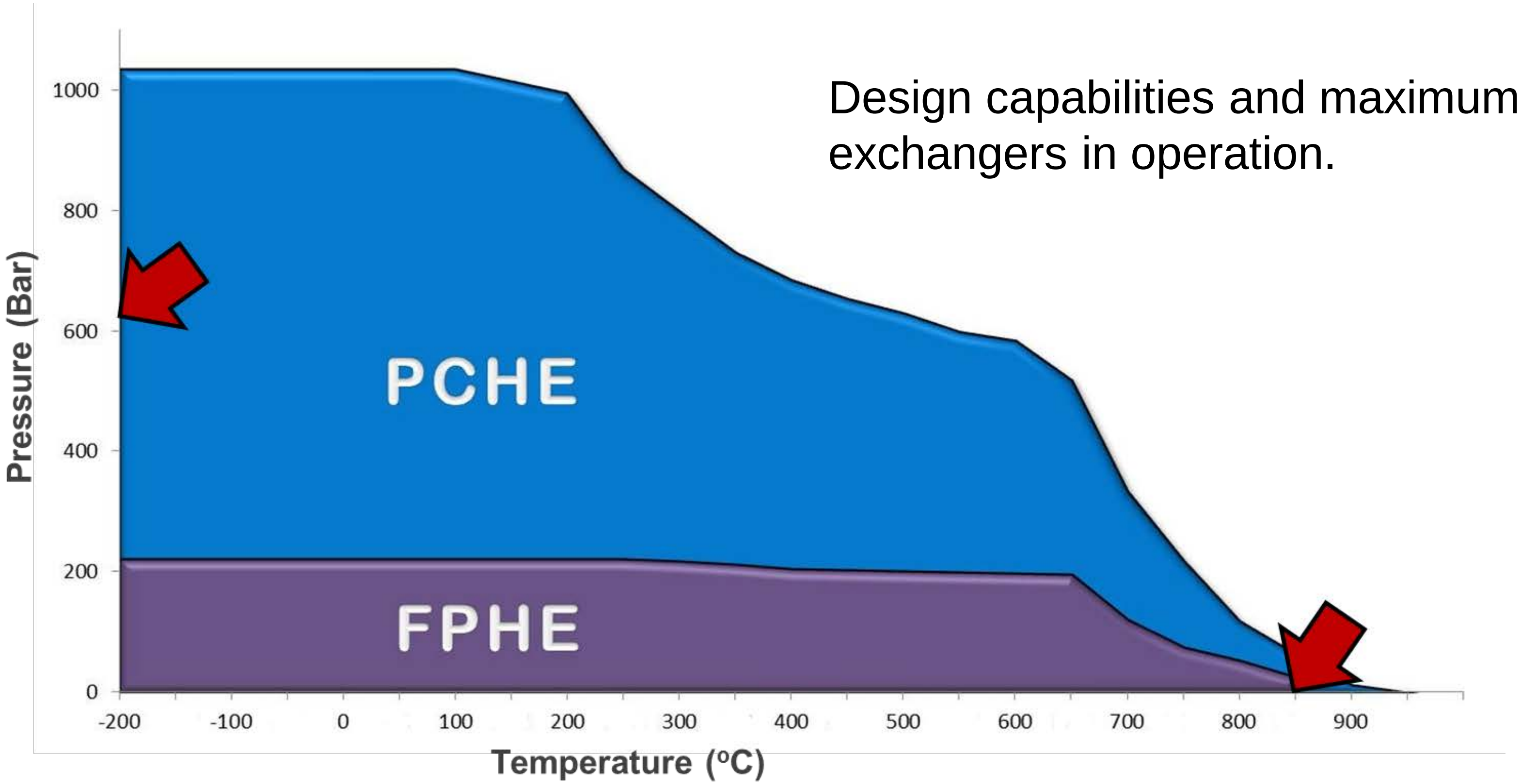






Cores are designed and values depend on thermal and hydraulic requirements

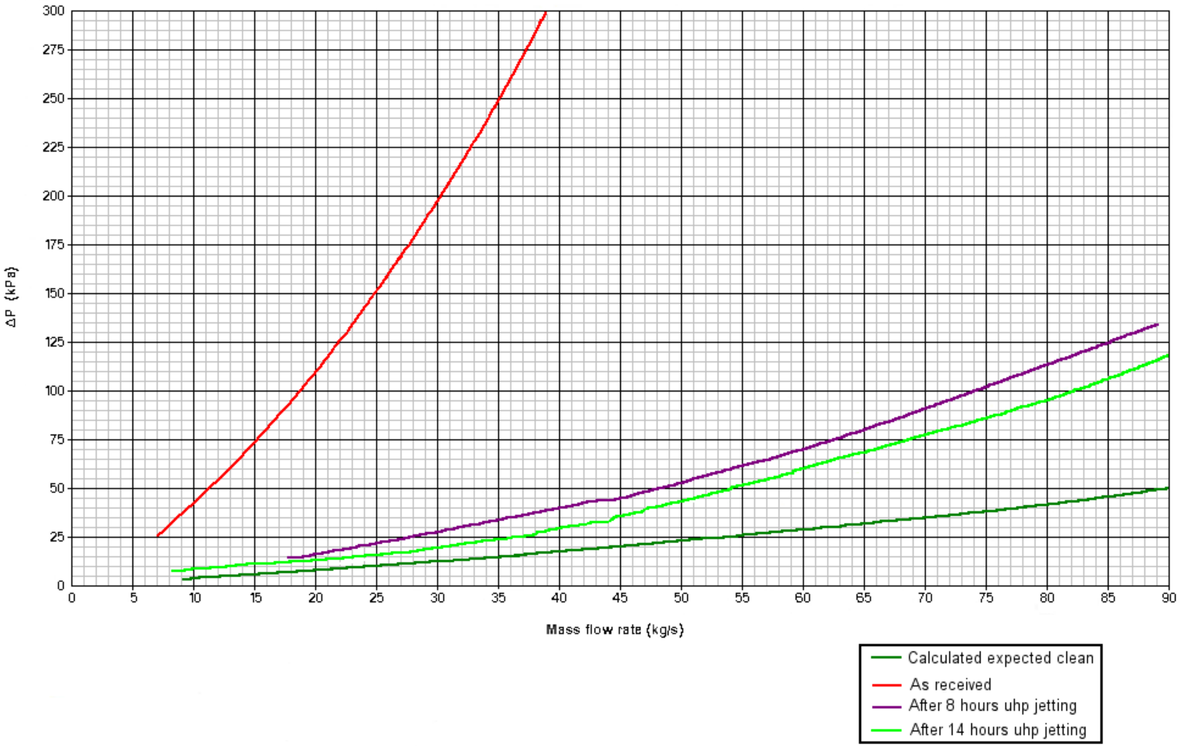
Operating Conditions

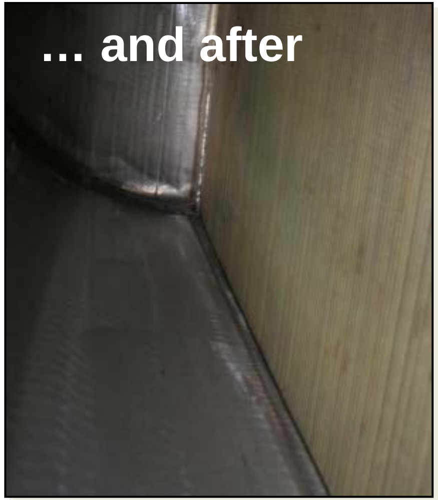


Design capabilities and maximum rated exchangers in operation.

Maintenance

- Mechanical: Ultra High Pressure (UHP) water jetting
- Chemical: Can be used with UHP or standalone





Design & Test for Heat Exchangers in the sCO_2 Brayton Cycle

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Content

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1

HXs Design Consideration

- Heater
- Recuperator
- Cooler

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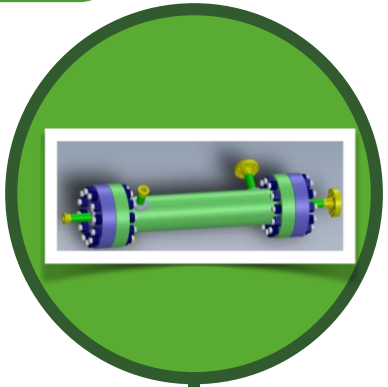
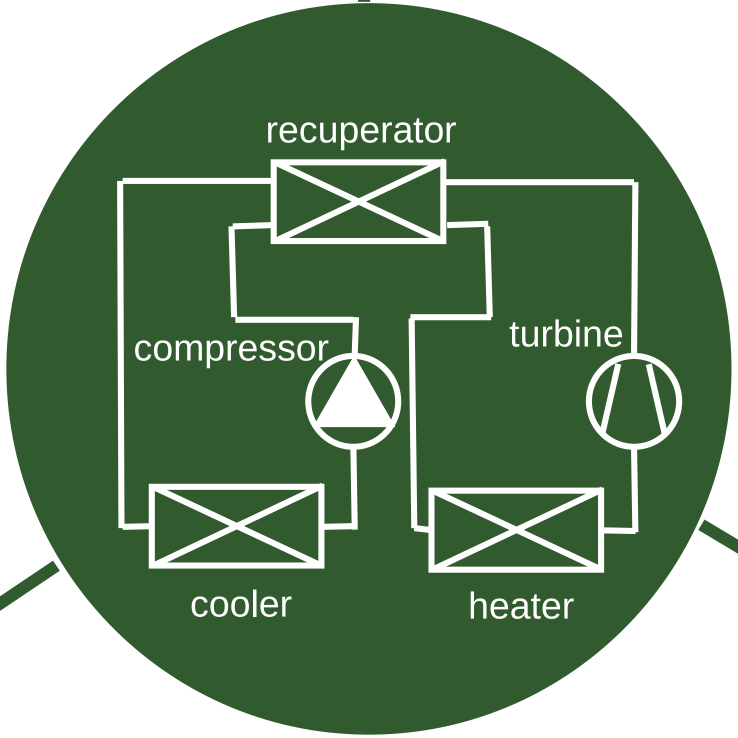
Standard sCO_2 Brayton Cycle



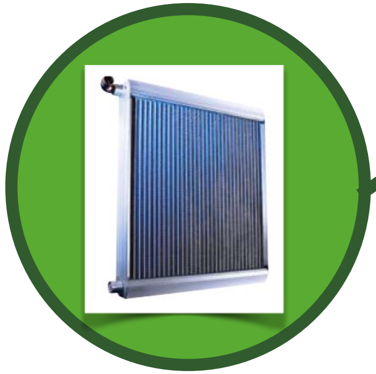
Recuperator

Counter-flow heat exchanger that increases the system efficiency by reusing energy in exhaust sCO_2 from turbine



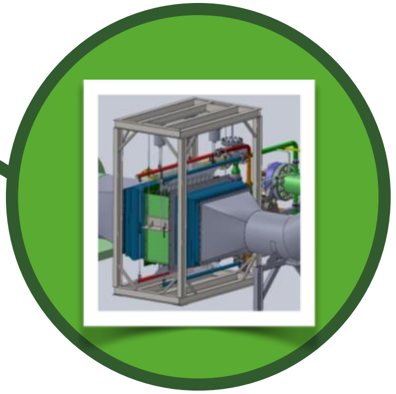
Cooler

Heat exchanger that utilizes water (counter flow) or air (cross flow) to cool sCO_2 for compression



Heater

Cross-flow counter current heat exchanger that takes combustion gas to heat sCO_2 to high temperature



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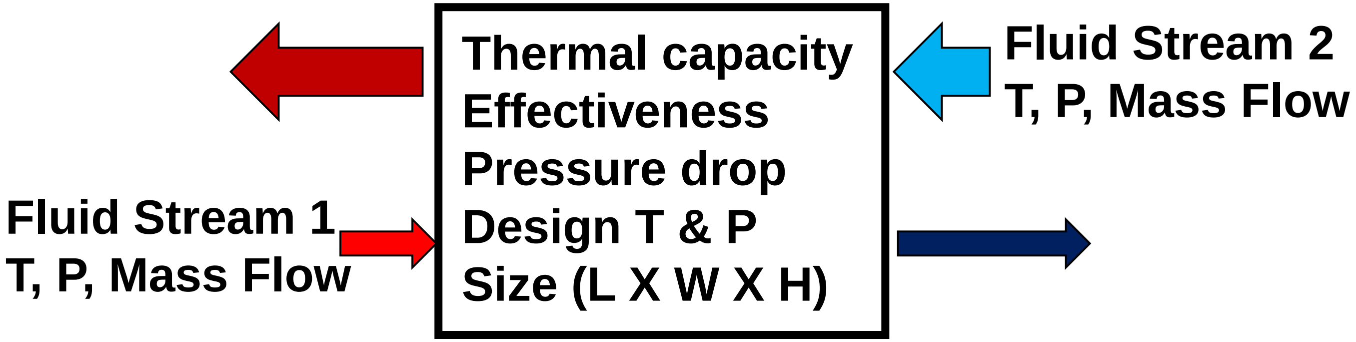
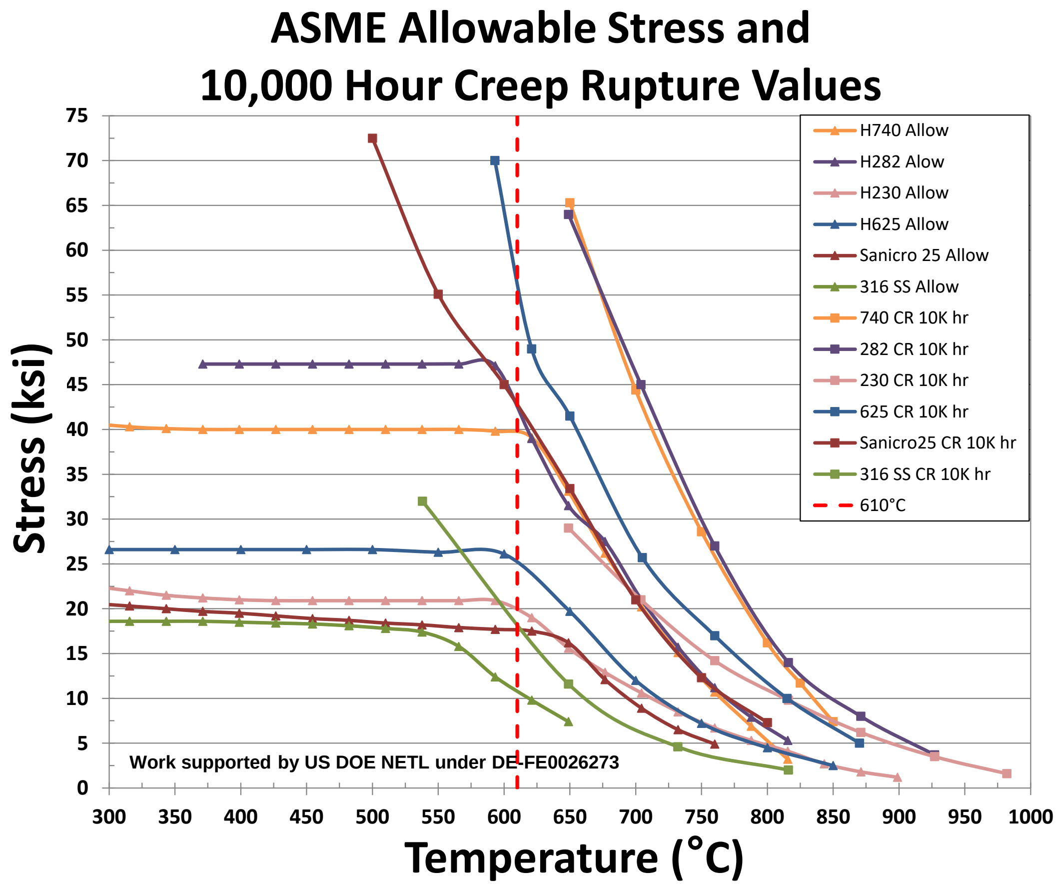
HX design specification questions

Goal: Meet performance requirements and provide margin of safety while minimizing over design.





Material of Construction

- Physical Properties
- Corrosion
- Contamination potential

Design T** increases **material strength drops & corrosion rates increase

ASME Code Stamp/Design

Design T ↑ ***$*** ↑

Fouling Factor

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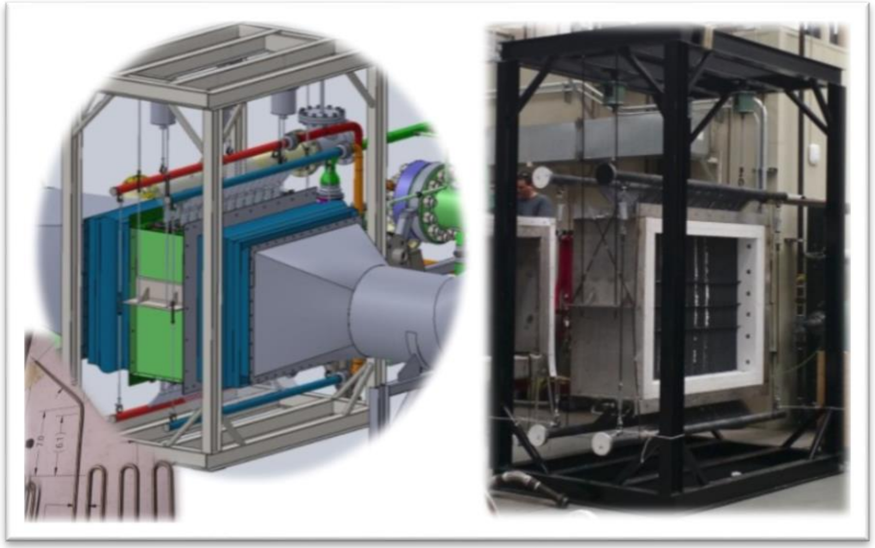
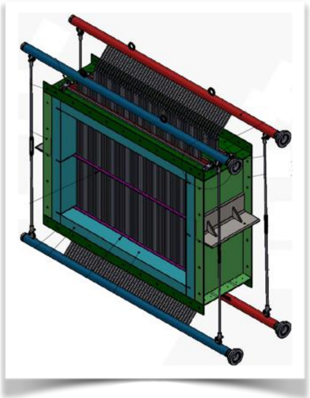
Heater Design Considerations



Design Conditions:

Combustion Gas
Max Temperature: 870°C

sCO_2
Max Temperature:715°C
Pressure: 280 bar



1. **Material Selection**

   High strength at high temperature

   Design to creep/rupture strength rather than yield strength

2. **Corrosion**

   Select materials that can stand carbon corrosion and combustion gas corrosion

3. **Thermal Expansion**

   Design the structure to allow free thermal expansion under high temperature

4. **Air Side Pressure Drop**

   Air side pressure drop has to be under limit to ensure overall efficiency

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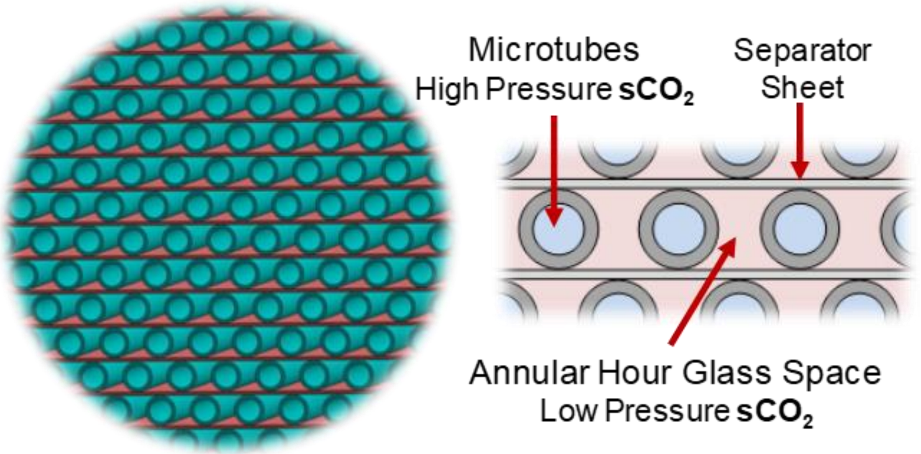
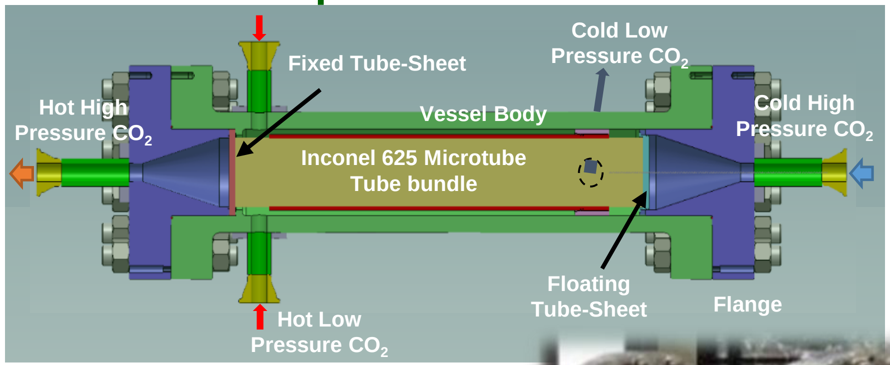
Recuperator Design Considerations



Flanged Pressure Vessel

Horizontal Separators - Counter-current



Design Conditions:

Max Temperature: 575°C

Pressure: 280 bar / 100bar

1 Material Selection

- Nickel-Alloys to hold pressure under high temperature. (Inconel 625 / Stainless Steel 316H)
- Design to yield strength or creep/rupture strength, depending on the metal and the design conditions
- Carbon corrosion resistant

2 Thermal Expansion

Design the structure to allow free thermal expansion under high temperature, such as floating head

3 High Efficiency

Recuperator has to have high efficiency (>90%) to maximize the efficiency of the whole cycle

4 Lower Cost

Reduce capital investment

5 Easy Maintenance

Replaceable tube bundle and removable end cap

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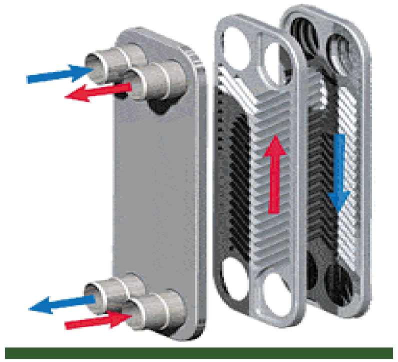
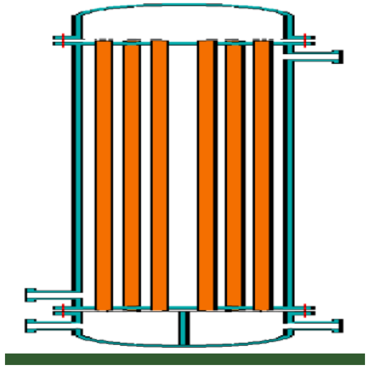
Cooler Design Considerations

Finned-Tubes
Air cool

Micro-channel
Air cool



Brazed-Plate
Water cool



Microtube
Water cool

Design Conditions:
Max Temperature: up to 100°C
Pressure: 100bar

1 Material Selection

More flexible due to low temperature. No one material is perfect for all applications. Tradeoffs in cost vs. reliability depends on water quality.

2 Corrosion and Erosion

Apart from corrosion issue, erosion should also be taken into account.

3 Easy Maintenance

Water-cool heat exchanger requires regular maintenance.

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2

Benefit of Compact Microtube HX

- High Performance
- Smaller Footprint
- Lighter Weight

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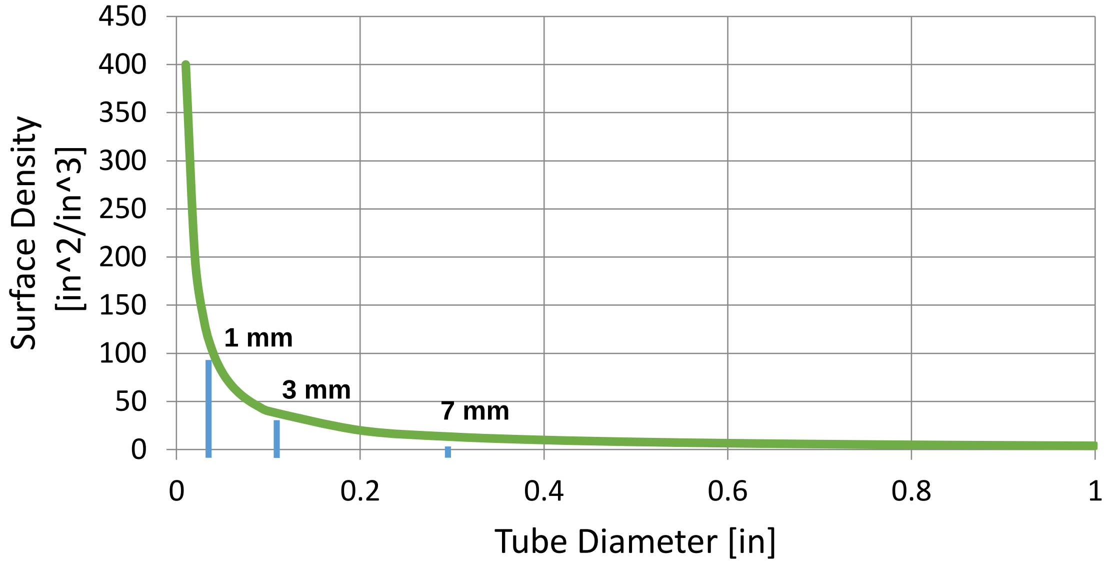
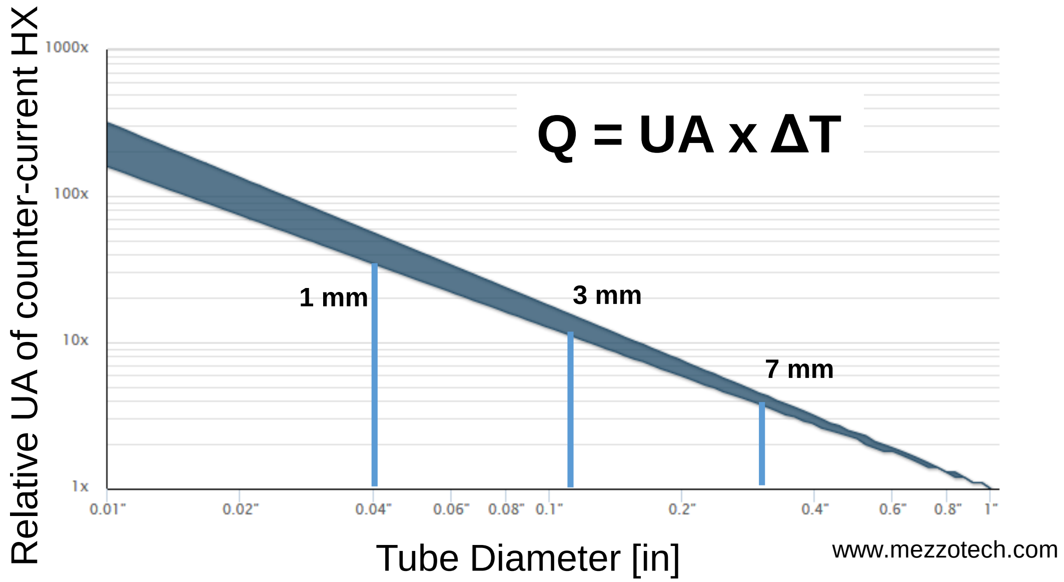
Microtube Improves Performance

Surface Density and Heat transfer coefficient of heat exchangers are significantly improved by using microtube





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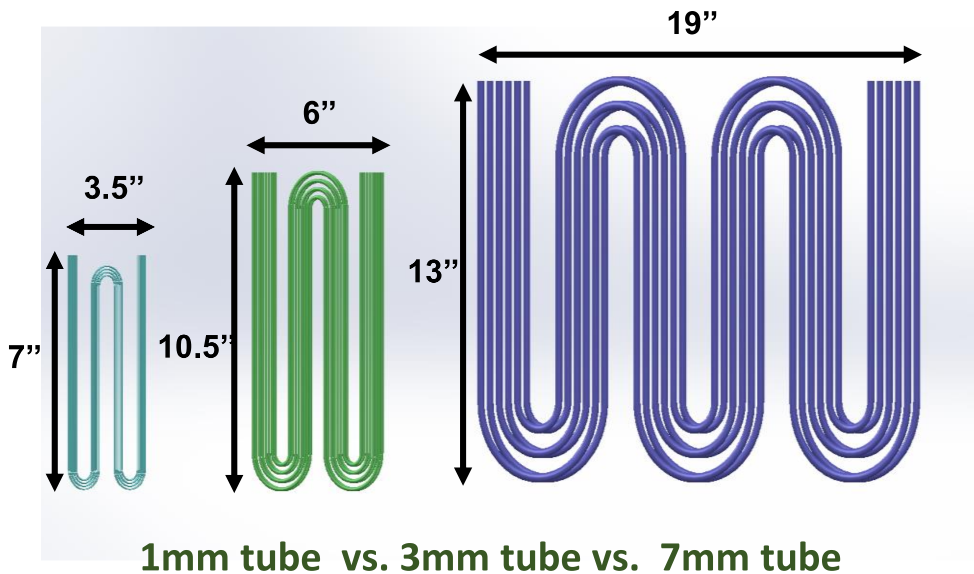
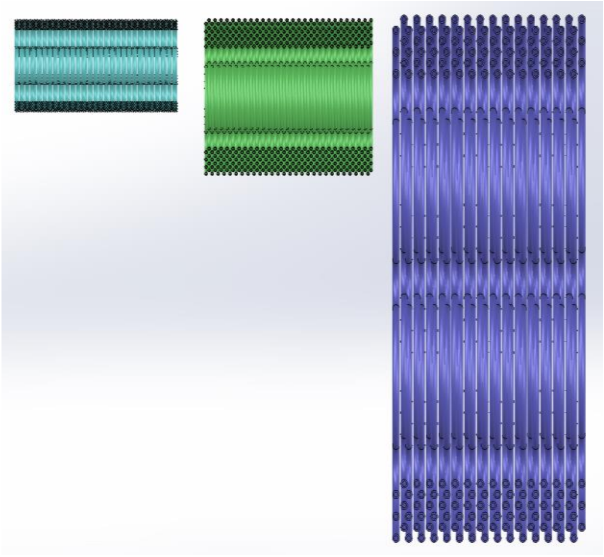
Cross Flow, Counter-current Microtube Heater



1mm tube vs. 3mm tube vs. 7mm tube

Figs shows the overall size comparison of microtube and conventional tube air to CO_2 cross flow heat exchangers with different tube sizes with the same capacity, effectiveness and air side pressure drop

	1mm	3mm	7mm
Total Tube Length	16,800''	9,240''	7,020''
Tube Number	600	220	90
Bundle Weight	4.5 lb	20 lb	90 lb
Surface Density	46 in^2/in^3	17 in^2/in^3	7 in^2/in^3
Efficiency	89%	89%	89%



1mm tube vs. 3mm tube vs. 7mm tube

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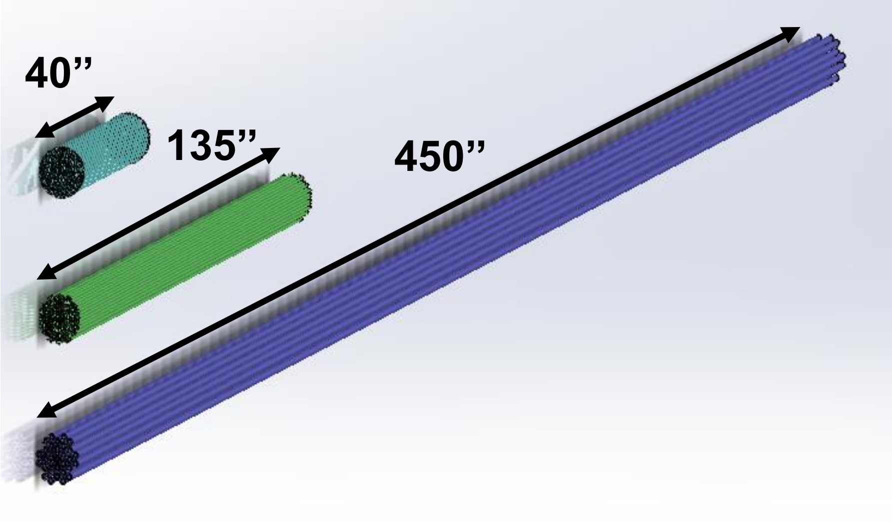
Counter Flow Microtube Recuperator



1mm tube vs. 3mm tube vs. 7mm tube

Figs shows the overall size comparison of microtube and conventional tube counter -current heat exchangers with different tube sizes with the same capacity, effectiveness and pressure drop

With the same performance, microtube counter-current heat exchanger is much more compact and lighter in weight

	1mm	3mm	7mm
Tube Length	40''	135''	450''
Tube Number	1500	175	30
Bundle Weight	17 lb	59 lb	244 lb
Surface Density	76 in^2/in^3	30 in^2/in^3	12 in^2/in^3
Efficiency	97%	97%	97%

1mm tube vs. 3mm tube vs. 7mm tube

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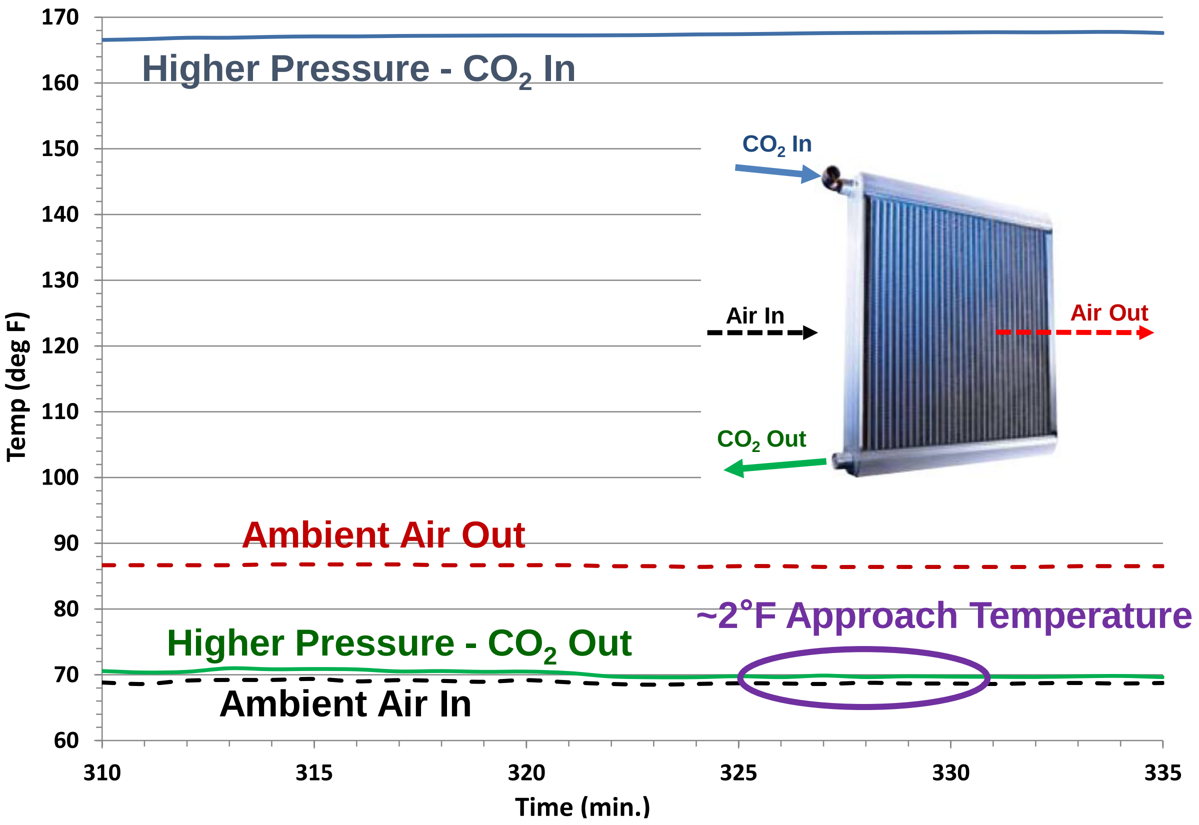
Mircotube/Micro-channel Cooler



At Thar's test facility, air and CO_2 approaching temperature as low as 2°F was achieved using micro-channel coil.

Micro-channel coils are generally 40% smaller, 40% more efficient, and use 50% less refrigerant than standard tube and fin coils. Air side pressure drop is also lower

3

sCO_2 HX Model Selection

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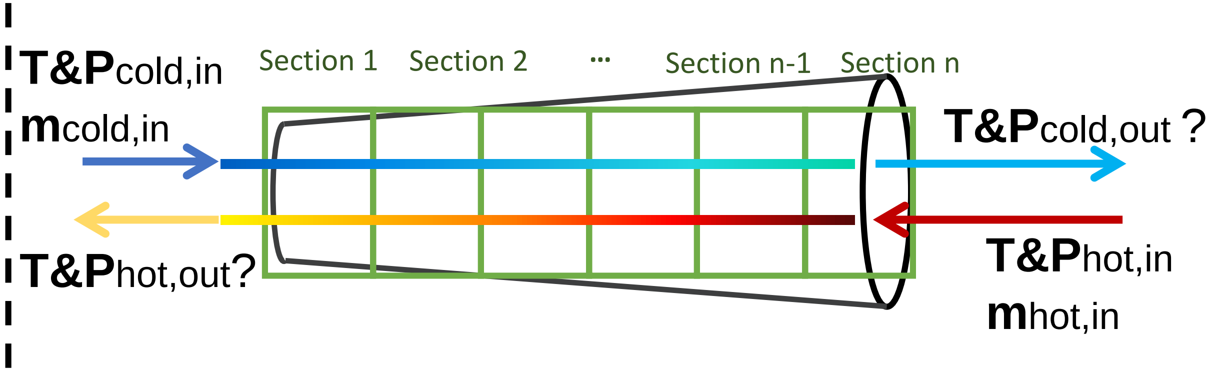
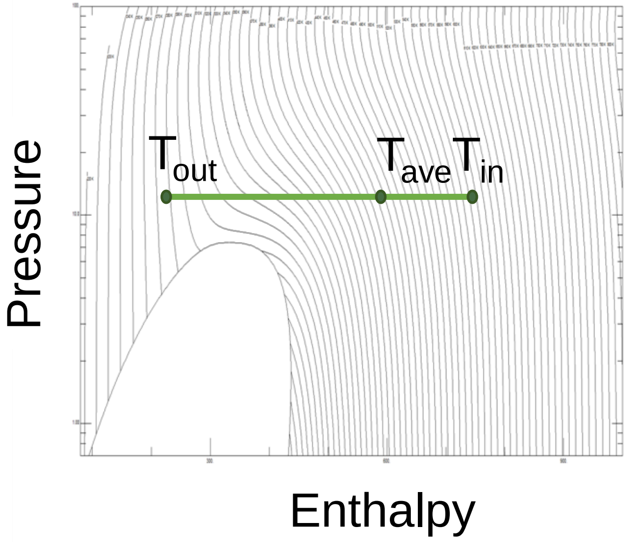
Heat Exchanger Calculation Method

CO_2 properties change dramatically with little variation in supercritical region

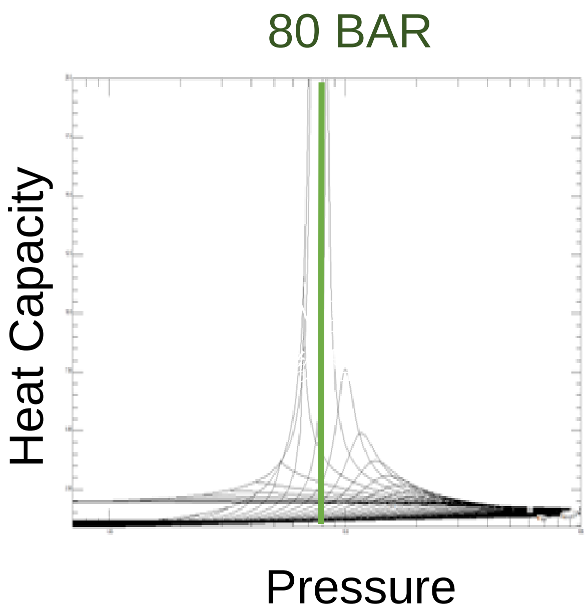
Used discretized model for sCO_2 heat transfer calculation

- Break the heat exchanger into n sections
- Calculate average properties of each section
- Interactively calculate the overall performance

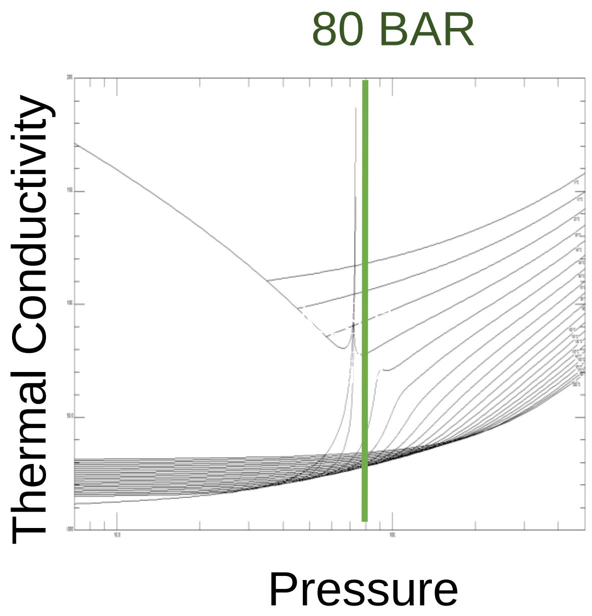




CO_2 P-H Diagram



CO_2 Cp-P Diagram



CO_2 K-P Diagram

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Heat Transfer Equations

Models selected from established heat transfer and pressure drop equations for the best accuracy compared to testing data

1 CO_2 Side Nusselt Number

Petukhov (1970)

$$Nu_{CO_2} = \frac{\left(\frac{f}{2}\right) \times Re \times Pr}{1.07 + \frac{900}{Re} - \frac{600}{1+10\mathrm{Pr}} + 12.7\left(\frac{f}{2}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}} - 1\right)}$$

2 Air Side Nusselt Number

Martin (2002)

$$Lq = \begin{cases} 0.92Hg \times Pr_{air}(i) \times \left(\frac{\frac{4X_T}{\pi} - 1}{X_L}\right) & X_L \geq 1 \\ 0.92Hg \times Pr_{air}(i) \times \left(\frac{\frac{4X_T X_L}{\pi} - 1}{X_L X_D}\right) & X_L < 1 \end{cases}$$

$$Nu_{air}(i) = 0.404 \times Lq^{\frac{1}{3}}$$

3 CO_2 Side Pressure Drop

Bhatti and Shah (1987)

$$f = 0.00128 + 0.1143Re^{-0.311}$$

$$\Delta p_{co_2} = f \frac{L}{d_o} \frac{\rho \mu^2}{2}$$

4 Air Side Pressure Drop

Zukauskas (1988)

$$\Delta p_{air} = N_r X \left(\frac{\rho {u_m}^2}{2}\right) f$$

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4

sCO_2 HXs Test Loop and Data

Thar sCO_2 HX Test Loop vs. a standard sCO_2 Brayton Test Loop

Different from Standard Loop

- Pump used in place of a compressor
- Turbine is replaced by back pressure regulator

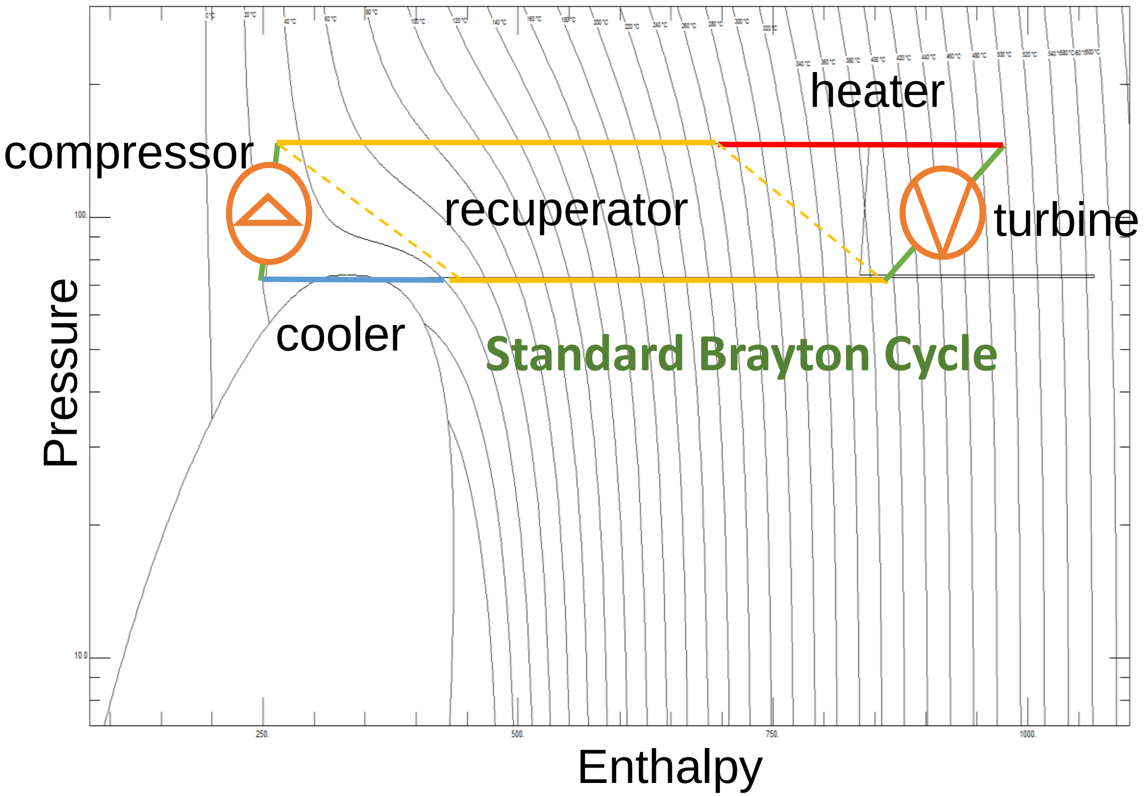
Test Condition

Supercritical Carbon Dioxide

- Operating Pressure: 255bar / 87bar
- Operating Temperature: 570°C

Combustion Gas

- Maximum Temperature: 750°C
- Maximum Flow: 250 scfm @ 750°C



Thar Loop Compares to Standard Brayton Cycle

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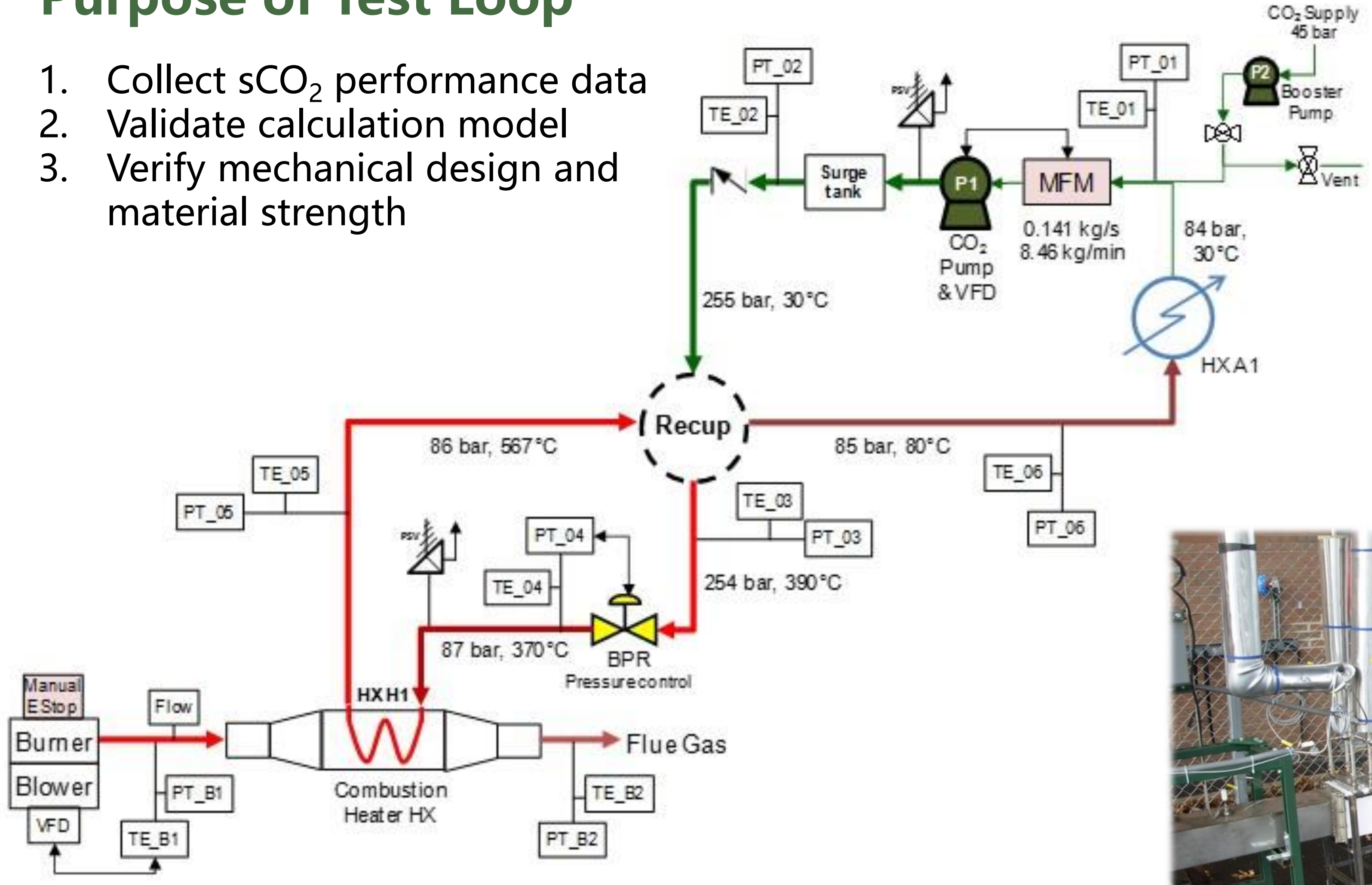
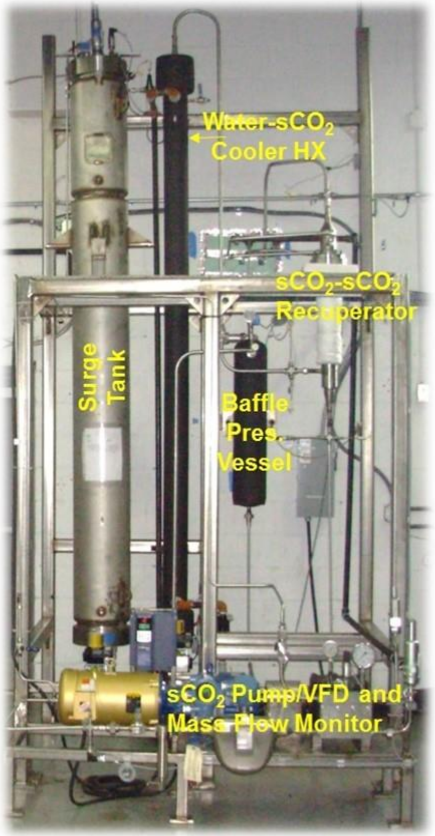
Thar sCO2 HX Test Loop

Purpose of Test Loop

1. Collect sCO_2 performance data
2. Validate calculation model
3. Verify mechanical design and material strength





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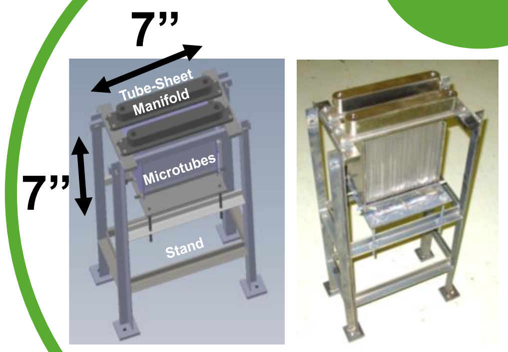
Three Heat Exchangers in Loop

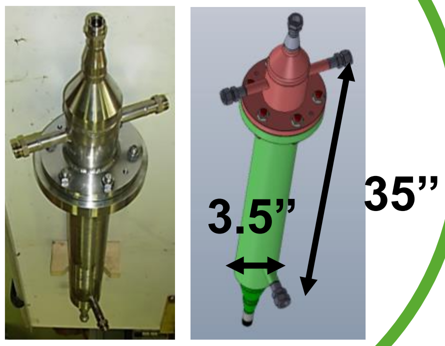
Heater



Microtube Cross-flow, Counter-current Heater

- Material: Inconel 625
- Design Max Temperature: 750°C
- Design Max Pressure: 280 bar

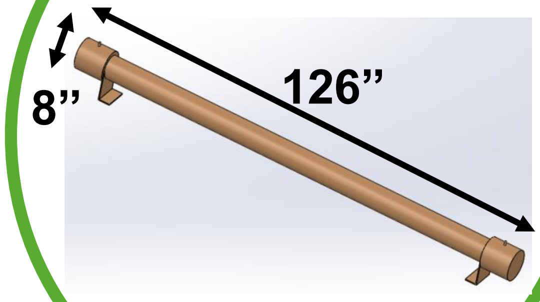
Recuperator



Microtube Counter-current Recuperator

- Material: Inconel 625 & SS 316H
- Design CO_2 High Side Pressure: 280 bar
- Design CO_2 Low Side Pressure: 100 bar
- Maximum Temperature: 575°C

Cooler



Counter-flow, Shell & Tube Water Cooler

- Material: Stainless Steel 304
- Design CO_2 Pressure: 100 bar

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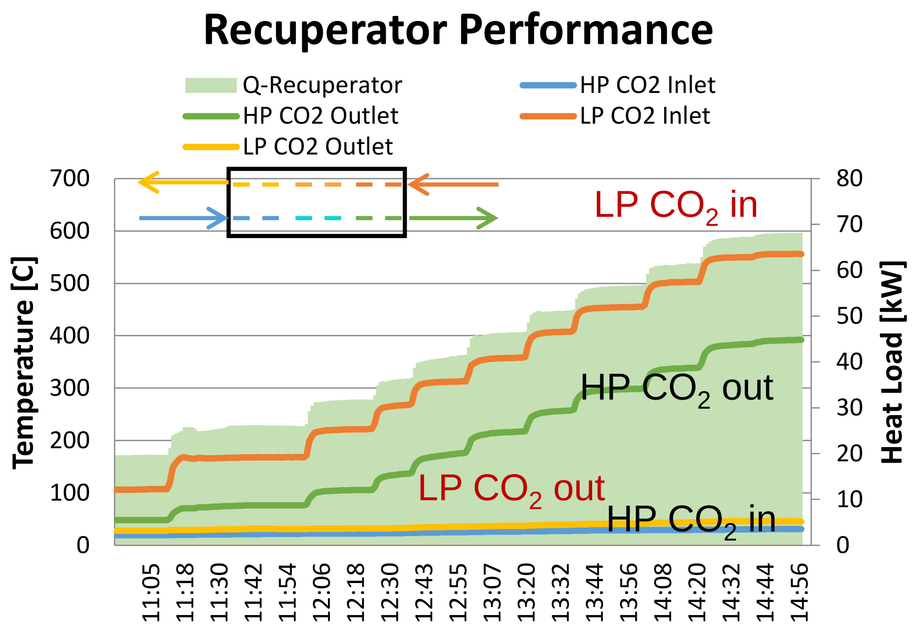
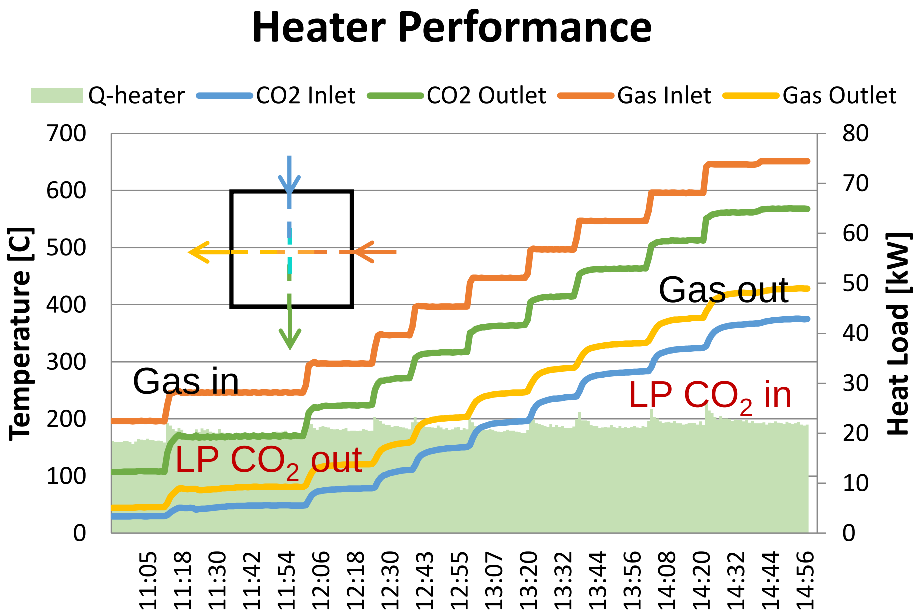
Data Collected

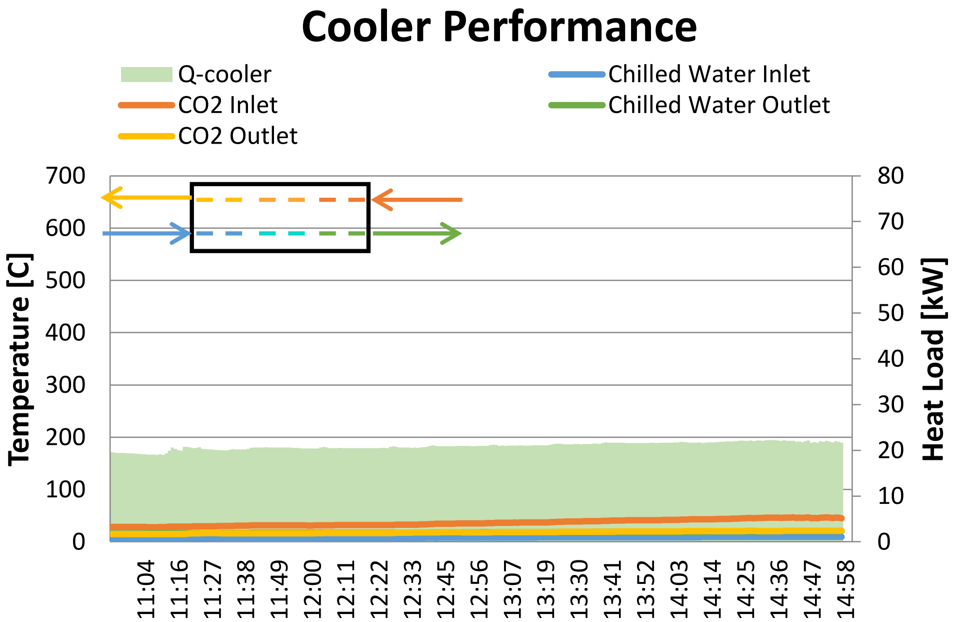
Test Condition

- CO_2 flow rate: 7kg/min
- Low Pressure Side: 75bar
- High Pressure Side: 150 bar
- Blower at 40 Hz constant

Figures show effect of increasing combustion gas temperature on heat exchangers' performance







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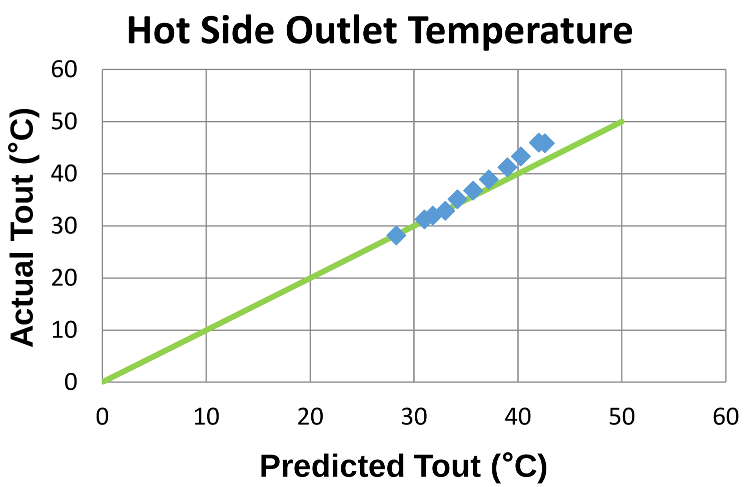
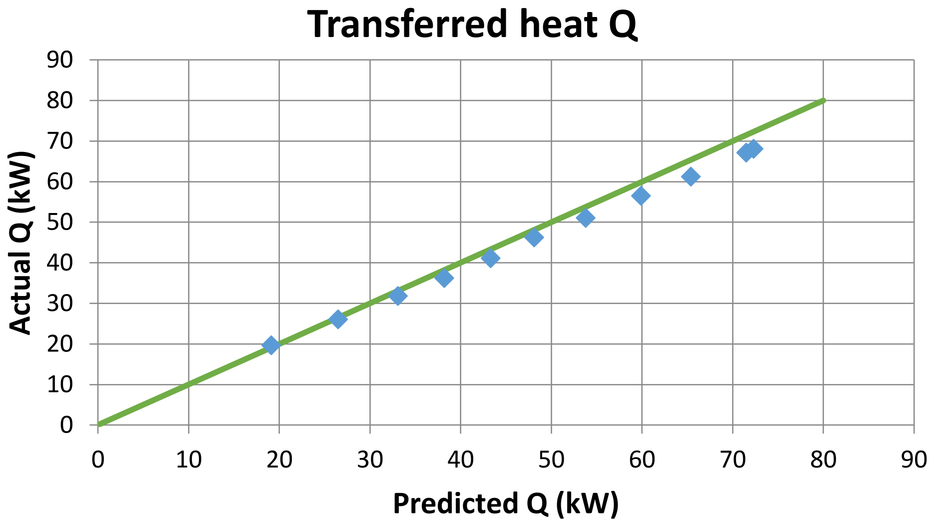
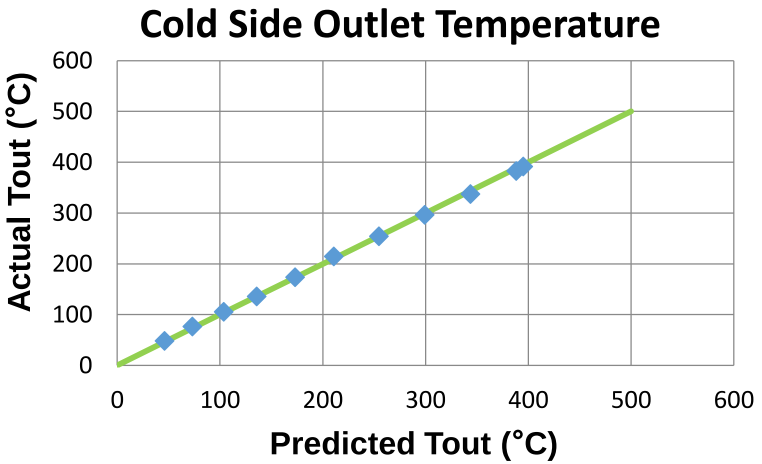
Comparison of Actual data vs. Prediction: Recuperator







Good linear relationship between actual data and calculated data

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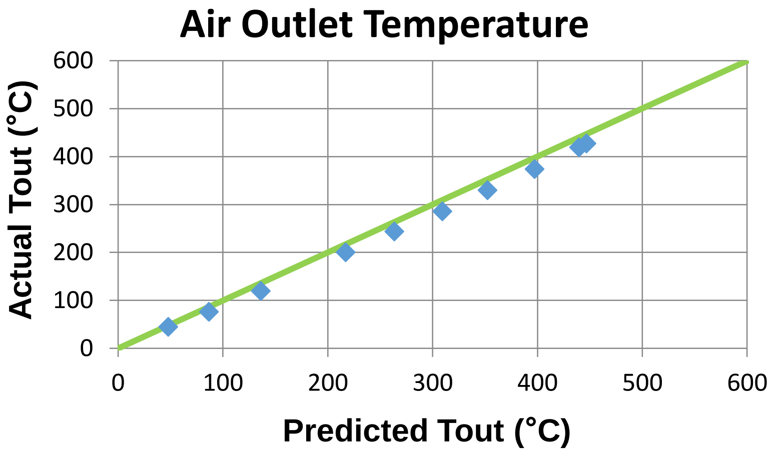
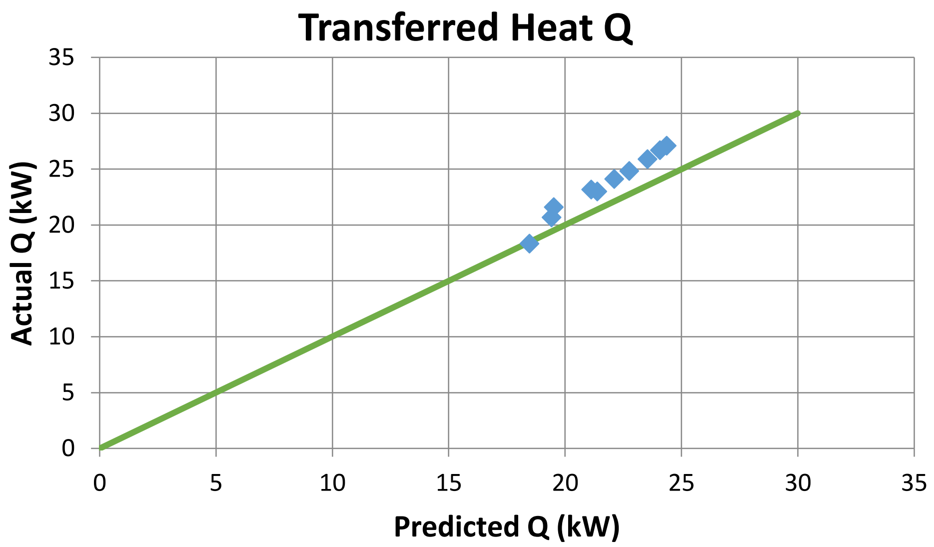
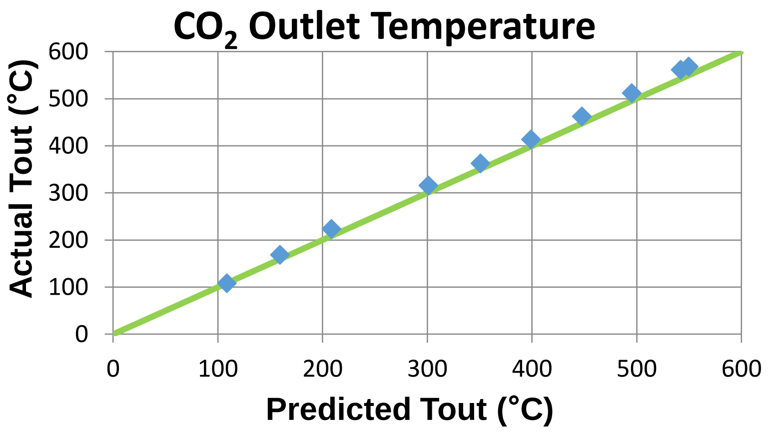
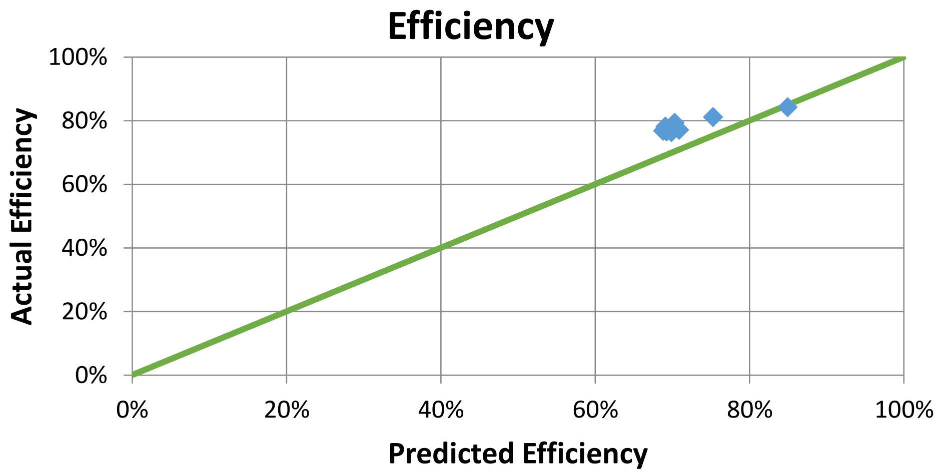
Comparison of Actual data vs. Prediction: Heater









Good linear relationship between actual data and calculated data
Actual performance ~10% better than prediction

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5

Summary

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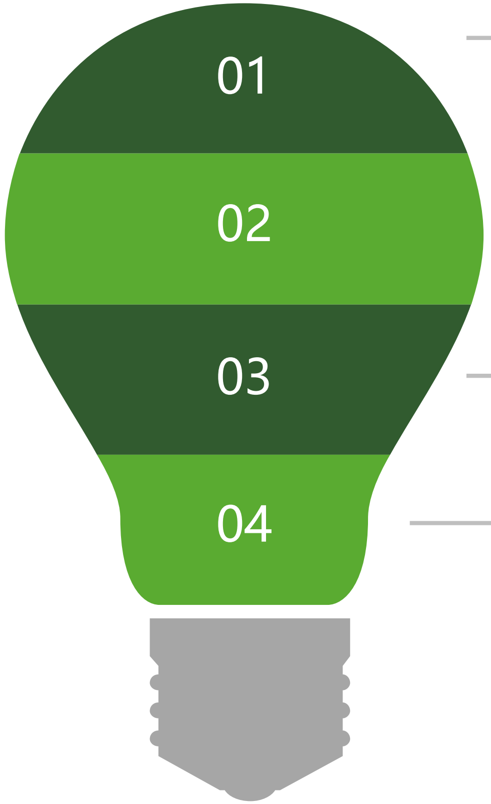
Summary



1 HXs Design Considerations

- Select high strength and corrosion resistance material
- Consider creep/rupture strength at high temperature
- Allow for thermal expansion
- Efficiency, cost, maintenance...

2 Use of Microtube

- Significantly improve thermal performance
- Smaller footprint and lighter

3 Heat Exchanger Calculation Model

- Discretized model increases accuracy
- Establish relationship between models and data

4 sCO_2 Brayton Cycle Testing Data

- Microtube heat exchangers were successfully evaluated at Brayton cycle T & P conditions
- Test data confirms sCO_2 microtube heat exchanger performance
- Good correlation between design & actual heat exchanger performance data

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THANK YOU!
QUESTIONS?

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Heat Exchanger Types

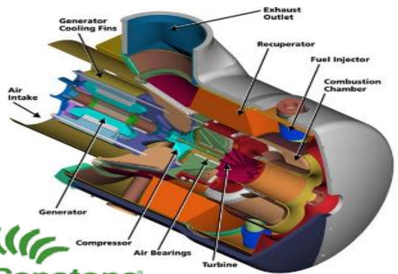
Continued

Shaun Sullivan

BraytonEnergy

sullivan@braytonenergy.com

Plate-Matrix Heat Exchangers – An Overview

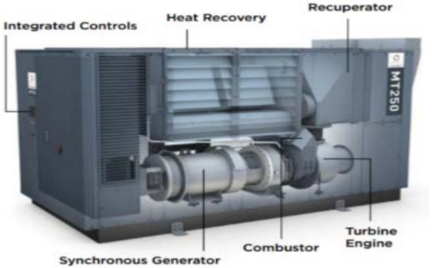


Capstone

(30, 65, and 200 kW)

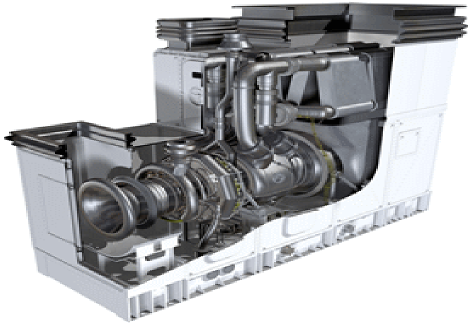
Ingersoll Rand

(70 and 250 kW)



FLEXENERGY

(250 and 333 kW)

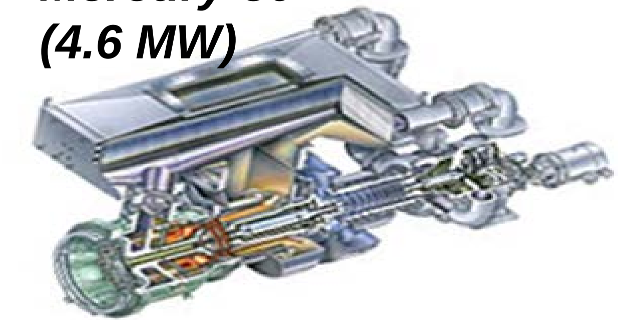


Rolls-Royce

WR-21 (25.2 MW)

Solar Turbines
A Caterpillar Company

Mercury-50
(4.6 MW)



The Plate-Matrix Unit Cell

External low-pressure matrices

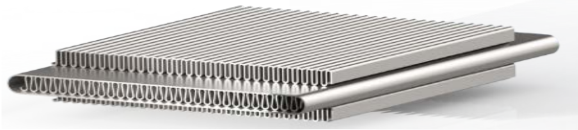
- Enhances the heat transfer of the low-pressure fluid as it flows between adjacent unit cells

Internal high-pressure matrix

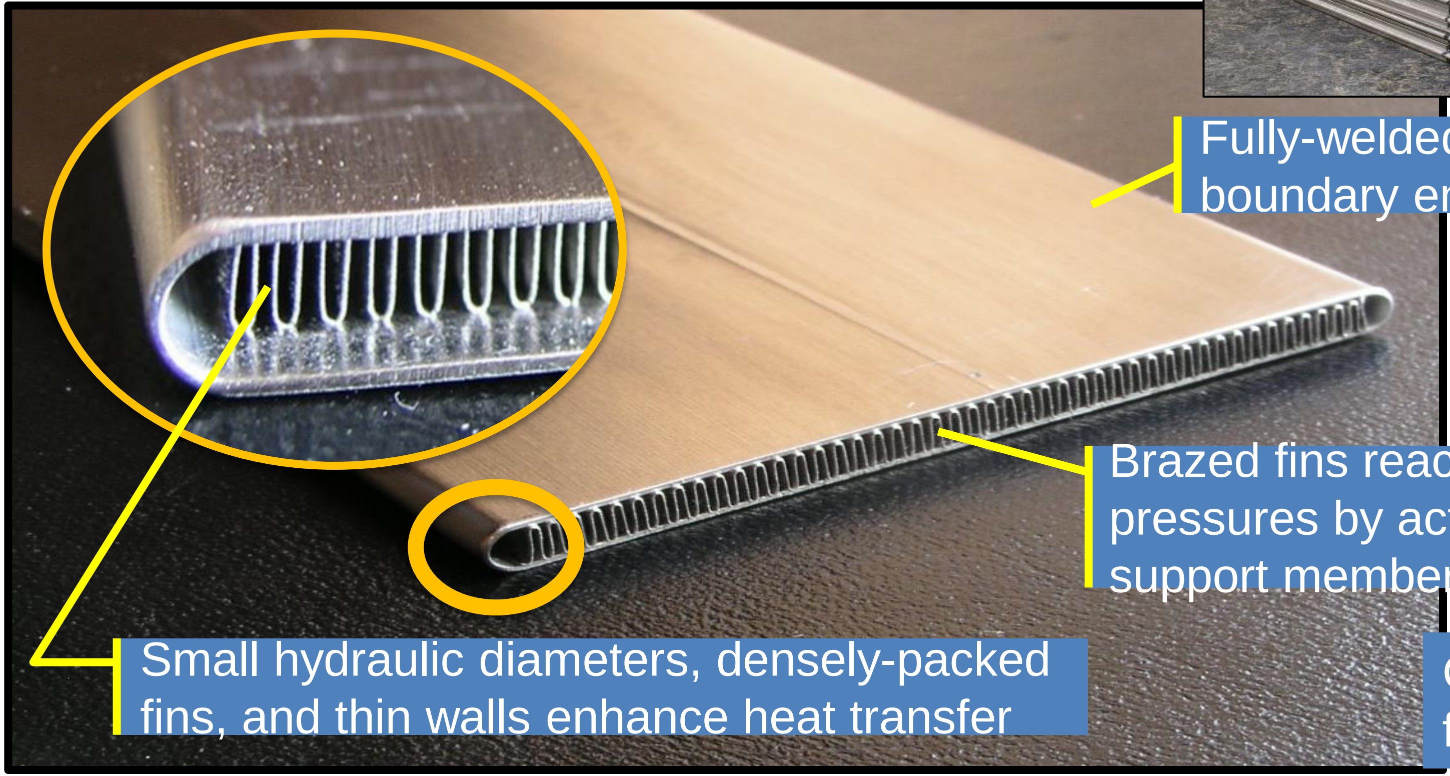
- Enhances the heat transfer of the high pressure fluid as it flows between the two parting plates
- Can serves as structural features for high-pressure (sCO_2) applications

Parting plates

- provide fluid boundary between the two flows



Unit Cell Design



Fully-welded pressure boundary ensures sealing

Individually tested for quality control

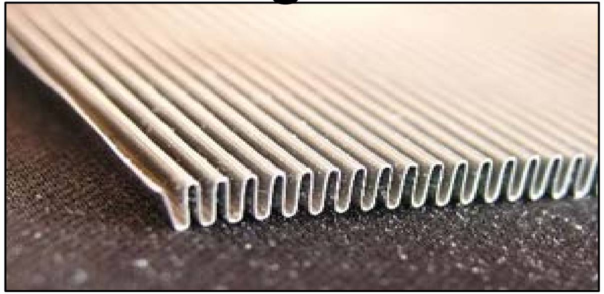
Brazed fins reac pressures by acting as tensile support members

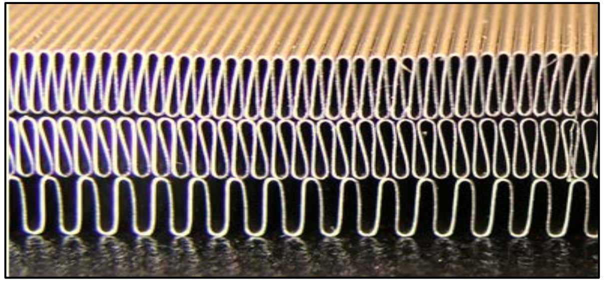
Small hydraulic diameters, densely-packed fins, and thin walls enhance heat transfer

Customizable fin geometry

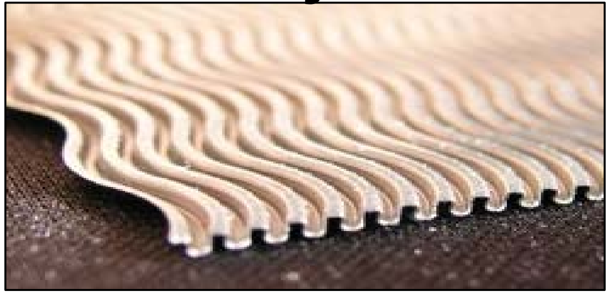
Heat Transfer Matrices

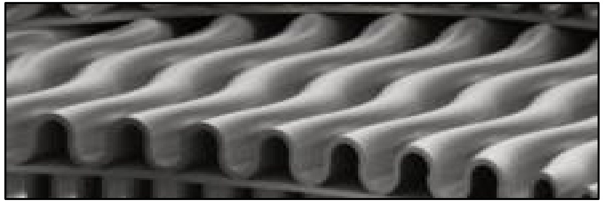
Straight Fin



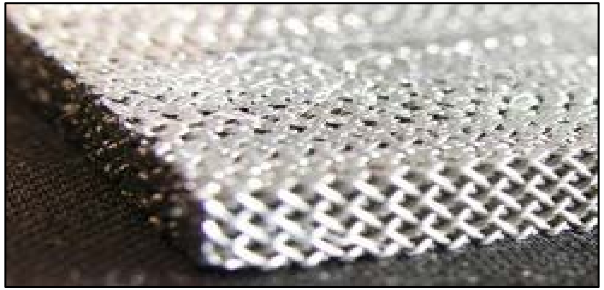


WavyFin



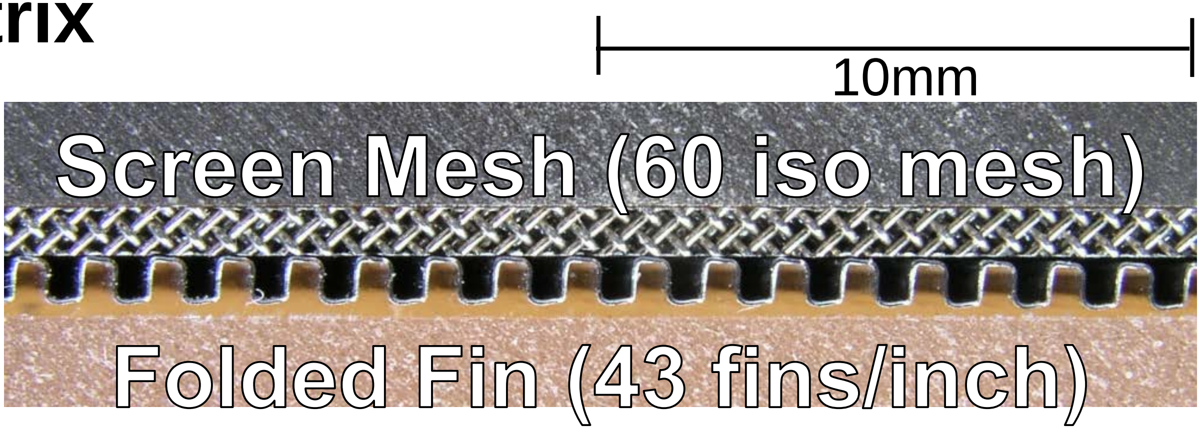


Wire Mesh



Choosing a Matrix

- Cost
- Mass
- Footprint
- Size (Volume)



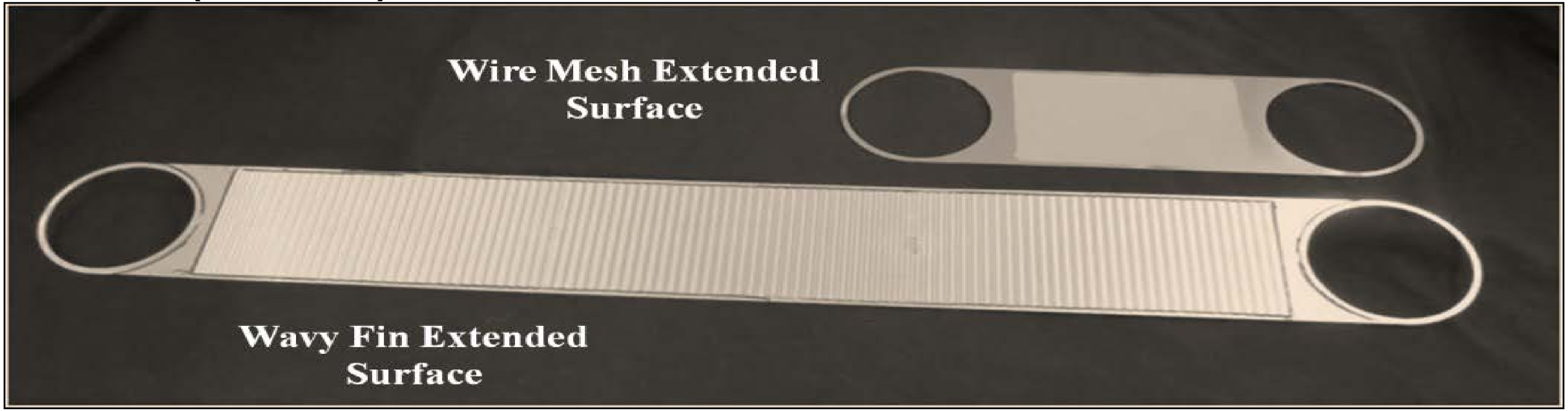
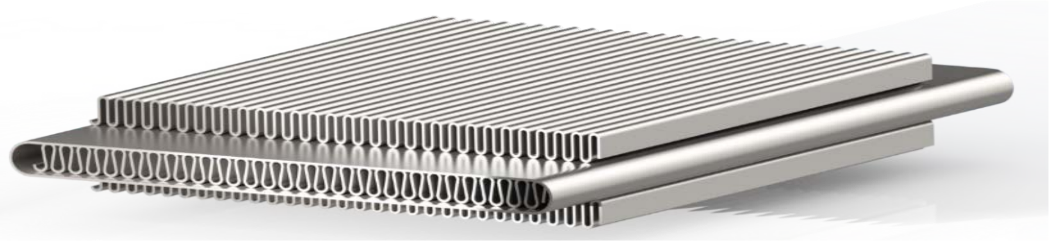


Plate-Matrix Heat Exchangers

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)

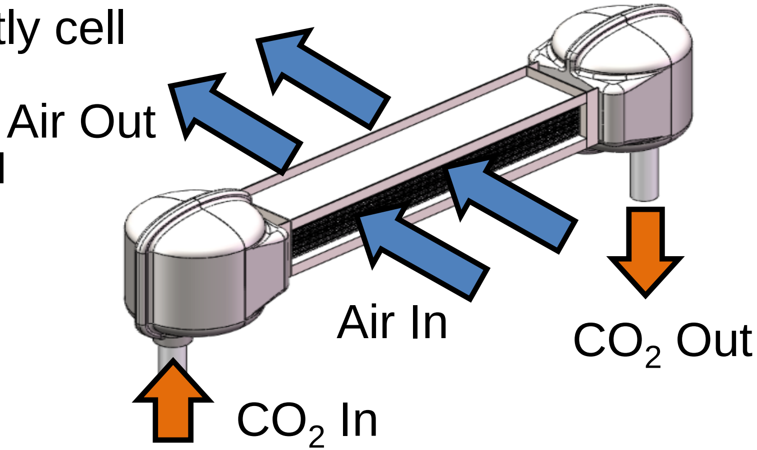


Plate-Matrix Heat Exchangers

Plate-Matrix Heat Exchanger Cell Counter Flows

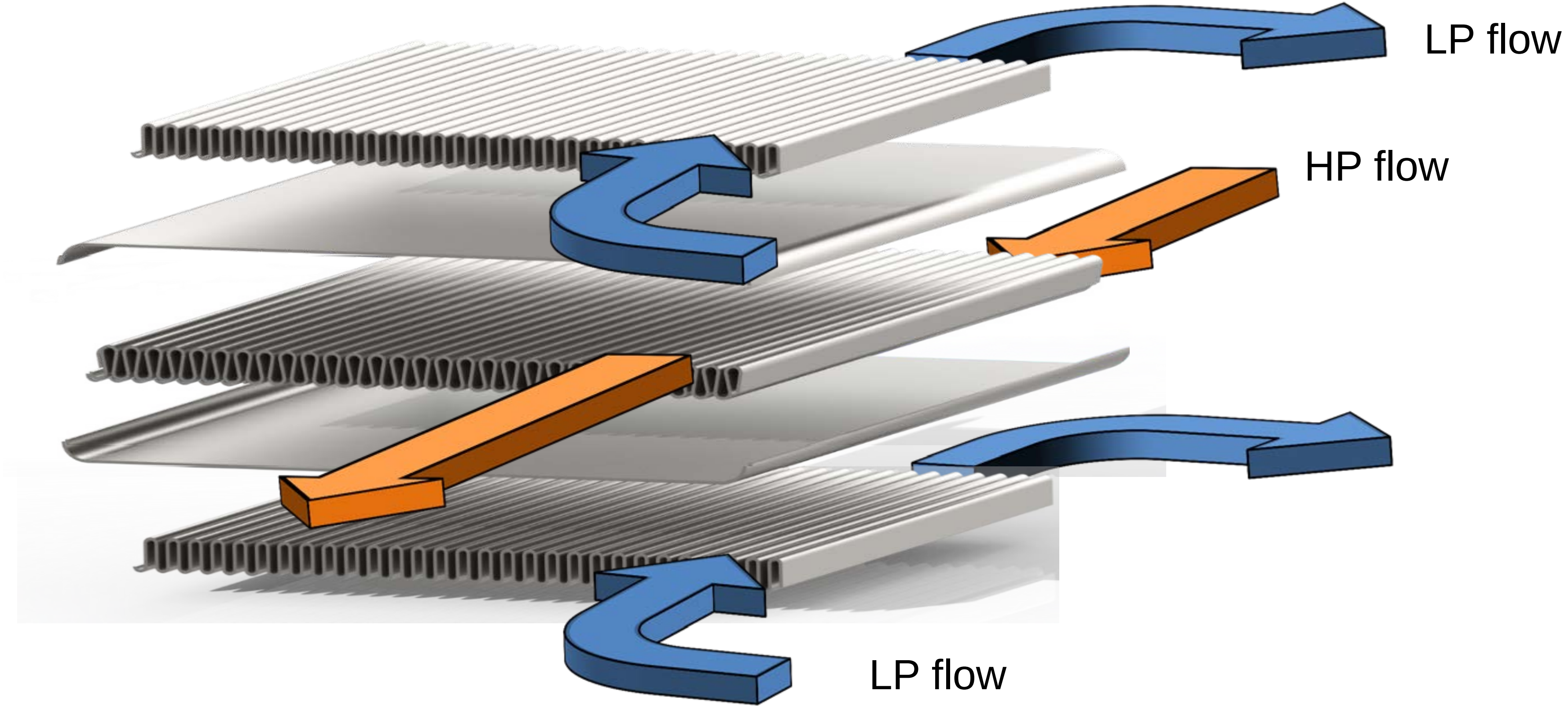
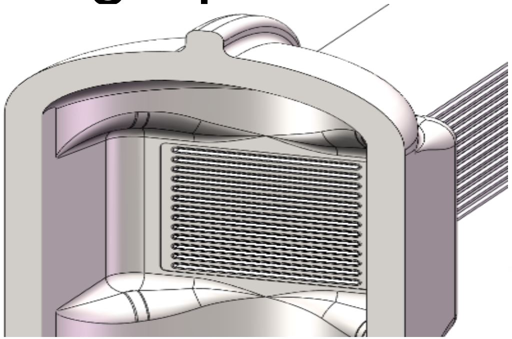


Plate-Matrix Heat Exchanger Manifolds

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



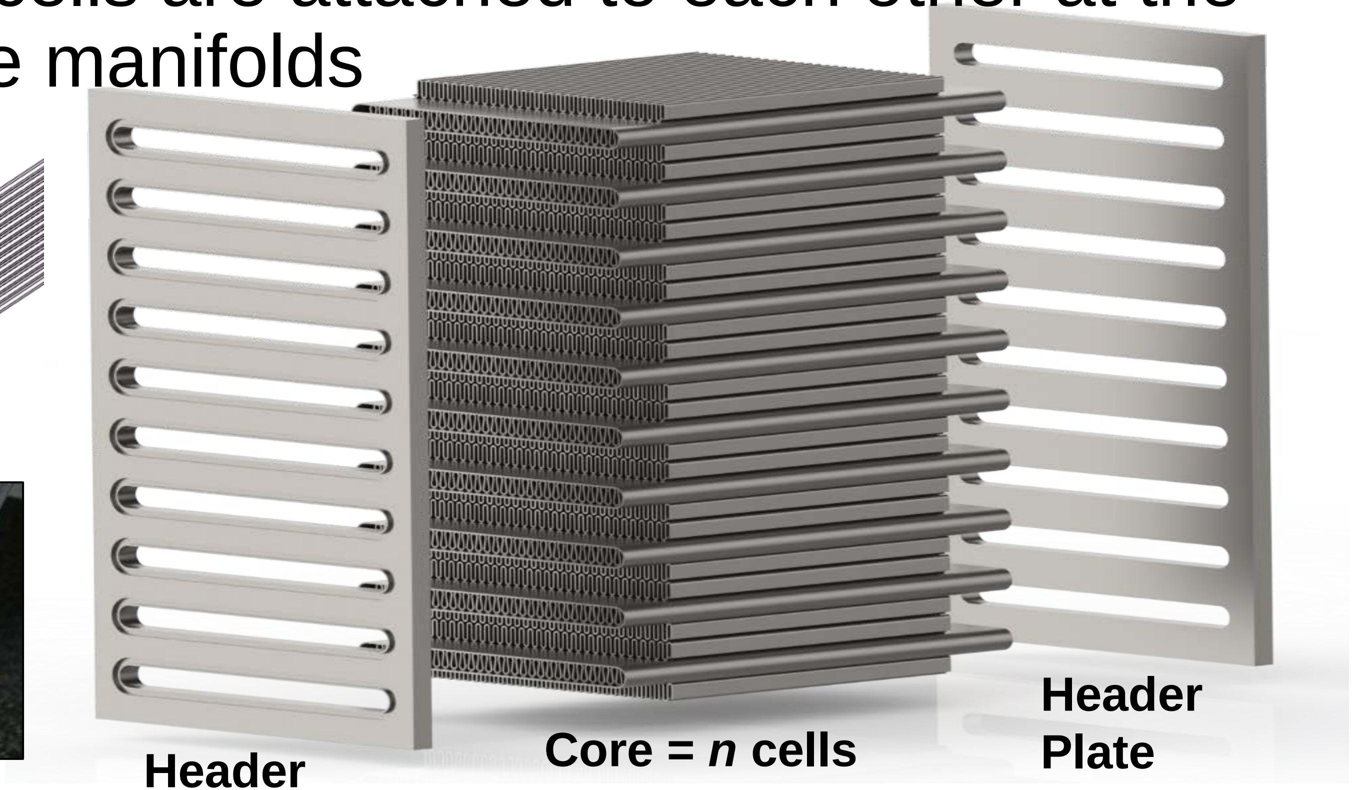
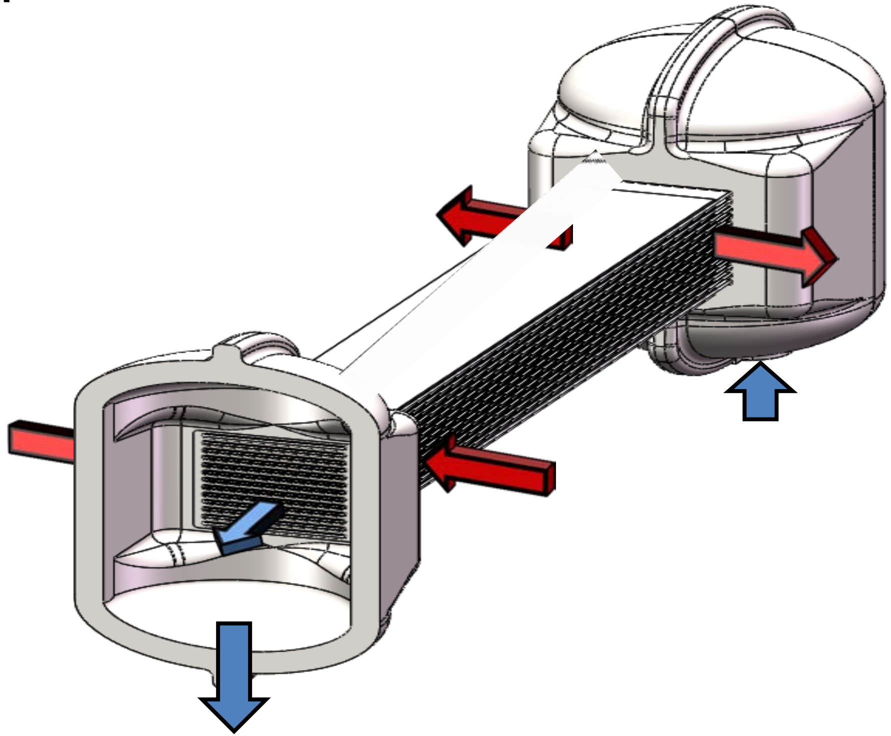


Plate-Matrix Heat Exchanger Cores

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



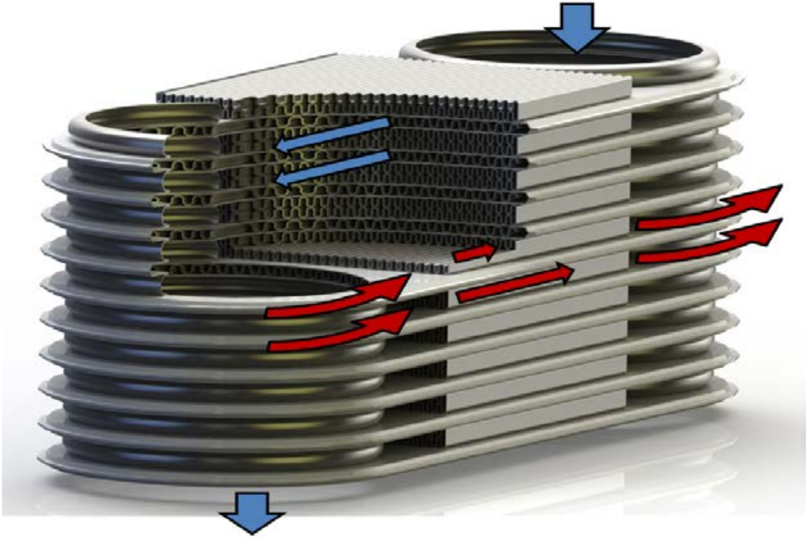


Plate-Matrix Heat Exchangers

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

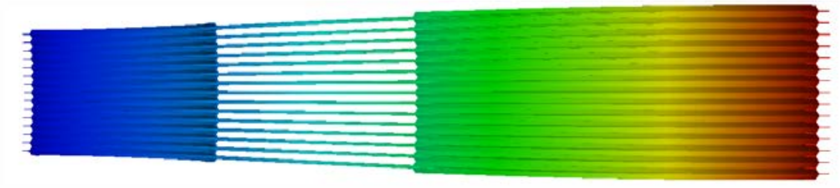
- Alternative high-pressure packaging designs may require ASME qualification

Plate-Matrix Heat Exchangers

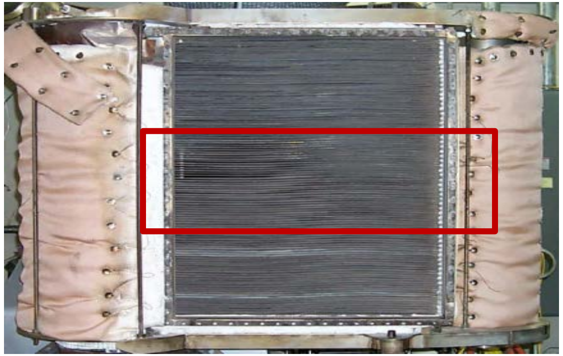
Thermo-Mechanical Strain Tolerance

- 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



Heat Exchanger Mechanical Design and Validation for S-CO_2 Environments

Shaun Sullivan

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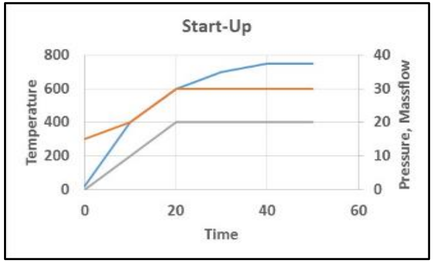
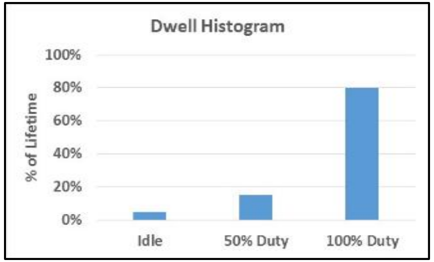
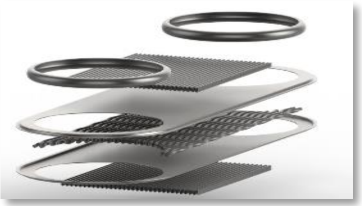
Design Methodology

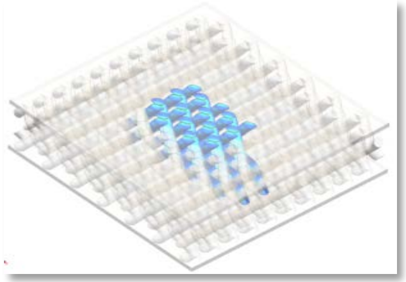
Mission Definition





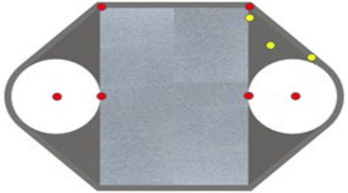
Mechanical Design and Simulations





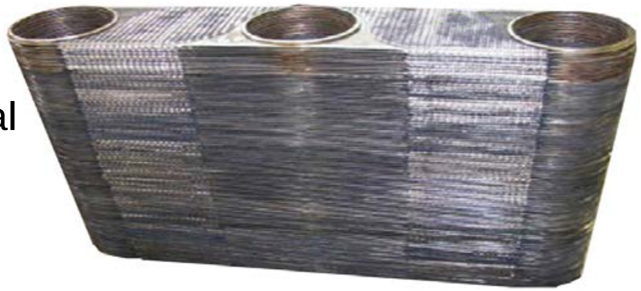
Configured and Processed Materials Characterization

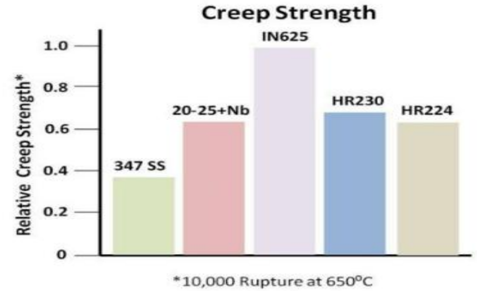
Thermal and Strain Validation & Endurance

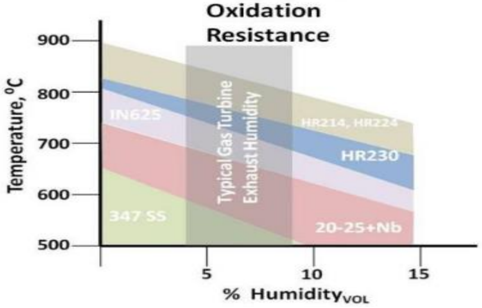


Requirements-to-Design Validation Method

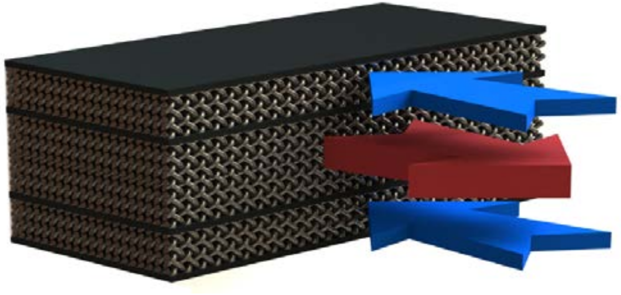
- Specify Requirements in terms of mission profiles
  - Including dwells and transient maneuvers
- Render thermal hydraulic design into mechanical design
- Initial analyses with substrate material properties:
  - temperature
  - stress/strain
  - durability
- Characterize as configured/processed materials 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.







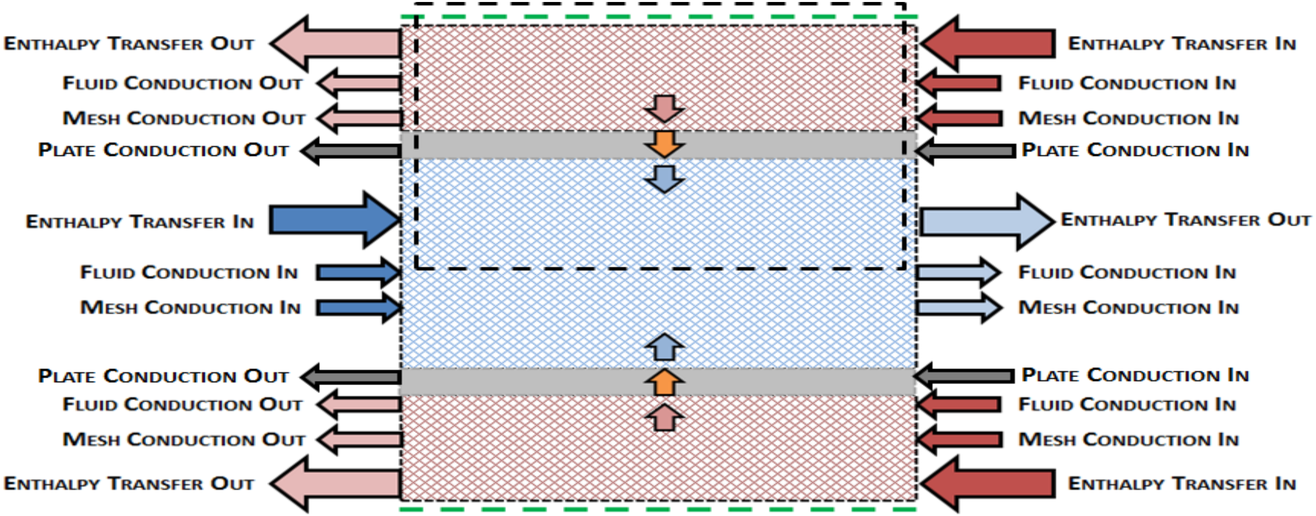
Heat Transfer Modeling



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

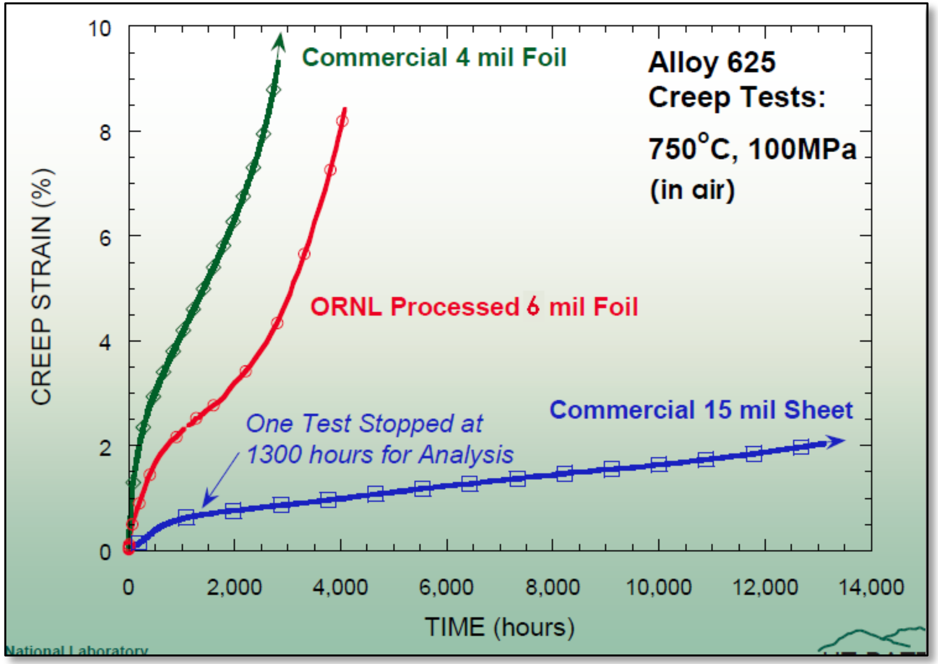
Enthalpy change is used to calculate the heat gain (or loss) so as to capture the significant pressure dependence of the internal energy of the fluid

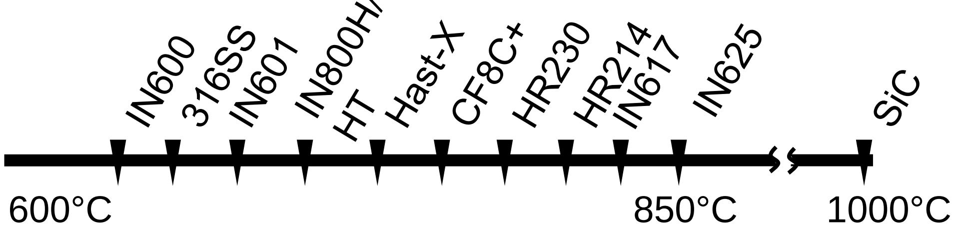
- Axial conduction losses – which may be significant in high-ε designs – are captured for both the parent material and the heat transfer enhancing structures



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?)

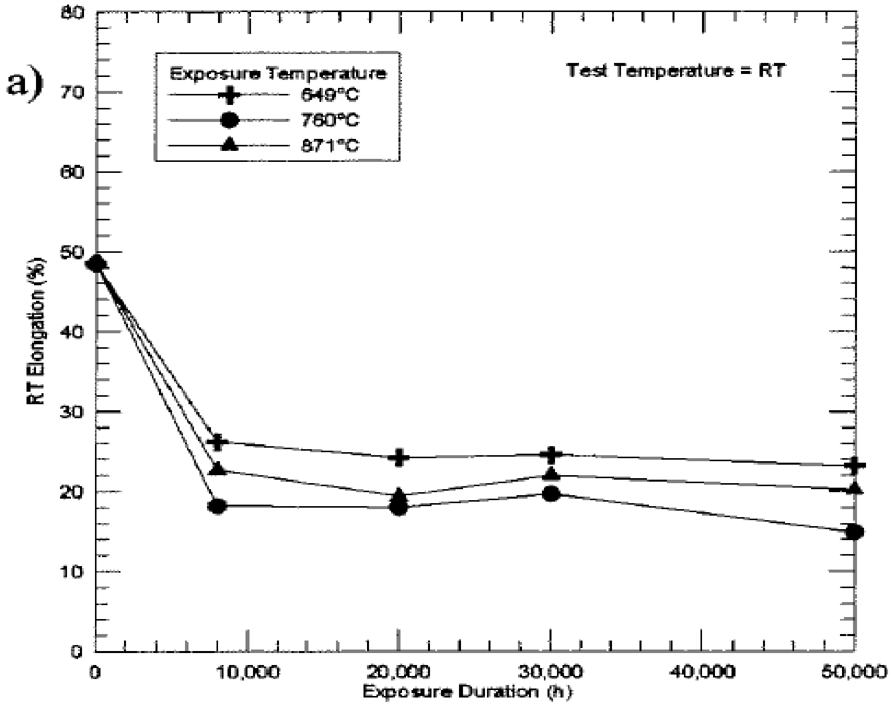




For 10,000 hour creep life at 70 MPa

Fatigue Considerations

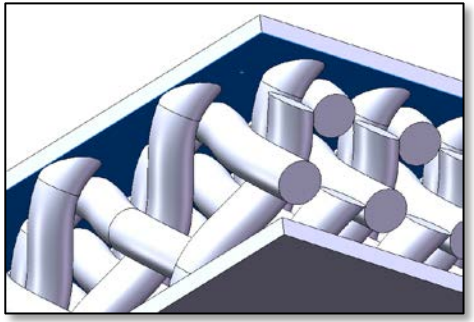
- Highly design dependent gradient selection for ΔT
- Structural compliance
  - Bigger is NOT stronger!
- Thick-thin avoidance
- Stress in weld-heat affected zones.
- Ductility – as processed, after aging

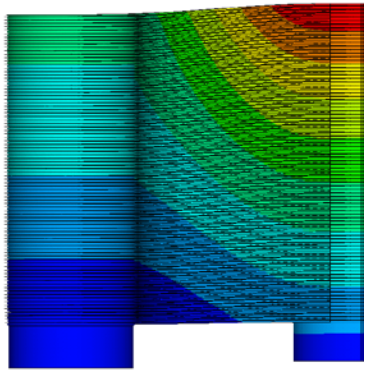


HR120 elongation with exposure at 649, 760 and 871°C. Source: Pike & Srivastava Haynes Int'l

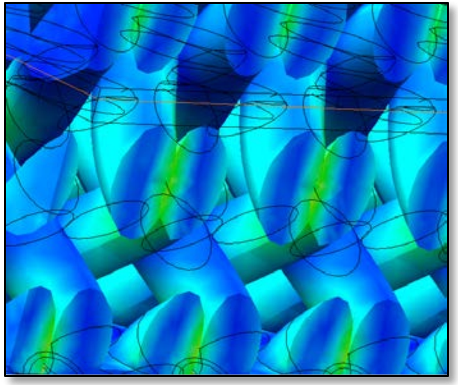
Simulations

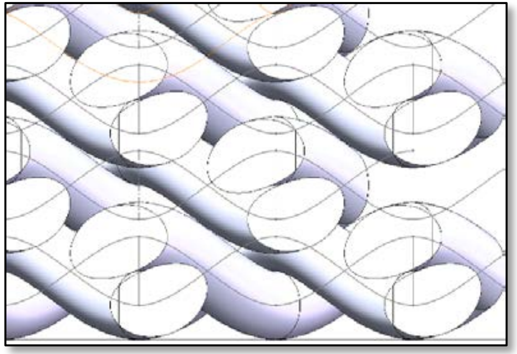
- Conduct thermal and structural FEA to determine temperature, stress, and strain
- Identify ‘control points; - details where damage may accumulate
- Perform initial life analyses to quantify creep, and fatigue





Core strain analysis

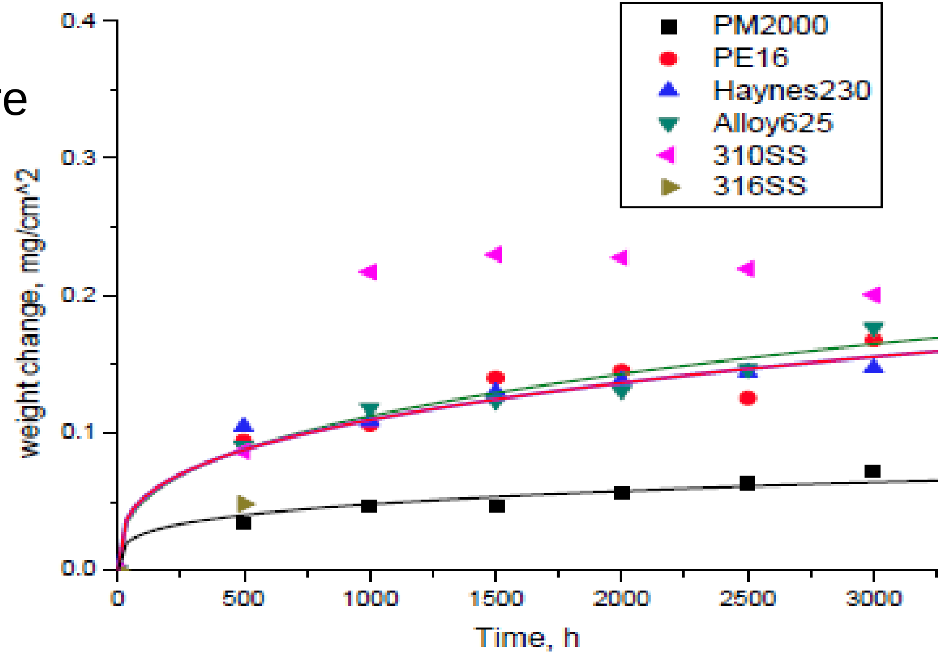




Wire-mesh analysis for creep and pressure-fatigue simulation.

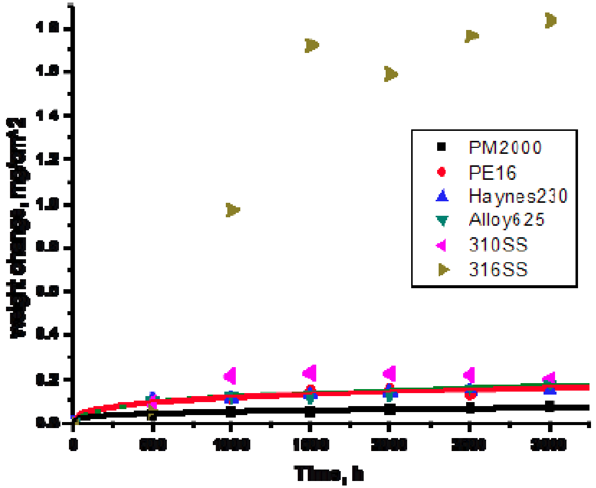
Corrosion Considerations

- 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^{0}C according to Sridharan et al.

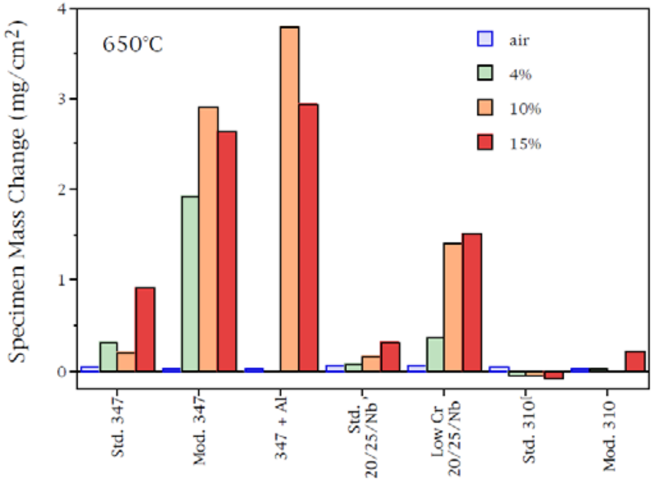


Source: Sridharan, Anderson, et al - University of Wisconsin, sCO_2 Power Cycle Symposium, Boulder, CO 2011

Type 310SS 650°C Oxidation sCO_2 vs. Air



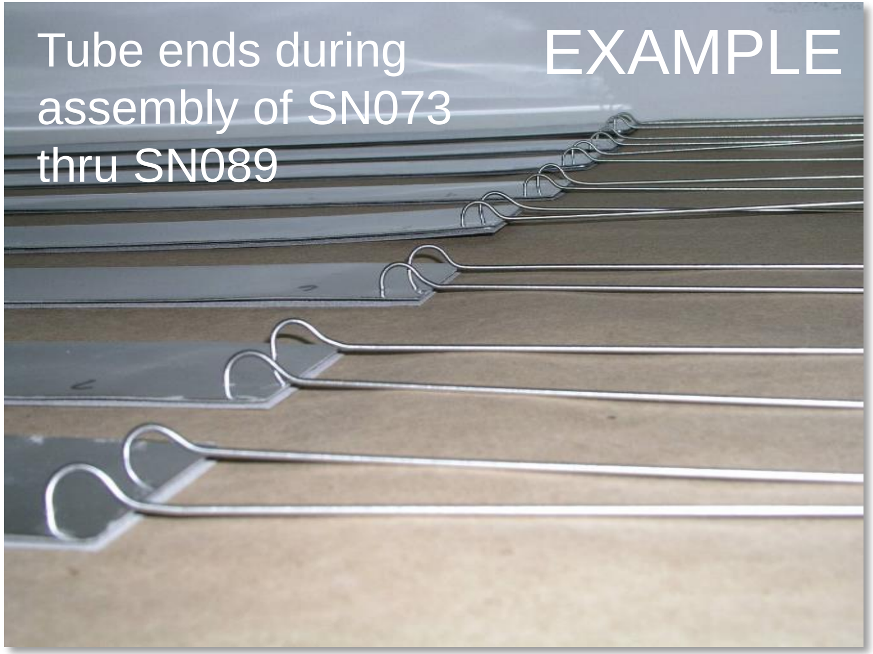
Sridharan, Anderson, University of Wisconsin, et al, sCO_2 Power Cycle Symposium, Boulder, CO 2011



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



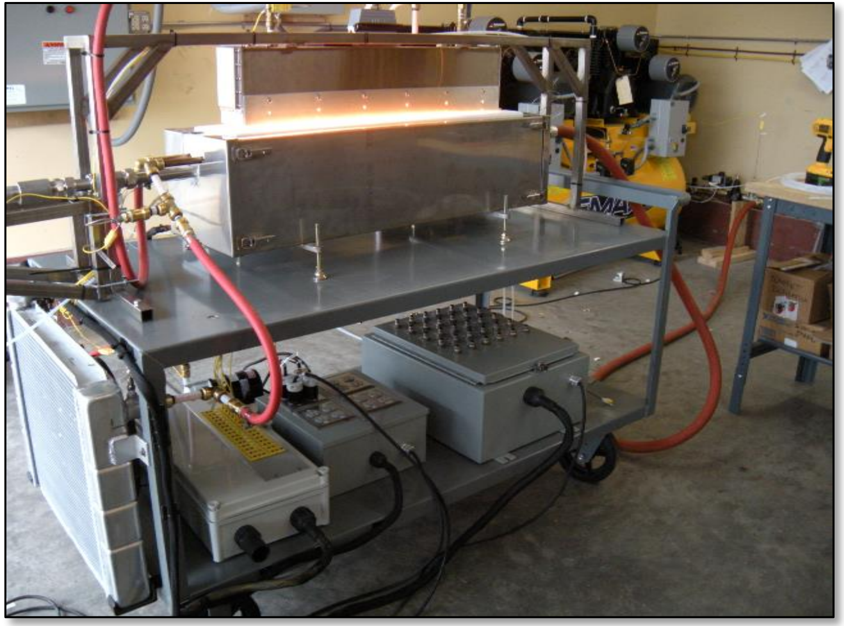
This final batch of heat exchanger cells were of high quality, leak tight and suitable for creep tests

- 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.

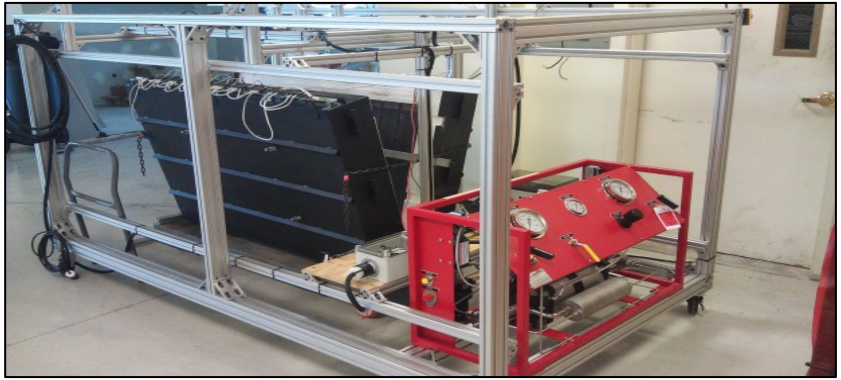
- sCO_2 pressurization for possible corrosion interaction

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

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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

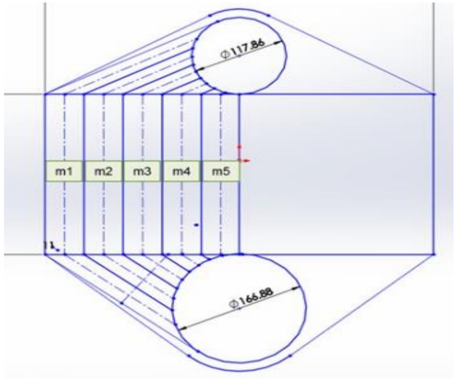
$$f = \frac{2\rho\Delta P}{G^2\beta t}\left(\frac{1-\varepsilon}{\varepsilon}\right)^{0.4}$$

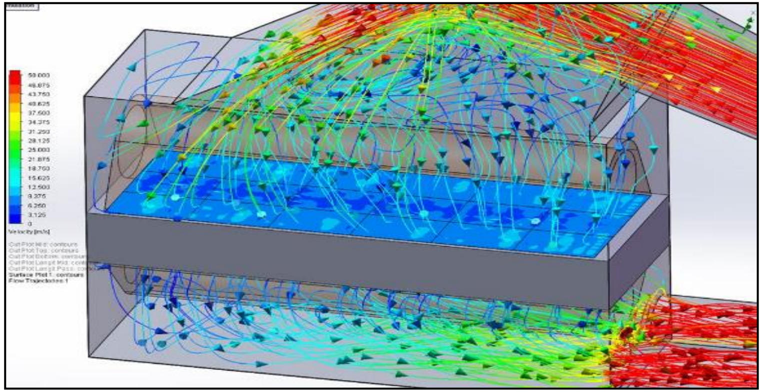
G = internal mass velocity
β = surface area/volume
ε = porosity

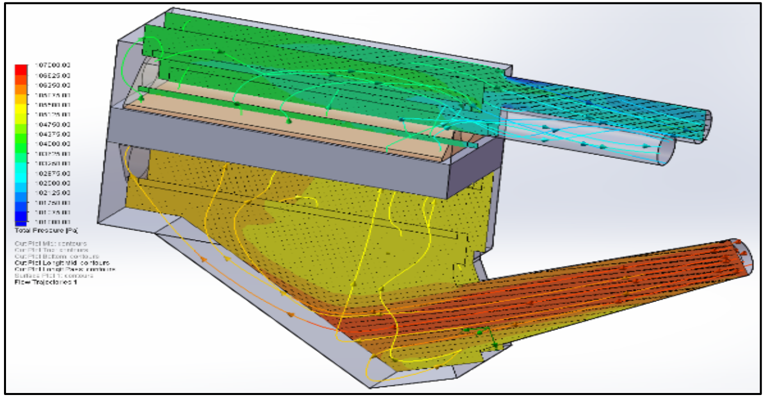
- CFD

Hydraulic Design – Flow Distribution

- Headered or unheadered, the net pressure loss along any given flow path 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







sCO_2 Heat Transfer

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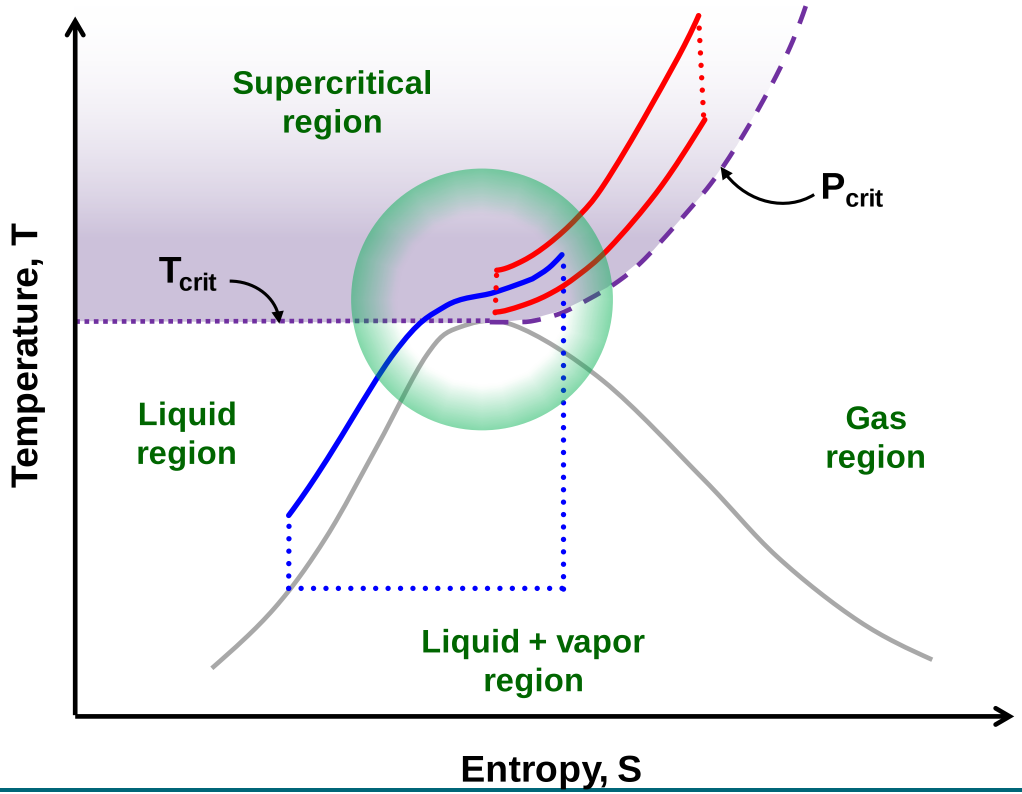
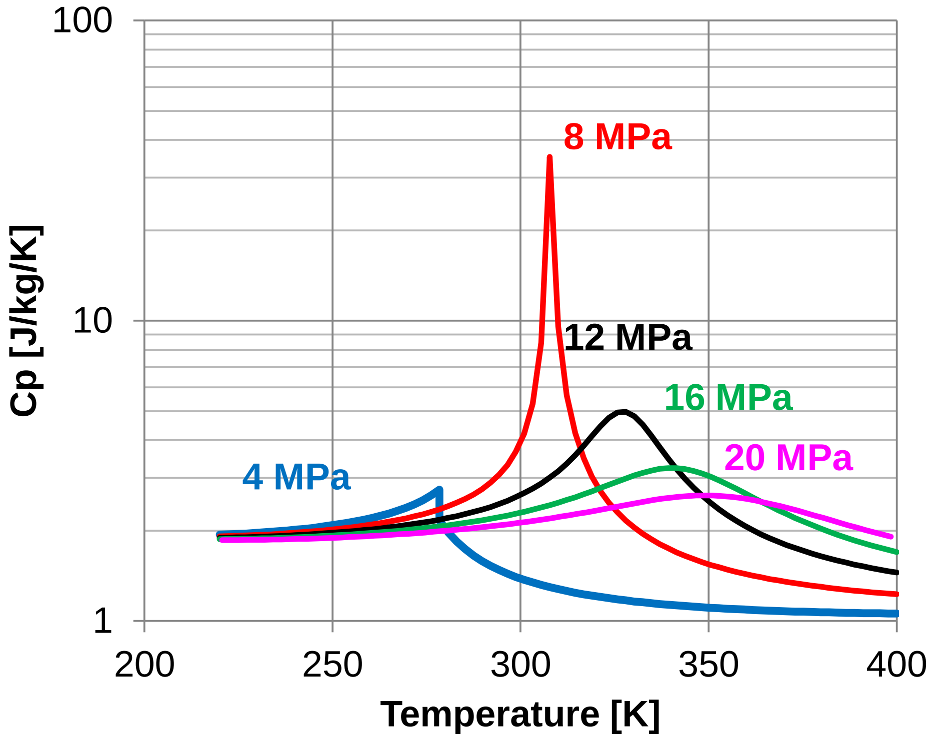
Fluid property effects near the critical point allow for less approximations in heat exchanger sizing

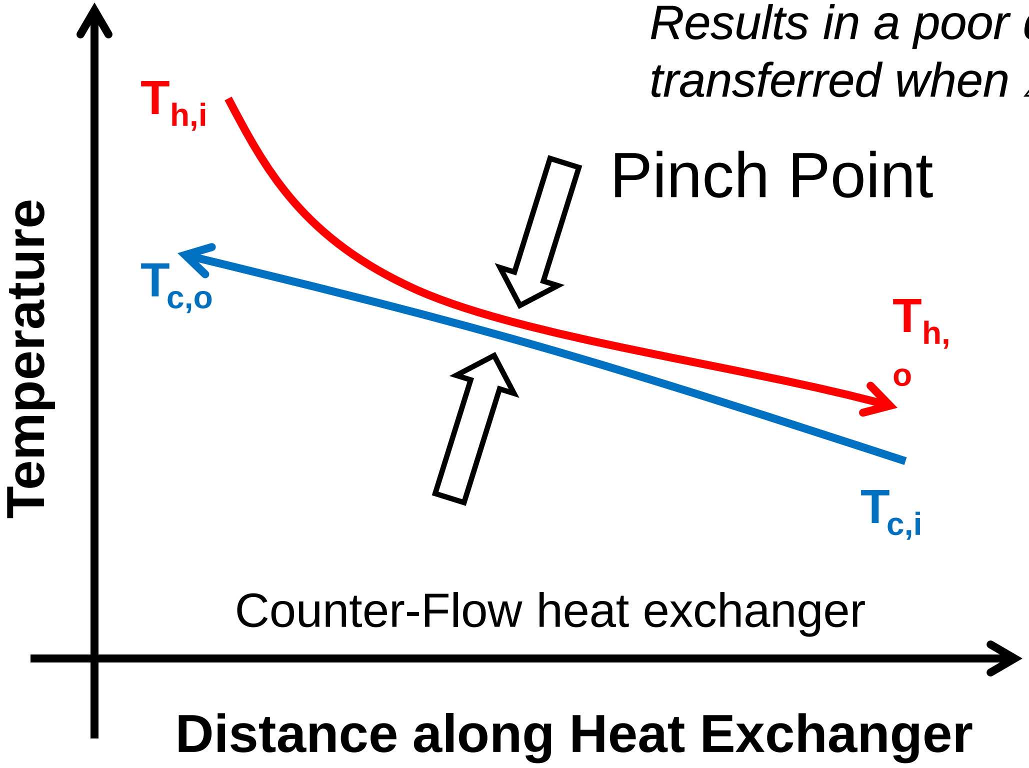


Fluid property effects near the critical point allow for less approximations in heat exchanger sizing



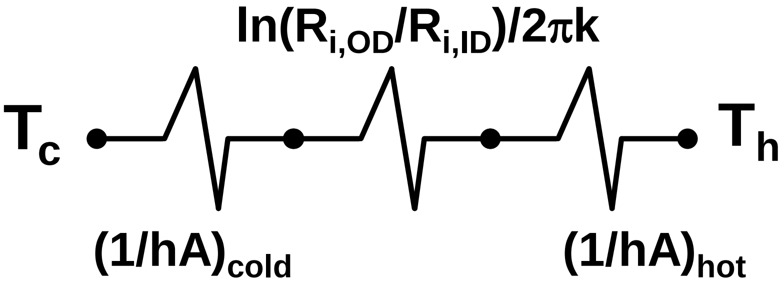
Real gas properties or phase change can create 'pinch' points in the temperature profile

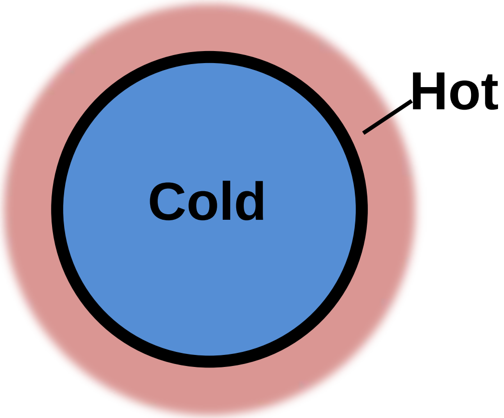
Results in a poor design because little-to-no heat is transferred when ΔT becomes very small



The calculated UA value can be used as a target value through the preliminary design process

Approximate the steady-state heat transfer path using thermal resistances





$$UA = \left(\frac{1}{(hA)_{cold}} + \frac{ln(R_i/R_o)}{2\pi k} + \frac{1}{(hA)_{hot}} \right)^{-1}$$

Typical approximations for heat exchanger sizing are not valid for near-critical sCO_2

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	
$\frac{1}{UA} = \frac{1}{(hA)_i} + \frac{ln\left(D_o/D_i\right)}{2\pi kL} + \frac{1}{(hA)_o}$	$h = f(Nu) = CRe^x Pr^y$

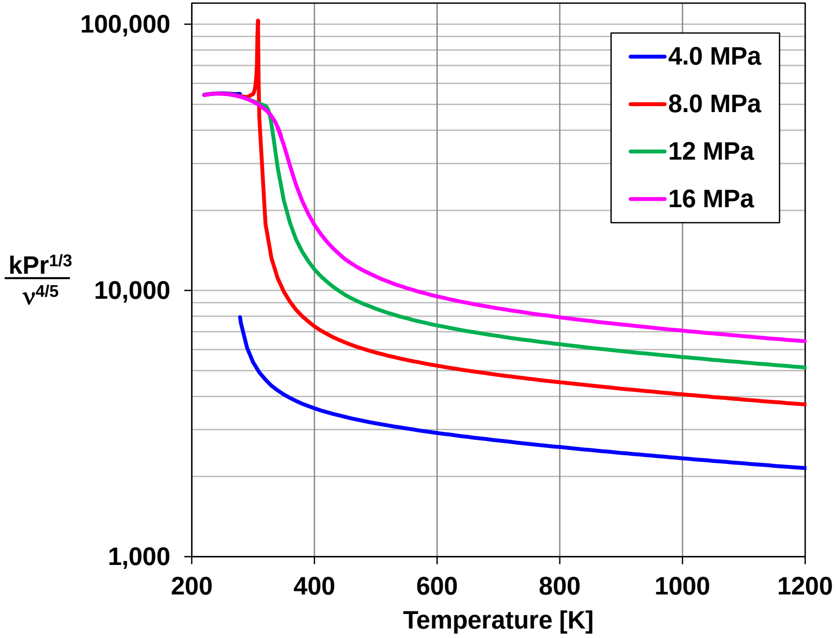
Typical correlations based on average fluid properties are not applicable near the critical point

Assume: x=4/5, y=1/3

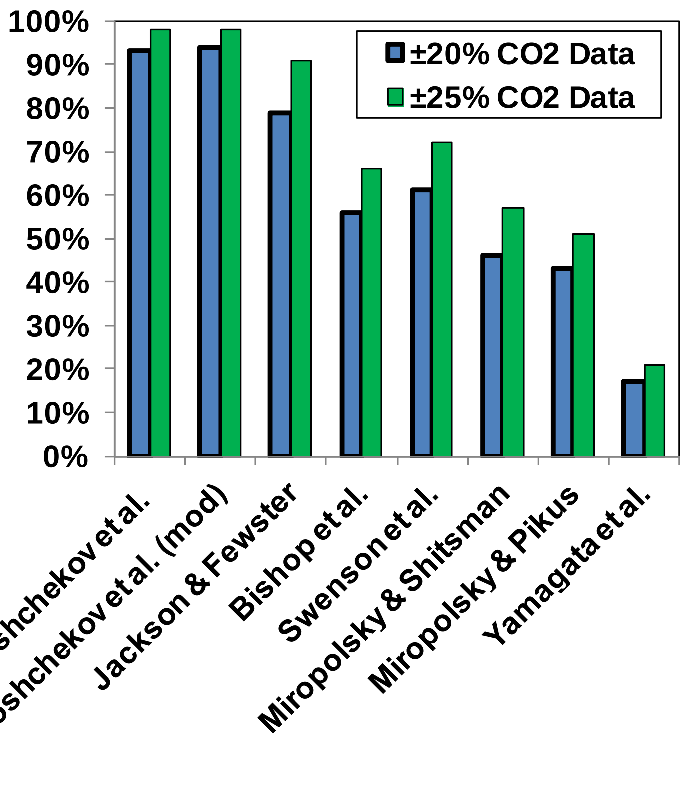
$$Q = hA\Delta T$$

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

$$h = fnc(k\nu^{-x}Pr^y)$$



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



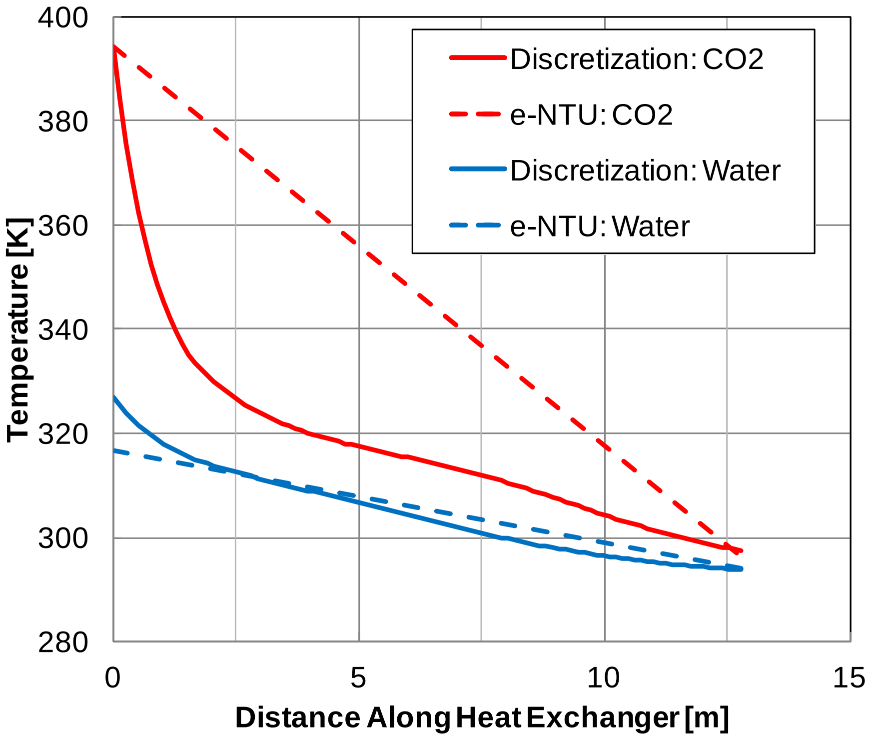
Test data screened for buoyancy

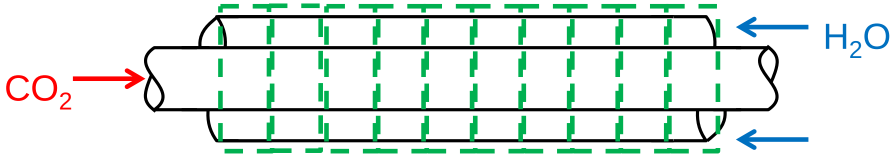
$$Nu_b = CRe_b^{m1} Pr_b^{m2} \left(\frac{\rho_w}{\rho_b}\right)^{m3} \left(\frac{\overline{C}_p}{C_{p_b}}\right)^{m4}$$

b = bulk
w = wall

[Values from Jackson 2013]

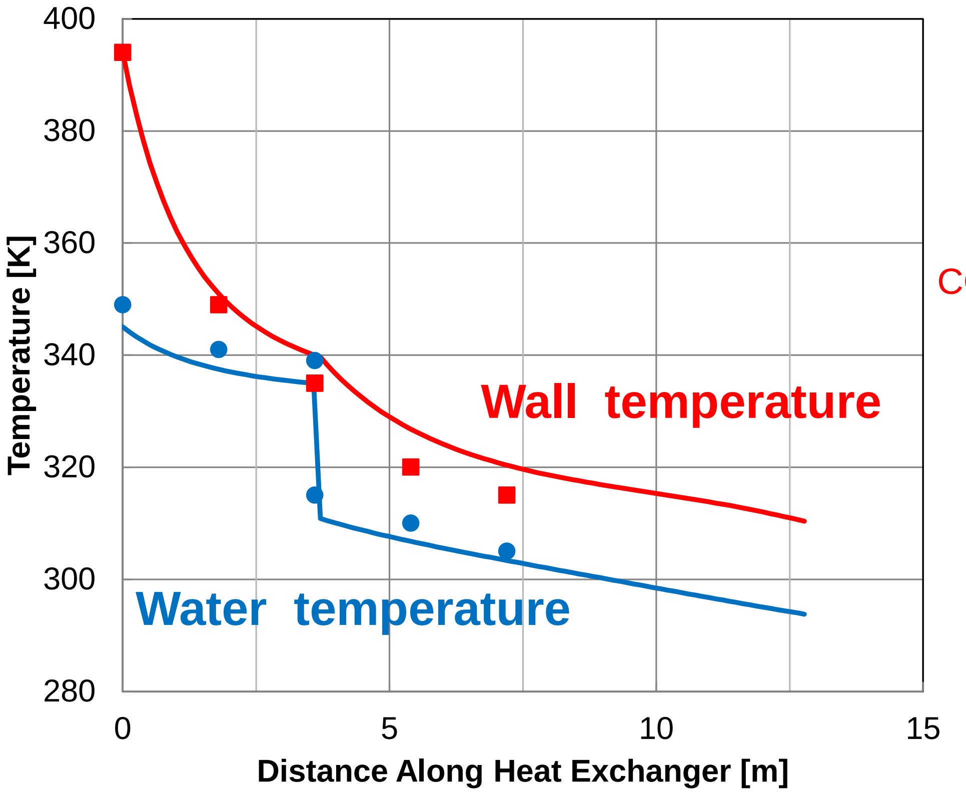
Discretizing the heat exchanger accounts for property differences that affect fluid temperature

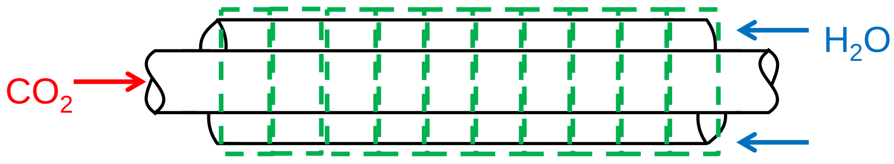




Operating conditions and geometry from:
Pitla, S., Groll, E., and Ramadhyani, S., 2001, "Convective Heat Transfer from In-Tube Cooling of Turbulent Supercritical Carbon Dioxide: Part 2—Experimental Data and Numerical Predictions," HVAC&R Research, ***7****(4), pp. 367–382.*

1D prediction methods match well with experimental measurements when the HX is discretized

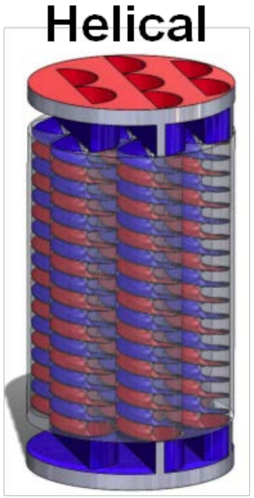


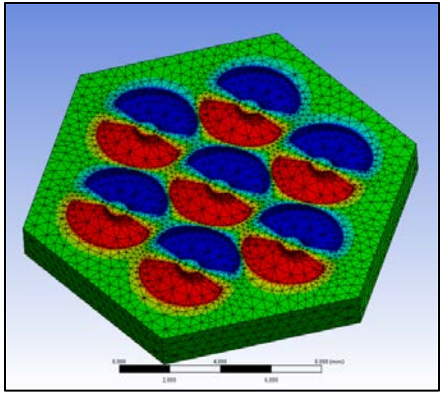


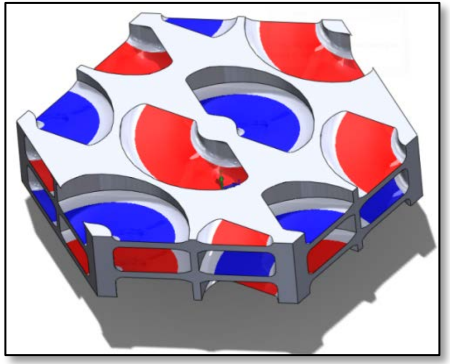
*Pitla, S., Groll, E., and Ramadhyani, S., 2001, "Convective Heat Transfer from In-Tube Cooling of Turbulent Supercritical Carbon Dioxide: Part 2—Experimental Data and Numerical Predictions," HVAC&R Research, **7**(4), pp. 367–382.*

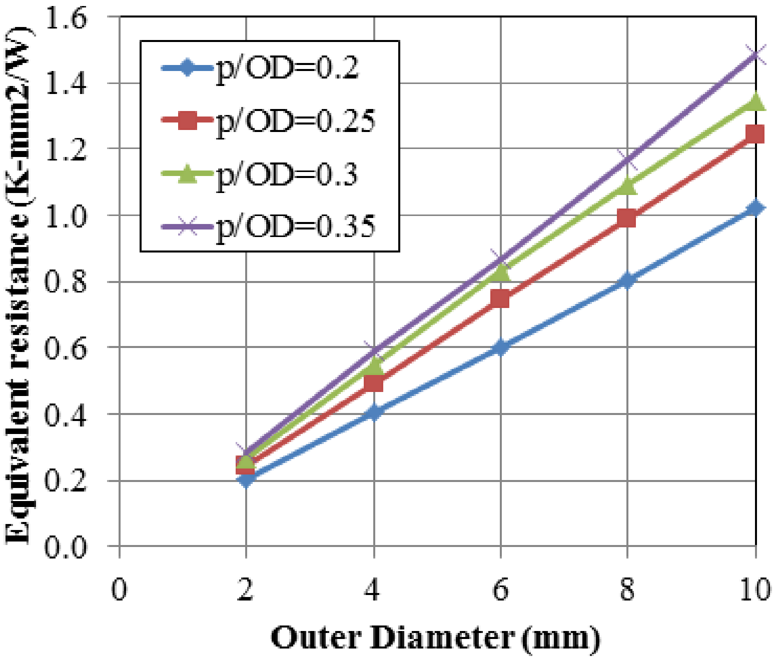
Detailed simulations may be needed for unconventional designs

Thermal modeling to inform 1D sizing models



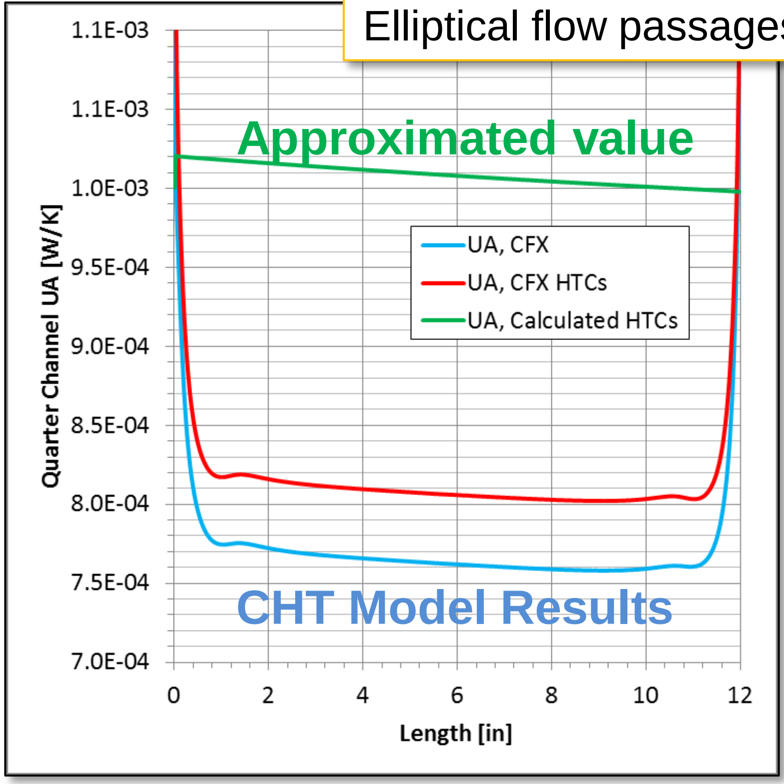




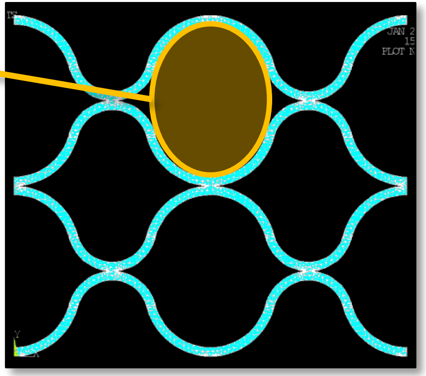


Courtesy Thar Energy, DE-FE0026273

CHT simulations can be used to check the validity of assumptions in the 1D design process



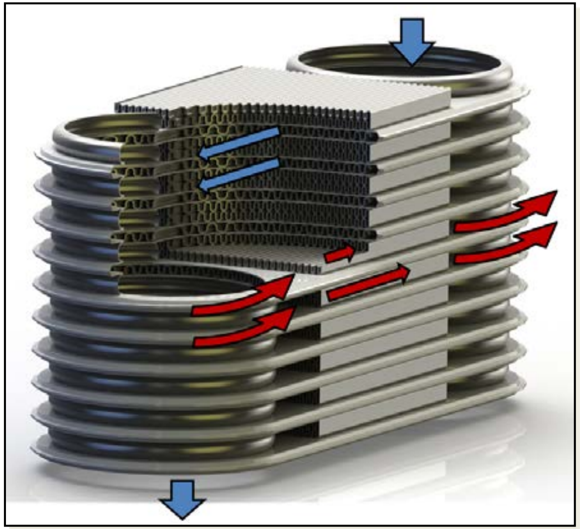
Elliptical flow passages assumed for HTC



A modification factor **~0.75** should be used for the approximate UA value

Courtesy Thar Energy, DE-FE0026273

Questions?



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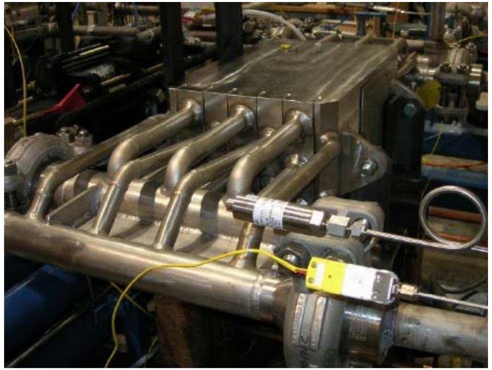
Backup Slides

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



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

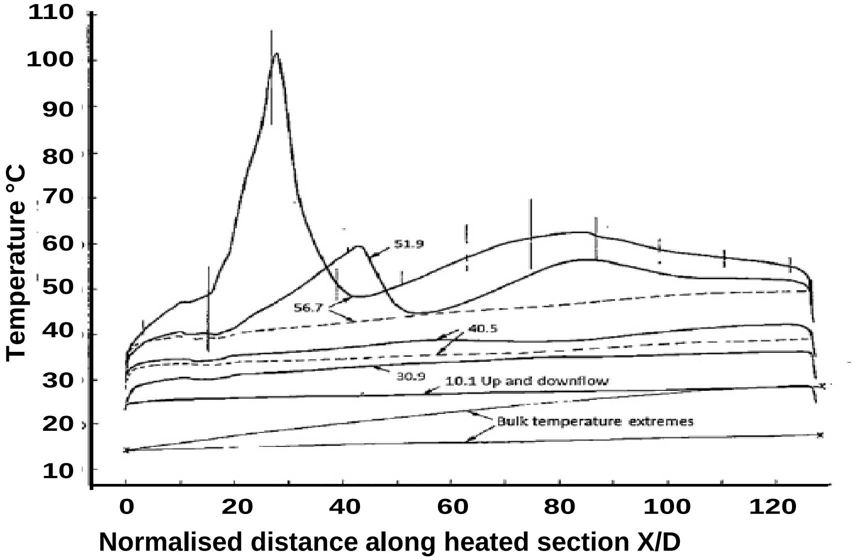


Fig. 4. Localized deterioration of heat transfer with upward flow; 19 mm diameter tube. Upflow is denoted by solid lines; downflow by broken lines, mass flow rate 0.160 kg/s; bulk inlet temperature 14 °C; wall heat flux as indicated, 30.9, 40.5, 51.9, 56.7 kW/m².

[Jackson 2013]

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

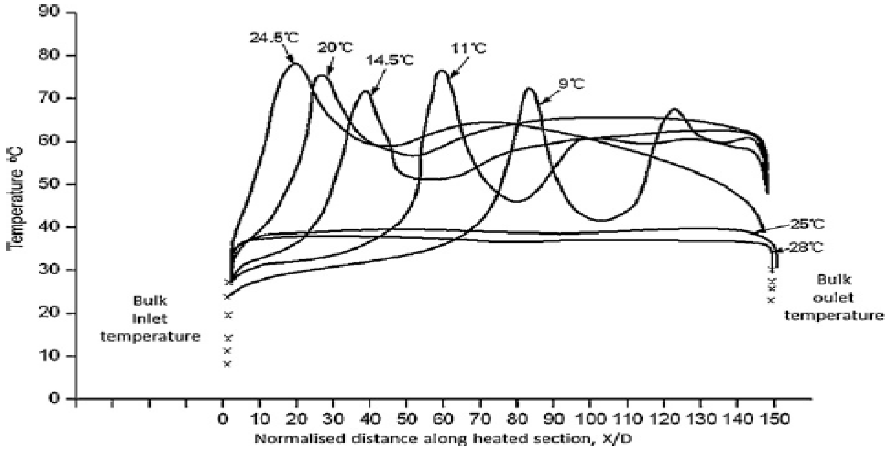


Fig. 8. Effect of reducing inlet fluid temperature, 8 mm diameter tube, upflow only. Pressure 7.58 MPa; inlet temperatures, 9 °C, 11 °C, 14.5 °C, 20 °C, 24.5 °C; mass flowrate 0.02 kg/s; wall heat flux 33.6 kW/m²; Re ~ 4 × 10⁴.

[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 ($Re \sim 2.5 \times 10^5$)

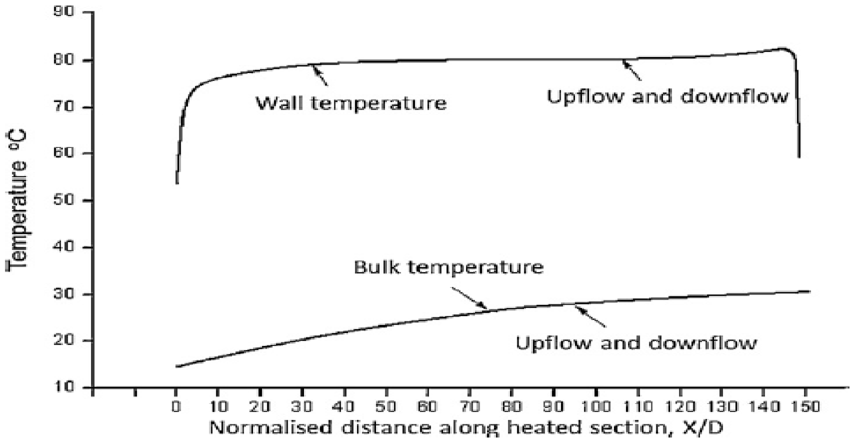


Fig. 9. Highest mass flow rate, 5mm diameter tube, pressure 7.58 MPar, upflow only. Mass flowrate 0.0645 kg/s; wall heat flux 455 kW/m^2.

[Jackson 2013]

The upward flow direction produces a peak wall temperature at a low mass flow

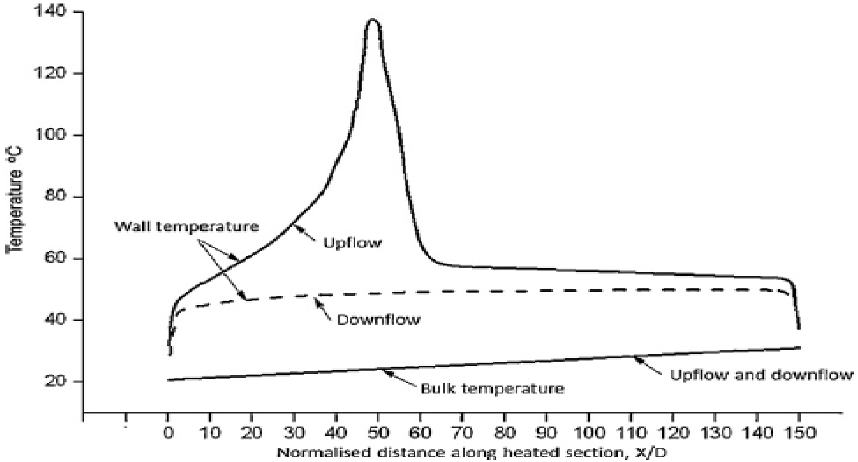


Fig. 10. Further reduction of flow rate, 5 mm diameter tube, upflow and downflow. Pressure 7.58 MPa; mass flow rate 0.0129 kg/s; wall heat flux 68 kW/m^2; $Re \sim 4 \times 10^4$.

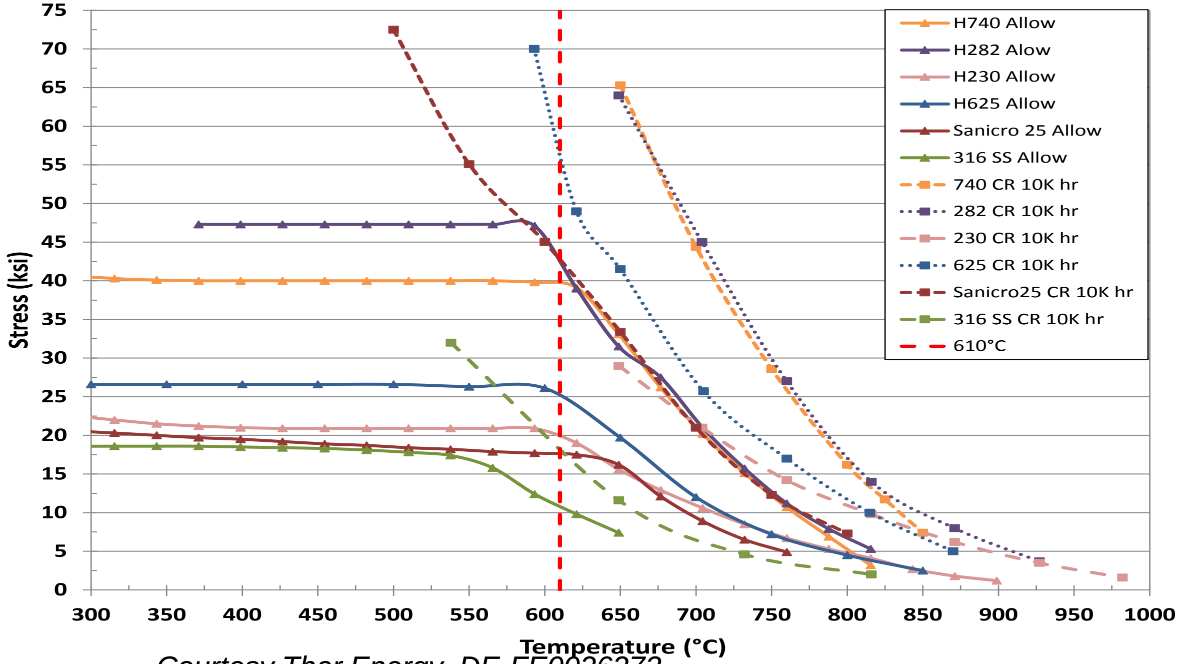
[Jackson 2013]

Materials Selection

- Usually material selection is from ASME code cases
- Material cost, strength, creep, corrosion allowance are factors in selection
- Material availability is also important:
    - Can the material be obtained in the desired form? (i.e. tubes, sheets, plates)

Material Strength

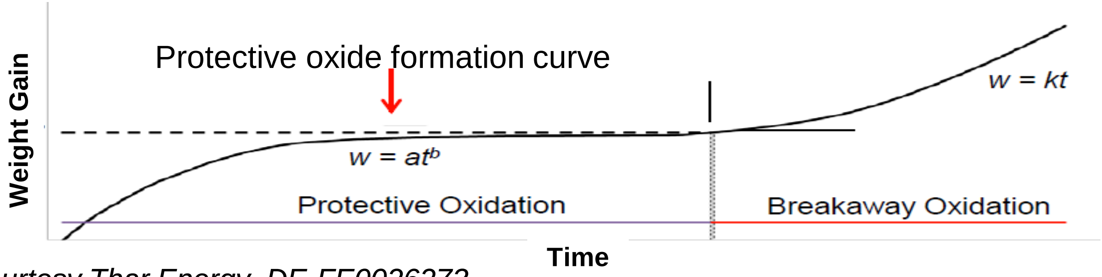
Required Material Thickness



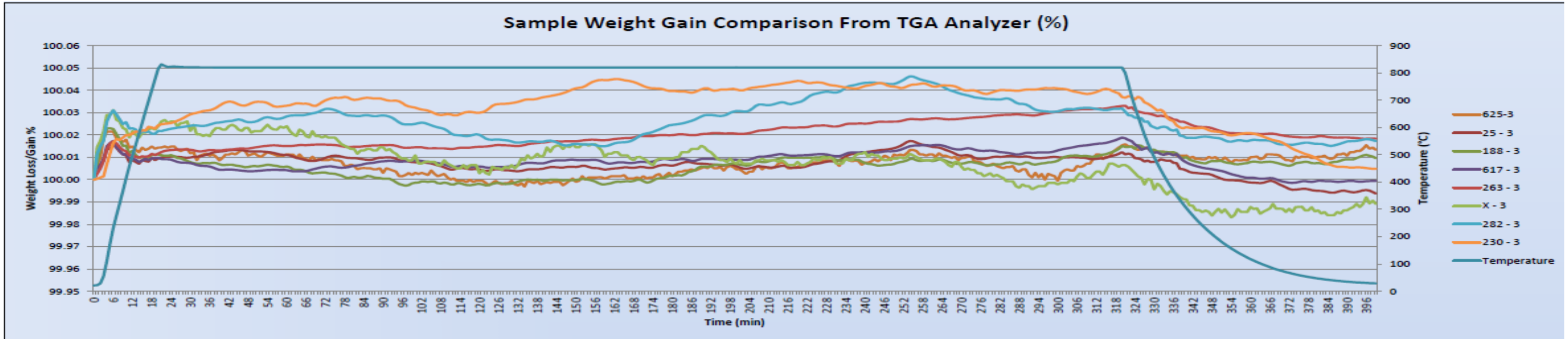
Courtesy Thar Energy, DE-FE0026273

Corrosion

Corrosion occurs over a long period of time to build an oxide layer on the heat transfer surfaces



Courtesy Thar Energy, DE-FE0026273



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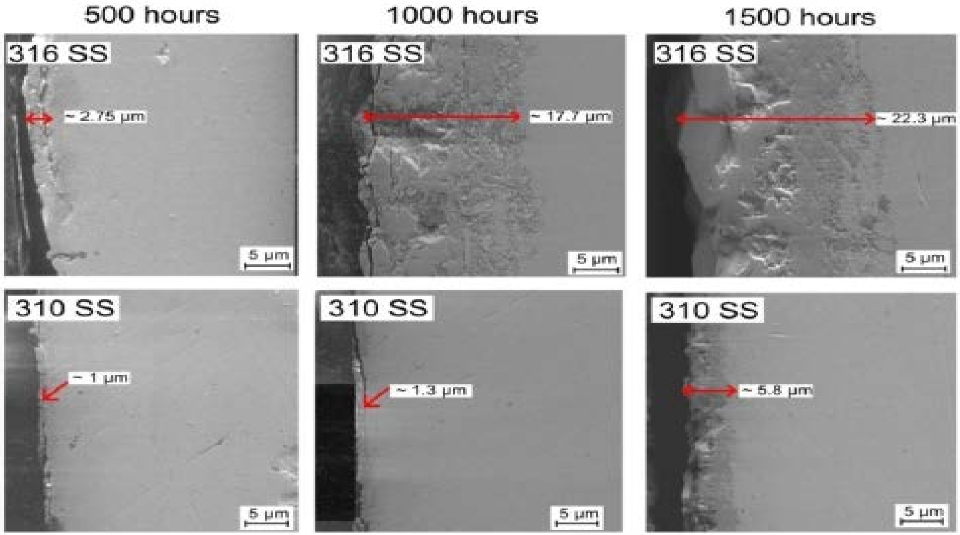
Corrosion

Corrosion of heat transfer surfaces

- Caused by water or process fluids that oxidize the heat exchanger material
- Corrosion allowance should be accounted for during design
- Careful material selection is i

Image from Premier Separator Services Limited



*Cao, G., Firouzdor, V., Sridharan, K., Anderson, M., and Allen, T. R., 2012, "Corrosion of austenitic alloys in high temperature supercritical carbon dioxide," Corrosion Science, **60**, pp. 246–255.*

Design conditions should include margin on top of the operating conditions

Condition	Temperature	Pressure	Comment
Operating	400°C	15 bar	Expected Operating Conditions
Design	430°C	17.25 bar	Add 30C margin and 5% for PSV

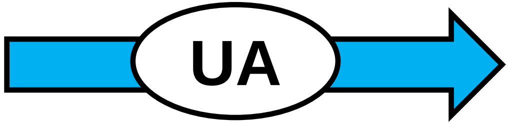
The design temperature and pressure will allow some margin for the actual operation of the heat exchanger

The design conditions may significantly affect the material selection and containment thickness

Guidance is available in ASME BPVC and in NORSOK P-001

Heat Exchanger Design Approach

Requirements for heat exchanger



Heat exchanger layout, sizing

- Fluid Inlet Conditions
- Fluid Outlet Conditions

Use the LMTD Approach to get UA

$$Q = UA\Delta T_{LM} = \dot{m}\left(h_{h,i} - h_{h,o}\right)$$

- Fluid inlet conditions
- Thermal duty (Q)

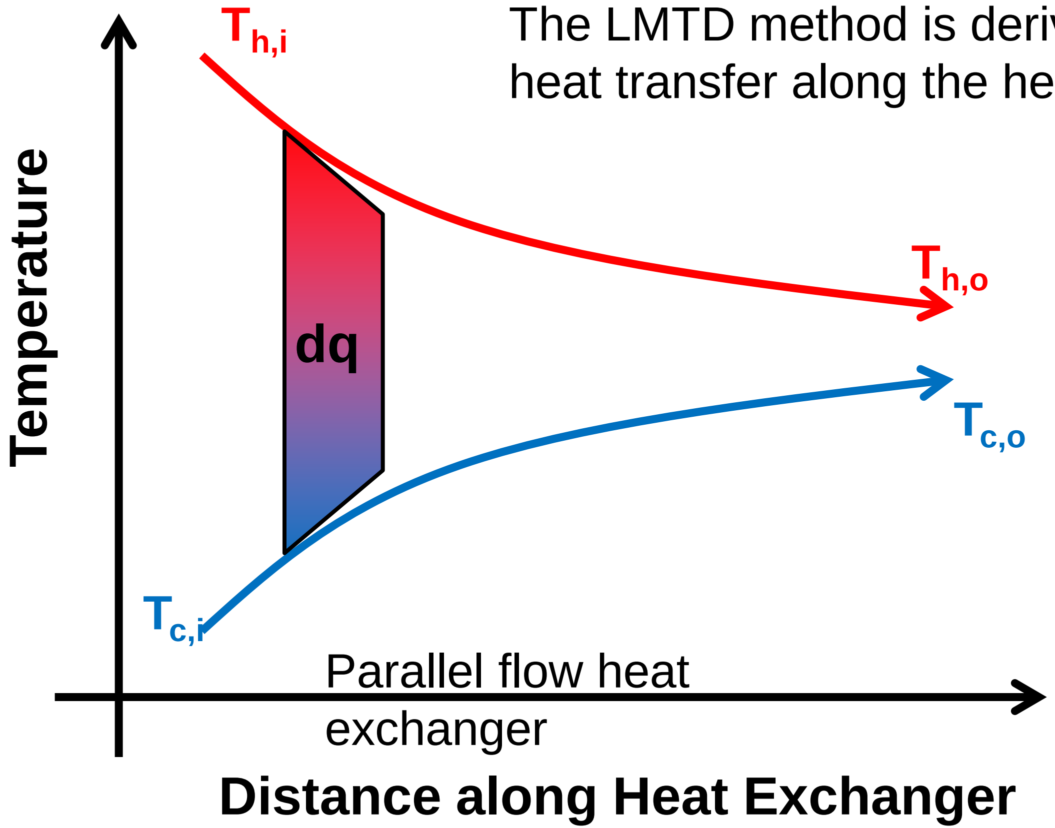
Use the ε-NTU Method to get UA

$$Q = \varepsilon Q_{max} \qquad UA = NTU \cdot C_{min}$$

$$NTU = fnc(\varepsilon, C)$$

No matter which approach is used, the same UA value will be calculated

Log Mean Temperature Difference (LMTD)



The LMTD method is derived from integrating the heat transfer along the heat exchanger

$$dq = U \Delta T dA$$

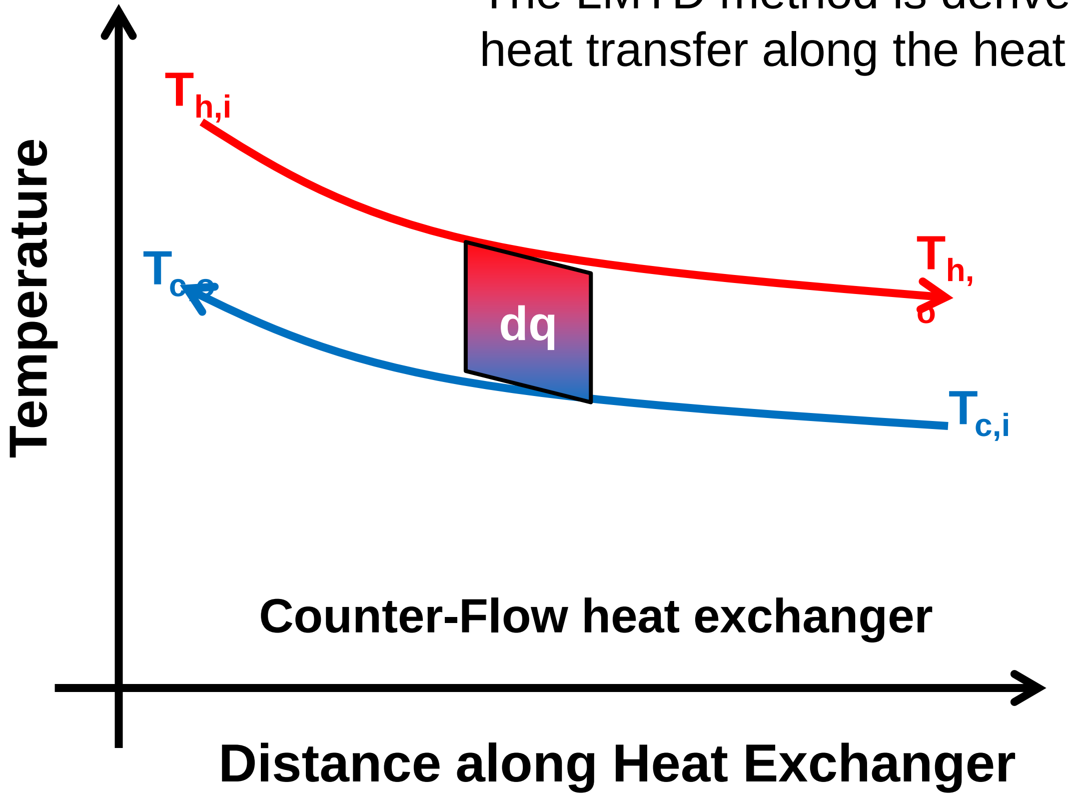
⇩

$$Q = UA \frac{\Delta T_2 - \Delta T_1}{ln(\Delta T_2 / \Delta T_1)}$$

Adapted from Incropera & DeWitt, Introduction to Heat Transfer, 4th Ed., 2002.

Log Mean Temperature Difference (LMTD)

The LMTD method is derived from integrating the heat transfer along the heat exchanger



dq= $U\Delta T dA$

$$Q = UA\frac{\Delta T_2 - \Delta T_1}{ln(\Delta T_2/\Delta T_1)}$$

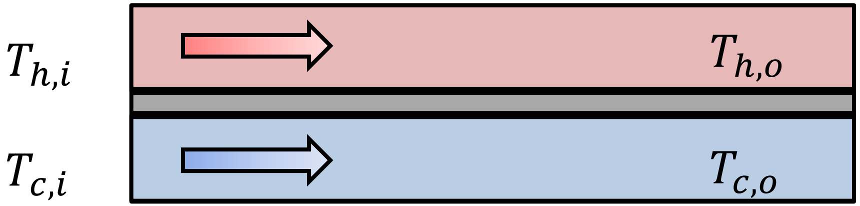
Adapted from Incropera & DeWitt, Introduction to Heat Transfer, 4th Ed., 2002.

Calculating UA by the LMTD Method

$$Q = UA\frac{\Delta T_2 - \Delta T_1}{ln(\Delta T_2/\Delta T_1)} \qquad Q = \dot{m}(h_{h,i} - h_{h,o})$$

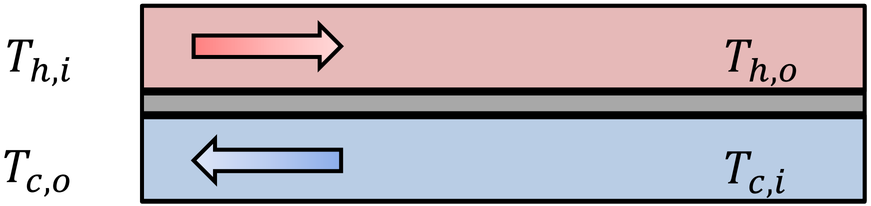
Parallel-Flow

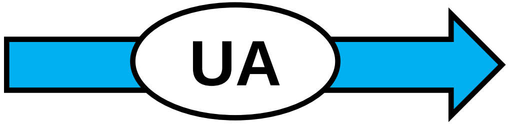
$$\Delta T_1 = \Delta T_{h,i} - \Delta T_{c,i}$$
$$\Delta T_2 = \Delta T_{h,o} - \Delta T_{c,o}$$



Counter-Flow

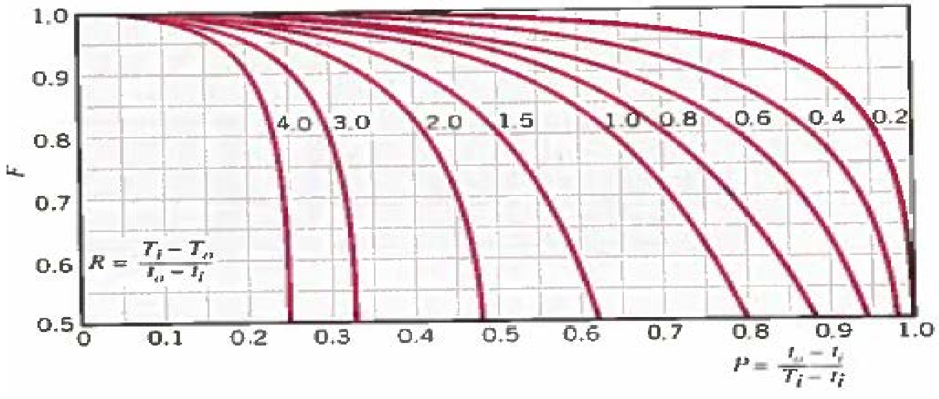
$$\Delta T_1 = \Delta T_{h,i} - \Delta T_{c,o}$$
$$\Delta T_2 = \Delta T_{h,o} - \Delta T_{c,i}$$

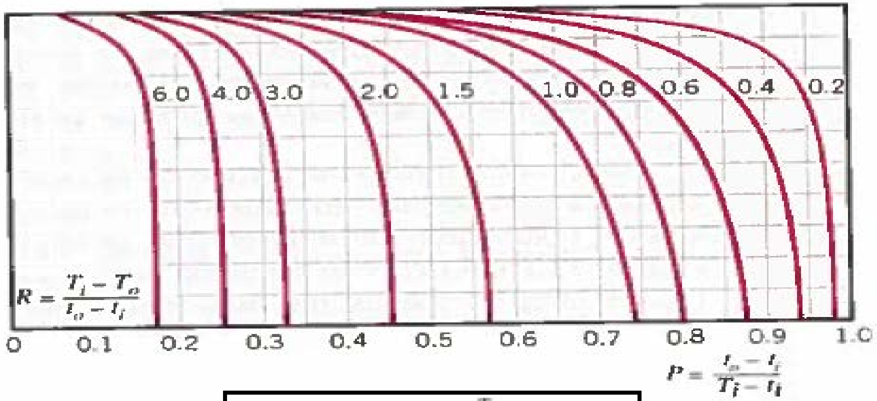


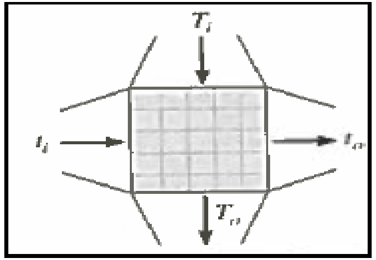


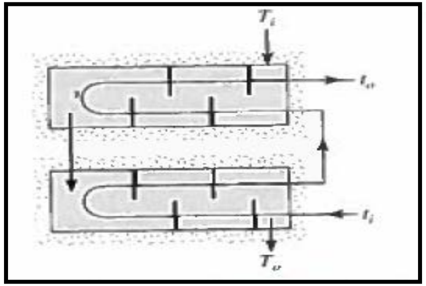
Variations on parallel-flow and counter-flow require a correction Factor

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









Figures from Incropera & DeWitt, Introduction to Heat Transfer, 4th Ed., 2002.

The effectiveness (ε) -NTU Method

- Fluid inlet conditions
- Thermal duty (Q)

$$Q = \varepsilon Q_{max} \qquad UA = NTU \cdot C_{min}$$

$$NTU = fnc(\varepsilon, C)$$

$$\varepsilon = \frac{Q}{Q_{max}}$$

Q_{max}

- Maximum heat transfer possible between the two fluids
- Limited by the fluid with the ***least*** thermal capacitance

$$C_h = \dot{m}_h C_{p,h} \qquad C_c = \dot{m}_c C_{p,c}$$

$$Q_{max} = min(C_c, C_h) * \left(T_{h,i} - T_{c,i}\right)$$

The effectiveness (ε) -NTU Method

- Fluid inlet conditions
- Thermal duty (Q)

$$Q = \varepsilon Q_{max}$$

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

$$NTU = fnc(\varepsilon, C)$$

$$NTU = fnc(\varepsilon, C)$$

NTU = Net Transfer Units

- Derivation provided in most heat transfer textbooks
- Calculated from ε and ratio of C_{min}/C_{max}

$C_r = C_{min}/C_{max}$

Parallel-Flow

$$NTU = \frac{-ln[1 - \varepsilon(1 + C_r)]}{(1 + C_r)}$$

Counter-Flow

$$NTU = \frac{1}{1 - C_r} ln\left(\frac{\varepsilon - 1}{\varepsilon C_r - 1}\right)$$

$C_r < 1$

The effectiveness (ε) -NTU Method

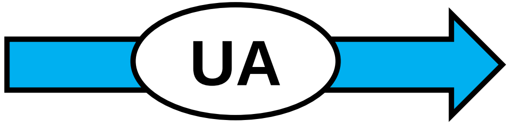
- Fluid inlet conditions
- Thermal duty (Q)

$$Q = \varepsilon Q_{max}$$

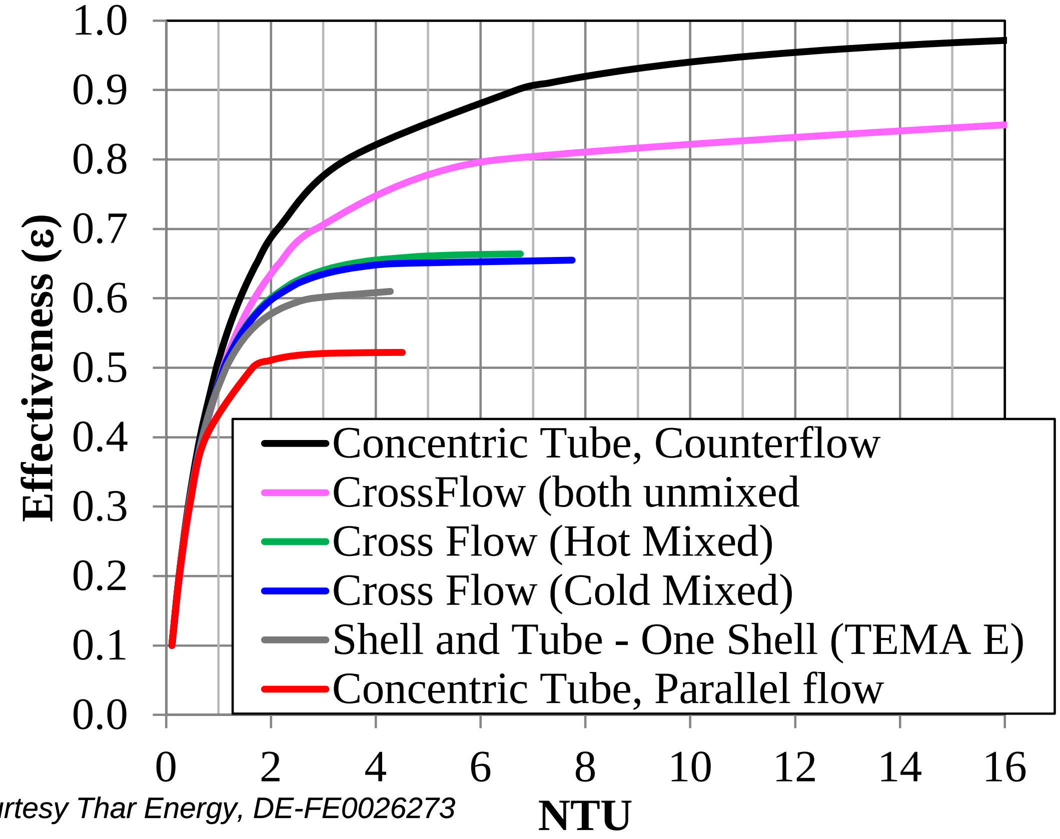
$$UA = NTU \cdot C_{min}$$

$$NTU = fnc(\varepsilon, C)$$

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



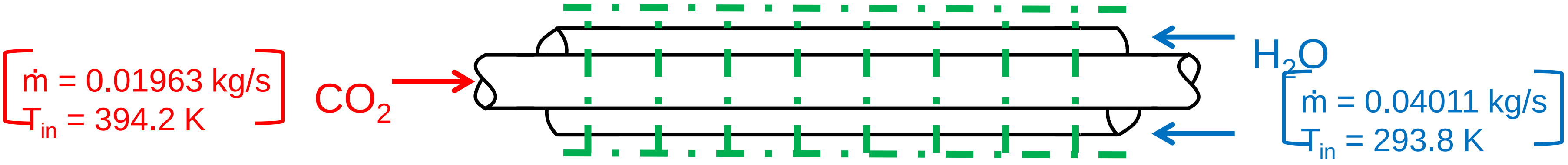
The ε-NTU method is a good way to quickly identify heat exchanger types



Courtesy Thar Energy, DE-FE0026273

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

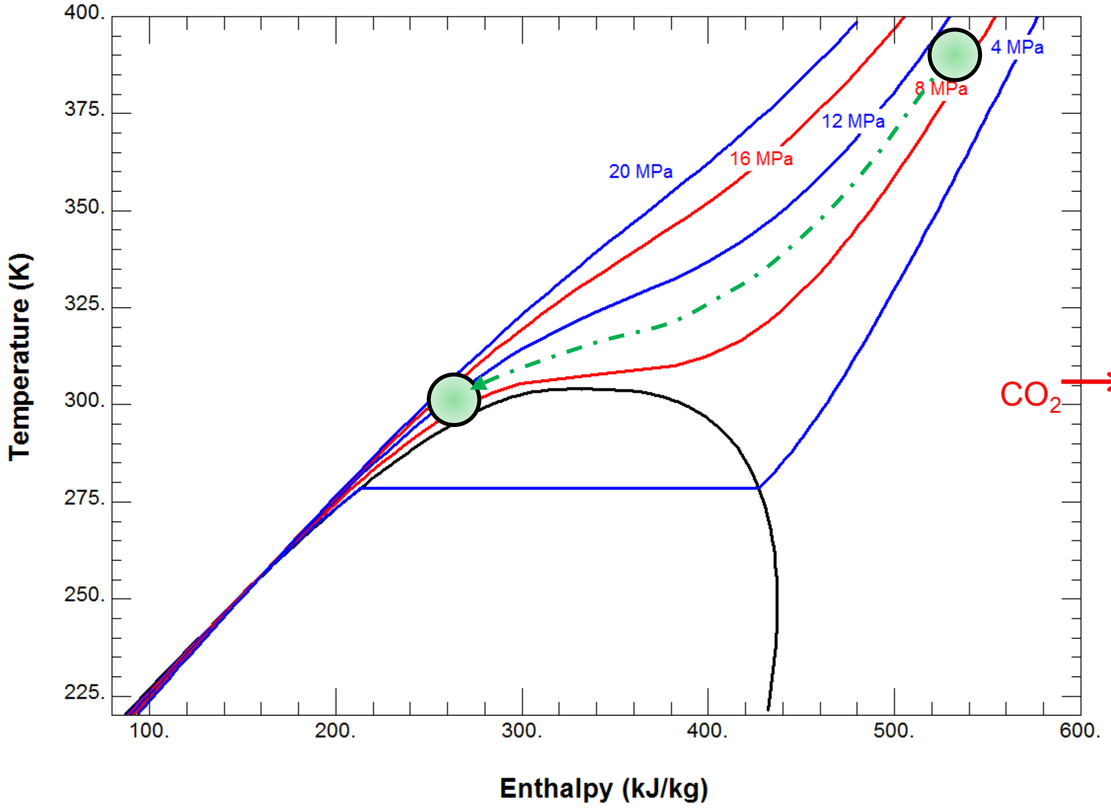
Validation is based on test data from [Pitla 2001]

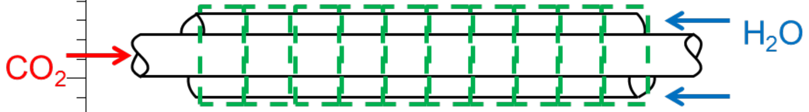


Assumptions:

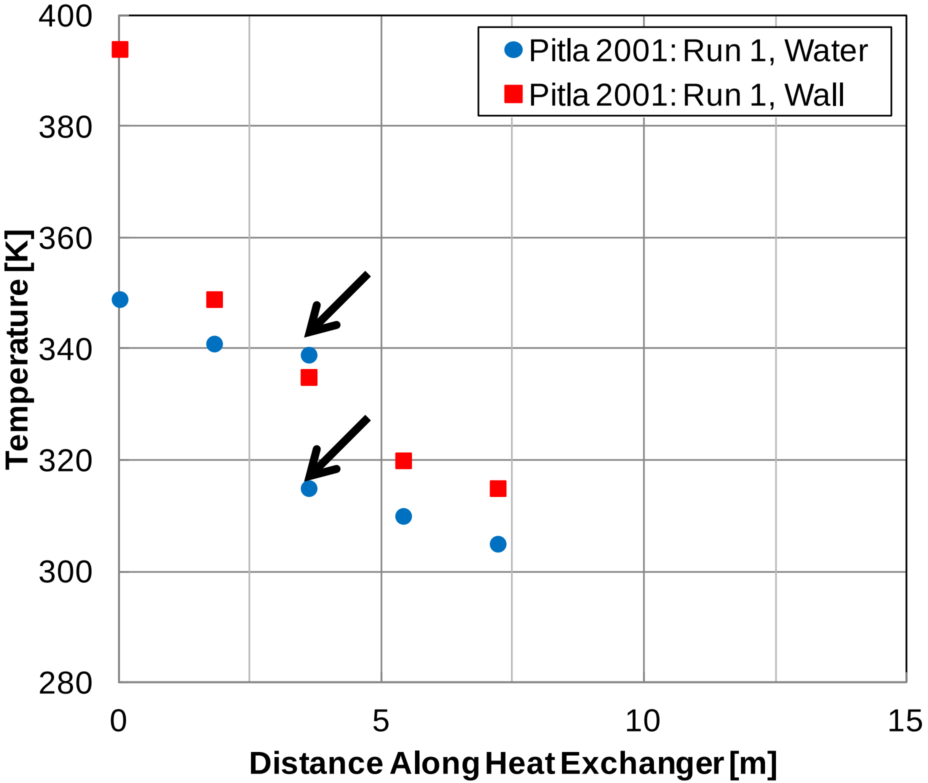
- one-dimensional
- steady-state
- frictionless flow

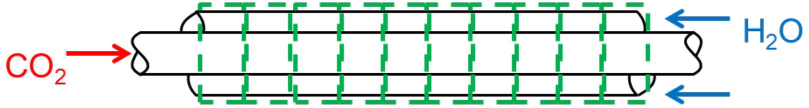
The test data is trans-critical





Water and CO_2 wall temperature is used for validation





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

ε-NTU Method (average fluid properties):

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

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

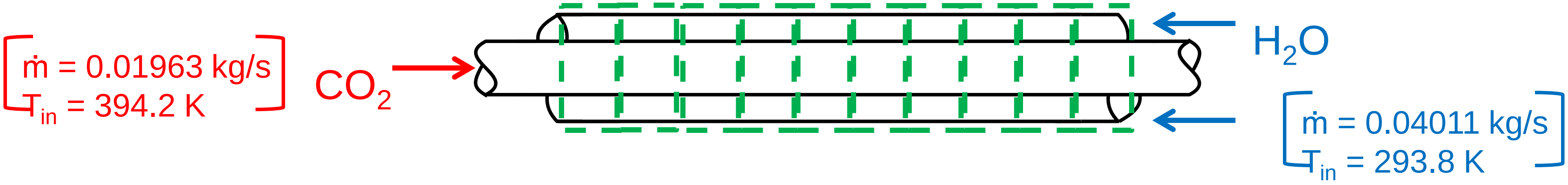
ε = Q/Qmax = 98.6%

A 1st order, backward difference discretization of the energy equation (100 elements):

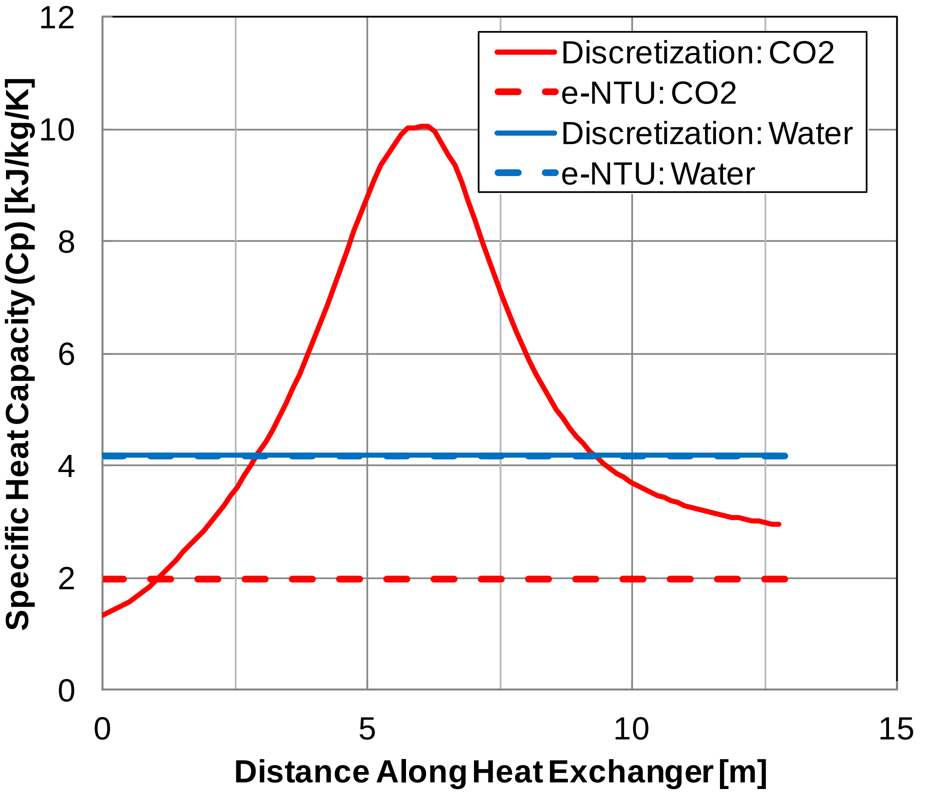
$$\left(h_{c,n}\right)^{i} = \left(h_{c,n-1}\right)^{i-1} + \frac{UA}{\dot{m}_c}\left(T_{h,n} - T_{c,n}\right)^{i-1}$$

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

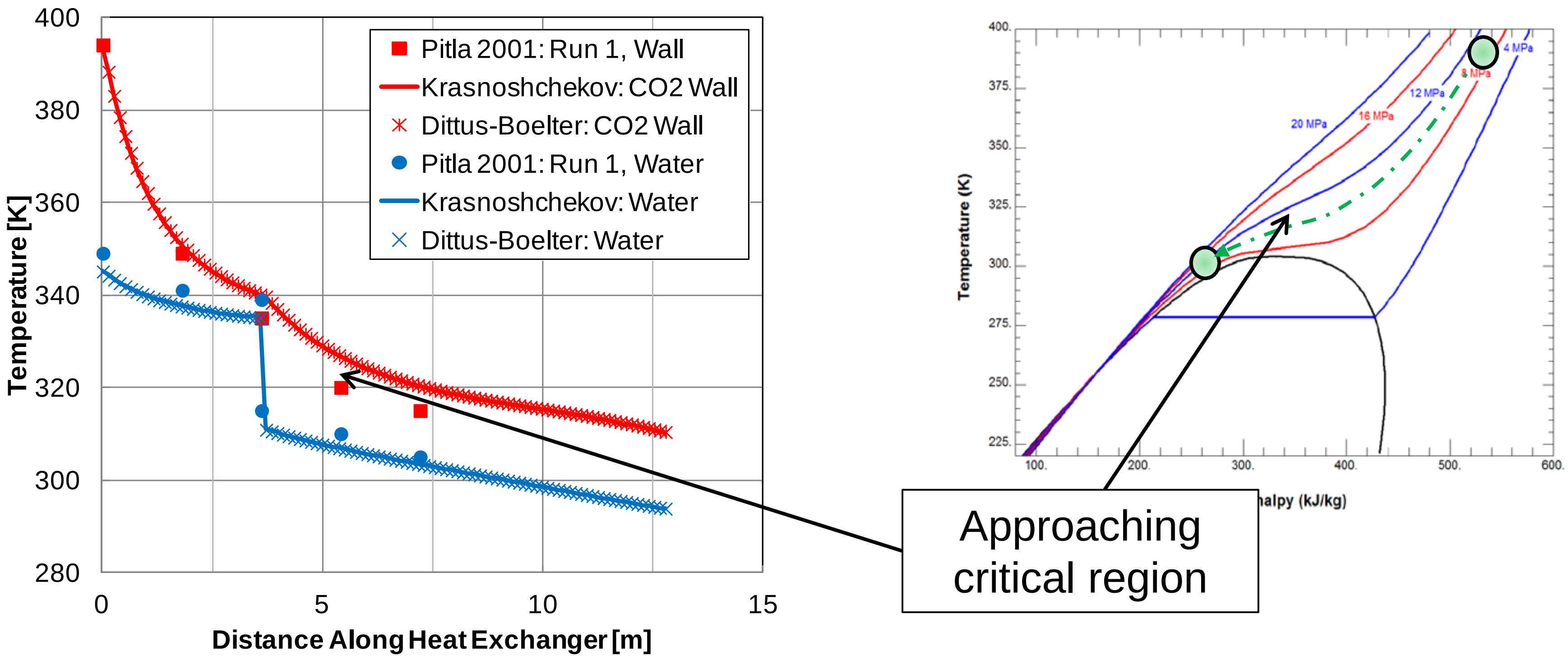
ε = Q/Qmax = 32.5%



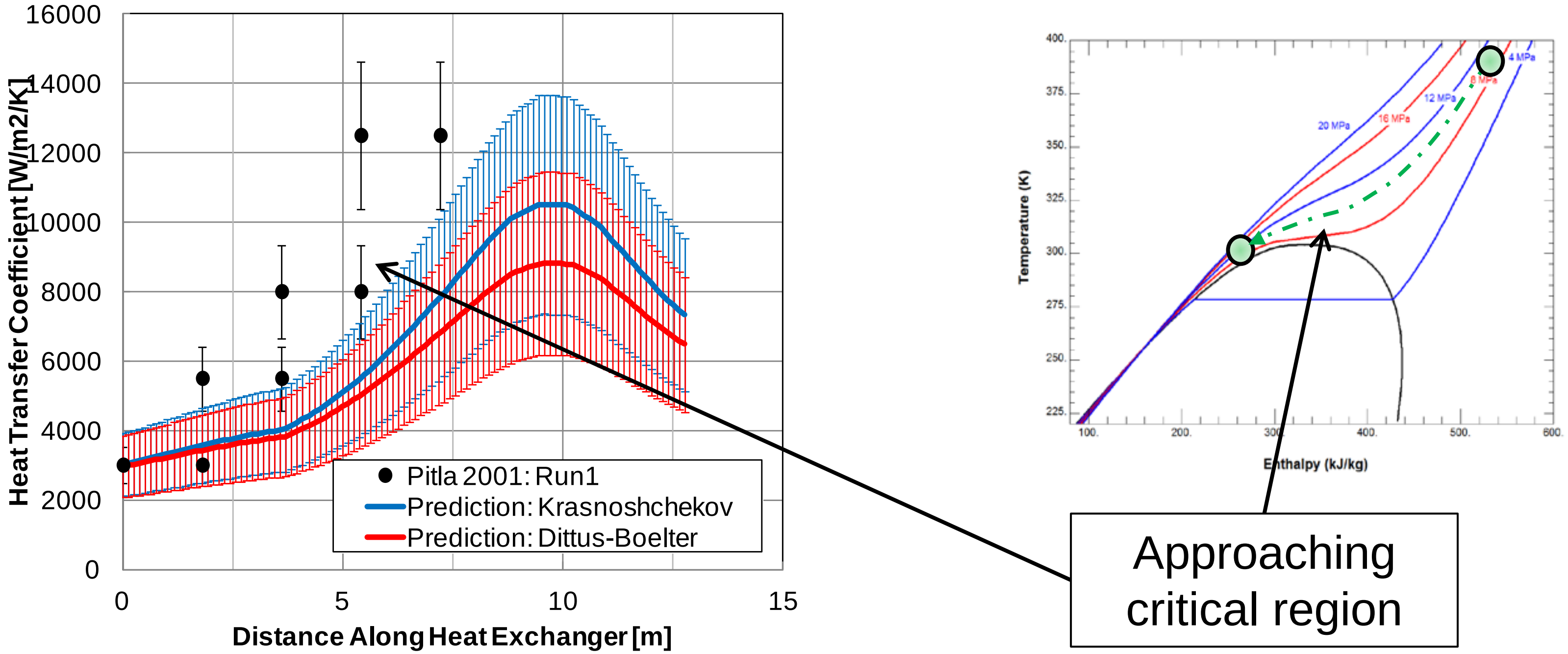
The heat exchanger should be discretized to accurately account for fluid property variations



Heat transfer variations from correlations can be negligible on temperature prediction

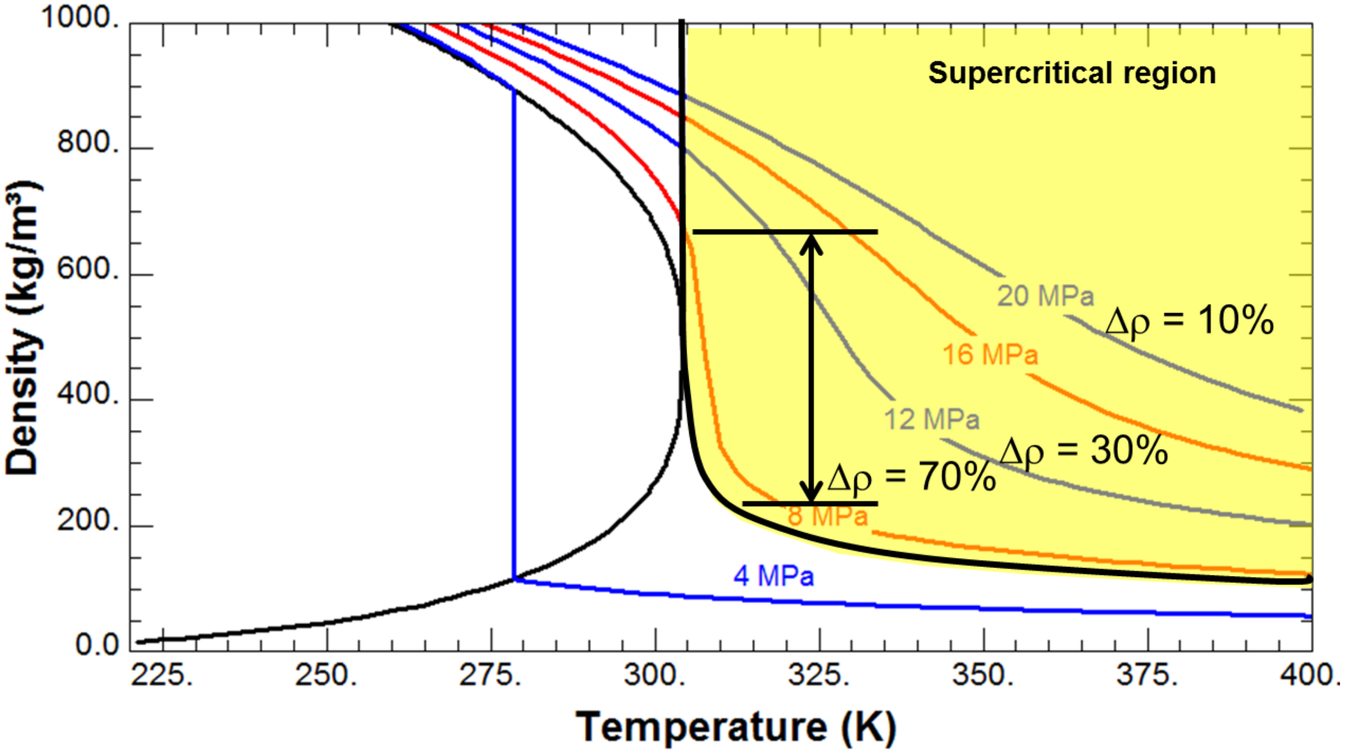


Property changes in the critical region cause heat transfer variations between correlations



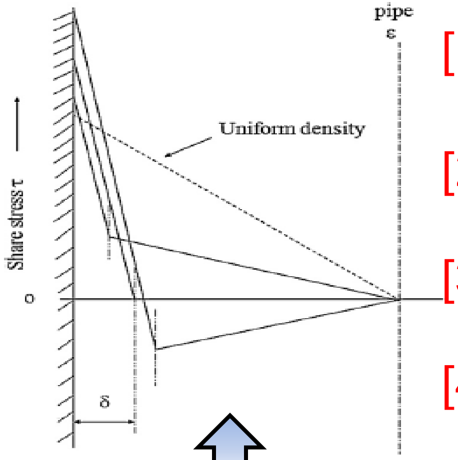
Note: 30% uncertainty bars applied to correlations

CO_2 density decreases near the critical point, which can induce buoyancy effects



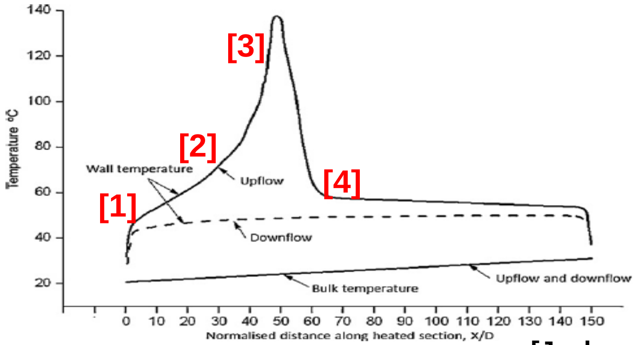
REFPROP (2007)

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



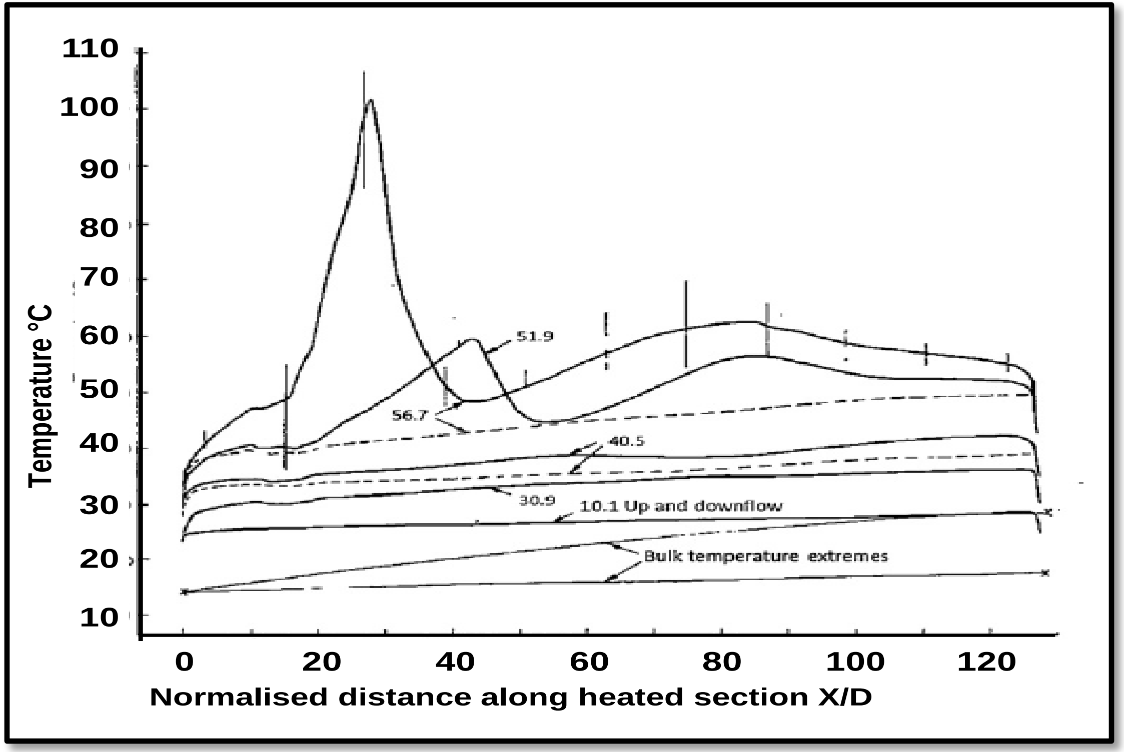
[Jackson 2013]

[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



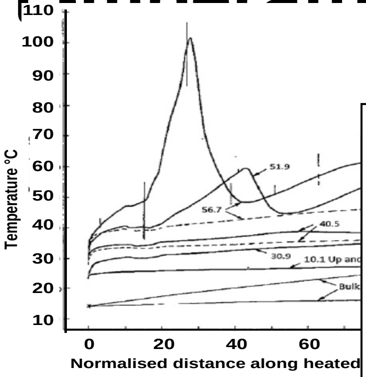
[Jackson 2013]

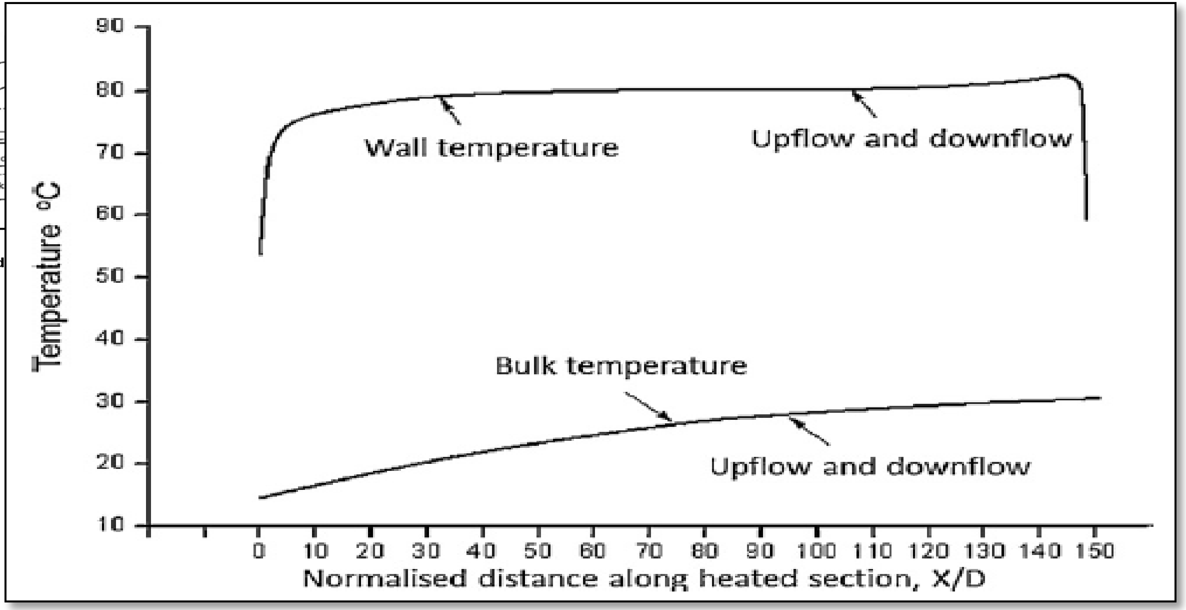
Flow direction and heat flux affect the wall temperature distribution



Figures from [Jackson 2013]

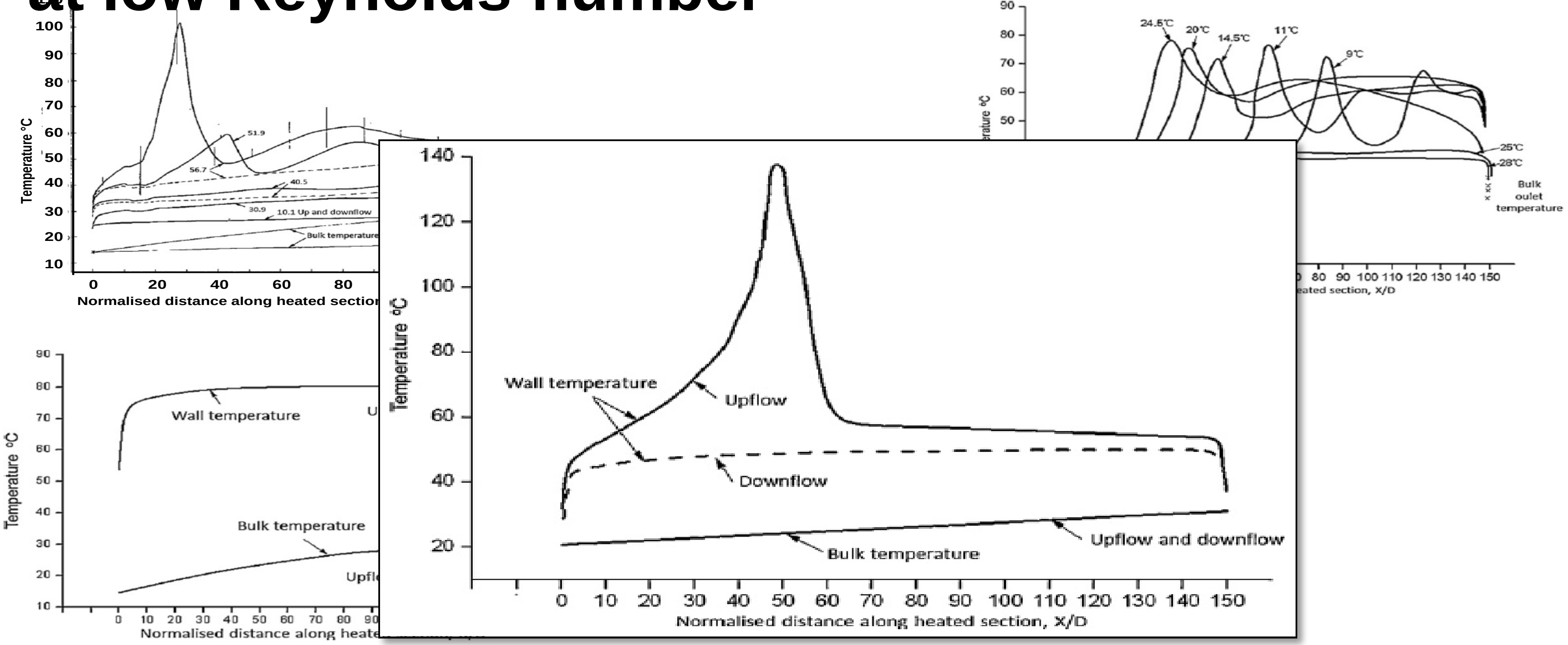
Upward and downward flow produces similar wall temperatures at high Reynolds number





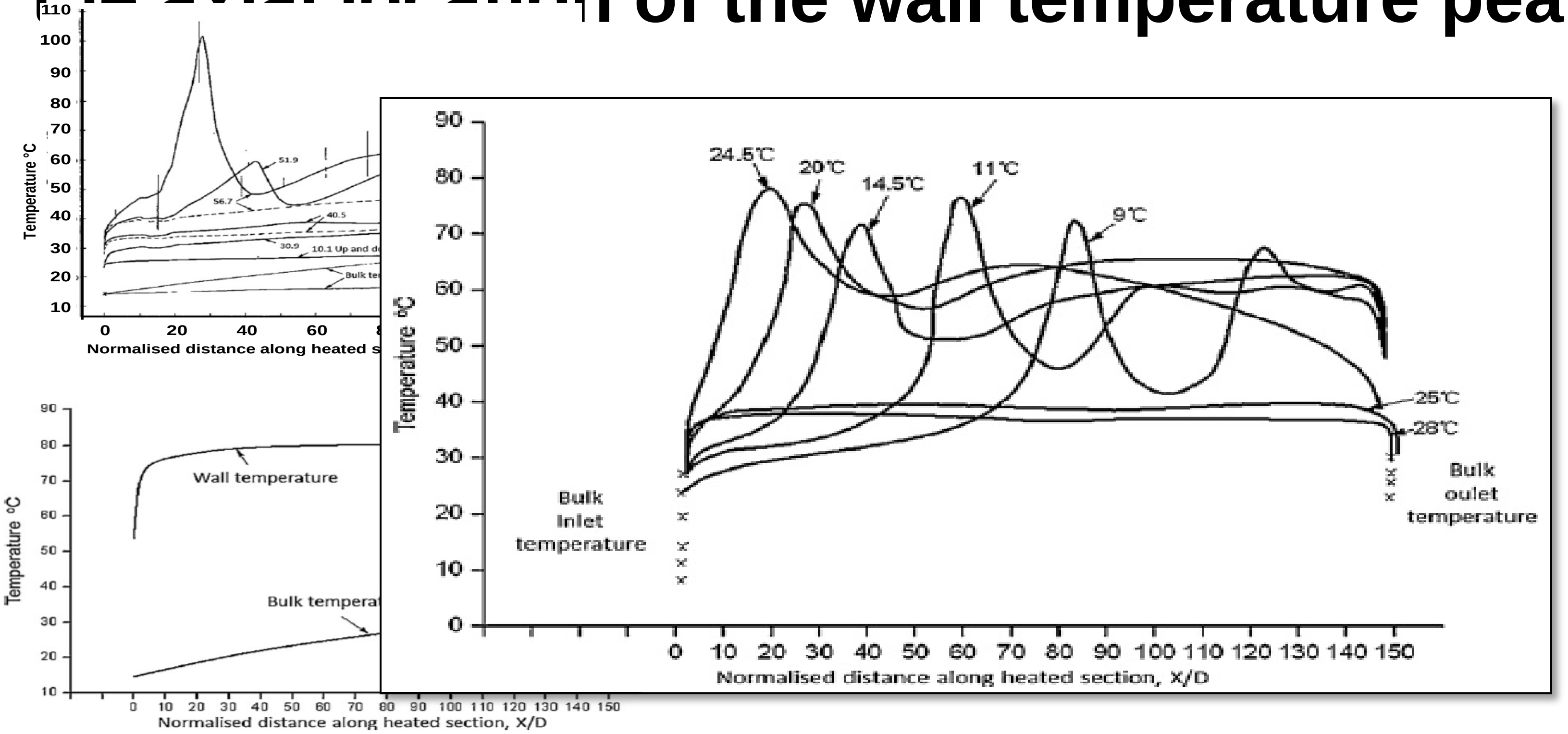
Figures from [Jackson 2013]

Upward flow produces a peak wall temperature at low Reynolds number



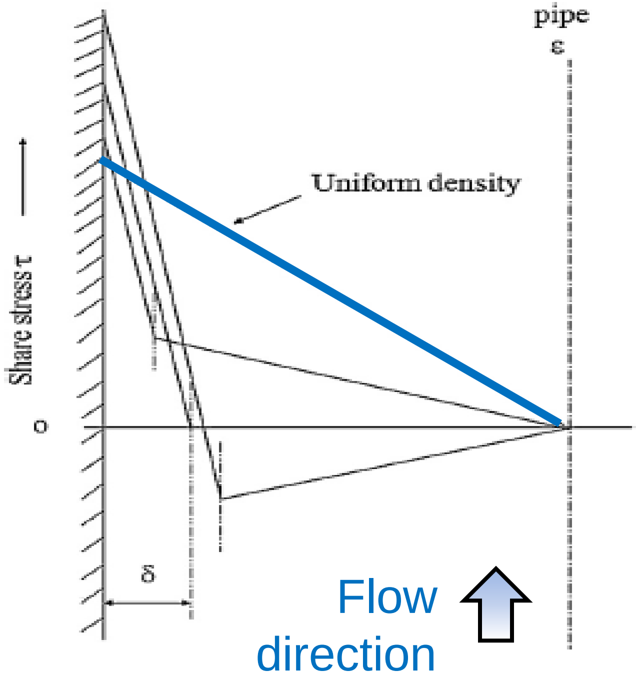
Figures from [Jackson 2013]

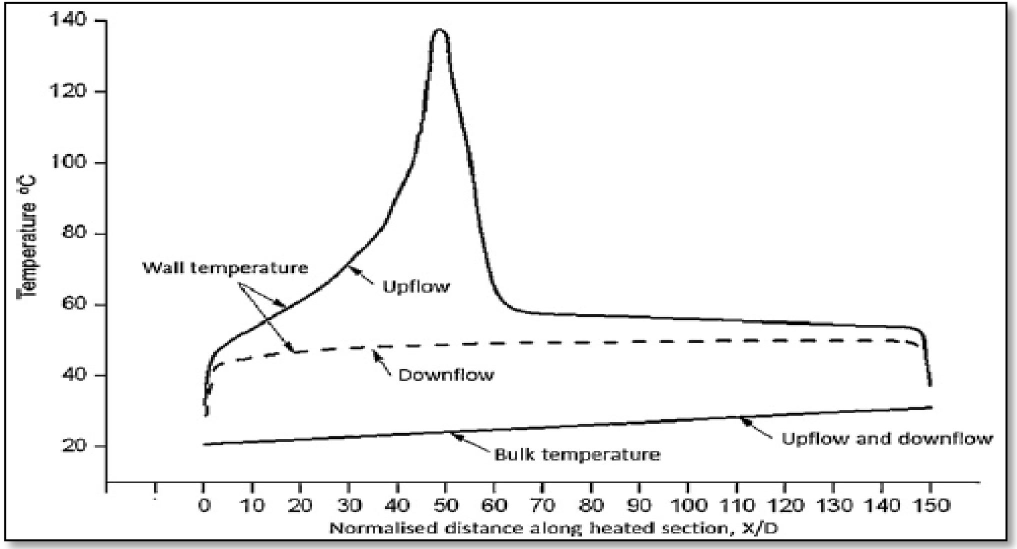
Inlet fluid temperature affects the axial location of the wall temperature peak



Figures from [Jackson 2013]

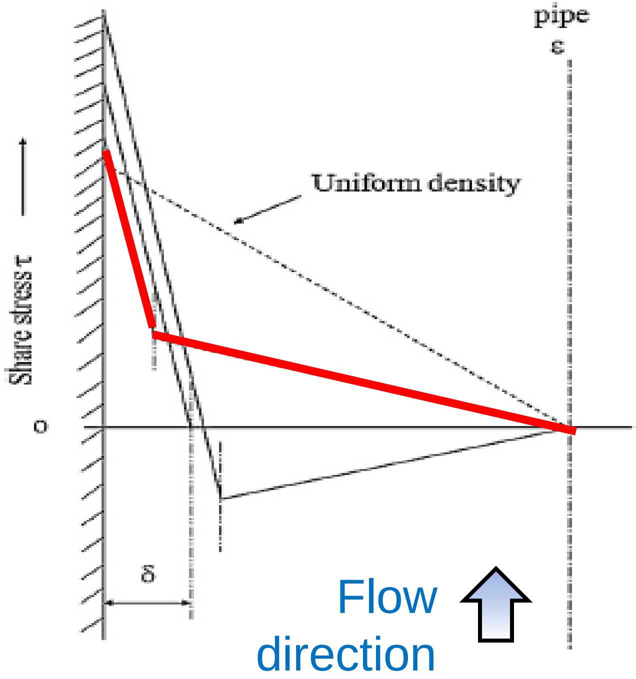
Heat transfer deteriorates and recovers due to buoyancy effects near the wall

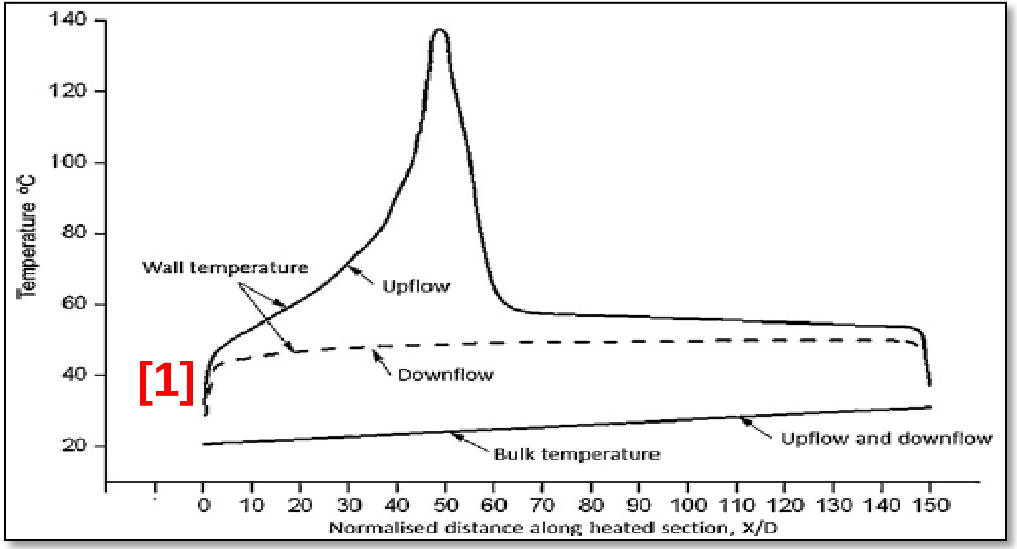




Figures from [Jackson 2013]

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

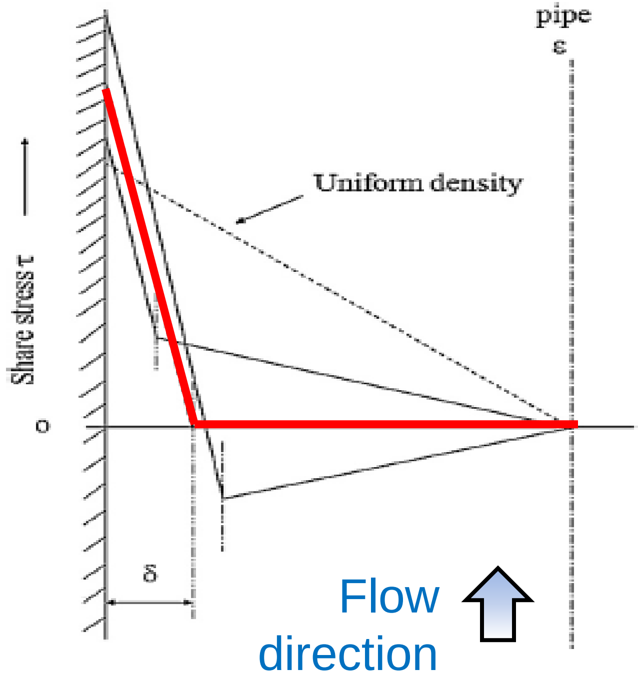


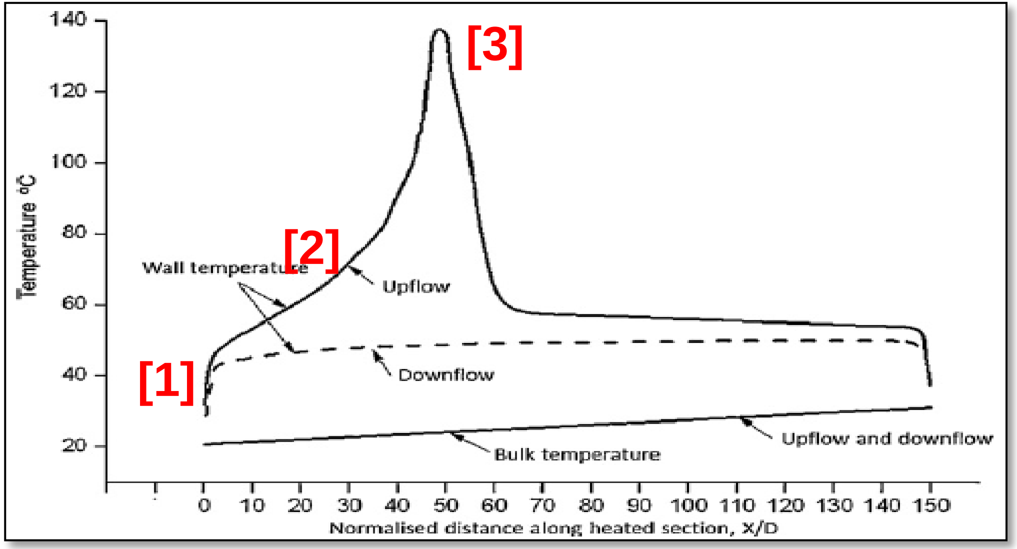


Figures from [Jackson 2013]

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

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

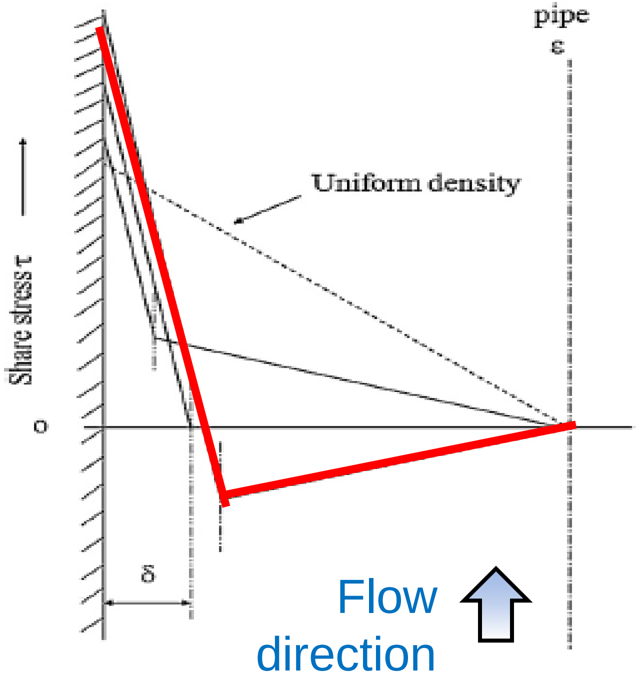


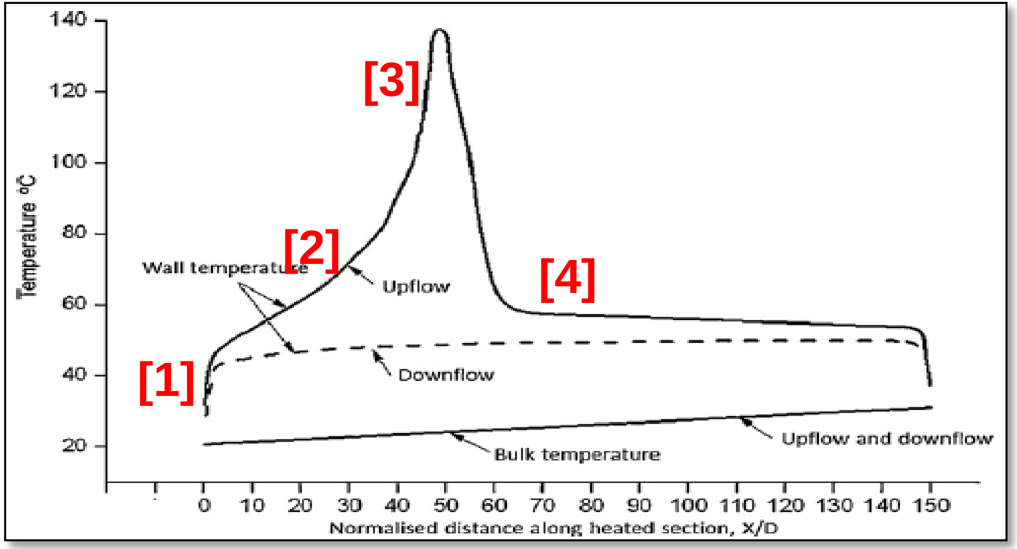


Figures from [Jackson 2013]

Growth of the buoyant wall layer causes the wall shear stress to decrease

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

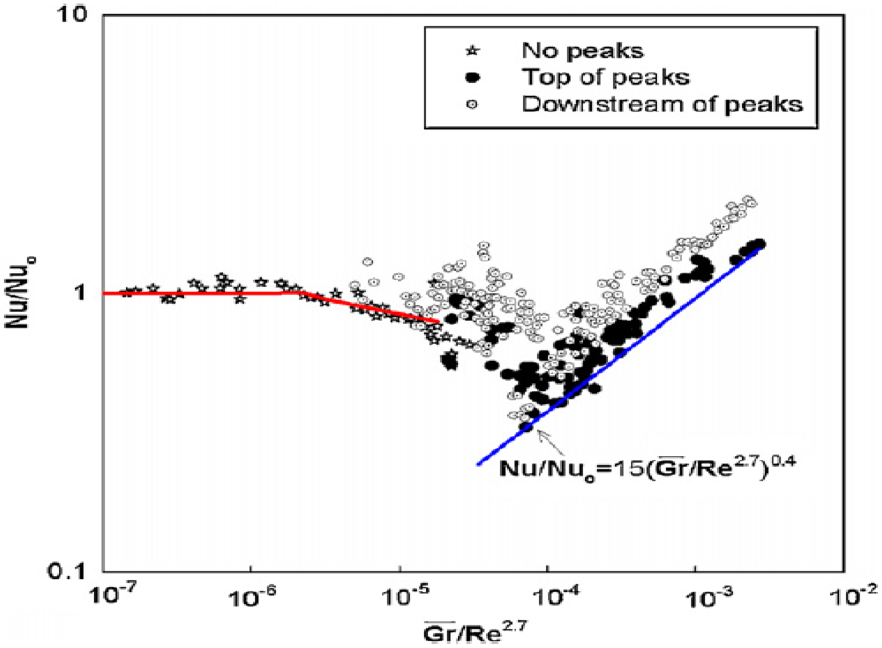




Figures from [Jackson 2013]

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

Buoyancy reduces or increases heat transfer in upward flow



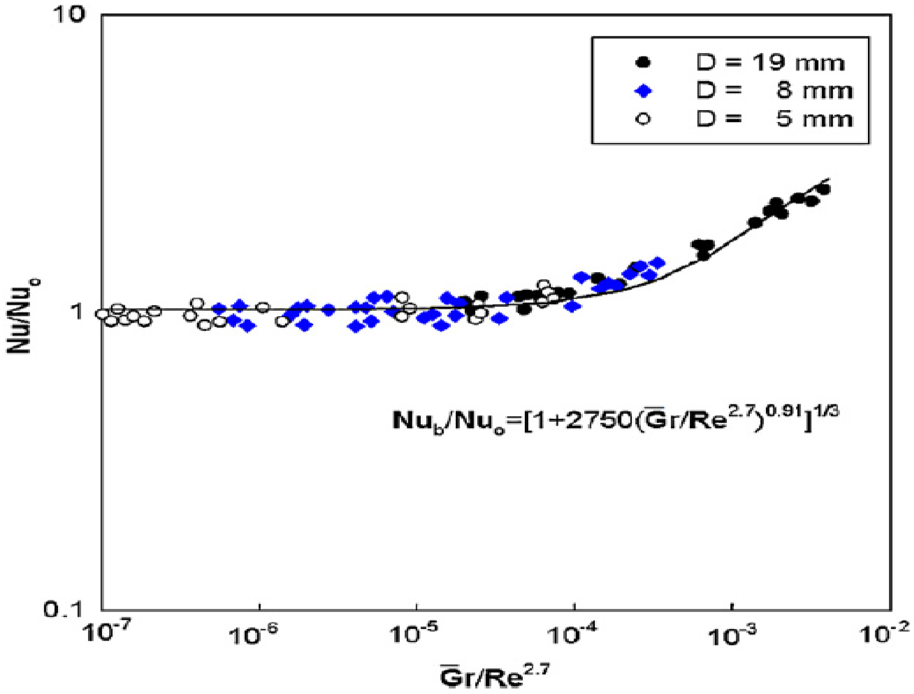
The onset of buoyant effects in upward flow: $\left(\frac{Gr_b}{Re_b^{2.7}}\right)\left(\frac{\mu_w}{\mu_b}\right)\left(\frac{\rho_b}{\rho_w}\right)^{1/2} > 10^{-5}$

[Jackson 1979a]

Nu_0 = Nusselt number for forced convection

[Jackson 2013]

Buoyancy in downward flow increases heat transfer by increasing the shear stress



Nu_0 = Nusselt number for forced convection

[Jackson 2013]

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