Tutorial:

Heat Exchangers for Supercritical CO2 Power Cycle Applications

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Tutorial:
Heat Exchangers for Supercritical CO2 Power Cycle Applications

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The following slides present an overview of heat exchangers in supercritical CO₂ applications:

- General heat exchanger overview and design trades
- Specific heat exchangers for sCO₂
- Heat exchanger mechanical design for sCO₂
- 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

![Diagram of a Rankine cycle](image)
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.
Some Conventional Heat Exchanger Layouts

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

[3] Stuhrlingenterprise
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

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:
The sky is the limit for heat exchanger concepts

Courtesy Thar Energy, DE-FE0026273
How Much Heat Transfer Area in a Heat Exchanger?

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

\[ D_h = \frac{3333}{\beta} \]
Design Trades for Heat Exchangers

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Heat exchanger design considerations

$s\text{CO}_2$ 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 $\Delta T$ becomes very small.

![Diagram showing temperature profiles for a counter-flow heat exchanger with pinch points at $T_{h,i}$ and $T_{h,o}$ for hot stream and $T_{c,i}$ and $T_{c,o}$ for cold stream. The diagram highlights the 'pinch' points where the temperature profiles of the hot and cold streams become parallel, indicating inefficient heat transfer.]
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.

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.

\[ D_h = \frac{3333}{\beta} \]
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

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 $\varepsilon$)
- Constraints ($\Delta P$, 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

PCHE
Printed Circuit Heat Exchanger

H²X
Hybrid

FPHE
Formed Plate Heat Exchanger
Main Components

- Etched plates
- Diffusion bonded core
- Headers, nozzles, flanges
- Formed plates
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

Current Typical Dimensions

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 sCO₂ Brayton Cycle

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Content

1. HXs Design Consideration
2. Benefit of Compact Microtube HXs
3. sCO₂ HXs Model
4. sCO₂ HXs Data
5. Summary
HXs Design Consideration

- Heater
- Recuperator
- Cooler
Standard sCO₂ Brayton Cycle

**Recuperator**
Counter-flow heat exchanger that increases the system efficiency by reusing energy in exhaust sCO₂ from turbine

**Cooler**
Heat exchanger that utilizes water (counter flow) or air (cross flow) to cool sCO₂ for compression

**Heater**
Cross-flow counter current heat exchanger that takes combustion gas to heat sCO₂ to high temperature
Goal: Meet performance requirements and provide margin of safety while minimizing over design.

Material of Construction
- Physical Properties
- Corrosion
- Contamination potential

ASME Code Stamp/Design

Fouling Factor

Design $T$ increases material strength drops & corrosion rates increase
Heater Design Considerations

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

**Design Conditions:**
- **Combustion Gas**
  - Max Temperature: 870°C
- **sCO₂**
  - Max Temperature: 715°C
  - Pressure: 280 bar
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

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

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

Lower Cost
- Reduce capital investment

Easy Maintenance
- Replaceable tube bundle and removable end cap
Cooler Design Considerations

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.

**Design Conditions:**
Max Temperature: up to 100°C
Pressure: 100bar
Benefit of Compact Microtube HX

- High Performance
- Smaller Footprint
- Lighter Weight
Surface Density and Heat transfer coefficient of heat exchangers are significantly improved by using microtube.
Figs shows the overall size comparison of microtube and conventional tube air to CO₂ cross flow heat exchangers with different tube sizes with the same capacity, effectiveness and air side pressure drop.

<table>
<thead>
<tr>
<th></th>
<th>1mm</th>
<th>3mm</th>
<th>7mm</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Total Tube Length</strong></td>
<td>16,800’’</td>
<td>9,240’’</td>
<td>7,020’’</td>
</tr>
<tr>
<td><strong>Tube Number</strong></td>
<td>600</td>
<td>220</td>
<td>90</td>
</tr>
<tr>
<td><strong>Bundle Weight</strong></td>
<td>4.5 lb</td>
<td>20 lb</td>
<td>90 lb</td>
</tr>
<tr>
<td><strong>Surface Density</strong></td>
<td>46 in²/in³</td>
<td>17 in²/in³</td>
<td>7 in²/in³</td>
</tr>
<tr>
<td><strong>Efficiency</strong></td>
<td>89%</td>
<td>89%</td>
<td>89%</td>
</tr>
</tbody>
</table>

1mm tube vs. 3mm tube vs. 7mm tube
Counter Flow Microtube Recuperator

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.

<table>
<thead>
<tr>
<th>Tube Length</th>
<th>1mm</th>
<th>3mm</th>
<th>7mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>1mm tube vs. 3mm tube vs. 7mm tube</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>40”</td>
<td>135”</td>
<td>450”</td>
<td></td>
</tr>
<tr>
<td>Tube Number</td>
<td>1500</td>
<td>175</td>
<td>30</td>
</tr>
<tr>
<td>Bundle Weight</td>
<td>17 lb</td>
<td>59 lb</td>
<td>244 lb</td>
</tr>
<tr>
<td>Surface Density</td>
<td>76 in²/in³</td>
<td>30 in²/in³</td>
<td>12 in²/in³</td>
</tr>
<tr>
<td>Efficiency</td>
<td>97%</td>
<td>97%</td>
<td>97%</td>
</tr>
</tbody>
</table>
Microchannel 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.

At Thar’s test facility, air and CO₂ approaching temperature as low as 2°F was achieved using micro-channel coil.
3

sCO₂ HX Model Selection
Heat Exchanger Calculation Method

**CO₂ properties change dramatically with little variation in supercritical region**

Used discretized model for sCO₂ heat transfer calculation
- Break the heat exchanger into n sections
- Calculate average properties of each section
- Interactively calculate the overall performance
Heat Transfer Equations

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

1. **CO₂ Side Nusselt Number**
   
   Petukhov (1970)
   
   \[ \text{Nu}_{\text{CO}_2} = \frac{ \left( \frac{f}{2} \right) \times \text{Re} \times \text{Pr} }{1.07 + \frac{900}{\text{Re}} - \frac{600}{1+10\text{Pr}} + 12.7 \left( \frac{f}{2} \right)^{\frac{3}{2}} (\text{Pr}^3 - 1)} \]

2. **Air Side Nusselt Number**
   
   Martin (2002)
   
   \[ \text{Nu}_{\text{air}}(\lambda) = 0.404 \times L_{\lambda}^{\frac{1}{3}} \]
   
   \[ L_{\lambda} = \begin{cases} 
   0.92H_g \times Pr_{\text{air}}(\lambda) \times \left( \frac{4X_T}{X_L} - 1 \right) & X_L \geq 1 \\
   0.92H_g \times Pr_{\text{air}}(\lambda) \times \left( \frac{4X_TX_L}{X_L - 1} \right) & X_L < 1 
   \end{cases} \]

3. **CO₂ Side Pressure Drop**
   
   Bhatti and Shah (1987)
   
   \[ f = 0.00128 + 0.1143 \text{Re}^{-0.331} \]
   
   \[ \Delta p_{\text{CO}_2} = f \frac{L \rho u^2}{d_e} \]

4. **Air Side Pressure Drop**
   
   Zukauskas (1988)
   
   \[ \Delta p_{\text{air}} = N_r \lambda \left( \frac{\rho_u^2}{2} \right)^f \]
sCO$_2$ HXs Test Loop and Data
Thar sCO₂ HX Test Loop vs. a standard sCO₂ 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
Purpose of Test Loop

1. Collect sCO₂ performance data
2. Validate calculation model
3. Verify mechanical design and material strength

Work supported by US DOE NETL under DE-FE0024012
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 CO₂ High Side Pressure: 280 bar
  - Design CO₂ Low Side Pressure: 100 bar
  - Maximum Temperature: 575°C

**Cooler**
- **Counter-flow, Shell & Tube Water Cooler**
  - Material: Stainless Steel 304
  - Design CO₂ Pressure: 100 bar

Work supported by US DOE NETL under DE-FE0024012 & DE-FE0025348
Data Collected

Test Condition

- CO₂ 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
Comparison of Actual data vs. Prediction: Recuperator

Good linear relationship between actual data and calculated data
Comparison of Actual data vs. Prediction: Heater

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

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

**Use of Microtube**
- Significantly improve thermal performance
- Smaller footprint and lighter

**Heat Exchanger Calculation Model**
- Discretized model increases accuracy
- Establish relationship between models and data

**sCO2 Brayton Cycle Testing Data**
- Microtube heat exchangers were successfully evaluated at Brayton cycle T & P conditions
- Test data confirms sCO2 microtube heat exchanger performance
- Good correlation between design & actual heat exchanger performance data
THANK YOU!
QUESTIONS?
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
WR-21 (25.2 MW)

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 serve as structural features for high-pressure (sCO₂) applications

Parting plates
• Provide fluid boundary between the two flows
Unit Cell Design

- Fully-welded pressure boundary ensures sealing
- Brazed fins react high internal pressures by acting as tensile support members
- Individually tested for quality control
- Customizable fin geometry

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

Plate-Matrix Heat Exchangers
Heat Transfer Matrices

Straight Fin  WavyFin  Wire Mesh

Plate-Matrix Heat Exchangers
Choosing a Matrix

- Cost
- Mass
- Footprint
- Size (Volume)

Screen Mesh (60 iso mesh)

Folded Fin (43 fins/inch)

Wire Mesh Extended Surface

Wavy Fin Extended Surface
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 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
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
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

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

Mission Definition

Start-Up

Temperature vs. Time

Dwell Histogram

% of Lifetime

Duty Cycle

Graphs and images related to temperatures, pressure, dwell histograms, and duty cycles are shown.
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 $\Delta T$
  - 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?)
Fatigue Considerations

- Highly design dependent gradient selection for $\Delta T$
- 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$_2$O$_3$ scale over chromia
- >20% Cr is key to oxidation resistance at 650$^\circ$C according to Sridharan et al.

1. 0.25 mg/cm² gain in sCO₂ vs. 0.045 in laboratory air after 1,000 hours
2. Aluminum addition with addition of humidity?


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

Type 310SS 650°C Oxidation sCO₂ vs. Air
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

\[ \Delta P_{\text{total}} = \Delta P_{\text{inlet manifold}} + \Delta P_{\text{entrance}} + \Delta P_{\text{internal flow}} + \Delta P_{\text{exit}} + \Delta P_{\text{outlet manifold}} \]

\[ \Delta P_{\text{internal flow}} = f \left( \frac{L}{D_h} \right) \frac{1}{2} \rho V^2 \]

\[ f = f(e, D_h, V, \rho, \mu) \]

\[ V = \frac{\dot{m}}{\rho A_f} \]

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 $\Delta h(T,P)$ to calculate energy gain or loss, instead of $mc_p$
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
  \[ \Delta P = \frac{Q \mu L}{k A_f} \]
  \( \mu \) = dynamic viscosity

• Wire-Mesh
  \[ f = \frac{2 \rho \Delta P}{G^2 \beta t \left( \frac{1 - \varepsilon}{\varepsilon} \right)^{0.4}} \]
  \( G \) = internal mass velocity
  \( \beta \) = surface area/volume

• 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
sCO₂ 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.

![Graph showing the change in specific heat capacity (Cp) with temperature at different pressures (4 MPa, 8 MPa, 12 MPa, 16 MPa, 20 MPa). The graph indicates a peak in Cp at critical points, with temperatures ranging from 200 K to 400 K and pressures ranging from 4 MPa to 20 MPa.](image-url)
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 $\Delta T$ becomes very small

Counter-Flow heat exchanger

Distance along Heat Exchanger

Temperature

Pinch Point

$T_{h,i}$, $T_{c,o}$, $T_{c,i}$, $T_{h,o}$
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.

\[
UA = \left( \frac{1}{(hA)_{\text{cold}}} + \frac{\ln(R_i/R_o)}{2\pi k} + \frac{1}{(hA)_{\text{hot}}} \right)^{-1}
\]
Typical approximations for heat exchanger sizing are not valid for near-critical sCO₂

**General equation**

**Heat transfer**

\[ Q = w(c_o - c_i) \]

\[ Q = \varepsilon C_{min}(T_{h,i} - T_{c,i}) \]

**Overall heat transfer coefficient**

\[
\frac{1}{UA} = \frac{1}{(hA)_i} + \frac{\ln(Do/D_i)}{2\pi kl} + \frac{1}{(hA)_o}
\]

**Typical approximation**

**Heat transfer**

\[ Q = wC_p(T_o - T_i) \]

\[ Q = UA\Delta T_{LM} \]

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

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

\[ h = f(Nu) = CR_{ex}Pr^y \]
Typical correlations based on average fluid properties are not applicable near the critical point

\[ Q = hA\Delta T \]

\[ h = f\left(\frac{k}{L} Re^x Pr^y\right) \]

\[ h = f_{nc}(k\nu^{-x} Pr^y) \]

Assume: \( x=\frac{4}{5}, \ y=\frac{1}{3} \)
Dittus-Boelter type correlations with property variation are valid when buoyancy is negligible

Test data screened for buoyancy

\[ \text{Nu}_b = C \text{Re}_b^{m_1} \text{Pr}_b^{m_2} \left( \frac{\rho_w}{\rho_b} \right)^{m_3} \left( \frac{C_p}{C_{pb}} \right)^{m_4} \]

\( b \) = bulk
\( w \) = wall

[Values from Jackson 2013]
Discretizing the heat exchanger accounts for property differences that affect fluid temperature

Operating conditions and geometry from:
1D prediction methods match well with experimental measurements when the HX is discretized.
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.

A modification factor \( \sim 0.75 \) should be used for the approximate \( UA \) value.

Elliptical flow passages assumed for HTC.

Courtesy Thar Energy, DE-FE0026273
Questions?


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.

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

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.

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 flow rate 0.02 kg/s; wall heat flux 33.6 kW/m²; Re ~ 4 x 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~2.5x10^5)

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

Fig. 9. Highest mass flow rate, 5mm diameter tube, pressure 7.58 MPa, upflow only. Mass flow rate 0.0645 kg/s; wall heat flux 455 kW/m².

Fig. 10. Further reduction of flow rate, 5mm diameter tube, upflow and downflow. Pressure 7.58 MPa; mass flow rate 0.0129 kg/s; wall heat flux 68 kW/m²; Re~4 x 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)

![Graph showing stress vs. temperature for different materials]

*Courtesy Thar Energy, DE-FE0026273*
Corrosion occurs over a long period of time to build an oxide layer on the heat transfer surfaces.

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 important to reduce corrosion allowance

Image from Premier Separator Services Limited

Design conditions should include margin on top of the operating conditions

<table>
<thead>
<tr>
<th>Condition</th>
<th>Temperature</th>
<th>Pressure</th>
<th>Comment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operating</td>
<td>400°C</td>
<td>15 bar</td>
<td>Expected Operating Conditions</td>
</tr>
<tr>
<td>Design</td>
<td>430°C</td>
<td>17.25 bar</td>
<td>Add 30°C margin and 5% for PSV</td>
</tr>
</tbody>
</table>

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

- Fluid Inlet Conditions
- Fluid Outlet Conditions

Heat exchanger layout, sizing

Use the LMTD Approach to get UA

\[ Q = UA \Delta T_{LM} = \dot{m}(h_{i} - h_{o}) \]

Use the ε-NTU Method to get UA

\[ Q = \varepsilon Q_{max} \]
\[ UA = NTU \cdot C_{min} \]
\[ NTU = fn(c(\varepsilon, C)) \]

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.

\[ dq = U \Delta T dA \]

\[ Q = U A \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)} \]

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

\[ dq = U \Delta T \, dA \]

\[ Q = U A \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} \]

Calculating UA by the LMTD Method

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

\[ Q = m(h_{h,i} - h_{h,o}) \]

**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

\[ Q = FUA\Delta T_{LM} \]

The effectiveness ($\varepsilon$) - NTU Method

- Fluid inlet conditions
- Thermal duty ($Q$)

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

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

$$Q_{\text{max}} = \min(C_c, C_h) \times (T_{h,i} - T_{c,i})$$

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

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

$$NTU = fnc(\varepsilon, C)$$
The effectiveness ($\varepsilon$) -NTU Method

- Fluid inlet conditions
- Thermal duty ($Q$)

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

$NTU = fnc(\varepsilon, C)$

NTU = Net Transfer Units

- Derivation provided in most heat transfer textbooks
- Calculated from $\varepsilon$ and ratio of $C_{\text{min}}/C_{\text{max}}$

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

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 ($\varepsilon$) - NTU Method

- Fluid inlet conditions
- Thermal duty ($Q$)

\[
Q = \varepsilon Q_{\text{max}} \quad \text{and} \quad UA = NTU \cdot C_{\text{min}}
\]

\[
NTU = fnc(\varepsilon, C)
\]
The $\varepsilon$-NTU method is a good way to quickly identify heat exchanger types.
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

CO₂

\[ \dot{m} = 0.01963 \text{ kg/s} \]
\[ T_{in} = 394.2 \text{ K} \]

H₂O

\[ \dot{m} = 0.04011 \text{ kg/s} \]
\[ T_{in} = 293.8 \text{ K} \]
The test data is trans-critical
Water and CO₂ wall temperature is used for validation

![Graph showing temperature vs distance along heat exchanger]

- Blue dots: Pitla 2001: Run 1, Water
- Red squares: Pitla 2001: Run 1, Wall

Temperature [K] vs Distance Along Heat Exchanger [m]

CO₂ and H₂O flows are indicated in the diagram.
Conventional heat exchanger calculation methods can be compared to a discretized enthalpy method. The ε-NTU Method (average fluid properties): 

\[ \text{NTU} = \frac{UA}{C_{\text{min}}} \] 

\[ \varepsilon = \frac{1 - \exp[-\text{NTU}(1 - C_r)]}{1 - C_r\exp[-\text{NTU}(1 - C_r)]} \] 

\[ \varepsilon = \frac{Q}{Q_{\text{max}}} = 98.6\% \]

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

\[ (h_{c,n}) = (h_{c,n-1})^{i-1} + \frac{\text{UA}}{\dot{m}c} (T_{h,n} - T_{c,n})^{i-1} \]

\[ \varepsilon = \frac{Q}{Q_{\text{max}}} = 32.5\% \]

\[ \dot{m} = 0.01963 \text{ kg/s} \] 

\[ T_{\text{in}} = 394.2 \text{ K} \]

\[ \dot{m} = 0.04011 \text{ kg/s} \] 

\[ T_{\text{in}} = 293.8 \text{ K} \]
The heat exchanger should be discretized to accurately account for fluid property variations

![Graph](image-url)

- Specific Heat Capacity (Cp) [kJ/kg/K]
- Distance Along Heat Exchanger [m]

- Discretization: CO2
- e-NTU: CO2
- Discretization: Water
- e-NTU: Water
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
CO₂ density decreases near the critical point, which can induce buoyancy effects
Heat transfer deteriorates and recovers due to buoyancy effects near the wall

[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
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

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

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

Figures from [Jackson 2013]
Heat transfer deteriorates and recovers due to buoyancy effects near the wall

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

Figures from [Jackson 2013]
Heat transfer deteriorates and recovers due to buoyancy effects near the wall.

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

\[ \frac{\text{Nu}}{\text{Nu}_0} = 15 \left( \frac{\text{Gr}}{\text{Re}^{2.7}} \right)^{3.4} \]

The onset of buoyant effects in upward flow:

\[ \frac{\text{Gr}_b}{\text{Re}_b^{2.7}} \left( \frac{\mu_w}{\mu_b} \right) \left( \frac{\rho_b}{\rho_w} \right)^{1/2} > 10^{-5} \]

[Jackson 1979a]

\[ \text{Nu}_0 = \text{Nusselt number for forced convection} \]

[Jackson 2013]
Buoyancy in downward flow increases heat transfer by increasing the shear stress

\[ \frac{Nu_b}{Nu_0} = \left[1 + 275C\left(\frac{Gr}{Re^{2.7}}\right)^{0.91}\right]^{1/6} \]

*\( Nu_0 = \) Nusselt number for forced convection

[Jackson 2013]


