Increasing Main Cooler Thermal Performance for sCO$_2$ Power Cycles

Matthew Searle
NETL Support Contractor

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Authors and Contact Information

Matthew Searle$^{1,2}$; Doug Straub$^1$

$^1$National Energy Technology Laboratory, 3610 Collins Ferry Road, Morgantown, WV 26505, USA
$^2$NETL Support Contractor, 3610 Collins Ferry Road, Morgantown, WV 26505, USA
Introduction and Motivation

- sCO₂ power cycle performance is highly dependent on ambient temperature (Wright, 2011; Conboy, 2015).
- Reducing CO₂ cooler outlet temperature increases cycle efficiency and lowers levelized cost of electricity (LCOE) (Pidaparti, 2020).
- Heat transfer enhancement integrated via monolithic additive manufacturing (AM), is a pathway to lower cost heat exchangers.
- AM heat exchangers may be cost competitive with printed circuit heat exchangers (PCHE) for small-duty applications (Robey, 2022).

6 °C reduction* -> + 1.4% point in efficiency
- 3.8% in LCOE

* Provided effective cooler and heat rejection temperature (Pidaparti, 2022)
Materials and Methods

- Shell and tube heat exchanger constructed with conventional tube.
- Inlet flows instrumented with mass flow, pressure, and temperature.
- Outlet flows instrumented with temperature and differential pressure.
- Wall temperature measurements on shell side.

### Table

<table>
<thead>
<tr>
<th></th>
<th>CO₂ (Tube)</th>
<th>Water (Shell)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Inlet Temperature (K)</strong></td>
<td>$T_{t,i}$</td>
<td>$T_{s,i}$</td>
</tr>
<tr>
<td><strong>Inlet Pressure (MPa)</strong></td>
<td>$P_{t,i}$</td>
<td>$P_{s,i}$</td>
</tr>
<tr>
<td><strong>Mass Flow Range (kg/s)</strong></td>
<td>$m_t$</td>
<td>$m_s$</td>
</tr>
<tr>
<td><strong>Reynolds Number Range, (-)</strong></td>
<td>$Re_t$</td>
<td>$8 \times 10^4$ to $1.3 \times 10^5$</td>
</tr>
<tr>
<td><strong>Mass Flow Range (kg/s)</strong></td>
<td>$m_s$</td>
<td>$0.016–0.126$</td>
</tr>
</tbody>
</table>
• Script reduces data to determine local heat transfer coefficients.
  • Assumption of uniform heat flux (see the following slide on approach verification).
  • Heat duty calculated from inlet and outlet conditions. Friction factor from pressure drop.
  • Uncertainty analysis by Kline and McClintock approach.

• Two test articles:
  • Conventional commercial tubing.
  • Additively manufactured (AM) tubing with square cross-section and rib turbulators.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Exchanger Length</td>
<td>$L$</td>
<td>1.04 m</td>
</tr>
<tr>
<td>Conventional Tube Outer Diameter</td>
<td>OD</td>
<td>9.5 mm</td>
</tr>
<tr>
<td>Conventional Tube Hydraulic Diameter</td>
<td>$D_h$</td>
<td>7.04 mm</td>
</tr>
<tr>
<td>Square Tube Inner Hydraulic Diameter (side wall length)</td>
<td>$D_h$</td>
<td>4.98 mm</td>
</tr>
<tr>
<td>Rib Height</td>
<td>$e$</td>
<td>0.39 mm</td>
</tr>
<tr>
<td>Rib Angle</td>
<td>$\alpha$</td>
<td>60°</td>
</tr>
<tr>
<td>Rib Pitch to Height</td>
<td>$P/e$</td>
<td>10</td>
</tr>
<tr>
<td>Rib Height to Hydraulic Diameter</td>
<td>$e/D_h$</td>
<td>0.078</td>
</tr>
</tbody>
</table>
Verification of Test Approach

Uniform Heat Flux Assumption

- Full conjugate model: convection in shell (water) and tube (CO₂) and conduction in wall (316 Stainless Steel).
- Reynolds averaged Navier Stokes (RANS).
- k-ω shear stress transport (SST) turbulence model.
- REFPROP 10.0 property data provided at 50 points spanning temperature range in CO₂.
- 0.5 mm mesh (heat transfer coefficient resolved to within 1.6%).
- Inlet/outlet conditions and geometry match experiments.
- Adiabatic shell outer wall. No axial heat flux in pipe at inlet/outlet.

![Diagram showing shell, tube, and wall with mesh representation.]
Local Surface Temperatures

\[ \theta_w = \frac{T_w - T_{s,i}}{T_{t,i} - T_{s,i}} \]

\[ Re_t = 1 \times 10^5 \]

30% increase in \( \theta_w \) (decrease in \( T_w \)) for \( x > 0.6 \) m
Local Heat Transfer Coefficients

Smooth conventional tube

\[ Re_t = 1 \times 10^5 \]

\[
\begin{array}{c}
\text{Smooth conventional tube} \\
\text{Square tube with ribs}
\end{array}
\]

\[ Re_t = 1 \times 10^5 \]
Comparison to Correlation

$Re_t = 1 \times 10^5$

$T_{pc} = 307.8$ K

Smooth conventional tube

$Nu_b = 0.023 \text{Re}^{0.8} \text{Pr}^{n}$

Dittus-Boelter

$Nu_b = \begin{cases} 
0.14 \text{Re}_b^{0.69} \text{Pr}_b^{0.66} & \frac{T_{pc}}{T_b} < 1 \\
0.013 \text{Re}_b^{0.35} \text{Pr}_b^{-0.05} & \frac{T_{pc}}{T_b} \geq 1 
\end{cases}$

Yoon (2003)

Square tube with ribs

$St_{2} = \frac{f_i \lambda}{2 \left[ G(e^*) - R(e^*) \right] \sqrt{\frac{f_i}{2} + 1}}$

Han & Park (1988)
Average Heat Transfer Coefficient, $\bar{h}$, and Friction Factor, $f$, Results

$\bar{h}$ increases by 70% to 180%.

$Q''$, tube wall heat flux  
$G$, tube mass flux  
$Re_t = 1 \times 10^5$
Heat Exchanger Effectiveness and Cycle Efficiency

\[ \varepsilon = \frac{(T_{t,i} - T_{t,o})}{(T_{t,i} - T_{s,i})} \]

Data adapted from Pidaparti (2022)

+13% (0.072) in \( \varepsilon \) -> +0.6% point in cycle efficiency
Conclusions

• For $T/T_{pc} < 1.05$, Yoon correlation was more accurate than Dittus-Boelter.
• Angled rib tubes had 70% to 180% larger average heat transfer coefficient than smooth tube.
• Heat exchanger effectiveness due to angled rib turbulators was greater at low water flow rates and high sCO$_2$ flow rates.
• Angled ribs yielded a 13% increase in heat exchanger effectiveness (0.072 increment) at tube Reynolds number equal to $1.3 \times 10^5$.
• The 0.072 increment in effectiveness yields 0.6% point improvement in cycle efficiency.
Acknowledgments

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References


