A Comparison of Supercritical Carbon Dioxide Power Cycle Configurations with an Emphasis on CSP Applications

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Supercritical Carbon Dioxide Background

- Thermodynamic properties of CO₂ deviate from normal trends near the critical point (31°C, 7.4 MPa)
  - Increased density allows for high back work ratio in the cycle, which boosts efficiency
  - Increased thermal capacitance results in pinch points in heat exchangers and also impacts heat rejection decisions
Attractive Features of s-CO$_2$ Brayton Cycle

- Simpler cycle design than steam Rankine
- Potentially achieves higher efficiency than steam Rankine
- High density working fluid yields compact turbomachinery
- Viable turbine designs from 10 to 300 MWe
- Low-cost, low-toxicity, low-corrosivity fluid; thermally stable at temperatures of interest to CSP (550°C to 750°C)
- Single phase reduces operational complexity; integrates well with sensible heat storage in CSP systems
- Multiple high-efficiency cycle configurations possible to match to a specific application
Considerations for CSP Integration

Factors for integrating s-CO2 power cycles into CSP plants

1. Superior performance vs. steam Rankine at dry cooling
2. Economic integration of TES

\[ \text{Energy}_{TES} = \text{mass} \times \text{specific heat} \times (T_{\text{hot}} - T_{\text{cold}}) \]

For sensible heat storage:

- The required mass of HTF is proportional to the hot and cold tank temperatures.
- All else equal, a cycle with a larger temperature difference is preferred to a cycle with a smaller temperature difference.
Configurations

Simple

Recompression

Partial Cooling
Modeling Background

• One reheat stage increased efficiency around 1.2% for a recompression cycle (Dostal, 2004)
• Partial cooling cycle achieves competitive efficiency with the recompression cycle while offering larger HTF temperature differences (Dostal, 2011)
• Dry-cooled partial cooling and recompression cycles have potential to achieve > 50% efficiency (NREL, 2013)

Summary

• Recompression and partial cooling efficiencies are similar, but studies have not investigated the heat exchanger requirements

Objective

• Model the heat exchangers using a conductance (UA) model and compare the efficiency, HTF temperature difference, and other useful cycle performance metrics as a function of allocated conductance
Recuperator Modeling Approaches

1. Select HX effectiveness and minimum temperature difference
   - Simplest approach to approximate HX performance
   - Does not consider temperature profile within HX (e.g. pinch points)
   - Non-dimensional metric – does not correlate to HX size

2. Select HX conductance (UA)
   - Calculates HX performance based on approximation of HX size
   - Correlates performance and size without requiring specific physical dimensions – useful for relative comparisons
   - Does not capture effects of specific design decisions or varying fluid properties

3. Select a HX design
   - Requires realistic dimensions and heat exchanger material properties
   - Most complex and computationally expensive approach
   - Provides the best data with which to compare different cycles
### Design and Optimized Parameters for Case Studies

#### Design Parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine efficiency</td>
<td>93%</td>
<td>Projection of mature, commercial size axial flow turbine efficiency</td>
</tr>
<tr>
<td>Compressor efficiency</td>
<td>89%</td>
<td>Projection of mature, commercial size radial compressor</td>
</tr>
<tr>
<td>Heat exchanger effectiveness</td>
<td>97%</td>
<td>5°C minimum temperature difference, neglect pressure drops</td>
</tr>
<tr>
<td>Heat exchanger conductance (UA)</td>
<td>Varied MW/K</td>
<td>Neglect pressure drops</td>
</tr>
<tr>
<td>Turbine inlet temperature</td>
<td>650°C</td>
<td>SunShot target for CSP power tower outlet temperatures</td>
</tr>
<tr>
<td>Compressor inlet temperature</td>
<td>50°C</td>
<td>Possible under dry cooling with 35°C ambient temperature</td>
</tr>
<tr>
<td>Upper pressure</td>
<td>25 MPa</td>
<td>Upper limit given available and economic piping</td>
</tr>
<tr>
<td>Turbine Stages</td>
<td>2</td>
<td>One stage of reheat at average of high and low side pressures</td>
</tr>
<tr>
<td>Net power output</td>
<td>35 MW</td>
<td>Estimate of power cycle requirements for a 100 MW-thermal SunShot target power tower with a solar multiple of 1.5</td>
</tr>
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#### Optimized Parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Relevant Cycles</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure ratio (PR)</td>
<td>All</td>
</tr>
<tr>
<td>Fraction of total UA allocated to HTR</td>
<td>Recompression, Partial Cooling (not applicable for effectiveness approach)</td>
</tr>
<tr>
<td>Ratio of pressure ratios (rpr)</td>
<td>Partial Cooling (sets intermediate pressure)</td>
</tr>
</tbody>
</table>
Results – Recuperator Effectiveness Model

- Similar efficiencies for complex cycles
- Much larger recuperator for the recompression cycle
- Smaller pre-cooler for recompression cycle (rejects more heat at higher temperatures, Seidel 2010)

*Effectiveness model does not give complete picture*

\[ \varepsilon = 97\% \]
\[ \Delta T_{\text{min}} = 5^\circ\text{C} \]
Results – Recuperator Conductance Model

- Significantly different results when recuperator conductance is specified
- At smaller conductance values, the recompression cycle reverts to simple cycle behavior (Bryant 2011, Dyreby 2012)
- As conductance reaches largest values
  - Recompression cycle efficiency reaches partial cooling efficiency
  - Recompression $\Delta T$ decreases more rapidly than partial cooling $\Delta T$
Modeling Limitations

• Neglecting pressure drops
  ▪ Larger pressure difference in partial cooling cycle
  ▪ Lower densities in partial cooling cycle may require more or larger channels

• Conductance HX model does not consider the impact of absolute pressures and pressure differentials

• Does not consider impact of varying fluid properties on heat transfer coefficients

• Pre-cooler model is not optimized with the cycle design
  ▪ Trade-offs between cycle efficiency, pre-cooler size, and fan parasitics should be considered
Integration with Direct CO2 Receivers

• Ongoing research of direct s-CO2 receivers
  ▪ NREL, Brayton Energy, OSU/PNNL, CSIRO

• Potential advantages of integration with partial cooling cycle:
  1. Lower average temperature of receiver may help reduce thermal losses
  2. Enables longer flow paths in receiver
     1. Stabilizes mass flow rate through parallel tubes
     2. Reduces deviation of absorbed energy per tube
  3. Lower total mass flow rate reduces header piping sizes
  4. Greater potential to decrease the high pressure in the receiver
Conclusions

- Using a conductance model for the recuperator provides a more equivalent comparison than an effectiveness model.
- The partial cooling cycle outperforms the recompression cycle until large quantities of conductance are modeled.
- The partial cooling cycle offers a larger temperature difference across the primary heat exchanger, which is critical to TES integration in CSP systems. It also may offer benefits to direct receiver designs.
- Studies of first-principle models are available for simple and recompression cycles. This work suggests that similar studies would be worthwhile for the partial cooling cycle in order to better understand heat exchanger dimensions and off-design performance.
THANK YOU!