Dry Cooling System Options for the sCO$_2$
Brayton Cycle for Concentrating Solar Thermal Power

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Locations suitable for CSP, with good solar resource
- Arid locations – no cooling water
- Dry cooling – to ambient air

Supercritical CO\textsubscript{2} Brayton cycle:
- Higher cycle efficiency (>560 °C)
- More compact components
- Simpler plant configuration

Main compression in supercritical region, however:
- CO\textsubscript{2} properties change rapidly with small ΔT
- Cycle performance sensitive to CIT – > dependent on \( T_{\text{amb}} \)

Dostal, V., A Supercritical Carbon Dioxide Cycle for Next Generation Nuclear Reactors. 2004, MIT
Supplementary cooling systems, which can cool past $T_{amb} \rightarrow$ decoupling CIT from $T_{amb}$:
- Vapour compression refrigeration cycle
- Heat driven absorption cooling

For CIT = 33-34 °C with NDDCT $\rightarrow$ design point $T_{amb}$ 20-25 °C, above which $\rightarrow$ cycle performance deterioration

For Alice Springs, NT, this is 40-60% of a typical mean year
Introduction – Cooling system options

- Case 1 – NDDCT only
- Case 2 – NDDCT and VCRC
- Case 3a – NDDCT and LiBr-H₂O ARC – thermal storage as ARC heat source
- Case 3b – NDDCT and LiBr-H₂O ARC – cooling sCO₂ stream as ARC heat source

NDDCT: natural draft dry cooling tower
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VCRC: Vapour compression refrigeration cycle
Case 3b:
- Can provide constant CIT
- Requires work input
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ARC: Absorption refrigeration cycle

Case 3a:
- Can provide constant CIT
- Heat input as parasitic
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3b heat input from LTR outlet
• Small ↑ in CIT → large ↑ \( T_{in,ARC} \)
• CIT is variable
• Utilises waste heat

Modelling methodology

Modelling with IPSEpro v6.0:
• Standard models for power cycle components were used
  • Constant turbomachinery performance
  • Constant assumed pressure drop
  • CO₂ and R134a properties from REFPROP¹
  • LiBr-H₂O solution properties from Ref²

• Models developed for cooling system components
• 1-D models developed for NDDCT³ & ACC⁴ (VCRC)
  • To capture effect of Tₐmb, & cooling system inlet conditions on CIT
  • Cooling towers sized to provide reasonable performance

<table>
<thead>
<tr>
<th>Variable</th>
<th>Value</th>
<th>Unit</th>
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<tbody>
<tr>
<td>Net power output at design point</td>
<td>25</td>
<td>MWe</td>
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<tr>
<td>Turbine inlet temperature</td>
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<td>°C</td>
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<tr>
<td>Turbine inlet pressure</td>
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<td>MPa</td>
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<td>Turbine outlet pressure</td>
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<td>Turbine efficiency</td>
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<td>%</td>
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<tr>
<td>sCO₂ compressor efficiency</td>
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<td>%</td>
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<tr>
<td>VCRC compressor efficiency</td>
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<td>%</td>
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<tr>
<td>ACC fan efficiency</td>
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<td>%</td>
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<td>Heat exchanger pinch point, ∆Tₚp</td>
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<td>Heat exchanger pressure drop</td>
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<td>kPa</td>
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</table>

Results - Overall

- Case 1 – reasonably good $\eta_{th}$, but lowest $W_{net}$
- Case 2 – fairly good $\eta_{th}$ & $W_{net}$ up to ~35 °C – performance declines at high $T_{amb}$
  - Parasitic load could be supplied by solar PV – High $T_{amb}$ (supplementary cooling), typically coincide with good conditions for PV
- Case 3a – lowest $\eta_{th}$, highest (constant nominal) $W_{net}$
  - Surplus heat available in CSP system? High $T_{amb}$ → longer days (summer) → $Q_{ln,ARC}$ wouldn’t be parasitic → Wouldn’t affect $\eta_{th}$
- Case 3b – best overall performance
Results – Effect of selected CIT

Case 2 (VCRC):

Case 3a (ARC from TES):
Results – Case 3a vs. 3b

- $Q_{in, Gen} \& Q_{out, ARC-CT}$ for case 3a ~double case 3b values
- Case 3b significantly dampens CIT from $T_{amb}$
- Potential alternative solution hybrid 3a and 3b to reduce $\eta_{th}$ penalty
Conclusions

• Only Case 3b outperforms base case in terms of $\eta_{th}$
• All outperform base case in terms of $W_{net}$
• Case 2: More appealing if coupled with solar PV – cost dependent
• Case 3a: More feasible if supplementary cooling demand coincides with surplus heat availability – CSP sizing dependent
• Case 3b: Best overall performance & greatly dampens effects of $T_{amb}$ on CIT
  • Case 3a + 3b – > provide nominal $W_{net}$ with reduced $\eta_{th}$ penalty – Future work
• Optimal cooling system selection depends on:
  • Value to system of constant cycle operating with $T_{amb}$ – simplify control systems
  • Plant operating role within grid, eg. base load might favour Case 3a
  • Techno-economic optimisation results – Future work
Questions?
Cooling System Sizing/Setup

- NDDCT sized to achieve optimal CIT (34 °C) at ambient temperature = 20 °C
- NDDCT size is fixed for each case
- Case 2:
  - Refrigerant selected: R134a
  - MDACC is sized to give ITD = 20 °C, at ambient temperature = 40 °C

Case 2: refrigerant selection

Case 2: MDACC sizing

Direct NDDCT sizing diagram

Case 2: air cooled condenser heat transfer area (number of tubes) and initial temperature difference on VCRC COP, analysed at ambient temperature of 40 °C.

Results – Case 2

- Compressor has highest contributions
  - $W_{fan}$ increases with high $T_{amb}$ cases
- COP of the VRC is ~3 up to 10.
  - Doesn’t account for compressor off-design performance

Proportion of contributions dependent on MDACC sizing
Results – Case 3b

![Graph showing temperature changes with ambient temperature for Case 3b.]
Results – ambient temperature distribution

- For Alice Springs TMY temperature values:
  - Approx. 60% of the year is over 20 °C
    - Negatively affecting cycle performance
  - Approx. 40% is 20 °C to 30 °C
  - Approx. 20% is 30 °C to 40 °C