An Investigation of Turbomachinery Concepts for an Isothermal Compressor Used in an S-CO₂ Bottoming Cycle

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Waste heat source:
Manufacturing, gas turbine exhaust

- Waste heat (e.g. glass manufacturing, steel manufacturing, and gas turbine exhaust) can be utilized as a heat source for a work-generating power cycle (bottoming cycle) to improve the overall thermal efficiency.
- Various systems applicable as waste heat recovery systems, including steam Rankine, ORC, and s-CO₂.
- Supercritical CO₂ (s-CO₂) bottoming cycle achieves high efficiency, mainly due to lowered compression work near the critical point.
Background – S-CO₂ Cycle + Isothermal Compressor

- **S-CO₂ cycle** has reduced compression work near the critical point.
- Isothermal compression → compressing at constant temperature, through heat removal.
- Using an isothermal compressor can minimize the compression work, up to 50% [4].
- Partial heating cycle has been known as one of the high-performing waste heat recovery layouts [6].
- Isothermal compressor has been applied to partial heating cycle to show nearly 15% improvement in overall net work generated [4].
- Lowering compressor outlet temperature is beneficial for waste heat recovery, since it is not optimized for net efficiency but for net work (more heat input the better).

Question: How to realize the isothermal compressor in s-CO₂ power cycles?

<table>
<thead>
<tr>
<th>Type</th>
<th>Multistage + intercooling</th>
<th>Radial</th>
<th>Axial</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling method</td>
<td>Series of intercoolers</td>
<td>Internally-cooled</td>
<td>Impeller and shroud</td>
</tr>
<tr>
<td></td>
<td></td>
<td>diaphragm [7]</td>
<td>surface cooling</td>
</tr>
<tr>
<td>Diagram</td>
<td></td>
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</tr>
<tr>
<td>Comments</td>
<td>- Adiabatic compression</td>
<td>- Novel concept</td>
<td>- Cooled flow passes</td>
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<td></td>
<td>followed by intercoolers</td>
<td>- Developed by</td>
<td>through the impeller and</td>
</tr>
<tr>
<td></td>
<td>- Realistic option for</td>
<td>Southwest Research</td>
<td>shroud surfaces to</td>
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<td></td>
<td>realizing s-CO₂</td>
<td>Institute and</td>
<td>remove heat</td>
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<tr>
<td></td>
<td>isothermal compressor</td>
<td>Dresser-Rand</td>
<td>- Challenging cooling</td>
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<td></td>
<td>(e.g. commercialized by</td>
<td></td>
<td>heat flux levels due to</td>
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<td></td>
<td>MAN Turbo)</td>
<td></td>
<td>limited heat transfer</td>
</tr>
<tr>
<td></td>
<td>- Large pressure drop</td>
<td>- Removes heat of</td>
<td>area</td>
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<tr>
<td></td>
<td>expected in between</td>
<td>compression between each</td>
<td>- Close to real isothermal</td>
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<td></td>
<td>intercoolers</td>
<td>impeller</td>
<td>compression</td>
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<td></td>
<td>- Cycle re-optimization</td>
<td>- For CCS application</td>
<td></td>
</tr>
<tr>
<td></td>
<td>needed</td>
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</tbody>
</table>

Option 1

- Adiabatic compression followed by intercoolers
- Realistic option for realizing s-CO₂ isothermal compressor (e.g. commercialized by MAN Turbo)
- Large pressure drop expected in between intercoolers
- Cycle re-optimization needed

Option 2

- Novel concept
- Developed by Southwest Research Institute and Dresser-Rand
- Removes heat of compression between each impeller
- For CCS application

Option 3

- Cooled flow passes through the impeller and shroud surfaces to remove heat
- Challenging cooling heat flux levels due to limited heat transfer area
- Close to real isothermal compression

- Cooling done on the rotor and stator blade surfaces
- Concept from Frontline Aerospace IsoCool™
- Larger heat transfer area at each stage
- Axial compressor not yet realized for s-CO₂
Option 1: Multistage Compression with Intercooling

- Practical compressor stage number $\rightarrow$ 2–10 due to pressure drop (dP) of intercoolers and size
- $\Delta T_{out} \approx 2^\circ C$ for cases of stage number 5 $\rightarrow$ will influence cycle optimization results
- Optimal stage number at given conditions is 5, providing $\eta_{iso-c} = 85.8\%$ ($vs. 89\%$)

$$(\eta_{iso-c} = \frac{w_{ideal}}{w_{real}} = \frac{\sum m w_{x,i}}{\int_{1}^{2} v dP} = \frac{\sum_{i=1}^{m} (h_{x,i,isen} - h_{x,i-1})}{\eta_{S}} = \lim_{n\to\infty} \sum_{i=1}^{n-1} \Delta P_i \cdot \frac{v_{i+1}(T_{in}) + v_i(T_{in})}{2})$$

Fig. T-s diagram (left) and isothermal compressor efficiency (right) for multistage compression with intercooling

Fig. S-CO$_2$ partial heating cycle layout
**Option 2: Impeller Cooling - Conditions**

- CFD study conducted to evaluate the feasibility of the cooling method
- Assumptions: uses previous geometry from KAERI SCIEL compressor for preliminary study
- Adopted STAR-CCM+ software
- Used 3-D Reynolds-Averaged Navier-Stokes (RANS) simulation, and $k - \omega$ SST turbulence model (used for turbomachinery analysis)
- Created a property table for CO$_2$ from the NIST REFPROP database

### Design parameters

<table>
<thead>
<tr>
<th>Design parameters</th>
<th>Values</th>
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</thead>
<tbody>
<tr>
<td>Mass flow rate (kg/s)</td>
<td>3.2</td>
</tr>
<tr>
<td>RPM</td>
<td>70000</td>
</tr>
<tr>
<td>Inlet stagnation temperature (°C)</td>
<td>33</td>
</tr>
<tr>
<td>Inlet stagnation pressure (MPa)</td>
<td>7.8</td>
</tr>
<tr>
<td>Pressure ratio</td>
<td>1.8</td>
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<tr>
<td>Inlet diameter</td>
<td>23mm</td>
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<tr>
<td>Outlet diameter</td>
<td>46mm</td>
</tr>
<tr>
<td>Number of blades</td>
<td>16</td>
</tr>
<tr>
<td>Isentropic efficiency (%)</td>
<td>65</td>
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<tr>
<td>Compression process number</td>
<td>100</td>
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</tbody>
</table>

Table: Design parameters obtained from SCIEL design conditions and isothermal compressor methodology
**Option 2: Impeller Cooling - Methodology**

**Initial design conditions:**
Compressor inlet/outlet temperatures and pressures, mass flow rate

**Calculation of compressor geometry:**
Obtain the general information of compressor geometry under isothermal condition using reference design or 1-D design software

**Calculation of boundary profile and CFD Analysis:**
Obtain the boundary condition as a heat flux profile \( q''(r) \), using the infinitesimal approach, with respect to the compressor radius, and CFD analysis

**Performance analysis using CFD results:**
\[
\eta_{ISO} = \lim_{n \to \infty} \frac{\sum_{i=1}^{n} \Delta P_i \cdot \frac{v_i(T_{in}) + v_{i-1}(T_{in})}{2}}{T \cdot \omega} \\
(\Delta P_i = P_{x,i} - P_{x-1})
\]
Option 2: Impeller Cooling – CFD Results

- Converged for constant temperature thermal boundary condition on shroud and hub (constant T BC case)
- Total inlet temperature has been raised to 38°C in order to achieve better convergence for the results
- Two sets of thermal BC provided to the reference compressor: adiabatic (case 1), and constant temperature for hub and shroud surfaces at 35°C (case 2)
- Discharge temperature is not lowered sufficiently: 47.9°C for the adiabatic case and 46.3°C for constant T BC case
- Total pressure increases at the impeller tip: 9.5MPa for adiabatic case and 10.5MPa for constant T BC case
- Pressure and temperature fields change only locally with constant temperature BC → not enough heat removal due to high heat capacity of s-CO₂

Figs. Temperature and total pressure of SCIEL s-CO₂ compressor under adiabatic conditions (top left and right) and constant temperature condition at hub and shroud surfaces (bottom left and right)
Option 2: Impeller Cooling – Heat Flux Profile

- Under SCIEL compressor design conditions for reference comparison + new design parameters for isothermal compressor design
- Adopting the infinitesimal approach for the calculation of heat removal
- Able to obtain the profile of work and heat removed inside the isothermal compressor

![Fig. Geometry of SCIEL impeller with dimensions labeled](image1)

![Fig. Schematic of infinitesimal approach used to evaluate s-CO$_2$ isothermal compression](image2)

![Fig. Profile of infinitesimal work and heat using infinitesimal approach w.r.t. the isothermal compression index](image3)

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Fig. Schematic of infinitesimal approach used to evaluate s-CO$_2$ isothermal compression

Fig. Profile of infinitesimal work and heat using infinitesimal approach w.r.t. the isothermal compression index
**Option 2: Impeller Cooling – Heat Flux Profile**

- Infinitesimal heat flux profile = $dq(x)/dA(x)$
- Unfeasible heat flux values obtained at the entrance (compare with best technology for cooling flux)
- Rapid increase of $c_p (\frac{dh}{dT}) \rightarrow$ large heat needs to be removed to lower $\Delta T$ for isothermal
- The CFD analysis also does not converge with such high heat flux levels
- Difficult to realize isothermal compression by impeller/shroud cooling concepts (not enough surface area)

**Infinitesimal area profile of the compressor**

Infinitesimal area profile of compressor

**Fig. Infinitesimal area profile of the compressor**

**Fig. Calculated $c_p$ and heat flux profile within the isothermal compressor**

- Region of high $c_p$ value and heat flux

**Fig. Microchannel cooler showing heat exchanger zones**

- E.g. state-of-the-art electronics cooling heat flux level: 10MW/m²
### Option 3: Axial-type Compressor – Basic Design

#### Design Parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor inlet total temperature (°C)</td>
<td>32</td>
</tr>
<tr>
<td>Compressor inlet total pressure (MPa)</td>
<td>7.69</td>
</tr>
<tr>
<td>Total-to-total pressure ratio</td>
<td>2.6</td>
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<tr>
<td>RPM</td>
<td>3600</td>
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<tr>
<td>Compressor mass flow rate (kg/s)</td>
<td>1915</td>
</tr>
<tr>
<td>Realizable cooling flux level (MW/m²)</td>
<td>5</td>
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<tr>
<td>Compression process number</td>
<td>50</td>
</tr>
<tr>
<td>Isentropic efficiency (small stage efficiency) (%)</td>
<td>89</td>
</tr>
</tbody>
</table>

↑ Table. Design parameters at optimal point for recompression iso-Brayton MC main compressor from Wang (2005)

- No real axial s-CO₂ compressor has been designed and tested
- Reference design values used for the s-CO₂ axial compressor from Wang (2005)
- Using in-house KAIST-TMD code for 1-D meanline axial compressor design, a reference main compressor geometry information is obtained
- Surface area estimated from the provided geometry → axial-type s-CO₂ compressor can realize the surface cooling flux by the concept of **stator vane cooling**

↑ Figs. Schematic of axial compressor stator vane cooling concept from Frontline Aerospace IsoCool™ (left) and diagram of rotor and stator vanes (right)

↑ Fig. Turbomachinery geometry for s-CO₂ axial compressor for reference recompression main compressor (**red**: rotor, **blue**: stator)
Option 3: Axial-type Compressor – Heat Flux Profile

† Fig. Heat removed vs. isothermal compression index for axial-type isothermal compressor

Heat profile

Fig. Heat flux level with respect to stator numbers when vane number = 80

Region of high $c_p$ value and heat flux

Area profile
(assuming fixed stator vane number for all stators)

- Heat flux values still remain high especially for front stators, where the CO$_2$ is expected to be near the critical point (hence, high $c_p$)
- Axial compressor design allows the increase of surface area within the stator, high vane number $>$ 80 may allow enough cooling

For realistic levels,

$$\text{average heat flux} = 7.4 \text{MW/m}^2$$
Option 3: Axial-type Compressor – Realistic Isothermal Comp.

- Introducing realistic heat flux level at 5MW/m², realistic isothermal compression can be analyzed.
- Calculating specific heat removed by applying heat flux and stator surface area, the results yield 6.4kJ/kg (compared to 64.8kJ/kg for perfect isothermal compression).
- \( \eta_{iso-c} \) results in 75.9% for realistic isothermal compression, compared to 74.7% for adiabatic compression, and 88.9% for perfect isothermal compression.

Fig. T-s diagram explaining the thermodynamic pathway of realistic isothermal compression using the infinitesimal approach.

Fig. T-s diagram comparing the realistic isothermal compression to the adiabatic compression under the given design conditions of Table 1.
1. Three possible concepts to realize the isothermal compressor are investigated: multistage compressor with intercooler, radial-type compressor impeller cooling, and axial-type compressor with stator vane cooling.

2. Realistic concept of multistage compressor with intercooler is largely limited by pressure drop in the intercoolers and stage number, but it would be one of the realistic ways to apply the isothermal compressor.

3. For the radial-type compressor impeller cooling concept, rough estimate directs towards unfeasible cooling flux levels (several 100MW/m² range). Otherwise, constant temperature thermal boundary condition on the shroud and hub surfaces would not induce sufficient temperature drop to produce isothermal condition. Hence, axial-type compressor is instead recommended for investigation for larger heat transfer area.

4. Axial-type compressor with stator vane cooling concept is tested for conceptual study. Although high cooling flux levels exist at the entrance stators, heat can be removed at realistic levels when the vane number>80 for all the compressor stators.

5. Provided the realistic level of heat flux (1-5MW/m²), results of ‘realistic isothermal compression’ are calculated. $\eta_{iso-c}$ results in 75.9% for realistic isothermal compression, compared to 74.7% for adiabatic compression, and 88.9% for perfect isothermal compression.
References

THANK YOU FOR YOUR ATTENTION
Option 1: Multistage Compression with Intercooling

Observations:
- Realistic dP value will not achieve high cycle efficiencies → performance is sensitive to intercooler dP
- Multiple local maxima appear as stage number is increased, global maxima held at higher PR points
- Designing for $0 < \text{dP} < 30$ and stage number $> 7$ will bring efficiency gain
- Hence, concept of multistage compression with least intercooling dP is desirable
CO₂ T-s Diagram
CO$_2$ $c_p$-$T$ Diagram
Frontline Aerospace IsoCool™ Close-up