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An Investigation of Turbomachinery Concepts for an Isothermal Compressor Used in an S-CO_2 Bottoming Cycle

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CONTENTS

Background – Waste Heat Recovery Systems

Waste heat source:
Manufacturing, gas turbine exhaust

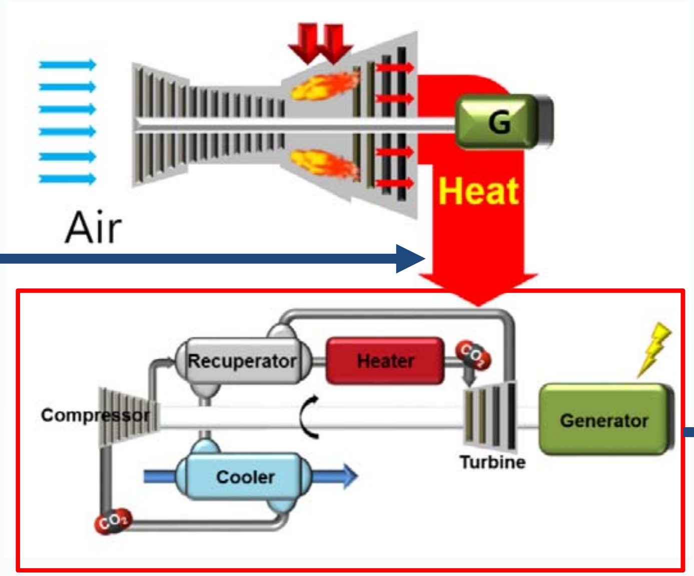

Fig. Bottoming cycle using waste heat source

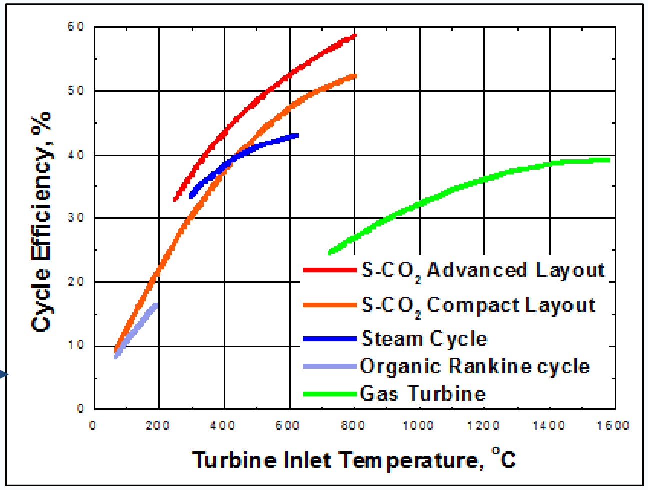

Fig. Cycle efficiency advantage for s-CO_2 cycles over other candidates

- 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-CO_2
- Supercritical CO_2 (s-CO_2) bottoming cycle achieves high efficiency, mainly due to lowered compression work near the critical point

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Background – S-CO_2 Cycle + Isothermal Compressor

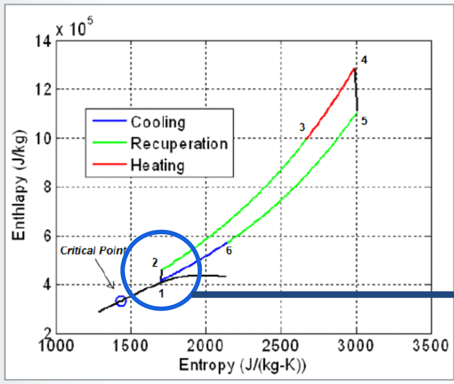

Fig. Minimizing compression work through isothermal compressor

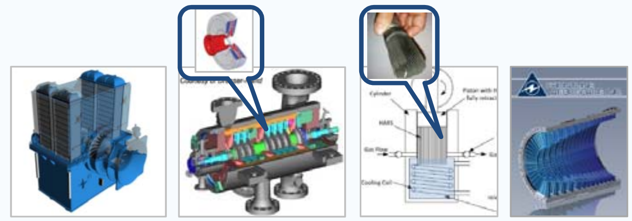

Fig. Various **isothermal compressor** concepts

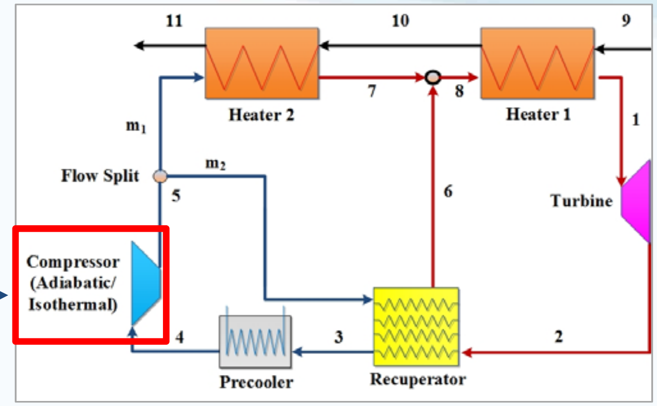

Fig. Configuration of partial heating cycle

- S-CO_2 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)

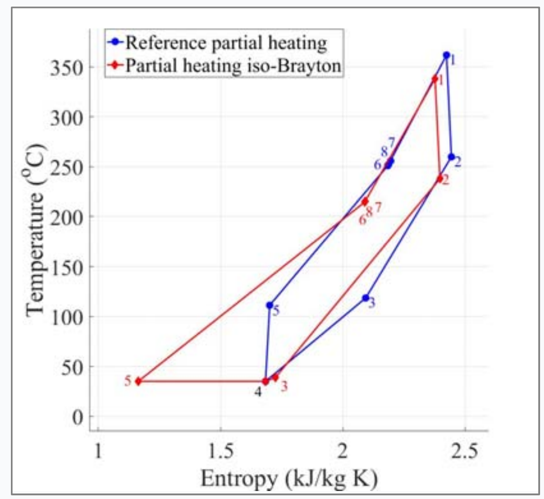

Fig. T-s diagram of partial heating cycle layouts →

[4] Heo, Jin Young, et al. "Thermodynamic study of supercritical CO_2 Brayton cycle using an isothermal compressor." Applied Energy 206 (2017): 1118-1130.

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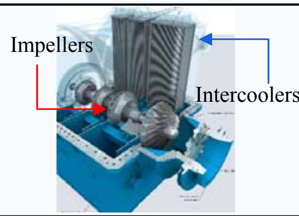
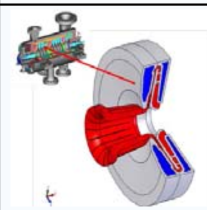
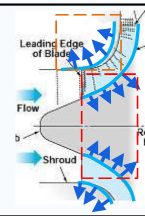
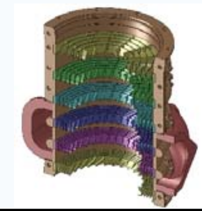
Feasibility Study – Options

Question: How to realize the isothermal compressor in s-CO_2 power cycles?

Type	Multistage + intercooling	Radial		Axial
Cooling method	Series of intercoolers	Internally-cooled diaphragm [7]	Impeller and shroud surface cooling	Stator blade cooling
Diagram				
Comments	- Adiabatic compression followed by intercoolers - Realistic option for realizing s-CO_2 isothermal compressor (e.g. commercialized by MAN Turbo) - Large pressure drop expected in between intercoolers - Cycle re-optimization needed	- Novel concept - Developed by Southwest Research Institute and Dresser-Rand - Removes heat of compression between each impeller - For CCS application	- 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-CO_2

Option 1   Option 2   Option 3

Option 1: Multistage Compression with Intercooling

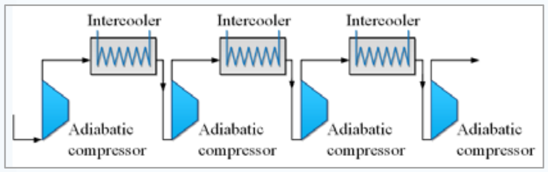


Fig. Multistage compression with intercooling (4 stages)

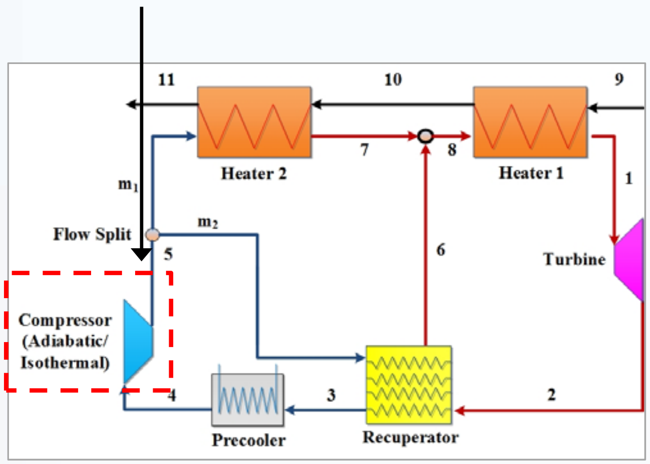


Fig. S-CO_2 partial heating cycle layout

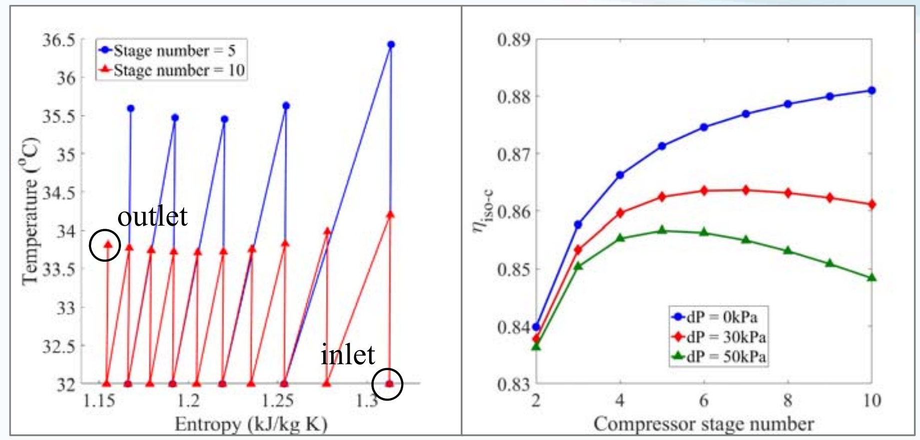
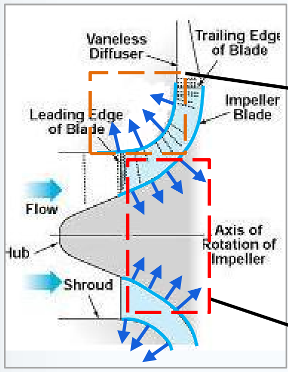


Fig. T-s diagram (left) and isothermal compressor efficiency (right) for multistage compression with intercooling

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

$$\left(\eta_{iso-c} = \frac{w_{ideal}}{w_{real}} = \frac{\sum_m w_{x,i}}{\int_1^2 v dP} = \frac{\sum_{i=1}^{m} \frac{\left(h_{x,i,isen} - h_{x,i-1}\right)}{\eta_s}}{\lim\limits_{n\to\infty} \sum_{i=1}^{n-1} \Delta P_i \cdot \frac{v_{i+1}(T_{in}) + v_i(T_{in})}{2}}\right)$$

Option 2: Impeller Cooling - Conditions

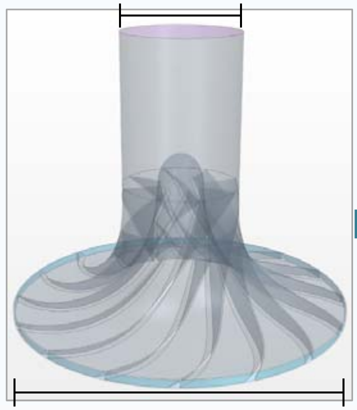


Cooling heat flux boundary condition on the **shroud surface**

← Fig. Diagram explaining the impeller cooling concept

Cooling heat flux boundary condition on the **impeller hub**

Inlet diameter



Outlet diameter

← Fig. Geometry of SCIEL impeller with dimensions labeled

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

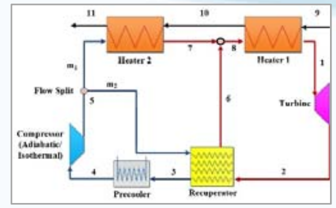
Table. Design parameters obtained from SCIEL design conditions and isothermal compressor methodology

- 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

Option 2: Impeller Cooling - Methodology

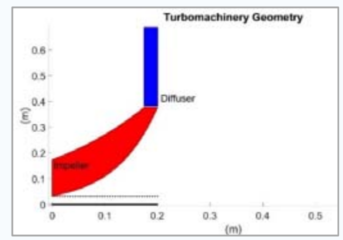
Initial design conditions:

Compressor inlet/outlet temperatures and pressures, mass flow rate



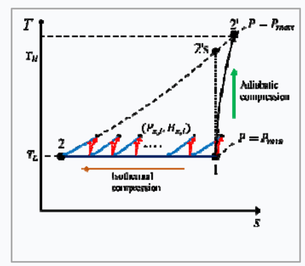
Calculation of compressor geometry:

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:

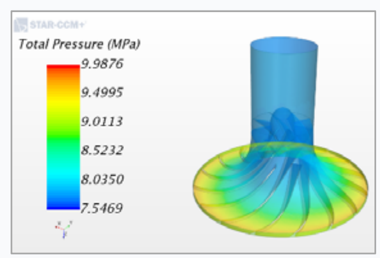
Obtain the boundary condition as a heat flux profile $q''(r)$, using the infinitesimal approach, with respect to the compressor radius, and CFD analysis



Performance analysis using CFD results:

$$\eta_{iso} = \frac{\lim_{n\to\infty} \sum_{i=1}^{n} \Delta P_i \cdot \frac{v_i(T_{in}) + v_{i-1}(T_{in})}{2}}{T \cdot \omega}$$

$$(\Delta P_i = P_{x,i} - P_{x-1})$$



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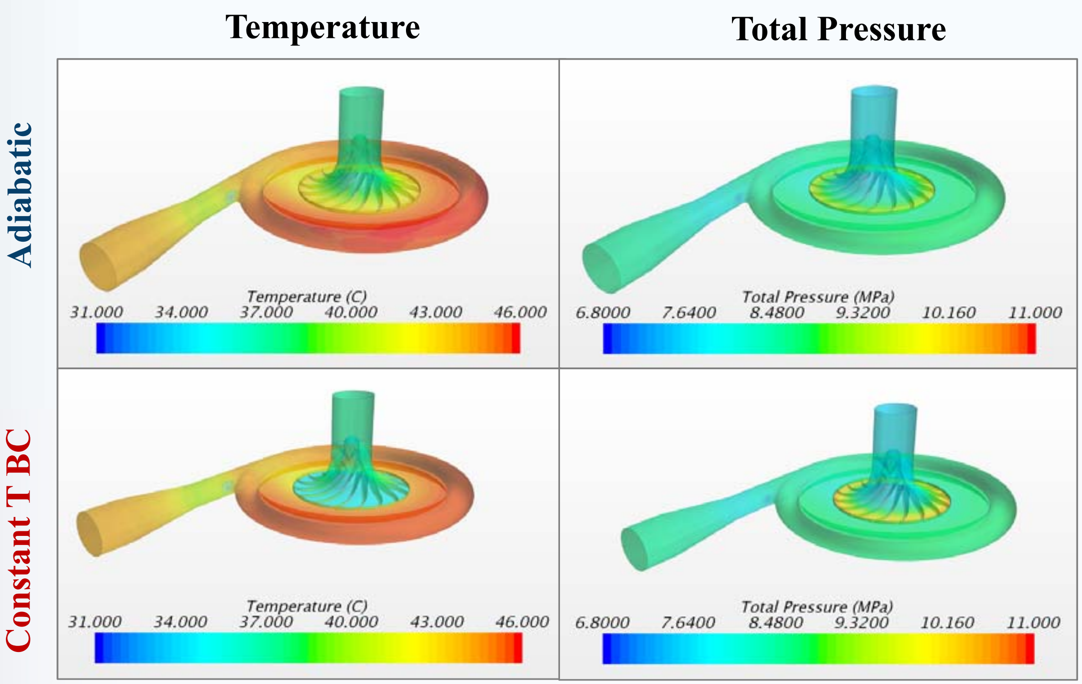
Option 2: Impeller Cooling – CFD Results

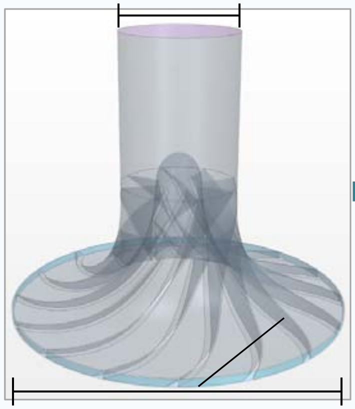


Figs. Temperature and total pressure of SCIEL s-CO_2 compressor under adiabatic conditions (top left and right) and constant temperature condition at hub and shroud surfaces (bottom left and right)

- 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-CO_2

Option 2: Impeller Cooling – Heat Flux Profile

Inlet diameter



← Fig. Geometry of SCIEL impeller with dimensions labeled

Outlet diameter

- 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

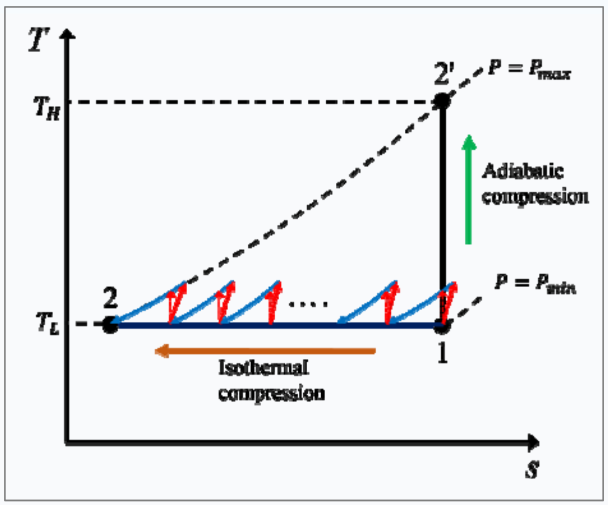


Fig. Schematic of infinitesimal approach used to evaluate s-CO_2 isothermal compression

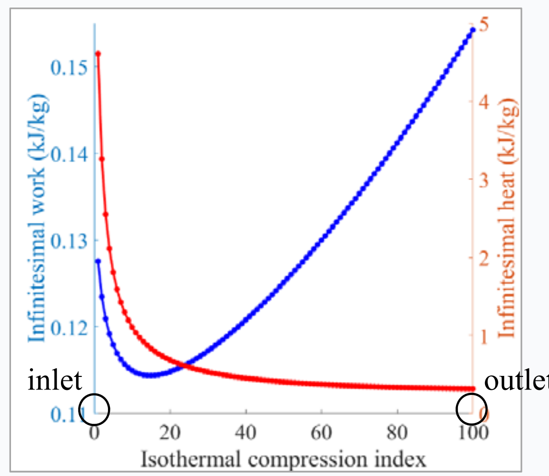


Fig. Profile of infinitesimal work and heat using infinitesimal approach w.r.t. the isothermal compression index

Option 2: Impeller Cooling – Heat Flux Profile

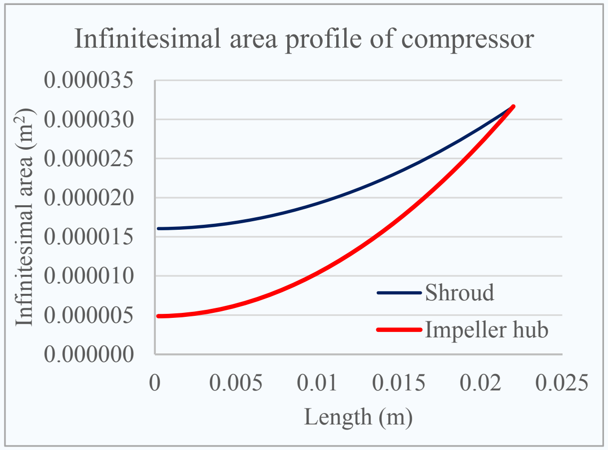


Fig. Infinitesimal area profile of the compressor

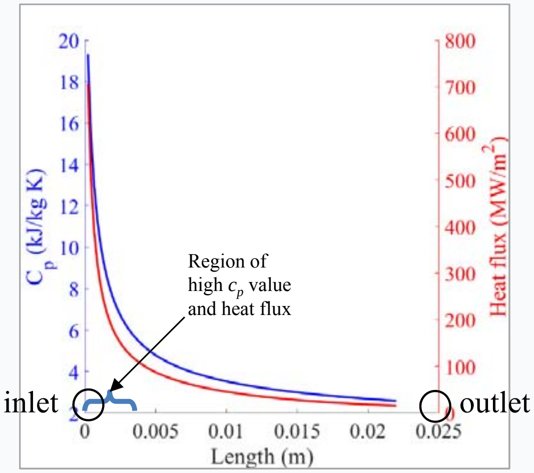


Fig. Calculated c_p and heat flux profile within the isothermal compressor

e.g. state-of-the-art electronics cooling heat flux level: **10MW/m²**

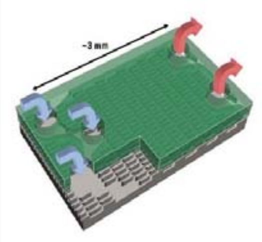
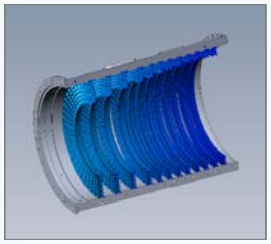
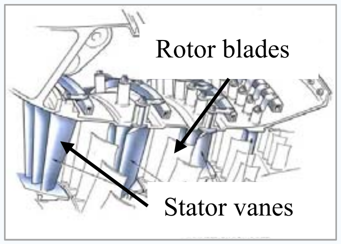


Fig. Microchannel cooler showing heat exchanger zones

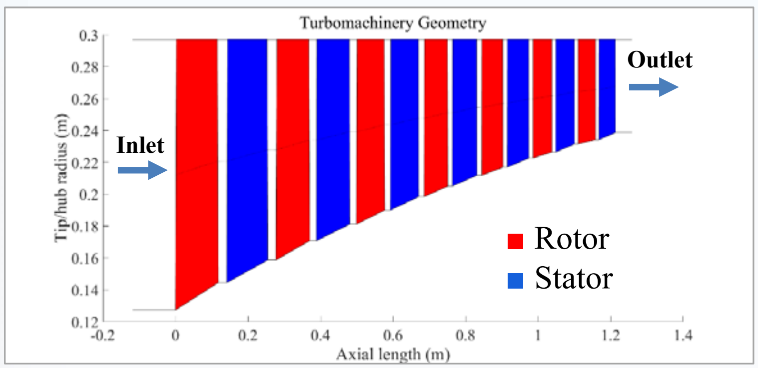
- Infinitesimal heat flux profile = $dq(x)/dA(x)$
- Unfeasible heat flux values obtained at the entrance (compare with best technology for cooling flux)
- Rapid increase of $c_p\left(=\frac{dh}{dT}\right)$→ large heat needs to be removed to lower ΔT for isothermal
- The CFD analysis also does not converge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

Option 3: Axial-type Compressor – Basic Design





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



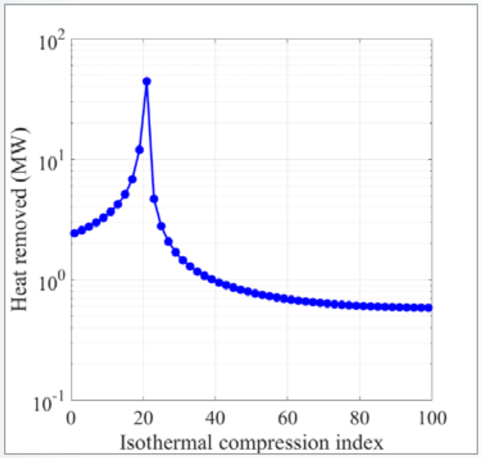
↑ Fig. Turbomachinery geometry for s-CO_2 axial compressor for reference recompression main compressor (red: rotor, blue: stator)

Design Parameters	Values
Compressor inlet total temperature (°C)	32
Compressor inlet total pressure (MPa)	7.69
Total-to-total pressure ratio	2.6
RPM	3600
Compressor mass flow rate (kg/s)	1915
Realizable cooling flux level (MW/m^2)	5
Compression process number	50
Isentropic efficiency (small stage efficiency) (%)	89

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

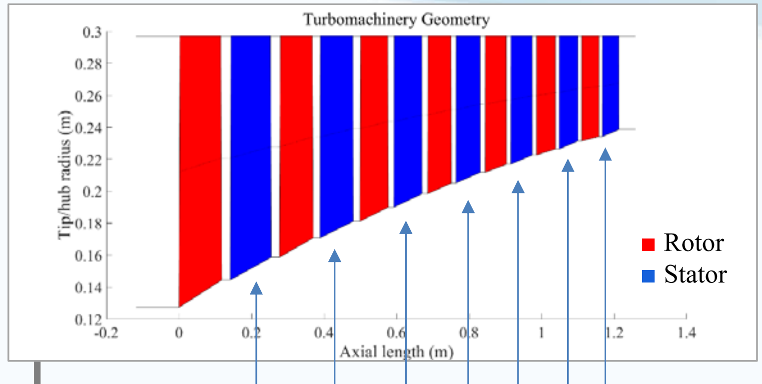
- No real axial s-CO_2 compressor has been designed and tested
- Reference design values used for the s-CO_2 axial compressor from Wang (2005)
- Using in-house KAIST-TMD code for 1-D meanline axial compressor design, a reference main compressor geometry information is obtained
- Surface area estimated from the provided geometry → axial-type s-CO_2 compressor can realize the surface cooling flux by the concept of **stator vane cooling**

Option 3: Axial-type Compressor – Heat Flux Profile



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

Heat profile



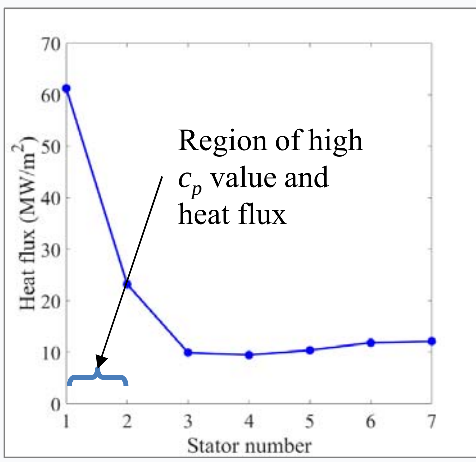
Stator vane cooling

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,

$$average\ heat\ flux = \mathbf{7.4 MW/m^2}$$



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

Option 3: Axial-type Compressor – Realistic Isothermal Comp.

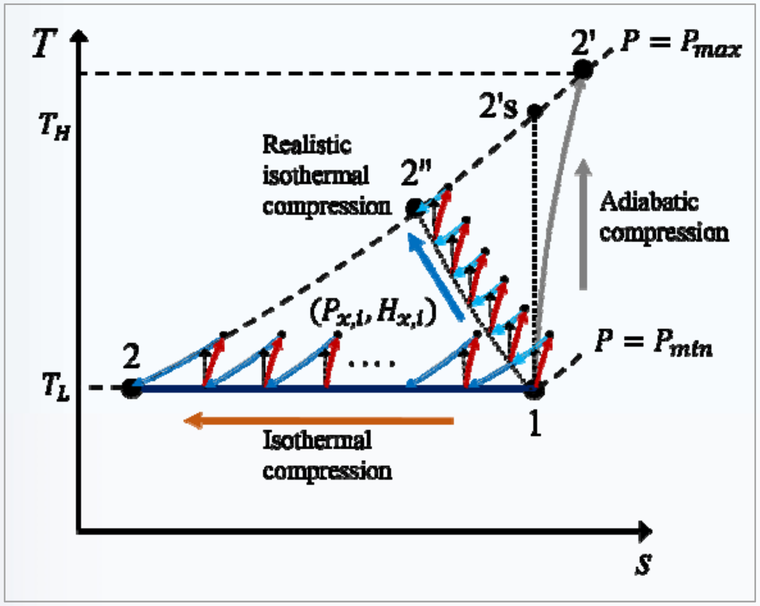


Fig. T-s diagram explaining the thermodynamic pathway of realistic isothermal compression using the infinitesimal approach

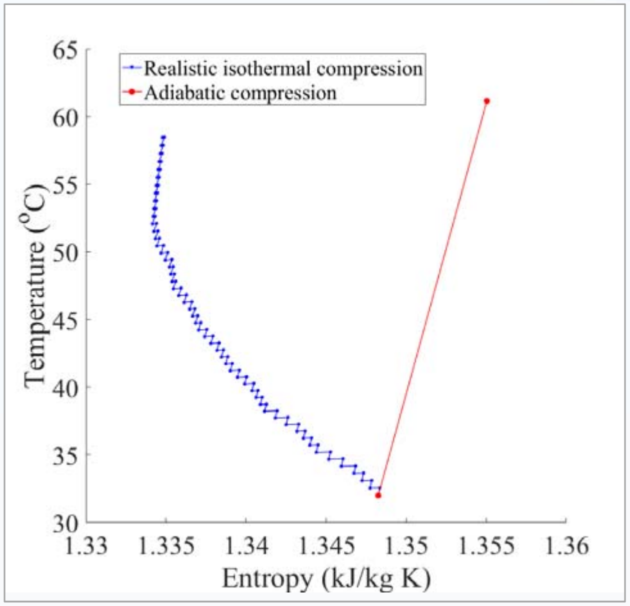


Fig. T-s diagram comparing the realistic isothermal compression to the adiabatic compression under the given design conditions of Table 1

- Introducing realistic heat flux level at 5MW/m^2, 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)
- η_{iso-c} results in **75.9%** for realistic isothermal compression, compared to 74.7% for adiabatic compression, and 88.9% for perfect isothermal compression

Conclusions and Further Works

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/m^2 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/m^2), results of 'realistic isothermal compression' are calculated. η_{iso-c} results in **75.9%** for realistic isothermal compression, compared to 74.7% for adiabatic compression, and 88.9% for perfect isothermal compression.

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References

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Option 1: Multistage Compression with Intercooling

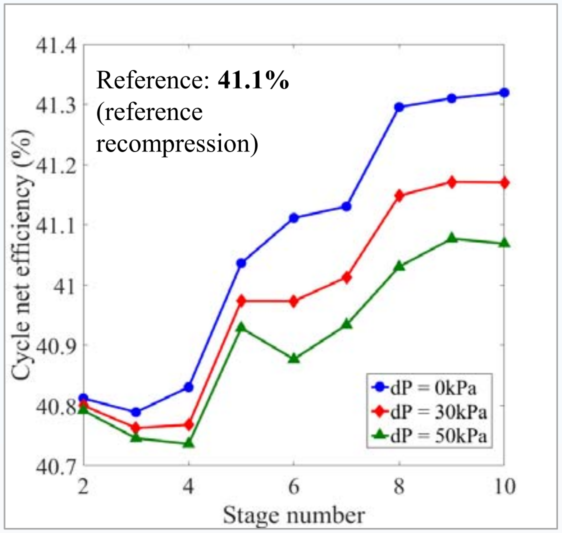

Fig. Graph of optimized cycle net efficiency with respect to stage number for three values of intercooler dP values

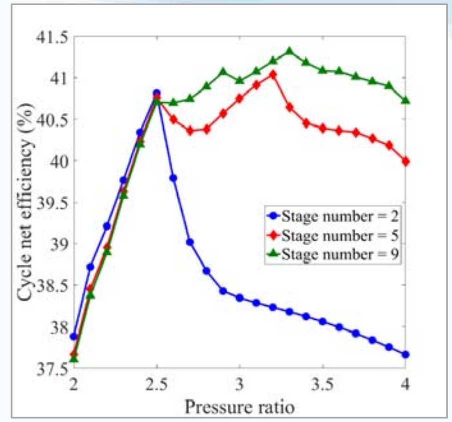

Fig. Graph of optimized cycle net efficiency with respect to pressure ratio for three values of stage number **(dP =0kPa)**

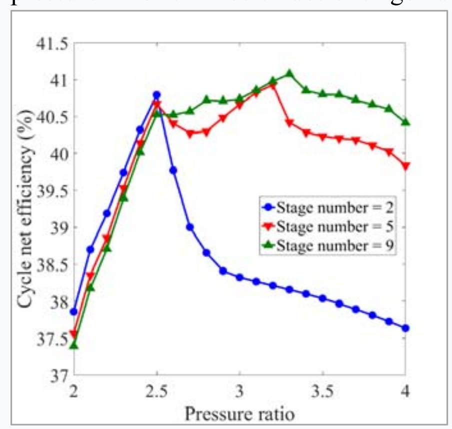

Fig. Graph of optimized cycle net efficiency with respect to pressure ratio for three values of stage number **(dP =50kPa)**

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 **$0 < dP < 30$** and **stage number>7** will bring efficiency gain
- Hence, concept of multistage compression with least intercooling dP is desirable

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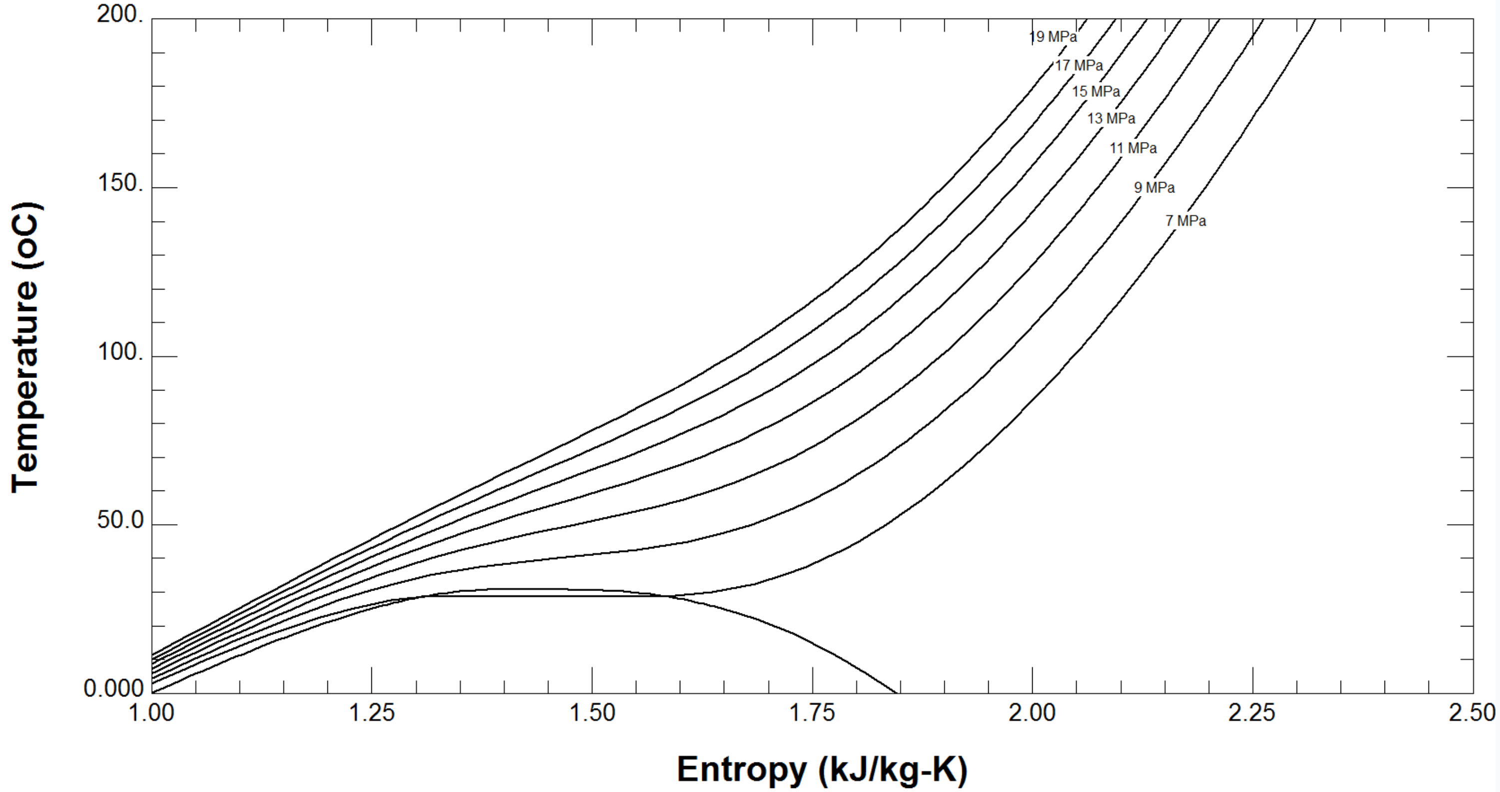
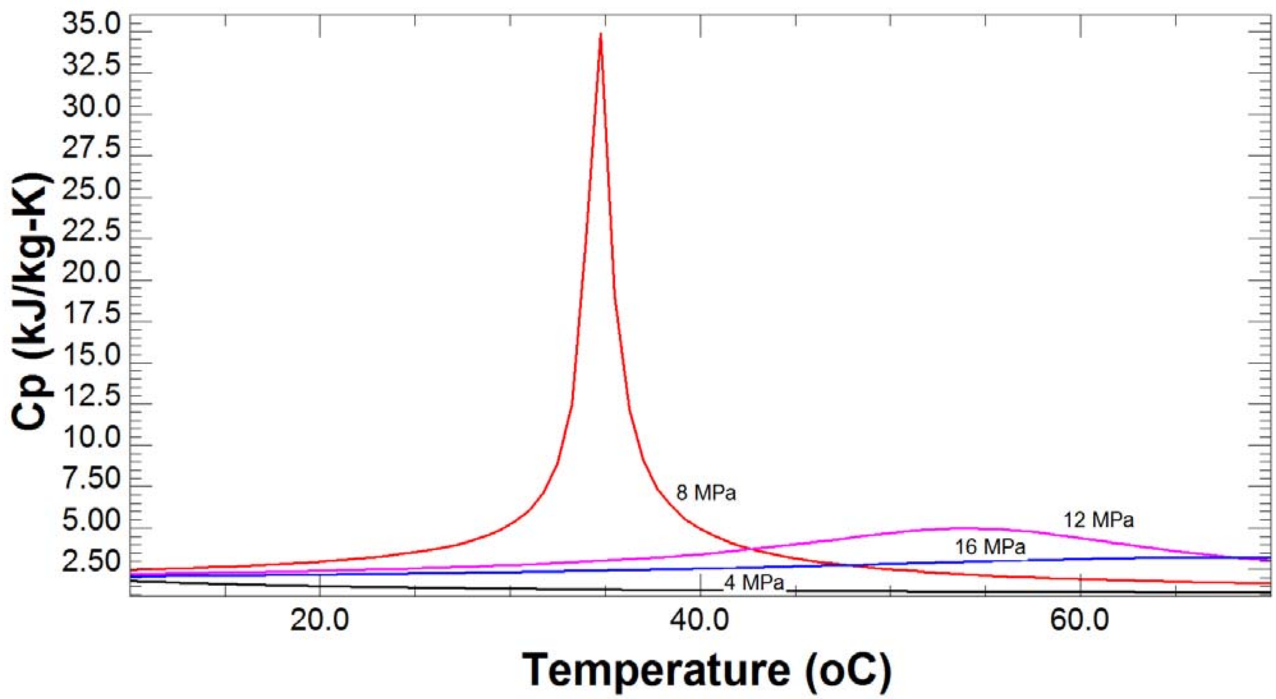
CO_2 T-s Diagram



CO_2 c_p-T Diagram



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Frontline Aerospace IsoCool^{TM} Close-up