Mapping the Design Space of a Recuperated, Recompression, Precompression Supercritical Carbon Dioxide Power Cycle with Intercooling, Improved Regeneration, and Reheat

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Outline

- Overview of Supercritical CO_2 Power Cycles
- Proposed System Layout
- Variable Property Heat Engine Cycle Analysis Code
- Heat Exchangers with Nonlinear and Dissimilar Specific Heats
- Results of the Design Space Exploration
- Conclusions

About Supercritical CO_2 (S-CO_2) Power Cycles

- Closed loop configuration.
- Main compressor inlet temperature and pressure are at or near the critical point.
- Carbon dioxide is the proposed working fluid because it is cheap, inert, and has a critical temperature of 304K (31°C), which is near typical ambient temperatures of $\sim$ 294K (21°C).
- High system pressures occur due to the high critical pressure of carbon dioxide (7.4 MPa).
- Possible applications:
  - Base load terrestrial electrical power generation
  - Marine, Aviation, and Spacecraft electrical power generation
- Possible Configurations:
  - Bottoming cycle using waste heat from a traditional open loop gas turbine (traditional Brayton cycle)
  - Primary cycle with nuclear and solar energy heat sources
  - Primary cycle with the combustion of fossil fuels as a heat source

Supercritical CO_2 Power Cycle - Strengths

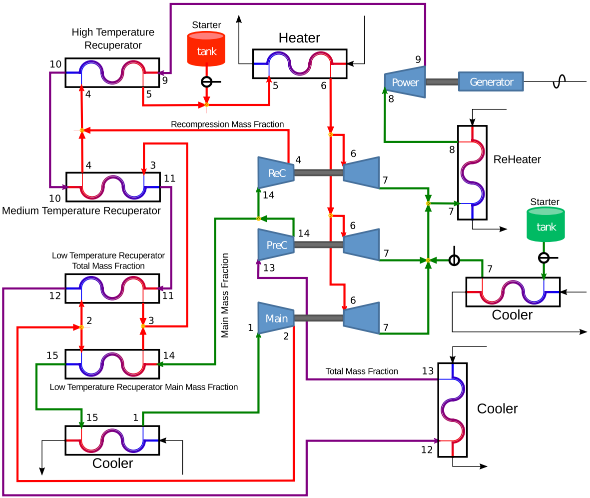
- Low Pressure Ratio (optimal overall pressure $\sim$ 3 to 8)
- Large amounts of recuperation possible.
- Low back work ratio
  - Decreased sensitivity of compressor/turbine efficiency on cycle efficiency.
  - S-CO_2 - $\sim$35%
  - Rankine - $\sim$2%
  - Open Loop Brayton - 40-80%
- High Power Density
  - High pressure and high molecular weight.
  - Fluid densities range from $\sim$23 kg/m^3 to $\sim$788 kg/m^3.
- Narrow heat addition and heat rejection temperatures does not require evaporative cooling, but still approximates a Carnot cycle better than an open loop Brayton cycle.
- High real cycle efficiency predicted
  - $>$50% @ 923K (650°C) turbine inlet temperature

Supercritical CO_2 Power Cycle - Weaknesses

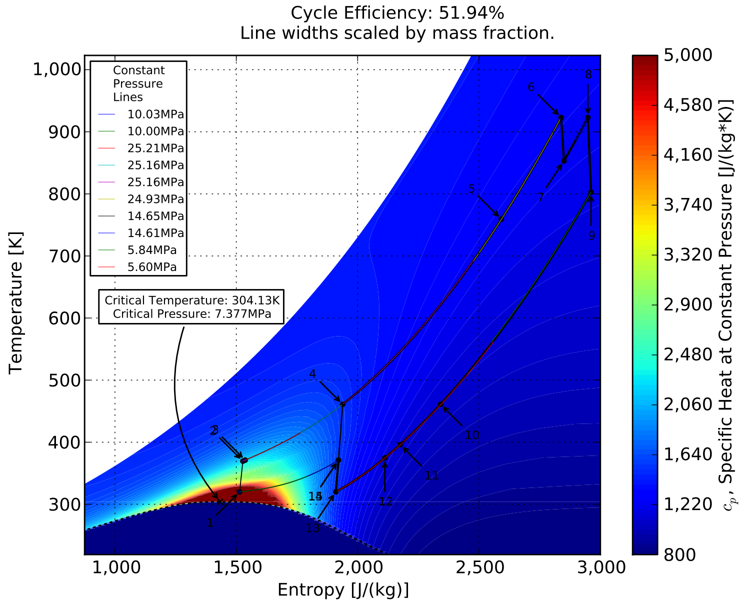
- Nonlinear specific heat mismatch causes difficulties exchanging heat between high and low pressure sides at lower temperatures.
- Closed loop design presents additional system complexities.
- High pressures present increased structural loading and seal leakage issues.
  - 20MPa to 30MPa maximum pressure typically proposed
- Nonlinear property variations near the critical point present turbomachinery design complications as well as challenges maintaining off design operability.
- High working fluid densities prohibit efficient low power, low speed, low cost prototypes to be developed.

Proposed System Layout

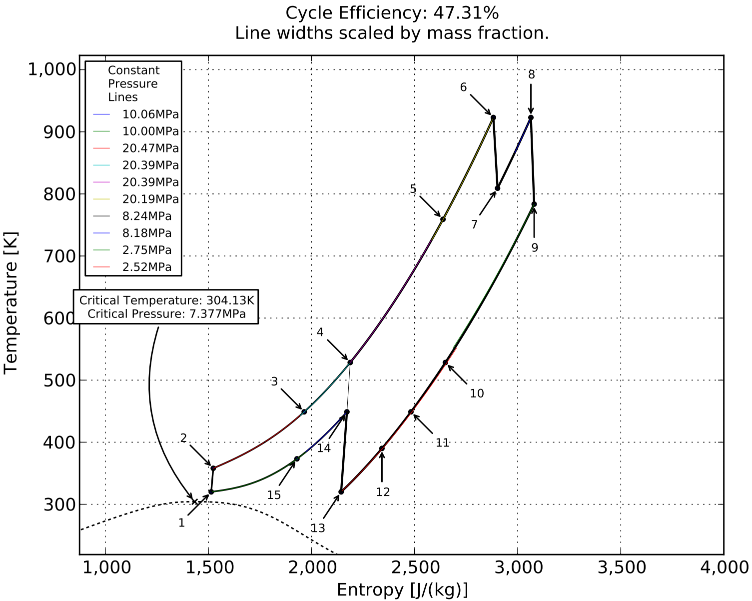
- Three compressors and several flow splits are used to help mitigate heat transfer issues due to specific heat mismatches.
- Four shafts are utilized to better match optimal operating speeds of each turbomachinery component.
- Due to the small size of the turbomachinery, as well as the use of multiple shafts, each assembly (except for the power turbine and generator) can be placed inside a pressure vessel to avoid the need for high speed, high pressure seals.
- Tanks and a blow down startup procedure are used to eliminate the need to attach a motor to the higher speed shafts.



Proposed System Layout - Temperature Entropy Diagram



Proposed System Layout - Temperature Entropy Diagram



Variable Property Heat Engine Cycle Analysis Code

- Cycle analysis code created from scratch.
- Developed with Python, NumPy, SciPy, and matplotlib.
- Variable fluid properties are utilized.
  - i.e. $h = h(T, p)$, $c_p = c_p(T, p)$, $s = s(T, p)$
  - Fluid property data used from REFPROP
- Specialized 1-D counterflow heat exchanger model was developed to account for variable fluid properties, yet maintaining high solution speed.
- Cycle iteratively solved for unknown pressures.
- Inputs include maximum temperature, minimum temperature, compressor pressure ratios, turbomachinery component efficiencies, heat exchanger pressure drop, main compressor inlet pressure, and mass fraction for flow splits.
- Design space for the inputs is explored in parallel and can run on as many processors as are available.

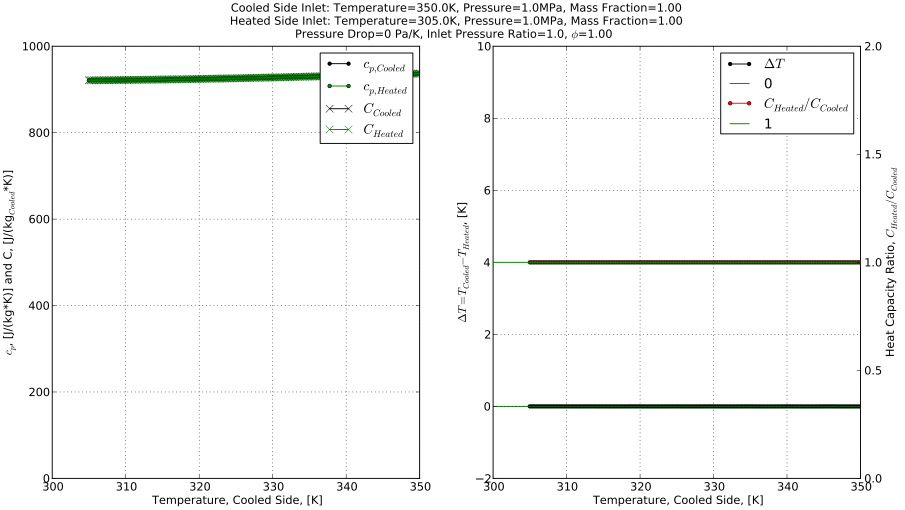
Variable Property Heat Engine Cycle Analysis Code Limitations and Assumptions

- Currently the code only supports gases and supercritical fluids. Liquids and and liquid vapor mixtures are not yet supported.
- Heat source currently modeled is that of a constant heat flux (i.e. solar) or a highly regenerated combustion system (heater efficiency is assumed to be 100%).
- Pumping power for the ambient pressure side of the heaters and coolers are assumed to be low.

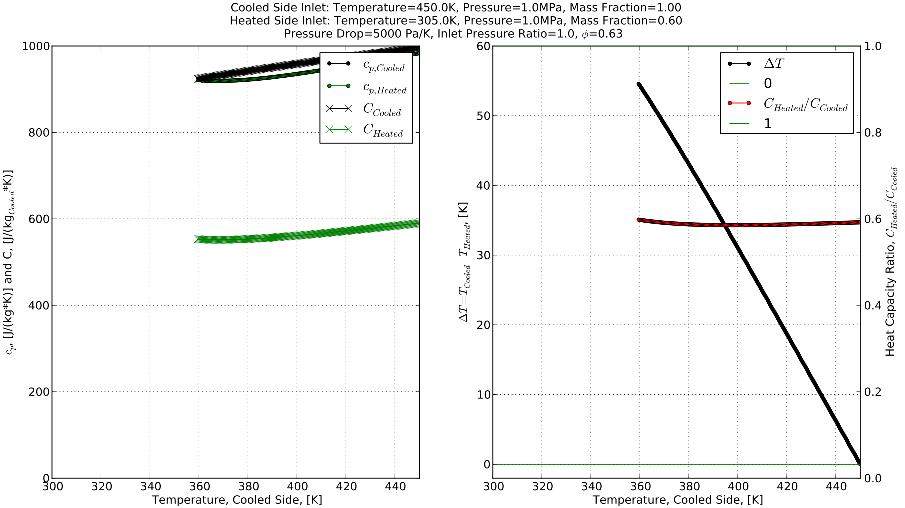
Heat Exchangers - Overview

- The current heat exchanger model assumes the limiting case where the convection coefficient is very high.
  - The temperature difference between the high pressure to the low pressure side of the heat exchanger is assumed to be purely due to specific heat mismatches.
  - At at least one point in the heat exchanger there will be approximately zero temperature difference between the high and low pressure side.
- Pressure drop
  - Pressure drop is not computed based on an assumed geometry, but is approximated to be linearly dependent upon temperature drop in the heat exchanger.
  - Temperature drop is assumed to be related to the length of the heat exchanger.
  - The linear relationship between temperature drop and pressure drop is another parameter varied as part of the design space exploration.
  - Pressure drop is assumed to be low, allowing the present approximation to be acceptable.

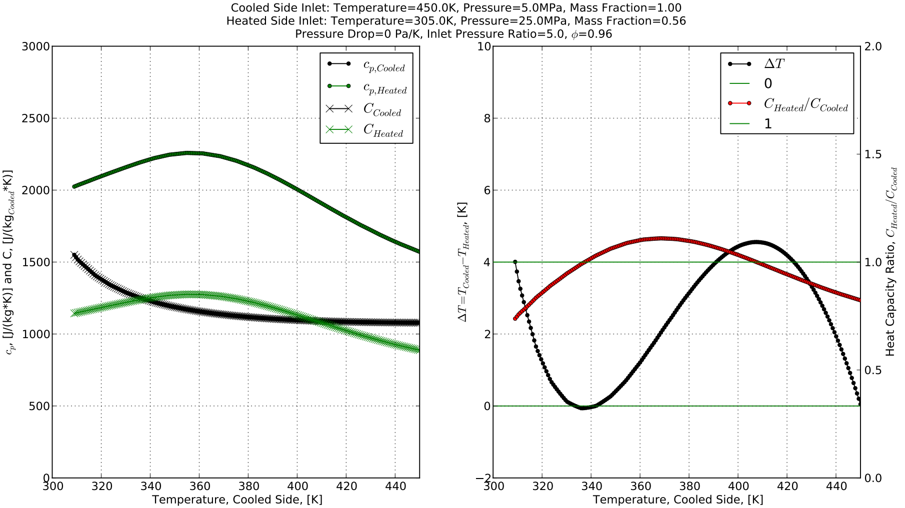
Heat Exchangers - Temperature and Specific Heat Variation



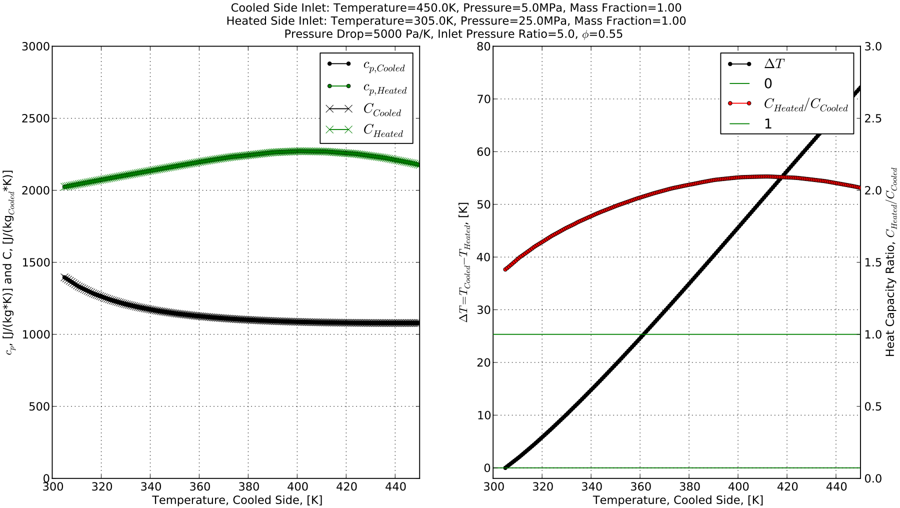
Heat Exchangers - Temperature and Specific Heat Variation



Heat Exchangers - Temperature and Specific Heat Variation



Heat Exchangers - Temperature and Specific Heat Variation



Design Space Exploration

Dataset I - 20,155,392 permutations

All Parameters - Coarse Exploration

Parameter	Minimum	Maximum	Number of Values	Value Plotted
PreCompressor Pressure Ratio	1.0	4.0	6	Optimal
Main Compressor Pressure Ratio	1.1	4.1	6	Optimal
Recompression Fraction	0.000	0.991	4	Optimal
Low Temperature Recuperator Main Fraction High Pressure Component Mass Fraction	0.001	0.991	4	Optimal
Main Compressor Inlet Pressure	6 MPa	10 MPa	6	Optimal
Maximum Temperature	798K	923K	3	923K
Minimum Temperature	320K	333K	3	320K
Main Compressor Isentropic Efficiency	0.75	1.00	4	0.85
PreCompressor Isentropic Efficiency	0.80	0.95	3	0.875
ReCompressor Isentropic Efficiency	0.80	0.95	3	0.875
Power Turbine Isentropic Efficiency	0.89	0.93	3	0.93
Main/Re/Pre Compressor Turbine Isentropic Efficiency	0.84	0.89	3	0.89
Heat Exchanger Pressure Drop	500 Pa/K	0 Pa/K	2	500 Pa/K

Design Space Exploration

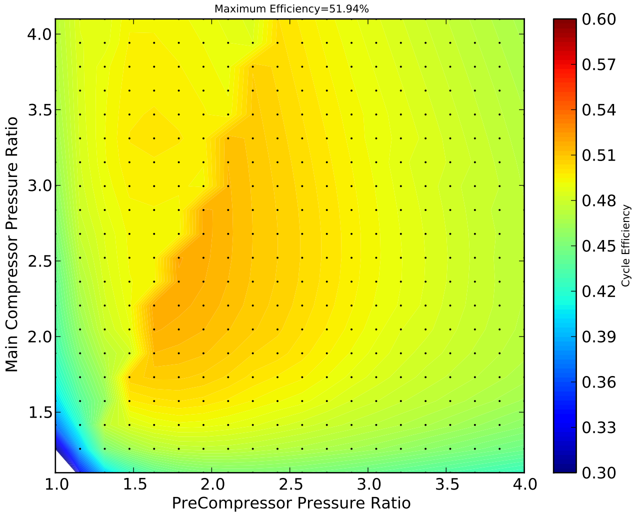
Dataset II - 1,800,000 permutations

Fixed Component Efficiencies and Max/Min Temp, Other Parameters Refined

Parameter	Minimum	Maximum	Number of Values	Value Plotted
PreCompressor Pressure Ratio	1.0	4.0	20	Optimal
Main Compressor Pressure Ratio	1.1	4.1	20	Optimal
Recompression Fraction	0.000	0.991	15	Optimal
Low Temperature Recuperator Main Fraction High Pressure Component Mass Fraction	0.001	0.991	15	Optimal
Main Compressor Inlet Pressure	6 MPa	10 MPa	20	Optimal
Maximum Temperature	923K	923K	1	923K
Minimum Temperature	320K	320K	1	320K
Main Compressor Isentropic Efficiency	0.85	0.85	1	0.85
PreCompressor Isentropic Efficiency	0.875	0.875	1	0.875
ReCompressor Isentropic Efficiency	0.875	0.875	1	0.875
Power Turbine Isentropic Efficiency	0.93	0.93	1	0.93
Main/Re/Pre Compressor Turbine Isentropic Efficiency	0.89	0.89	1	0.89
Heat Exchanger Pressure Drop	500 Pa/K	500 Pa/K	1	500 Pa/K

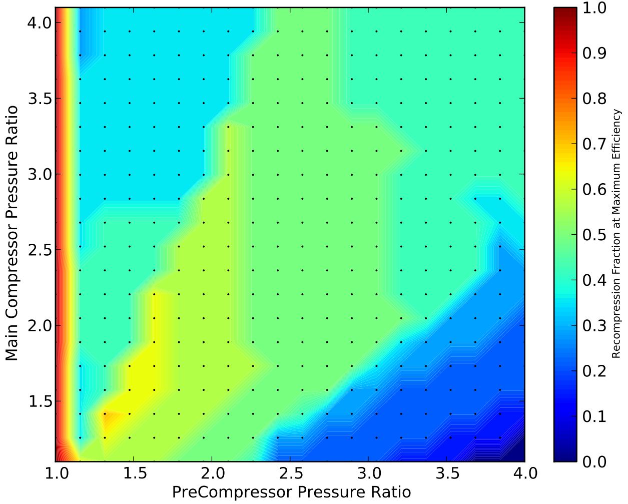
Design Space Exploration Results - Dataset II

Cycle Efficiency vs PreCompressor and Main Compressor Pressure Ratios



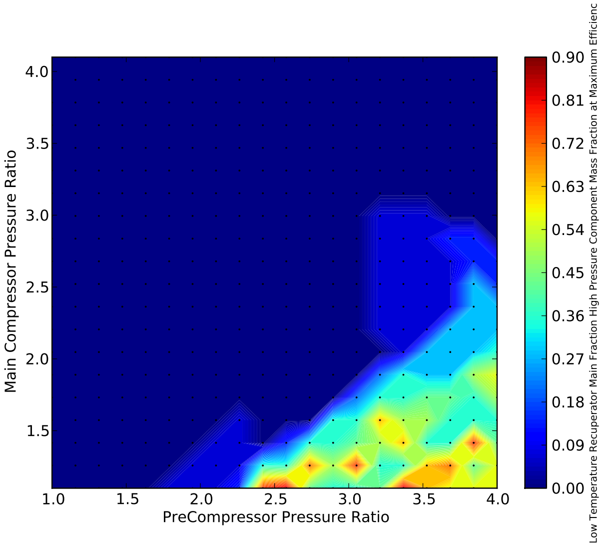
Design Space Exploration Results - Dataset II

Optimal Recompression Fraction vs PreCompressor and Main Compressor Pressure Ratios



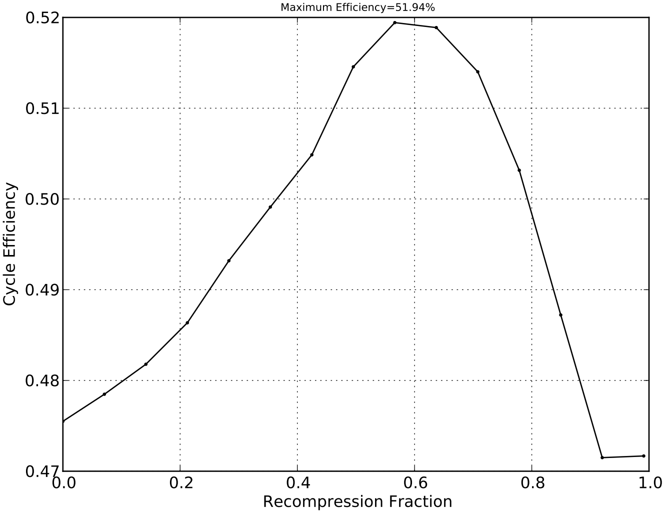
Design Space Exploration Results - Dataset II

Low Temperature Recuperator Main Fraction High Pressure Component Mass Fraction at Optimal Cycle Efficiency vs PreCompressor and Main Compressor Pressure Ratios



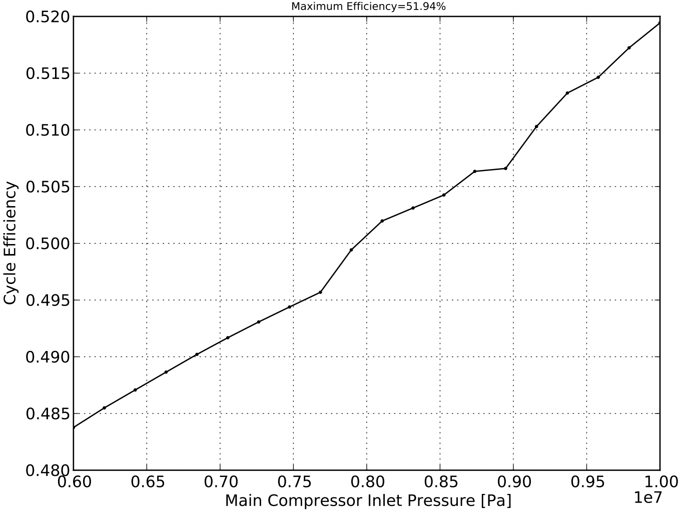
Design Space Exploration Results - Dataset II

Cycle Efficiency vs Recompression Fraction



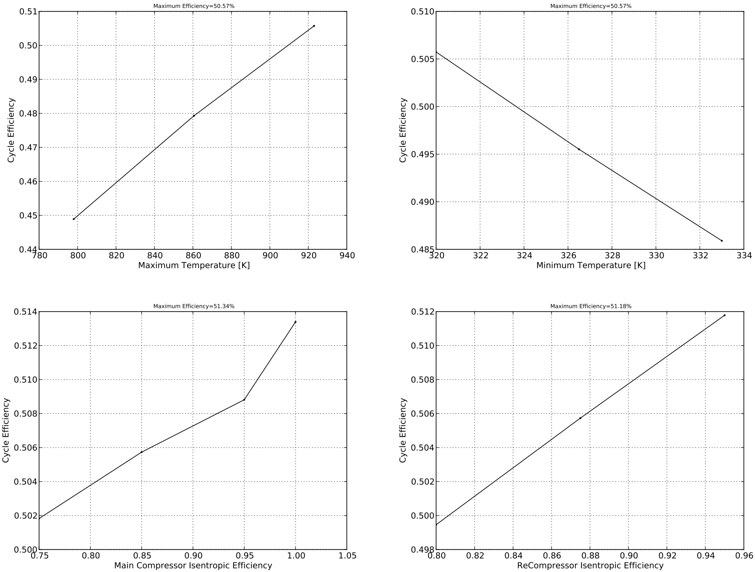
Design Space Exploration Results - Dataset II

Cycle Efficiency vs Main Compressor Inlet Pressure



Design Space Exploration Results - Dataset I

Cycle Efficiency vs Max and Min Temperature and Main and ReCompressor Efficiency



Web Based Graphical User Interface

Sensitivity Plots and Cycle Plot		Sensitivity Plots			
		Horizontal Axis	Vertical Axis	Contour Level	Vertical Axis
Independent Variable	Value Selected (selection ignored if variable is used for a sensitivity plot axis)	Contour and Line Plots	Contour Plot	Plot Value for Maximum Efficiency: Contour Plot	Plot Value for Maximum Efficiency: Line Plot
Dataset	20,155,392 permutations - All Parameters - Coarse Exploration				
PreCompressor Pressure Ratio	Value for Maximum Efficiency	○	○	○	○
Main Compressor Pressure Ratio	Value for Maximum Efficiency	○	●	○	○
Recompression Fraction	Value for Maximum Efficiency	○	○	○	○
Low Temperature Recuperator Main Fraction High Pressure Component Mass Fraction	Value for Maximum Efficiency	○	○	○	○
Main Compressor Inlet Pressure [Pa]	Value for Maximum Efficiency	○	○	○	○
Maximum Temperature [K]	923.0	○	○	○	○
Minimum Temperature [K]	320.0	○	○	○	○
Main Compressor Isentropic Efficiency	0.85	○	○	○	○
PreCompressor Isentropic Efficiency	0.875	○	○	○	○
ReCompressor Isentropic Efficiency	0.875	●	○	○	○
Power Turbine Isentropic Efficiency	0.93	○	○	○	○
Main/Re/Pre Compressor Turbine Isentropic Efficiency	0.89	○	○	○	○
Heat Exchanger Pressure Drop [Pa/K]	500.0	○	○	○	○
Sensitivity Plot Dependent Variable				Plot Value: Contour Plot	Plot Value: Line Plot
Maximum Cycle Efficiency				●	●

Cycle Plot			
Quantity	Horizontal Axis	Vertical Axis	Contour Level
None (loads quicker)			○
Temperature		●	○
Pressure	●		○
Enthalpy		○	○
Entropy	○		○
Density			○
CompressibilityFactor			○
cp			●
gamma			○

Conclusions

- Supercritical CO_2 Power Cycles have the potential for efficiencies of 51.94% with a maximum heat source temperature of 923K (650°C) and a minimum coolant temperature of 320K (47°C).
- A new system layout has been presented which may help to eliminate some of the design challenges with supercritical carbon dioxide engines.
- Highly nonlinear fluid properties present significant challenges in cycle and component design.
- A cycle analysis code has been developed, along with a web based interface for interactively exploring the design space. These tools can be continually expanded and improved to better understand supercritical carbon dioxide power cycles.

Questions?

http://AndySchroder.com/CO2Cycle/