DOCTORAL DISSERTATION:
A STUDY OF POWER CYCLES USING SUPERCRITICAL CARBON DIOXIDE AS THE WORKING FLUID

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OUTLINE

- Introduction
- Supercritical CO_2 Heat Exchanger and Cycle Analysis
- A Closed Loop Recuperated Lenoir Cycle using Supercritical CO_2
- Combined Cycle Engine Cascades
- Conjugate Heat Transfer With Supercritical CO_2
- Novelty of the Current Work
- Conclusions
- Recommended Future Work

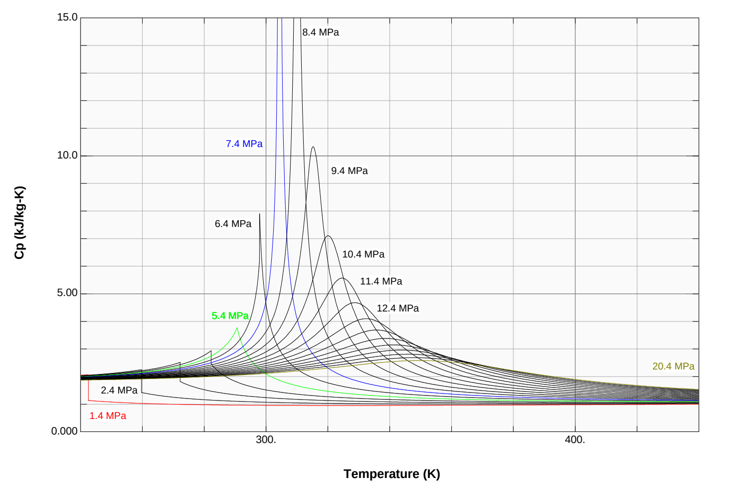
INTRODUCTION

- Supercritical Carbon Dioxide (S-CO_2) Power cycles can possess some favorable qualities of both the Rankine and Brayton cycles.
- S-CO_2 Power cycles are typically proposed as an alternative or compliment to traditional Rankine and Brayton cycle engines.
- Because of their complexity, a S-CO_2 engine has not yet been installed into production use.
- Ongoing research and development aims to make such engines a reality. The present work seeks to help those efforts.

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:
  - Combined cycle using waste heat from a traditional open loop gas turbine
  - Primary cycle with nuclear and solar energy heat sources

CARBON DIOXIDE – C_p VS TEMPERATURE



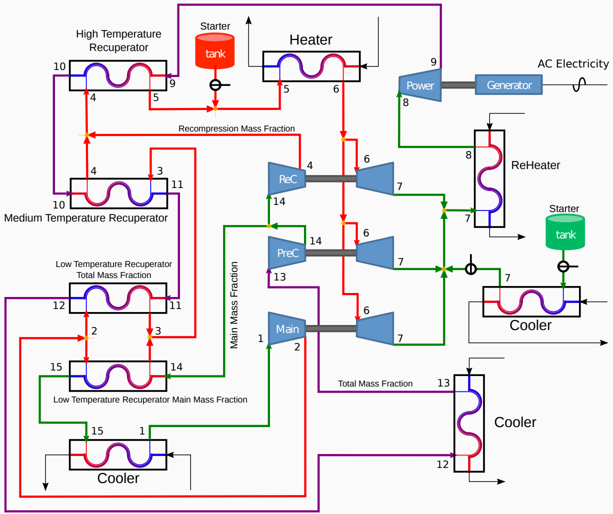
SUPERCRITICAL CO_2 POWER CYCLE - STRENGTHS

- Low Pressure Ratio
- Large amounts of recuperation possible.
- Low back work ratio: Decreased sensitivity of compressor/turbine efficiency on cycle efficiency.
- High Power Density
  - High pressure and high molecular weight.
  - Fluid densities range from $\sim$23 kg/m^3 to $\sim$788 kg/m^3.
- High exergy efficiencies.

SUPERCRITICAL CO_2 POWER CYCLE - WEAKNESSES

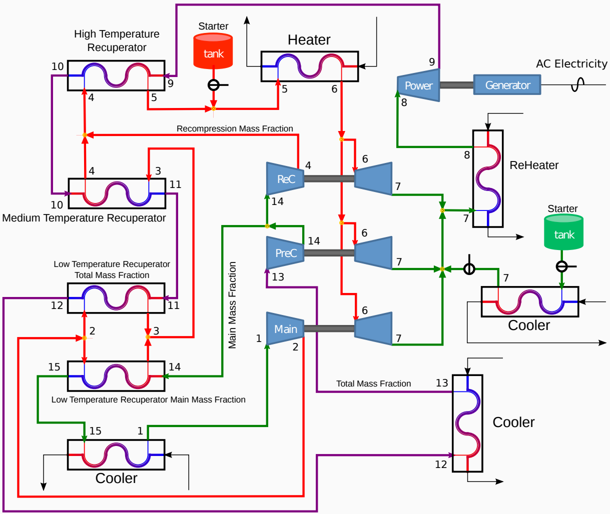
- Nonlinear specific heat mismatch causes difficulties exchanging heat between high and low pressure sides at lower temperatures.
- Heating power in recuperators can be 350% of the net output power and 180% of the input heating power.
- Closed loop design presents additional system complexities.
- High pressures present increased structural loading and seal leakage issues.
- 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.

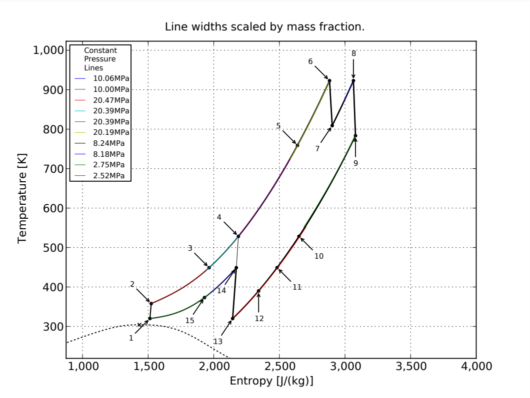
SUPERCRITICAL CO_2 HEAT EXCHANGER AND CYCLE ANALYSIS



- 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

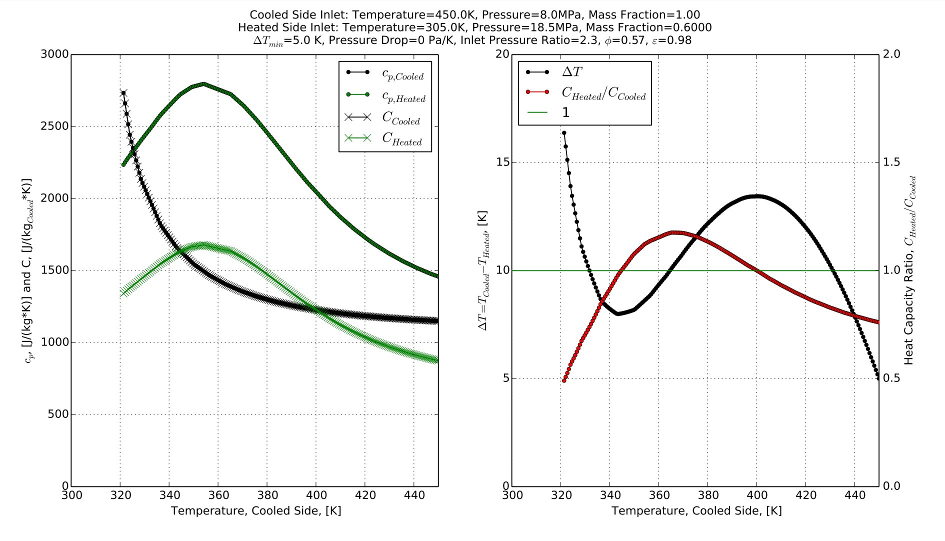




VARIABLE PROPERTY HEAT ENGINE CYCLE ANALYSIS CODE

- A thermodynamic cycle analysis code was created from scratch using Python.
- Variable fluid properties are implemented as a function of both temperature and pressure using REFPROP.
- 0-D counterflow heat exchanger model was developed to account for variable fluid properties, yet maintaining high solution speed.
- Design space for the inputs is explored in parallel and can run on as many processors as are available.

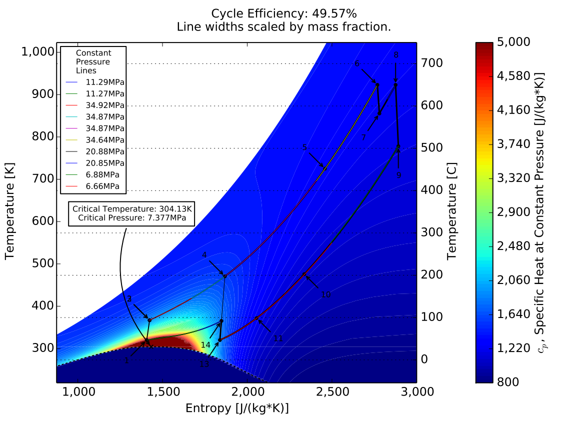
- Minimum temperature difference is defined instead of an effectiveness or surface area and convection coefficients.
- Pressure drop is not computed based on an assumed geometry, but is approximated to be linearly dependent upon temperature drop in the heat exchanger.
- Initial guess for the location of the minimum temperature difference and the corresponding unknown boundaries is made by comparing heat capacities of each fluid stream.
- A root finding technique is used with the initially guessed heat exchanger minimum temperature difference and unknown boundaries in order to find the actual minimum temperature difference and unknown boundaries.

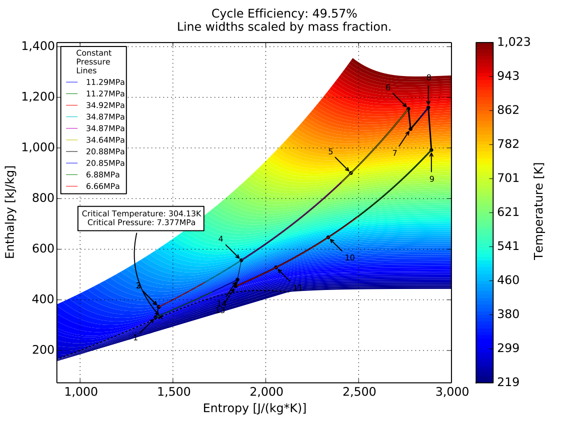

Cooled Side Inlet: Temperature=450.0K, Pressure=8.0MPa, Mass Fraction=1.00
Heated Side Inlet: Temperature=305.0K, Pressure=18.5MPa, Mass Fraction=0.6000
ΔT_{min}=5.0 K, Pressure Drop=0 Pa/K, Inlet Pressure Ratio=2.3, ϕ=0.57, ε=0.98
$c_{p,Cooled}$
$c_{p,Heated}$
C_{Cooled}
C_{Heated}
c_p, [J/(kg*K)] and C, [J/(kg$_{Cooled}$*K)]
Temperature, Cooled Side, [K]
ΔT
C_{Heated}/C_{Cooled}
1
$\Delta T = T_{Cooled} - T_{Heated}$, [K]
Heat Capacity Ratio, C_{Heated}/C_{Cooled}
Temperature, Cooled Side, [K]

CYCLE OPTIMIZATION CONSTRAINTS

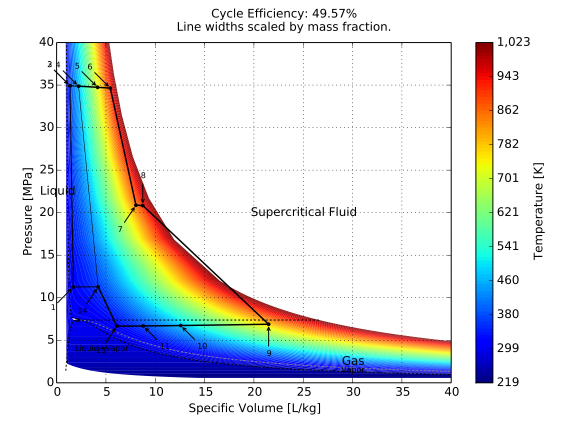
Parameter	Minimum	Maximum
PreCompressor Pressure Ratio	1.0	4.0
Main Compressor Pressure Ratio	1.1	4.1
Recompression Fraction	0.000	0.991
Low Temperature Recuperator Main Fraction High Pressure Component Mass Fraction	0.001	0.991
Main Compressor Outlet Pressure	2 MPa	35 MPa
Maximum Temperature	923 K [650°C]	923 K [650°C]
Minimum Temperature	320 K [47°C]	320 K [47°C]
Main Compressor Isentropic Efficiency	0.850	0.850
PreCompressor Isentropic Efficiency	0.875	0.875
ReCompressor Isentropic Efficiency	0.875	0.875
Power Turbine Isentropic Efficiency	0.930	0.930
Main/Re/Pre Compressor Turbine Isentropic Efficiency	0.890	0.890
Heat Exchanger Minimum Temperature Difference	5 K	5 K
Heat Exchanger Pressure Drop	500 Pa/K	500 Pa/K

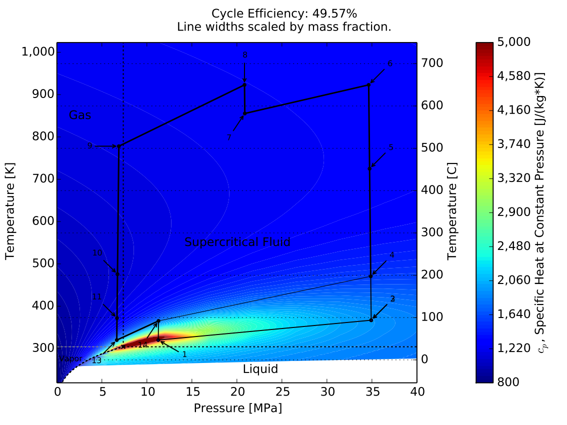
CYCLE T-S AND H-S DIAGRAMS



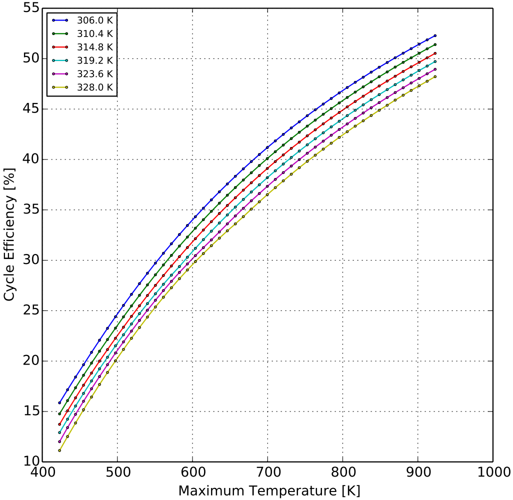


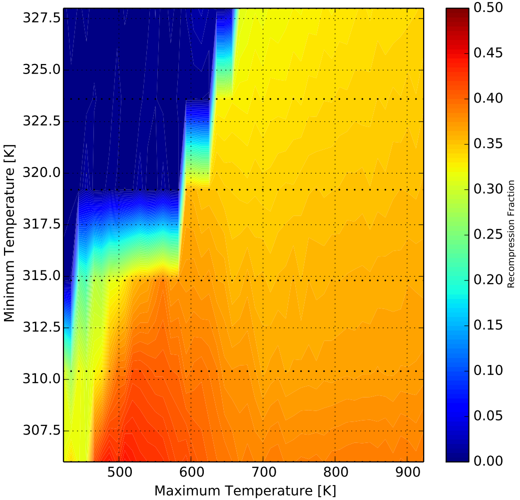
CYCLE P-V AND T-P DIAGRAMS



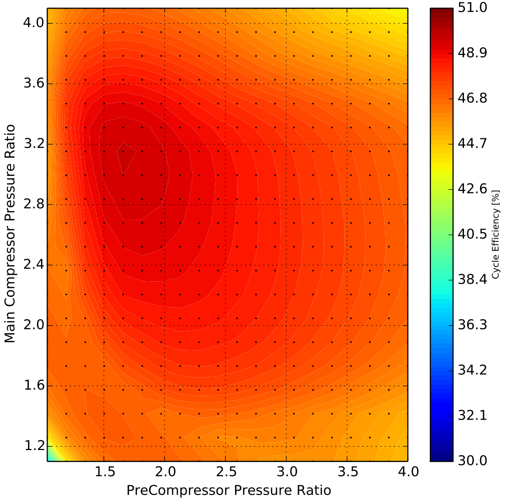


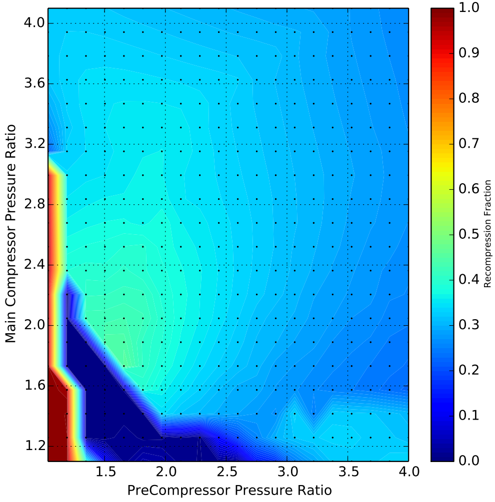
CYCLE EFFICIENCY & RECOMPRESSION FRACTION VS MAX & MIN TEMPERATURE



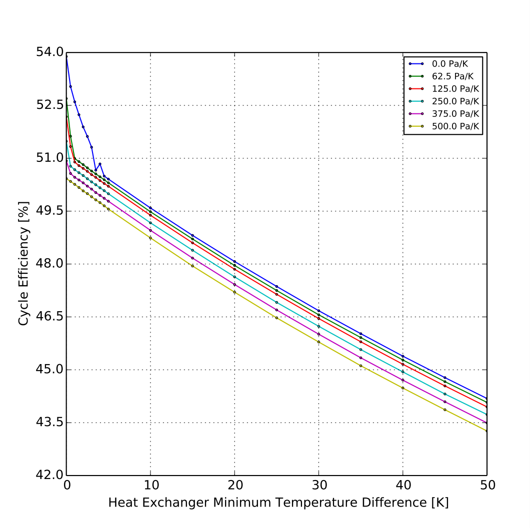


CYCLE EFFICIENCY & RECOMPRESSION FRACTION VS PRESSURE RATIOS

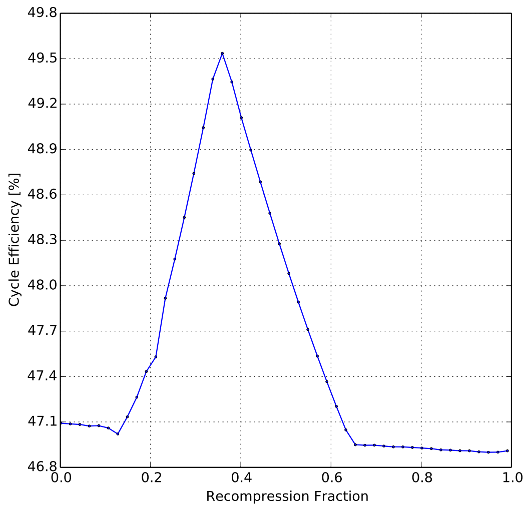


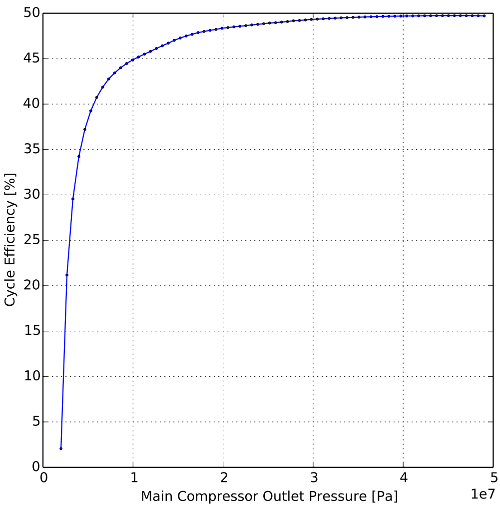


CYCLE EFFICIENCY VS HEAT EXCHANGER MINIMUM TEMPERATURE DIFFERENCE

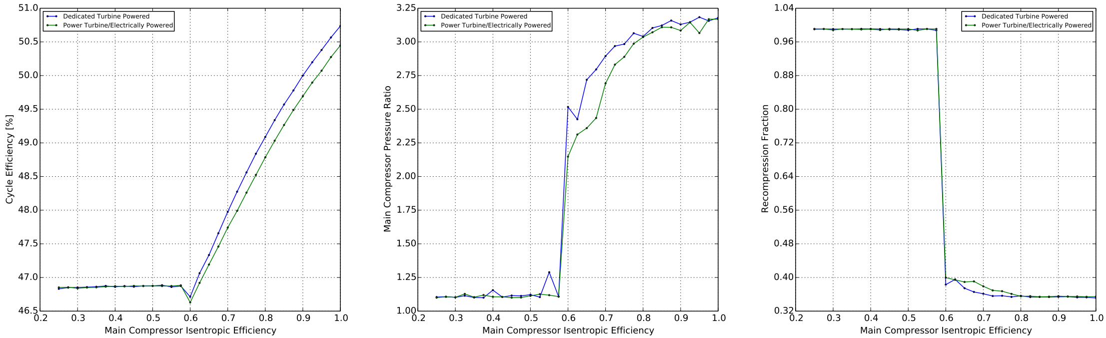


CYCLE EFFICIENCY VS RECOMPRESSION FRACTION & MAXIMUM PRESSURE

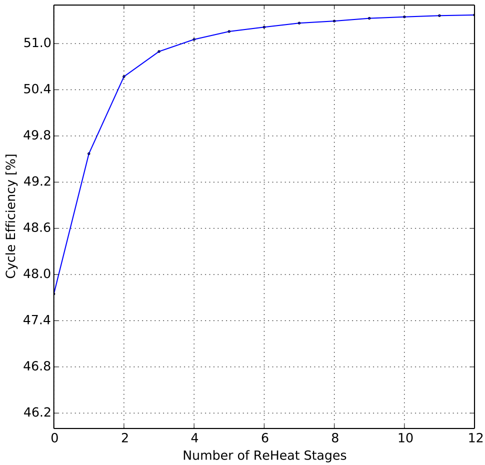




IMPACT OF THE MAIN COMPRESSOR EFFICIENCY AND POWER TAKE OFF POINT



CYCLE EFFICIENCY VS NUMBER OF REHEAT STAGES

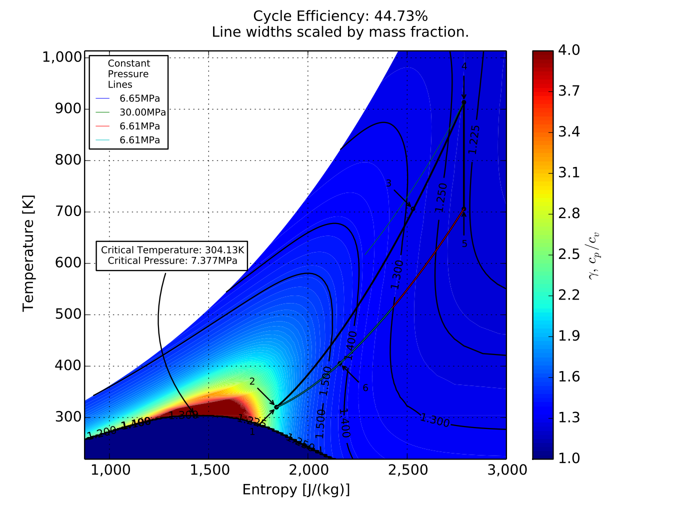


A CLOSED LOOP RECUPERATED LENOIR CYCLE USING SUPERCRITICAL CO_2

WITH CONSTANT VOLUME HEAT ADDITION

- A recuperated Lenoir cycle using supercritical carbon dioxide was studied.
- No other recuperated Lenoir cycle studies or Lenoir cycle studies with carbon dioxide have been identified.
- Efforts were inspired by the efficiency gains predicted for cycles that aim to approximate the Humphrey cycle, variation in fluid properties of carbon dioxide near the critical point, and the large amounts of recuperation used in the cycle presented previously.
- Cycle currently modeled using many moving chambers with pistons that are heated at constant volume and then expand allowing work to be done on the piston.
- Current analysis is an ideal cycle.
- The same minimum and maximum temperatures were used as in the previous studies (320 K [47°C] and 923 K [650°C]).

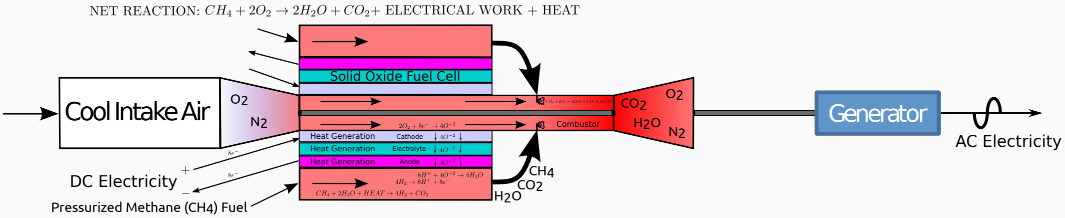
RECUPERATED LENOIR CYCLE - TEMPERATURE ENTROPY DIAGRAM

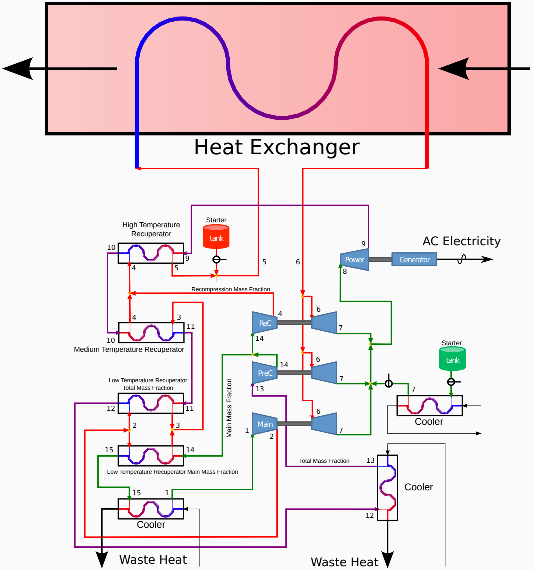


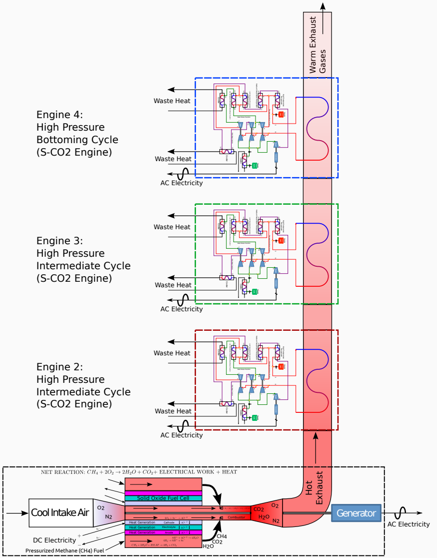
- The ratio of specific heats was too high, particularly at low temperatures, limiting the amount of recuperation possible.
- Low pressure was varied to find the optimal cycle efficiency.
- A significant amount of work was required to compress the fluid at constant pressure.
- Larger heat addition and heat rejection temperature ranges resulted in lower cycle efficiency.
- A more complex layout could be possible, improving the cycle efficiency, but the increased complexity coupled with the complex constant volume heat exchanger are believed to be less feasible and beneficial than increasing the amount of reheat and intercooling in the previously studied cycle.

COMBINED CYCLE ENGINE CASCADES

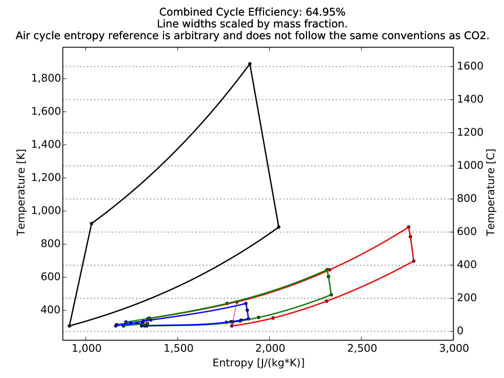
TOPPING CYCLE WITH FUEL CELL




Heat Exchanger
High Temperature Recuperator
Starter
tank
Recompression Mass Fraction
Medium Temperature Recuperator
Low Temperature Recuperator Total Mass Fraction
Low Temperature Recuperator Main Mass Fraction
Main Mass Fraction
Cooler
Power
Generator
AC Electricity
ReC
PreC
Main
Total Mass Fraction
Cooler
Waste Heat
Waste Heat

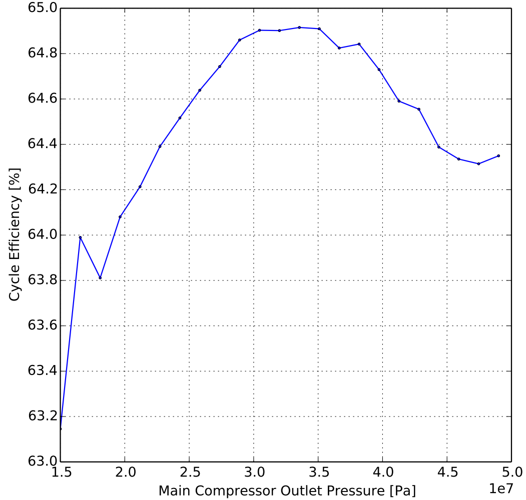


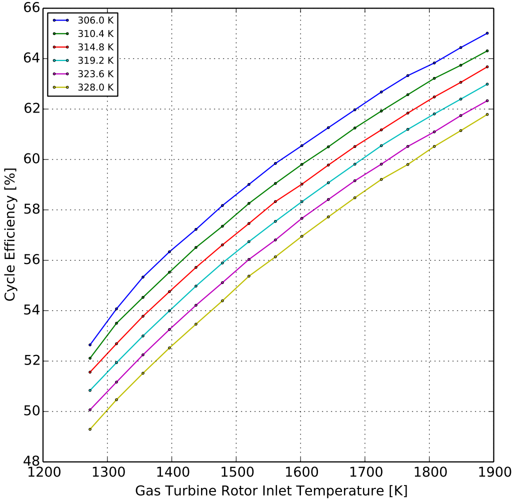
Engine 1: Low/Medium Pressure Topping Cycle (Airbreathing Gas Turbine with Fuel Cell)



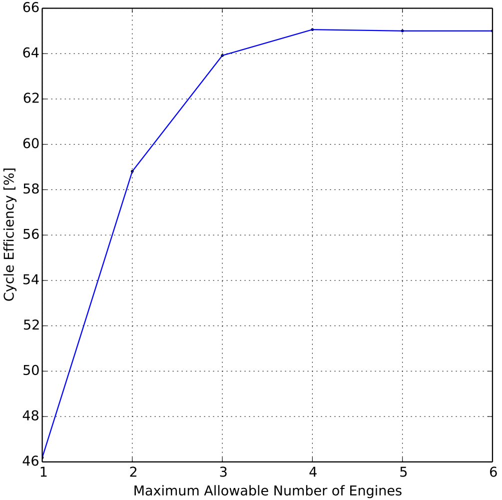
Engine		Work Fraction	Marginal Combined Cycle Efficiency	Engine Efficiency	Engine Exergy Efficiency
Type	Number	%	%	%	%
Gas Turbine	1	70.05	45.49	45.49	54.28
$S - CO_2$ Engine	2	18.60	12.08	49.59	75.02
$S - CO_2$ Engine	3	9.45	6.14	33.53	63.79
$S - CO_2$ Engine	4	1.90	1.23	14.14	46.10
Combined		100.00	64.95	64.95	77.5

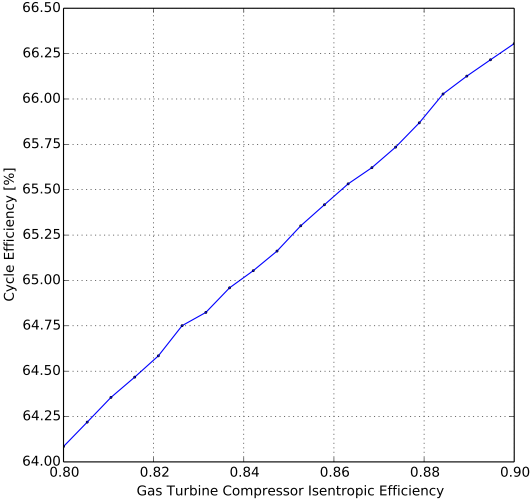
EFFICIENCY VS $S-CO_2$ ENGINE PEAK PRESSURE & TOPPING CYCLE TURBINE INLET TEMP



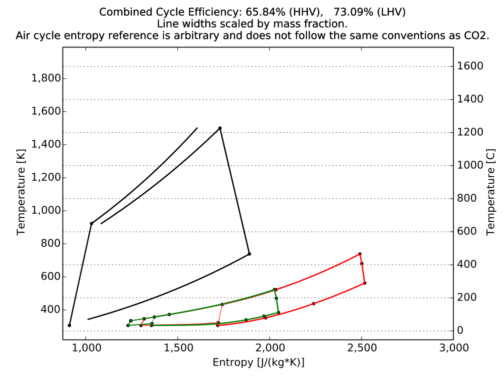


EFFICIENCY VS NUMBER OF ENGINES & TOPPING CYCLE COMP ISENTROPIC EFFICIENCY





COMBINED CYCLE ENGINE WITH FUEL CELL

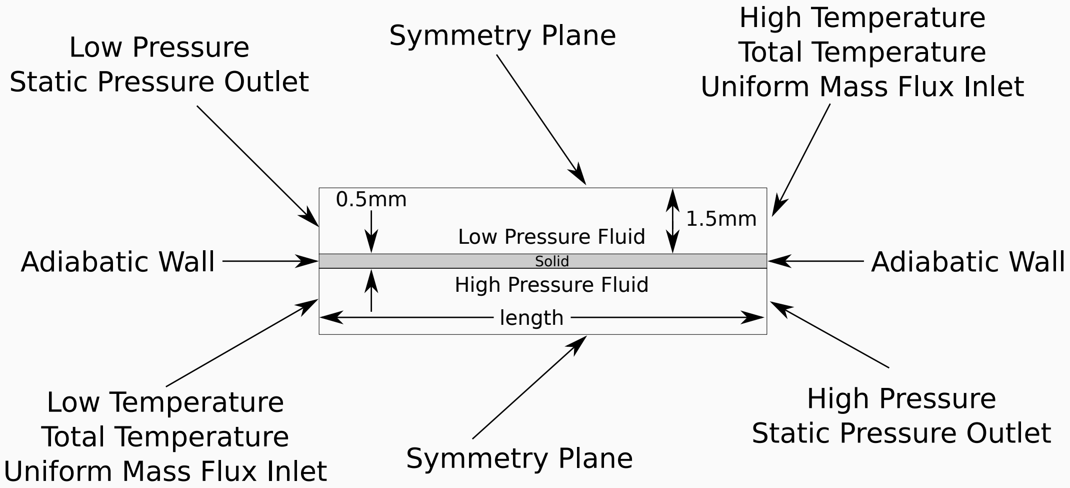


Engine		Work Fraction		Marginal Combined Cycle Efficiency				Engine Efficiency		Engine Exergy Efficiency
Type	Number	%		HHV, %		LHV, %		%		%
Fuel Cell	1	71.14	91.15	46.84	60.01	52.00	66.63	52.00 (LHV)	66.63 (LHV)	-
Gas Turbine		20.01		13.17		14.63		30.47 (LHV)		
$S - CO_2$ Engine	2	6.44		4.24		4.71		41.00		69.99
$S - CO_2$ Engine	3	2.41		1.59		1.76		23.02		55.52
Combined		100.00		65.84		73.09%				-

CONJUGATE HEAT TRANSFER WITH SUPERCRITICAL CO_2

CONJUGATE HEAT TRANSFER WITH SUPERCRITICAL CO_2

- Little experimental and theoretical research has been conducted related to supercritical carbon dioxide power cycle applications.
- Other efforts have focused on heat transfer with supercritical carbon dioxide and constant heat flux or constant temperature boundary conditions.
- Accurate understanding of real heat exchangers is critical in assessing real engine cycle performance, potentially more significant than the turbomachinery.


Symmetry Plane
Low Pressure
Static Pressure Outlet
High Temperature
Total Temperature
Uniform Mass Flux Inlet
0.5mm
1.5mm
Low Pressure Fluid
Adiabatic Wall
Solid
Adiabatic Wall
High Pressure Fluid
length
Low Temperature
Total Temperature
Uniform Mass Flux Inlet
Symmetry Plane
High Pressure
Static Pressure Outlet

2-D HEAT TRANSFER CASES

Case	Re_{D_h}, High Pressure Inlet	Viscous Model	Low Pressure Inlet Total Temperature	Low Pressure Outlet Static Pressure	High Pressure Inlet Total Temperature	High Pressure Outlet Static Pressure	High Pressure Mass Fraction	Length	Notes
I	10	Laminar	450 K	5 MPa	305 K	25 MPa	0.565	1 m	Low Re, Low ΔT_{min}
II	50	Laminar	450 K	5 MPa	305 K	25 MPa	0.565	1 m	Low Re, Medium ΔT_{min}
III	3,000	Turbulent	450 K	5 MPa	305 K	25 MPa	0.565	1 m	High Re, High ΔT_{min}
IV	4,000	Turbulent	450 K	5 MPa	305 K	25 MPa	0.565	1 m	High Re, High ΔT_{min}
V	3,000	Turbulent	450 K	5 MPa	305 K	25 MPa	0.565	10 m	High Re, Low ΔT_{min}
VI	3,000	Turbulent	700 K	1 MPa	600 K	5 MPa	1.000	10 m	Nearly Constant and Nearly Similar Specific Heats

Geometry Grids

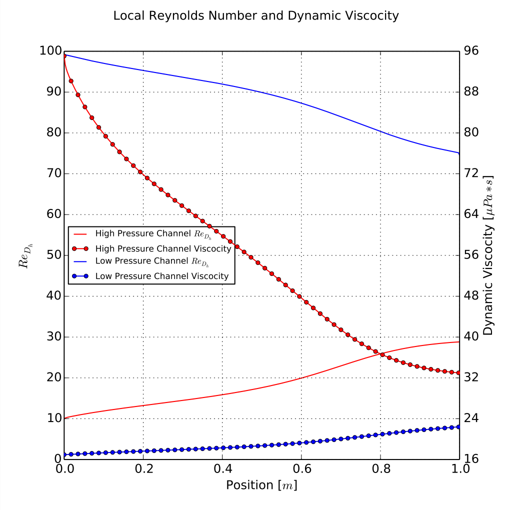
Grid	Top Half Channel Points	Top Channel First Point Spacing From Wall	Bottom Half Channel Points	Bottom Channel First Point Spacing From Wall	Solid Wall Points	Length Points	Total Points
00	41	1.00E-5 m (laminar), 2.50E-6 m (turbulent)	41	1.00E-5 m (laminar), 5.000E-6 m (turbulent)	17	2,609	258,291
11	21	2.00E-5 m (laminar), 5.00E-6 m (turbulent)	21	2.00E-5 m (laminar), 1.000E-5 m (turbulent)	9	1,305	66,555
22	11	4.00E-5 m (laminar), 1.000E-5 m (turbulent)	11	4.00E-5 m (laminar), 2.000E-5 m (turbulent)	5	653	17,631

Fluid Property Grids

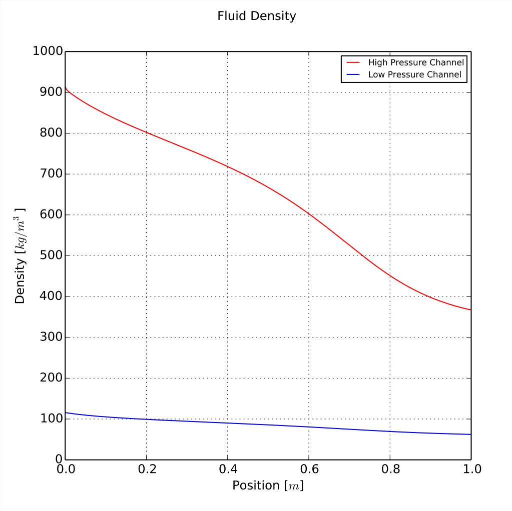
Grid Level	Minimum Temperature	Maximum Temperature	Temperature Points	Minimum Pressure	Maximum Pressure	Pressure Points	Total Points
00	304.22 K	500 K	3001	4.4 MPa	26.0 MPa	217	651,217
11	304.22 K	500 K	1501	4.4 MPa	26.0 MPa	109	163,609
22	304.22 K	500 K	751	4.4 MPa	26.0 MPa	55	41,305

CASE I: HIGH PRESSURE INLET RE$_{d_h}$=10, LAMINAR, 1M LONG

Reynolds Numbers and Average Dynamic Viscosities vs Length Position

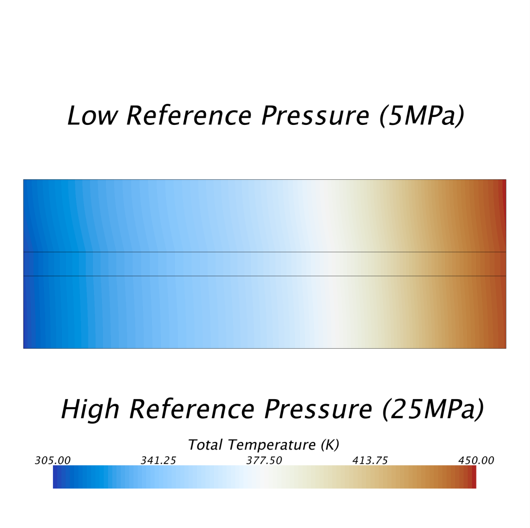


Average Densities vs Length Position



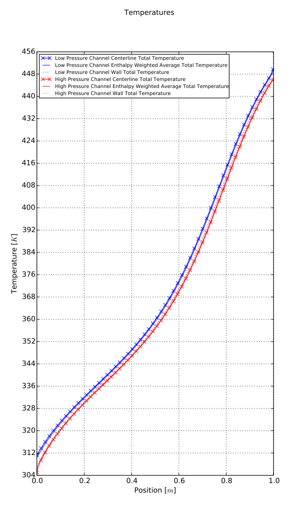
CASE I: HIGH PRESSURE INLET RE_{d_h}=10, LAMINAR, 1M LONG

Total Temperature Contours

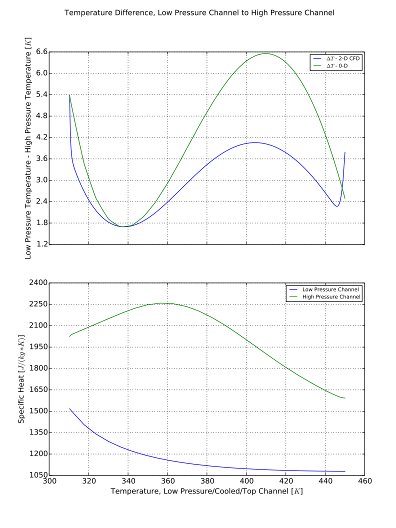


CASE I: HIGH PRESSURE INLET RE_{d_h}=10, LAMINAR, 1M LONG

Temperatures vs Length Position

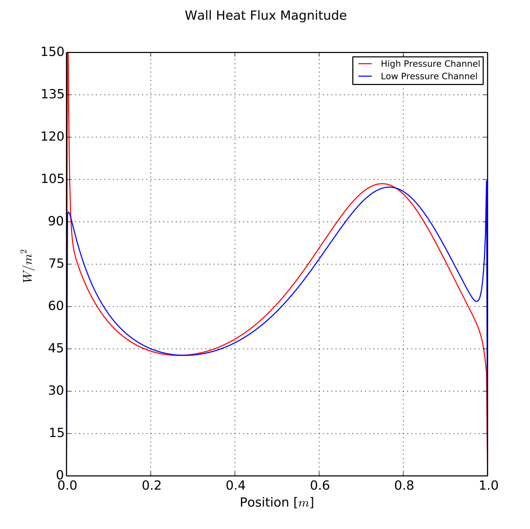


Average ΔT and Specific Heats vs Temperature on the Low Pressure Side

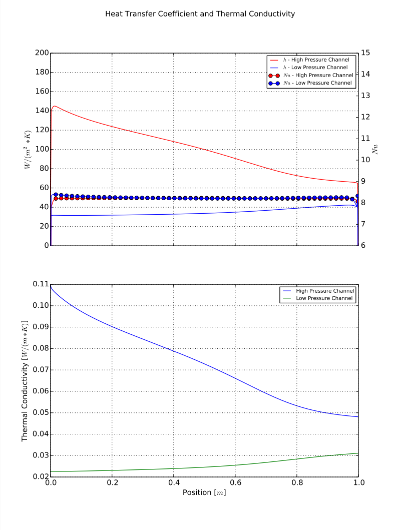


CASE I: HIGH PRESSURE INLET RE$_{d_h}$=10, LAMINAR, 1M LONG

Heat Fluxes vs Length Position

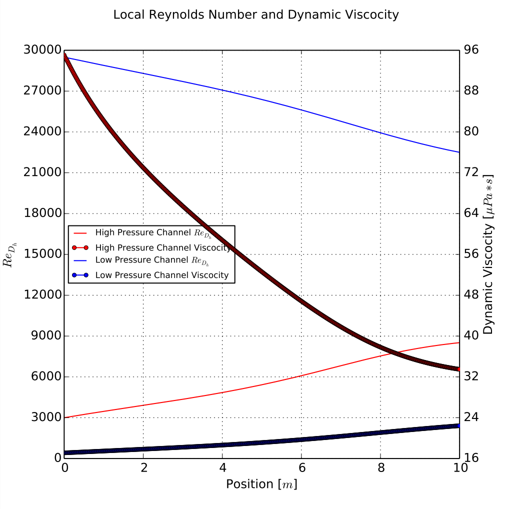


Heat Transfer Coefficients and Average Thermal Conductivities vs Length Position

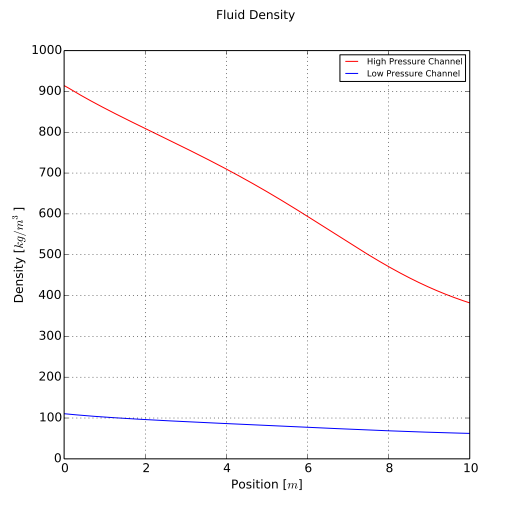


CASE V: HIGH PRESSURE INLET RE_{d_h}=3,000, TURBULENT, 10M LONG

Reynolds Numbers and Average Dynamic Viscosities vs Length Position

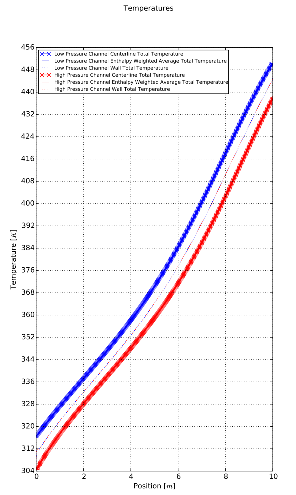


Average Densities vs Length Position

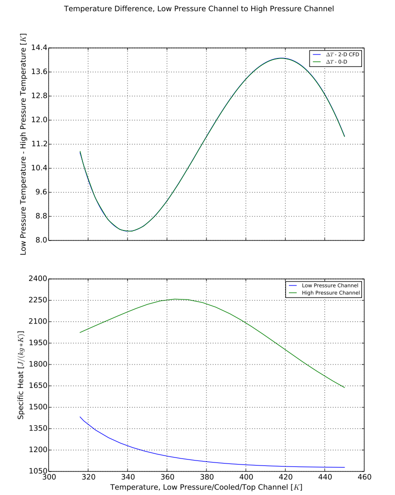


CASE V: HIGH PRESSURE INLET RE_{d_h}=3,000, TURBULENT, 10M LONG

Temperatures vs Length Position

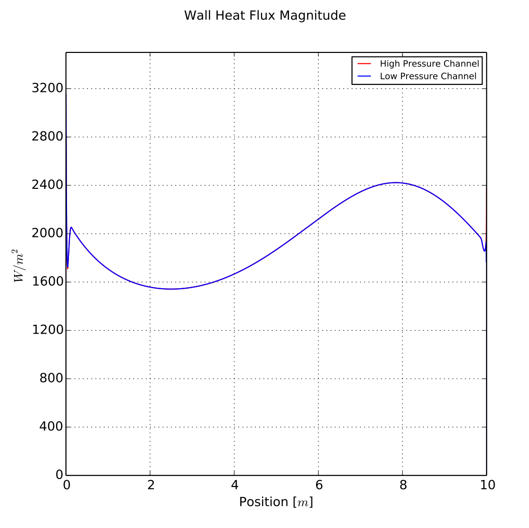


Average ΔT and Specific Heats vs Temperature on the Low Pressure Side

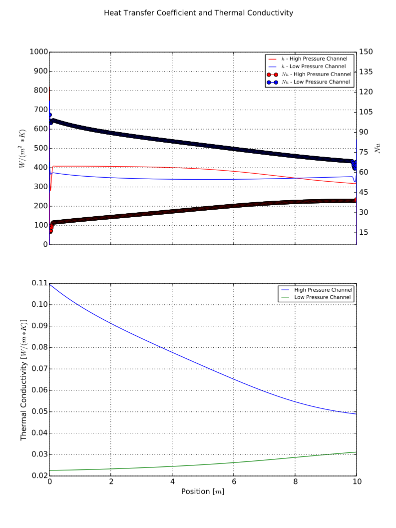


CASE V: HIGH PRESSURE INLET RE_{d_h}=3,000, TURBULENT, 10M LONG

Heat Fluxes vs Length Position



Heat Transfer Coefficients and Average Thermal Conductivities vs Length Position



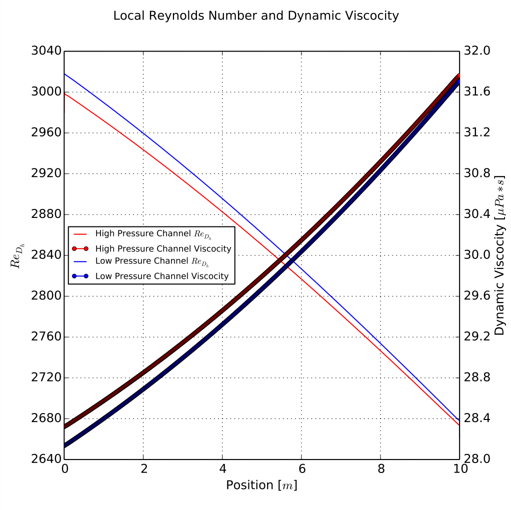
CASE VI: HIGH PRESSURE INLET Re_{d_h}=3,000, TURBULENT, 10M LONG, NEARLY CONSTANT C_p

Fluid Property Grid II

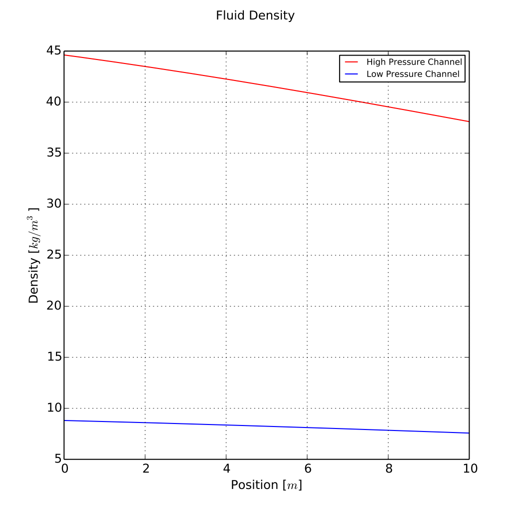
Grid Level	Minimum Temperature	Maximum Temperature	Temperature Points	Minimum Pressure	Maximum Pressure	Pressure Points	Total Points
00	590 K	710 K	3001	1 MPa	5 MPa	217	651,217

CASE VI: HIGH PRESSURE INLET RE_{d_h}=3,000, TURBULENT, 10M LONG, NEARLY CONSTANT C_p

Reynolds Numbers and Average Dynamic Viscosities vs Length Position

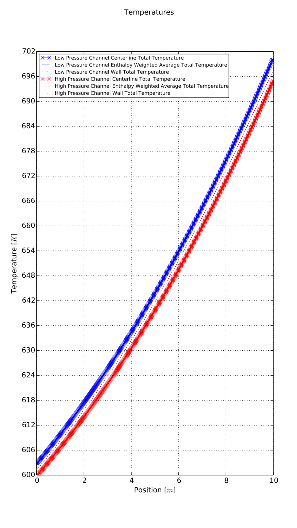


Average Densities vs Length Position

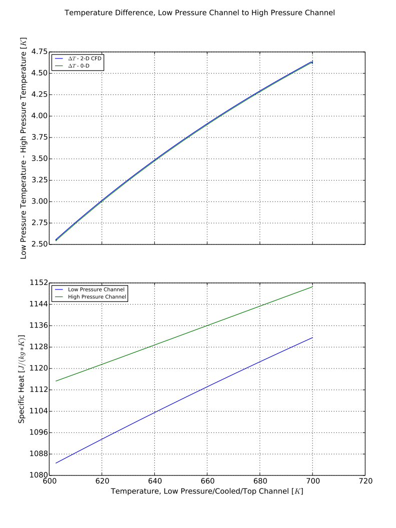


CASE VI: HIGH PRESSURE INLET RE_{d_h}=3,000, TURBULENT, 10M LONG, NEARLY CONSTANT C_p

Temperatures vs Length Position

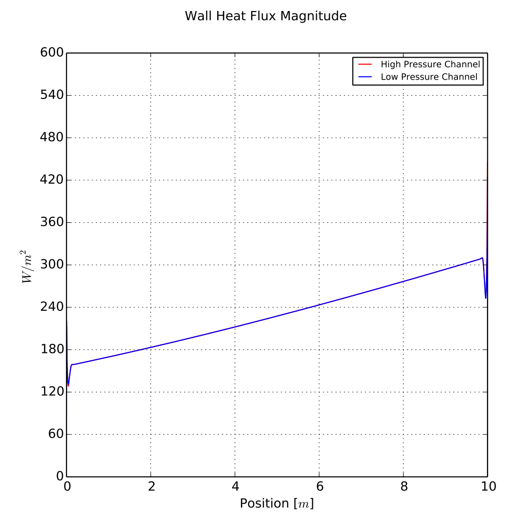


Average ΔT and Specific Heats vs Temperature on the Low Pressure Side

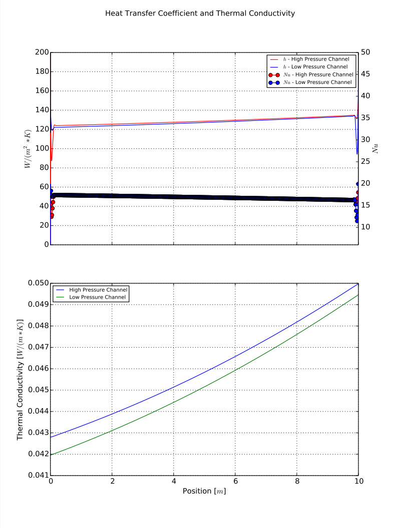


CASE VI: HIGH PRESSURE INLET RE_{d_h}=3,000, TURBULENT, 10M LONG, NEARLY CONSTANT C_p

Heat Fluxes vs Length Position



Heat Transfer Coefficients and Average Thermal Conductivities vs Length Position



NOVELTY OF THE CURRENT WORK

NOVELTY OF THE CURRENT WORK

- A new shaft layout and startup procedure are presented.
- With the multi-shaft configuration, turbomachinery can be placed inside pressure vessels to avoid high pressure ratio seals.
- A new variable property cycle analysis code was developed.
- The design space of the proposed cycle layout has been optimized and explored in detail in a very general manner.
- A cycle efficiency of 49.57% has been predicted with a turbine inlet temperature of 923 K [650°C] and a heat rejection temperature of 320 K [47°C].
- The significance of implementing multiple reheat stages in the turbine on cycle efficiency were explored.
- A closed loop recuperated Lenoir cycle using supercritical carbon dioxide was investigated.

NOVELTY OF THE CURRENT WORK

- Combined cycle configurations using supercritical carbon dioxide power cycles in conjunction with a fuel cell and gas turbine has been explored and optimized.
- A combined cycle efficiency of 64.95% was predicted for the combined cycle without a fuel cell with a turbine inlet temperature of 1,890 K [1,617°C] and a rejected heat temperature of 306 K [33°C].
- A combined cycle efficiency of 73.09% was predicted for the combined cycle with a fuel cell with a rejected heat temperature of 306 K [33°C].
- Two dimensional conjugate heat transfer was studied with a simple channel geometry using supercritical carbon dioxide and variable fluid property formulations.
- Averaged two dimensional results were in close agreement with the zero dimensional heat exchanger solver, validating its applicability in the cycle analysis code.

CONCLUSIONS

CONCLUSIONS

- Supercritical CO_2 Power Cycles have the potential for high efficiencies at low turbine inlet temperatures.
- Highly nonlinear fluid properties present significant challenges in cycle and component design.
- Appropriate modeling of heat exchangers is critical in assessing correct cycle performance.
- In order to use a 0-D heat exchanger model, a sufficiently long heat exchanger is assumed.
- Further investigations of the recuperated Lenoir cycle are not recommended.
- Supercritical carbon dioxide power cycles can be very beneficial in combined cycle configurations, provided multiple supercritical carbon dioxide power cycles are used and each cycle is optimized individually.

RECOMMENDED FUTURE WORK

RECOMMENDED FUTURE WORK

- Allow for variable turbomachinery efficiencies which are dependent on the inlet conditions and pressure ratio.
- Improve pressure drop relationships for heat exchangers in the 0-D heat exchanger solver.
- Support condensation and boiling in heat exchangers.
- Further investigate the use of CoolProp as a replacement for REFPROP.
- Incorporate a cost model into the cycle optimization process.
- Conduct numerical simulations of more realistic heat exchanger geometries.
- Conduct preliminary design and numerical simulations of turbomachinery components.

Questions?