Thermal Optimization and Comparative Analysis of Supercritical CO_2 and Steam Rankine Cycles Across Source and Sink Conditions

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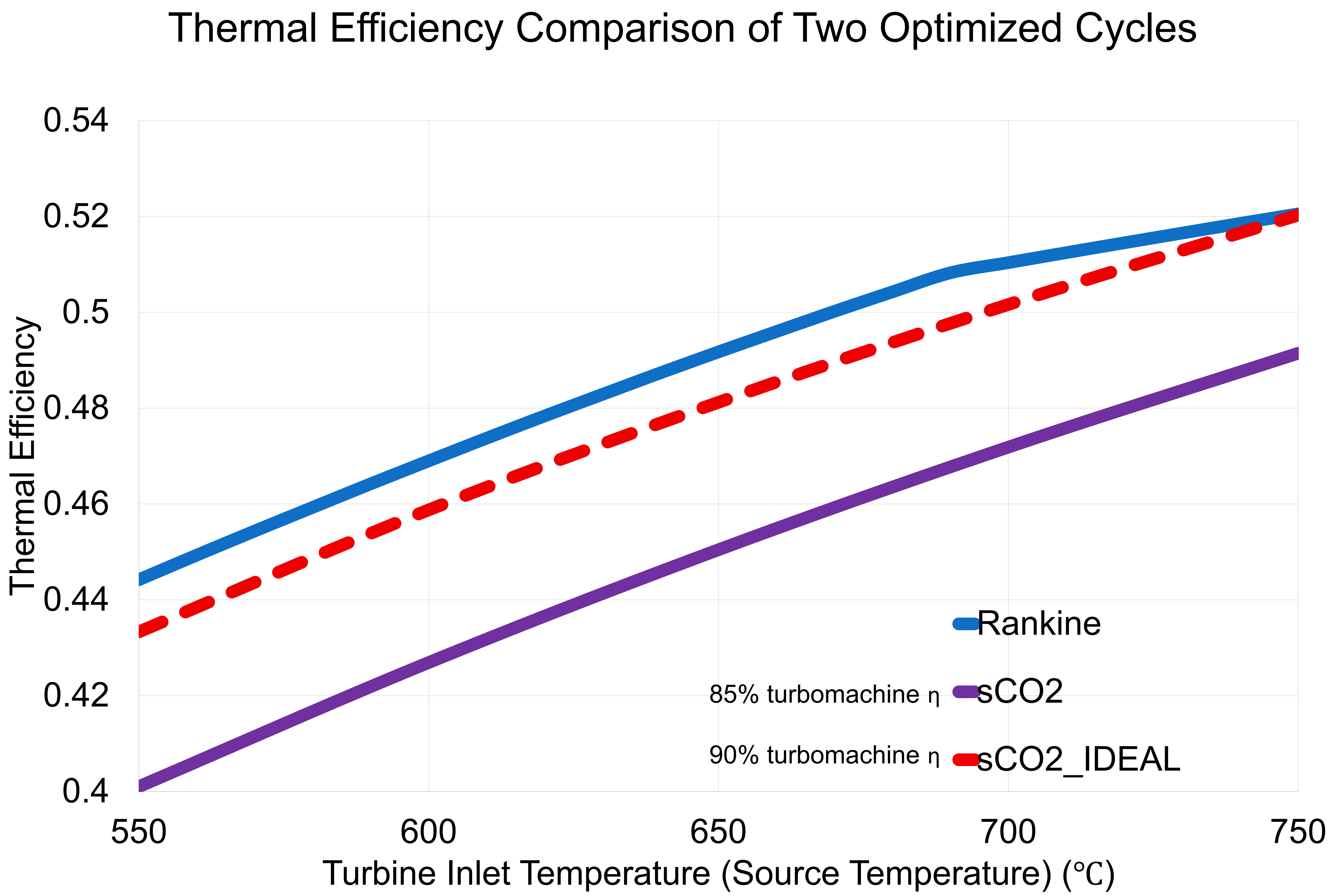
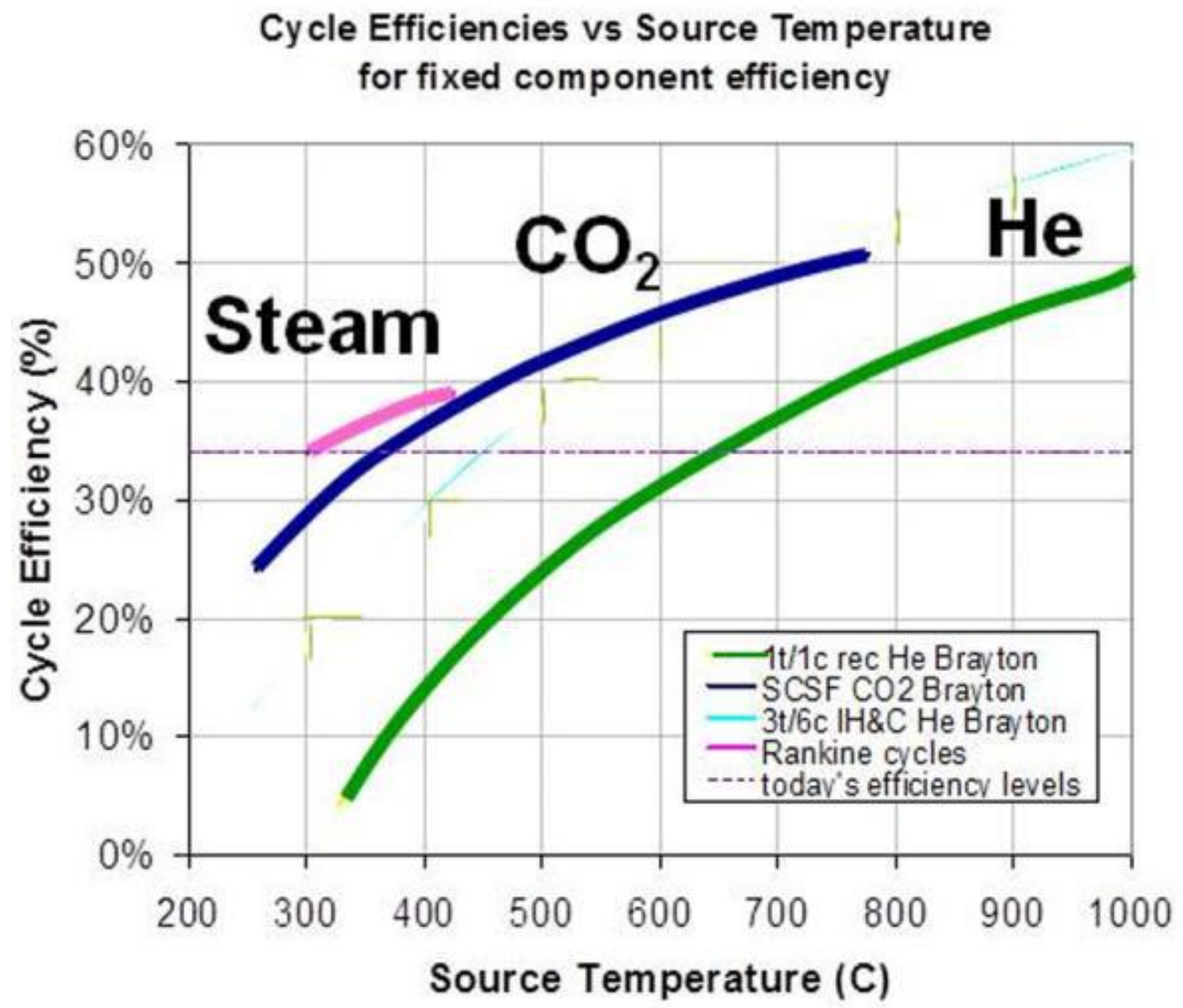
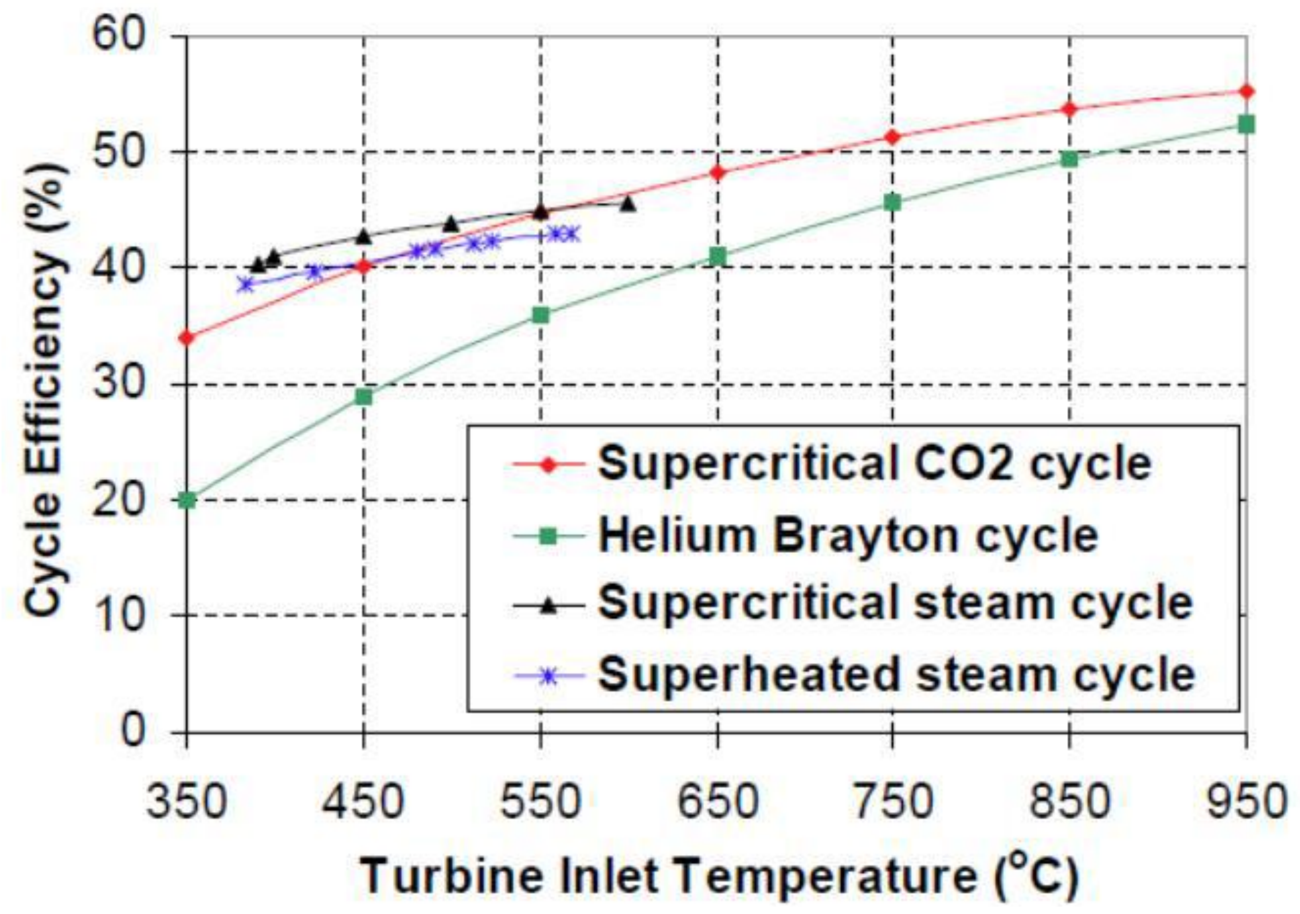
Previous conventional wisdom about the efficiency benefits of sCO_2 cycles must be challenged!







Fleming, D., Holschuh, T., Conboy, T., Rochau, G., & Fuller, R. (2012, June). Scaling considerations for a multi-megawatt class supercritical CO2 Brayton cycle and path forward for commercialization. In *Turbo Expo: Power for Land, Sea, and Air* (Vol. 44717, pp. 953-960). American Society of Mechanical Engineers.

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Preface

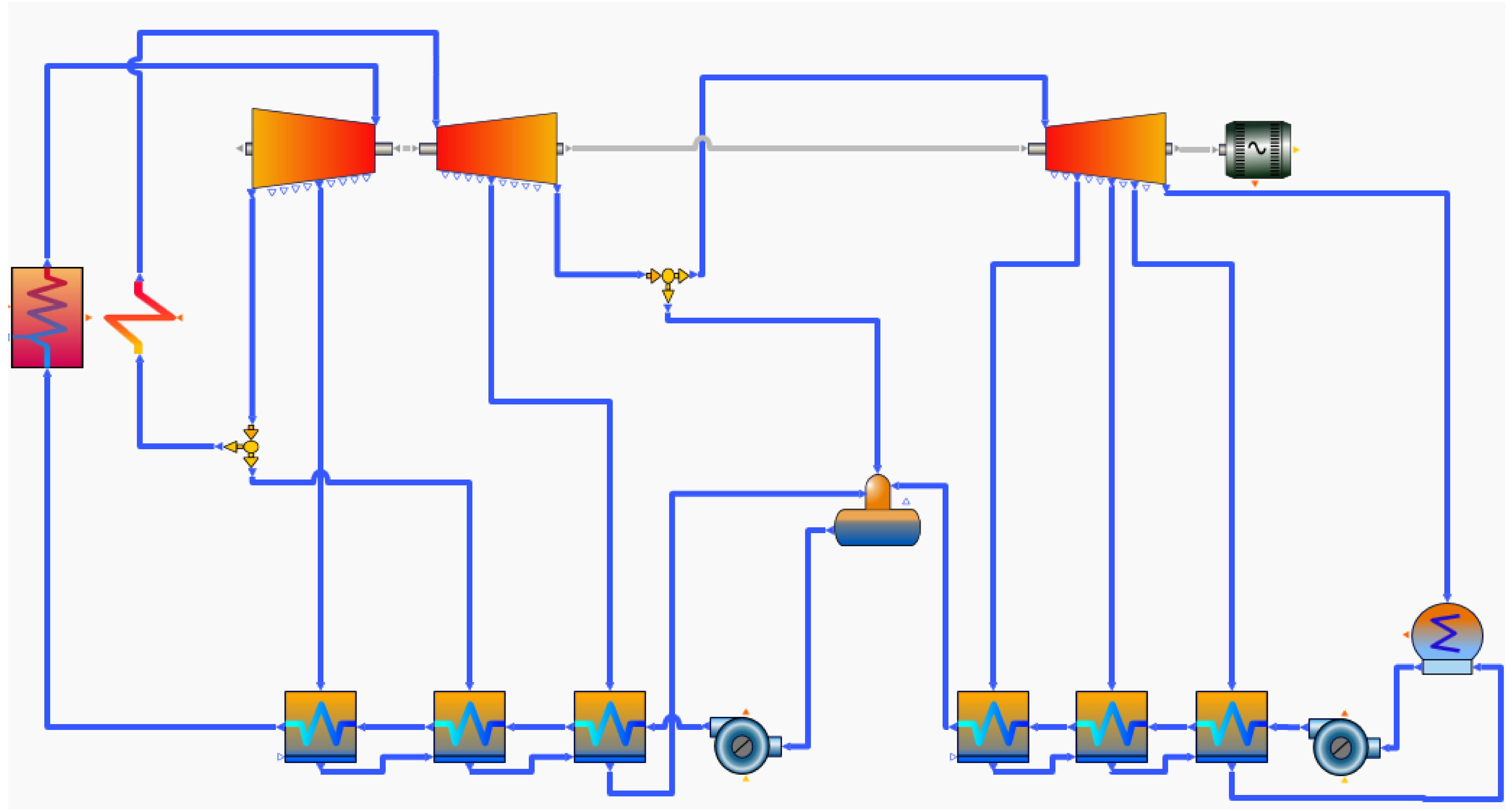
The previous slide challenges the conventional wisdom about the efficiency benefits of the sCO_2 power conversion cycle in comparison to the incumbent water-steam Rankine cycle. It should be taken with the limited range of applicability under which the comparison was made. It is NOT meant to suggest that there are not other advantages that can be leveraged and that there are good technical use cases for each cycle operating on power generation applications.

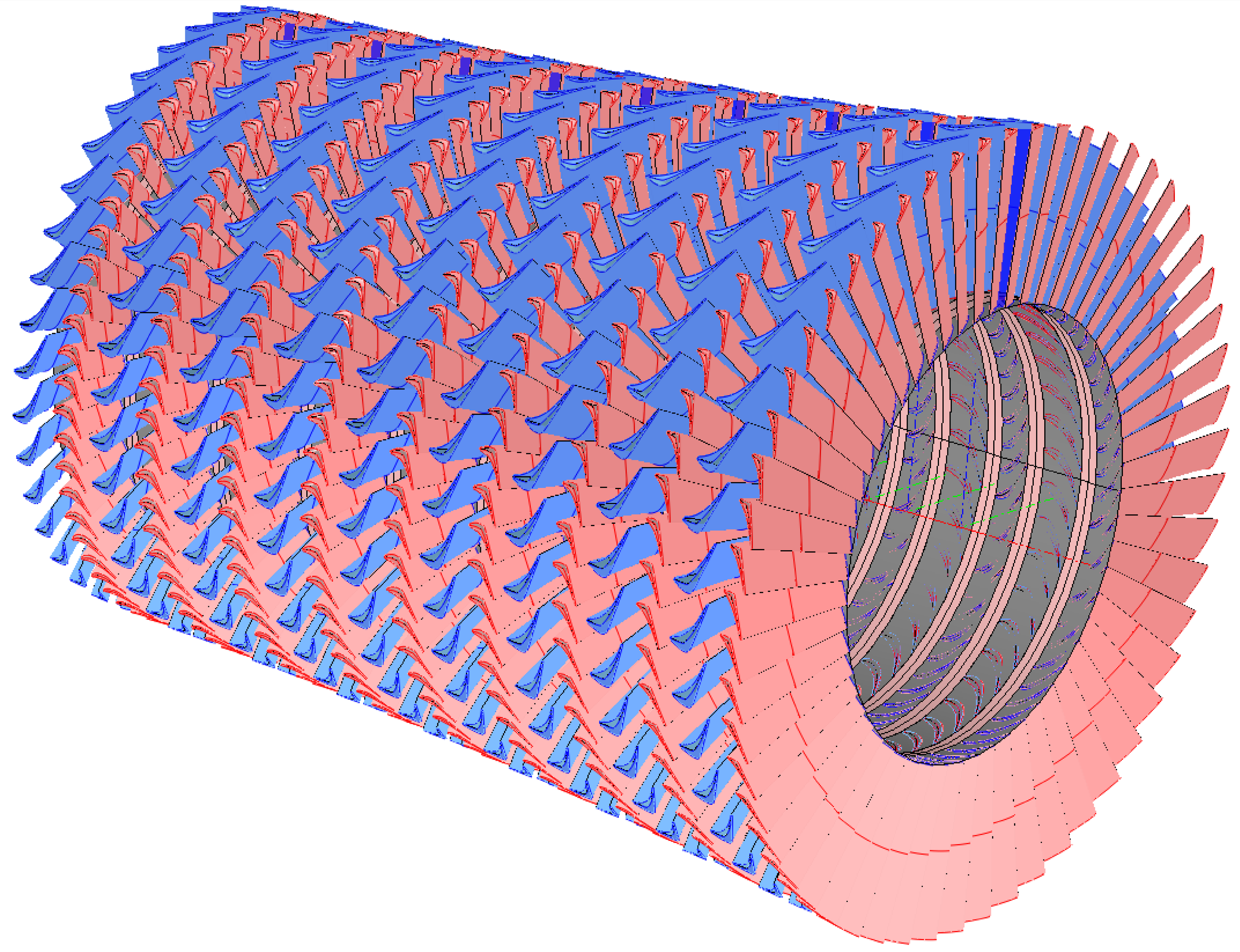
Topics Worthy of Consideration

- Focus on PRIMARY heat source, not waste recovery (more on this later)
- Particular power cycle applications, such as waste heat recovery may indeed show better efficiency
- Consideration for the capital costs
- Consideration for materials and volume and footprint of the facility
- Considerations for deployment as a power/propulsion drive for transportation applications
- Integration capabilities for energy storage may be different (another topic that is explored within this research)
- Applications of both heat pumping and power production and other hybridized/combined systems
- Resource usage that impacts the cycle; for example, the usage of water for cooling (sink)
- Our case relied strongly on a CST application, but generally is applicable to certain other scenarios.

Agenda

1. Methodology
2. Source Temperature Variability
    1. Steam Rankine PCS
    2. sCO2 Brayton PCS
3. Sink Temperature Variability
    1. Steam Rankine PCS
    2. sCO2 Brayton PCS
4. Equipment Scale and Operational Analysis
    1. Turbomachinery
    2. Heat Exchangers
5. Revisit Source Temperature Variability
6. Cold Storage Study
7. Conclusions

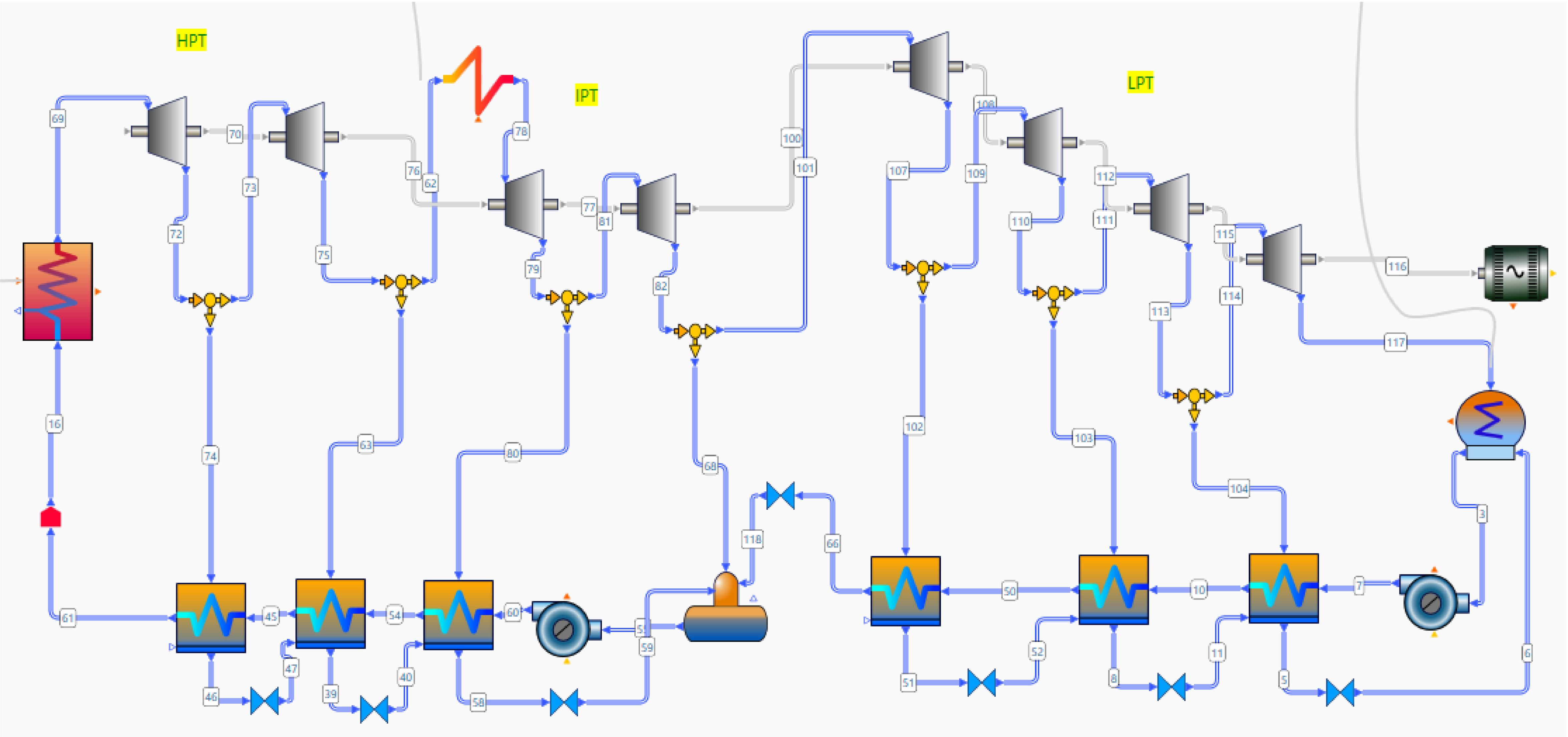


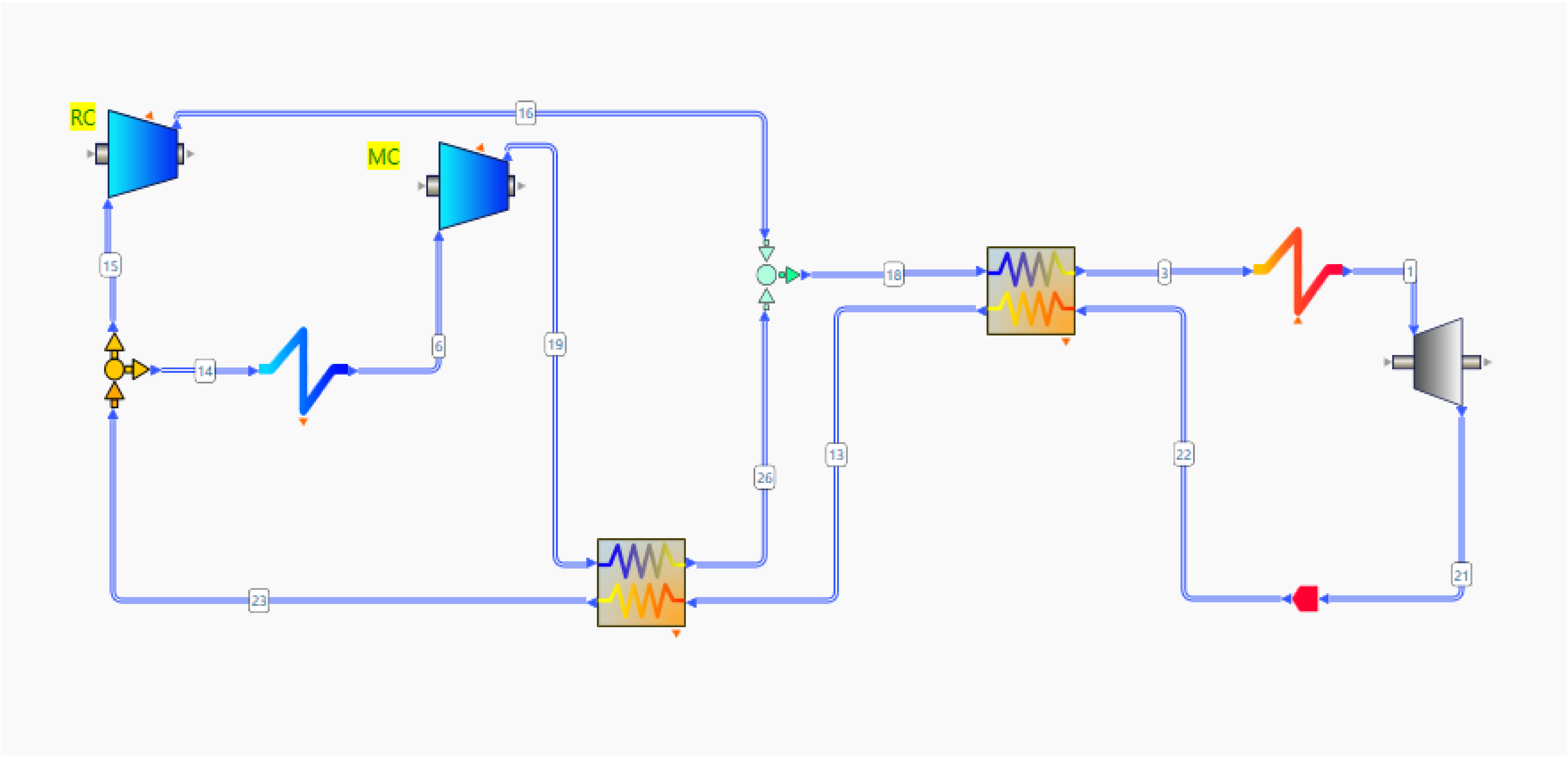


METHODOLOGY

Cycle Configurations

- Primary simulation tool is AxSTREAM™ System Simulation
- Essentially a lumped-parameter 0D thermofluid network model
- Each component obeys fundamental thermodynamic and fluid dynamic physical laws, including energy, mass, and momentum conservation. Components characterized by performance parameters which were either estimated or derived from internal or literature sources.
- Two cycle configurations compared:
  - Steam Rankine: Single reheat subcritical or supercritical, with 7 regenerative stages of feedwater heating. It is the typical minimum complexity configuration (top).
  - sCO2: Re-Compression Brayton Cycle (RCBC). Chosen based on previous research (by others) asserting such configuration as the prime candidate balancing cycle performance and complexity. (Bottom)

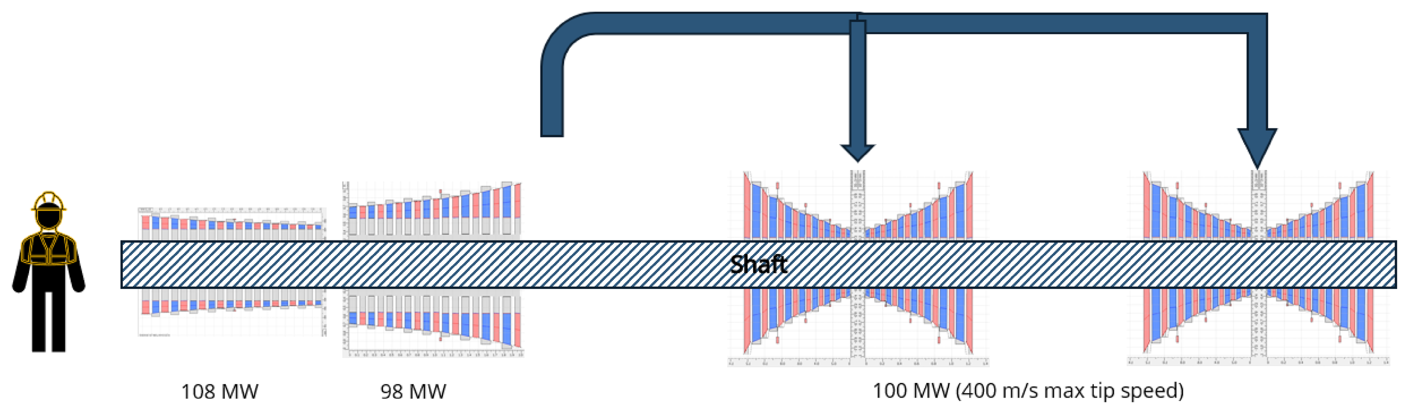




Turbomachinery Performance

A note about efficiencies:

- Steam turbines operating in Rankine cycles are well characterized and understood, as there has been almost 200 years of practice to rely on.
- The steam turbine efficiencies are representative of real-world tested performance after a typical “new and clean” break-in period, so represent the typical state of the art accounting for long-term performance with initial degradation from factory conditions assuming good operating practice.
  - HP: Includes typical throttling and control stage. Has highest impact of clearances due to small initial blade size. Degradation from new and clean can be substantial.
  - IP: Typically best efficiency with fewest degradation mechanisms. No control stage and easily manageable exhaust velocities. Major degradation mechanisms are seal wear and blade deposits from reheat de-superheat spray (temperature control of hot reheat).
  - LP: Most challenge with spanwise distribution and moisture effects on last stages. Exhaust losses typically substantial due to combination of high spanwise distribution/ stratification and near sonic outlet flow. Lowest effect of clearances but also need highest clearances due to movement. Degradation from moisture erosion and seal wear can be substantial.



Turbomachinery	Isentropic Efficiency			Cycle
HP Turbines	88%			Steam Rankine
IP Turbines	94%			Steam Rankine
LP Turbines	91%			Steam Rankine
	Practical	Optimistic	Specific	
Compressor	85%	90%	80%	sCO_2 Brayton
Recompressor	85%	90%	84%	sCO_2 Brayton
Gas Turbine	85%	90%	90%	sCO_2 Brayton

sCO2 Turbomachinery:

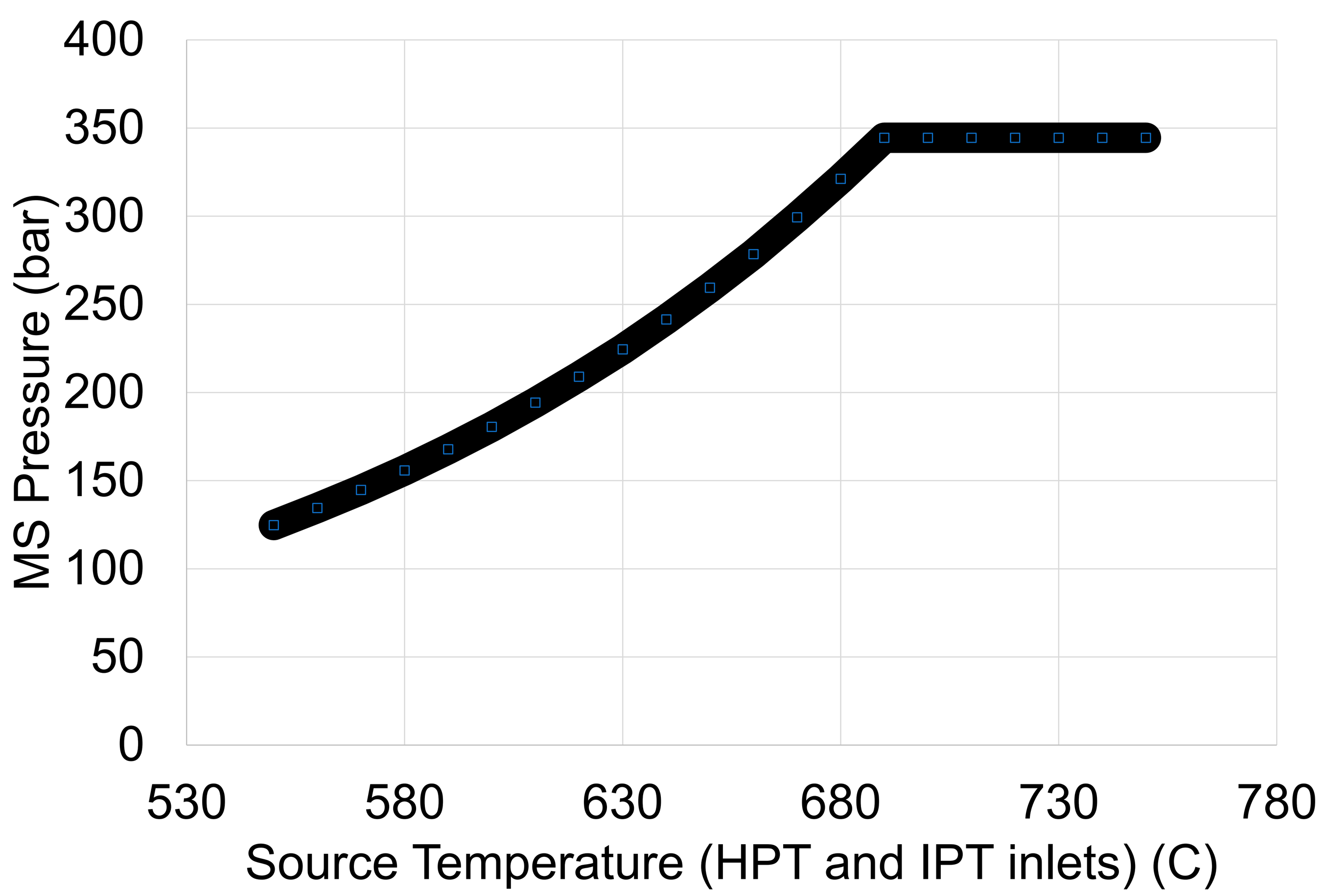
- Technology is much less mature – Is there a utility-scale thermal power plant operating today?
- Little documented effects and challenges of long-term operation
- Predictable Technical Aspects:
  - Small pressure ratio
  - High overall pressure – need to seal through shaft (or hermetic!)
  - Cylinder size comparable to HP steam turbine
  - Required to have control
  - Compressors must operate stably across load range – likely a compromise between efficiency and stability / operating range
  - Transcritical effects have to be factored in, especially main compressor
    - Is it a compressor or pump? Features that ensure safe and stable operation will also result in performance tradeoffs.

Rankine Cycle Setup

Baseline boundary conditions for steam Rankine cycle simulation model

Component	Defined BC value
Steam Generator Final Temperature	550C
MS Pressure	124.8 bar
Reheater Final Temperature	550C
Condenser / LP Exhaust total pressure	P_{sat}(42°C) = 0.083 bar
Exhaust Quality	95%
Condensate pump	12 bar
Boiler Pressure Drop	8 bar
Reheater Pressure Drop	2 bar

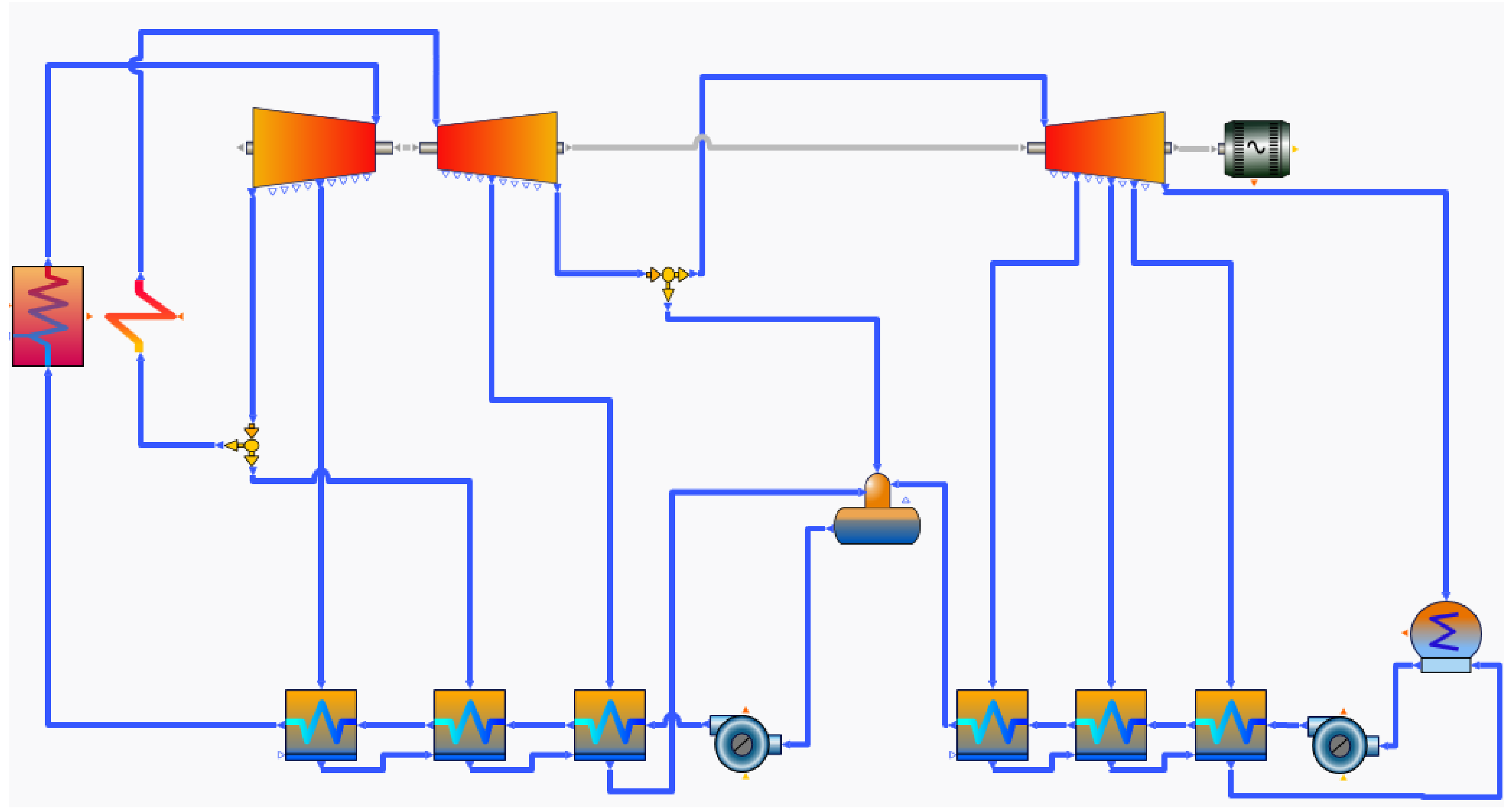
MS Pressure vs MS Temperature to Maintain Target Exhaust Quality



Key optimization for this cycle is top pressure control (MS Pressure) and extraction setup. Most are physically constrained by expansion processes and fundamental limitations.

$$x = \left(\frac{P_{MS}}{P_{exhaust}}\right)^{(1/N)}$$

$$x = pressure\ ratio\ between\ extractions$$
$$N = number\ of\ intervals\ (8)$$



Component	TTD (C)	Extraction MFR (kg/s)
LPFWH1	5	4.02 (2.4%)
LPFWH2	5	6.62 (3.8%)
LPFWH3	5	7.91 (4.3%)
DA	0	3.22 (3.7%)
HPFWH1	5	8.03 (4.1%)
HPFWH2	2	19.95 (9.1%)
HPFWH3	0	30.17 (12.1%)

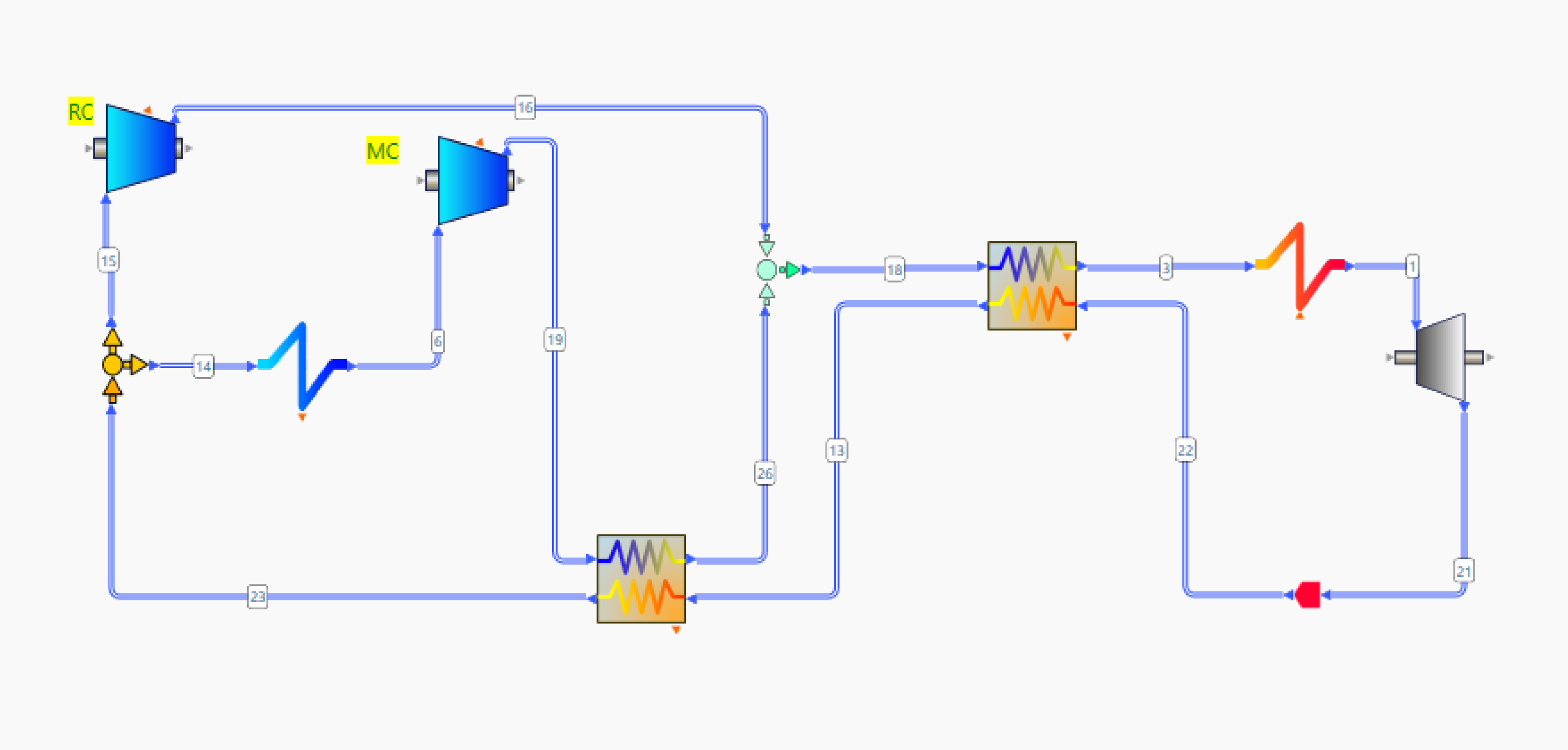
Pressure Stage	i	Extracts to	Value in bar
Main Steam		--	124.8
HP turbine extraction	1	FWH#7	50.012
HP exhaust	2	FWH#6	20.042
IP turbine extraction	3	FWH#5	8.031
IP Exhaust / Crossover	4	DA – FWH#4	3.218
LP turbine extraction 1	5	FWH#3	1.290
LP turbine extraction 2	6	FWH#2	0.5168
LP turbine extraction 3	7	FWH#1	0.2071
LP Exhaust	8	Condenser	0.083

sCO_2 Cycle Setup

Fundamental baseline settings

Component	Defined BC value
Gas Heater Final Temperature - TIT	550C
Turbine Outlet Pressure	108 bar
Split Ratio	0.7
Compressor pressure ratio	3.0
Compressor Outlet Pressure	300 bar
Condensation Temperature	42 C

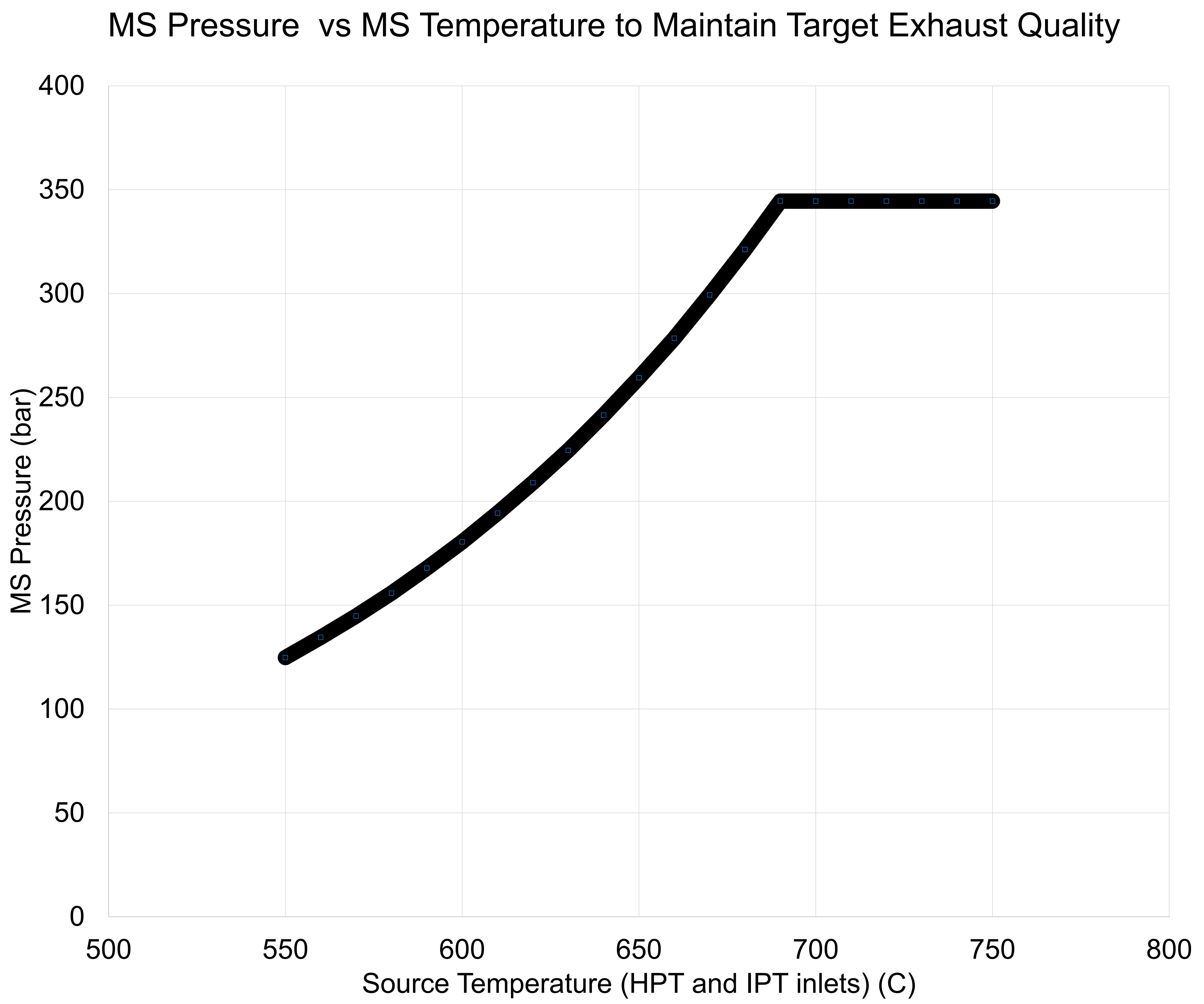
Component	Pressure Drop (bar)
HT Recuperator Hot branch	2
HT Recuperator Cold branch	1
LT Recuperator Hot Branch	5
LT Recuperator Cold Branch	1
Heater	5
Cooler	1



- More variables available to control and not directly tied to fundamental process constraints:
  - Pressure ratio
  - Split ratio
  - Bottom/Top pressure (it can be a trans-critical and/or condensing cycle)
- This is where we must specifically optimize the sCO2 cycle at each operational point (source and sink temperature)
- Some of this work repeats previous results by other researchers, but we must perform the analysis to ensure the results hold for the implementation we have

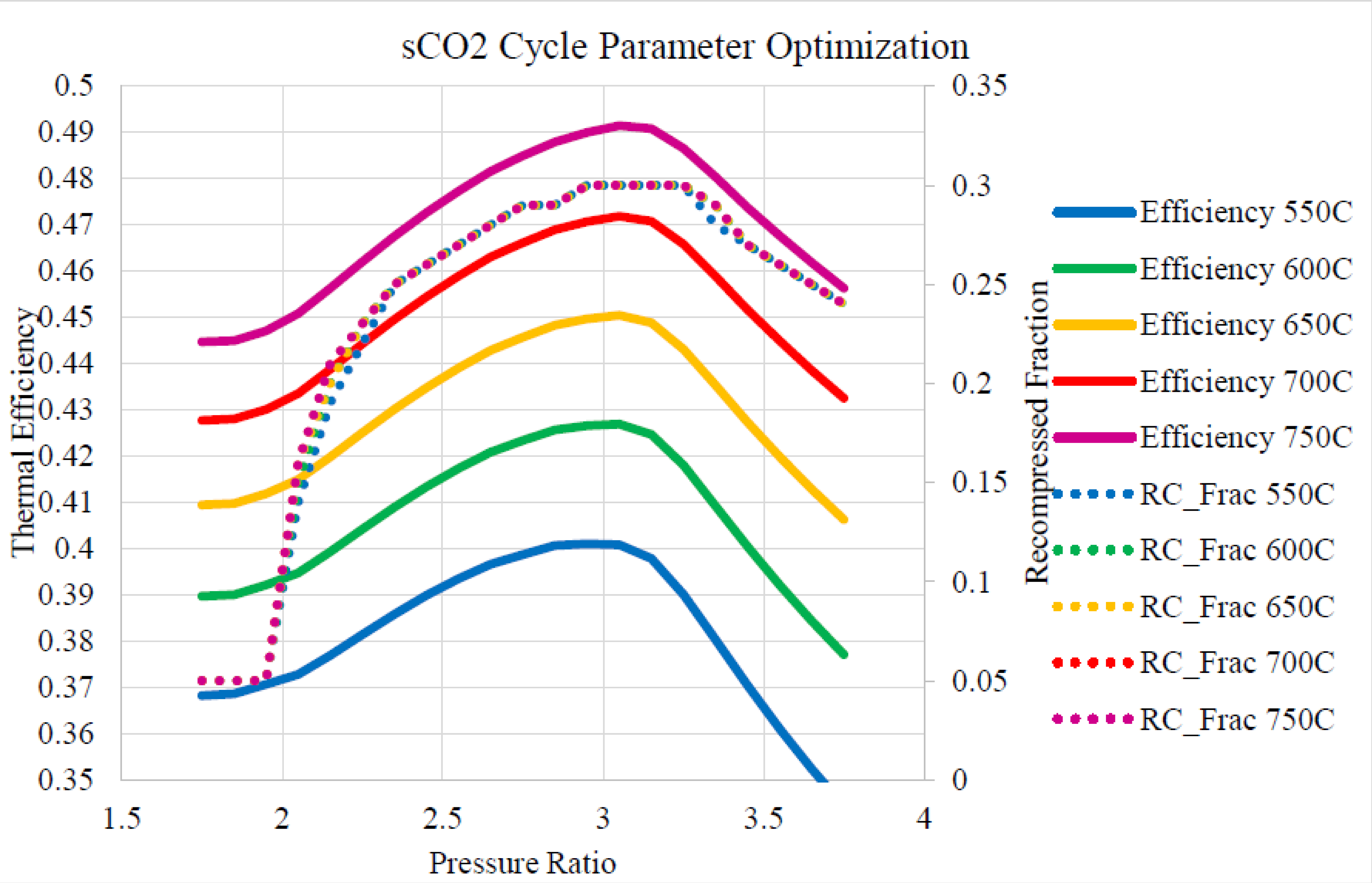
SOURCE TEMPERATURE VARIATION

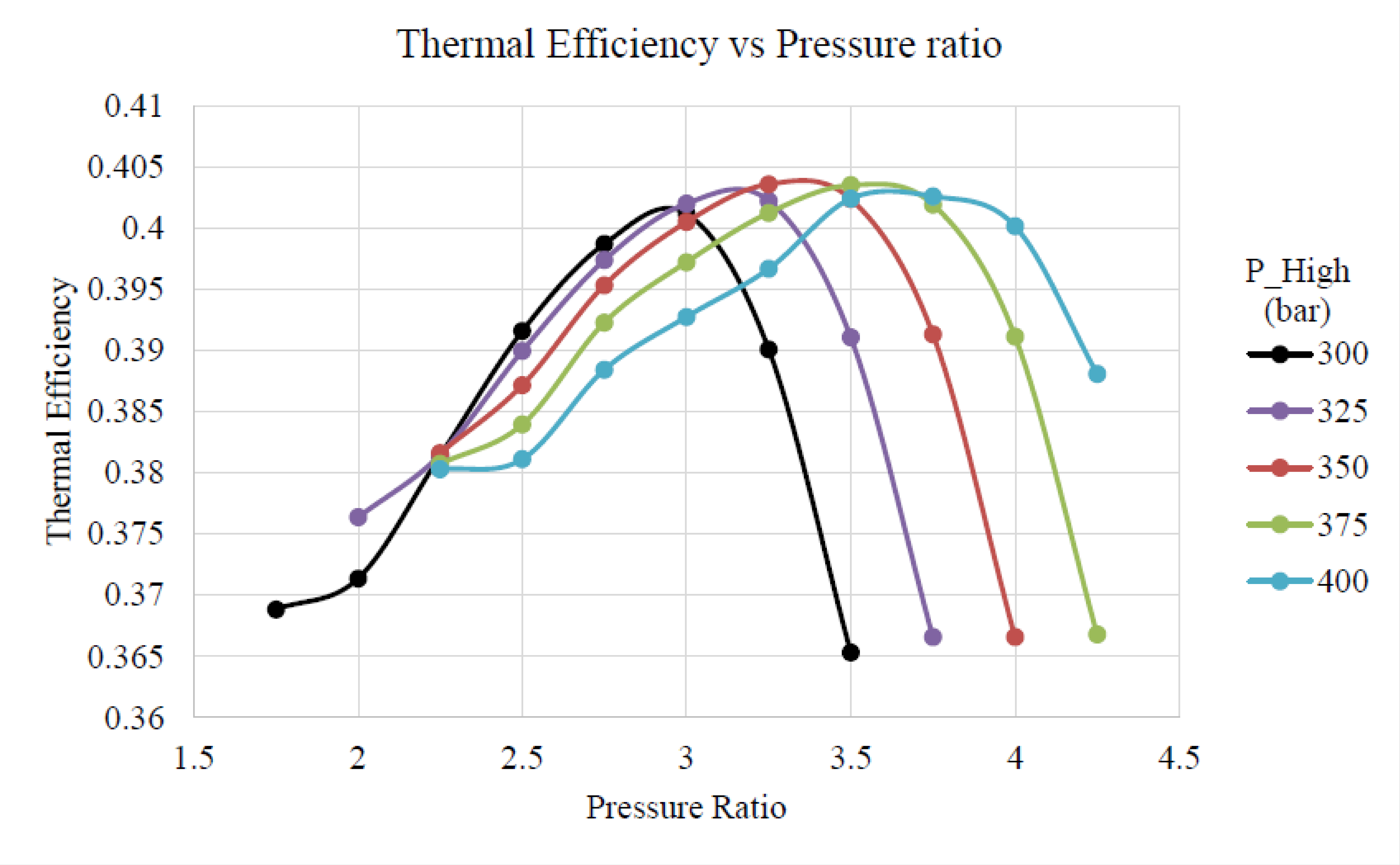
Steam Rankine



- Rankine cycle is condensing cycle, so need to ensure appropriate expansion end line point (target a vapor quality)
- Only real pertinent design variable is maximum cycle pressure – approximately MS pressure
- MS Pressure is determined to obtain 95% quality at LP turbine exhaust (practical limit that enables efficient energy extraction with tolerable damage mechanics)

sCO2

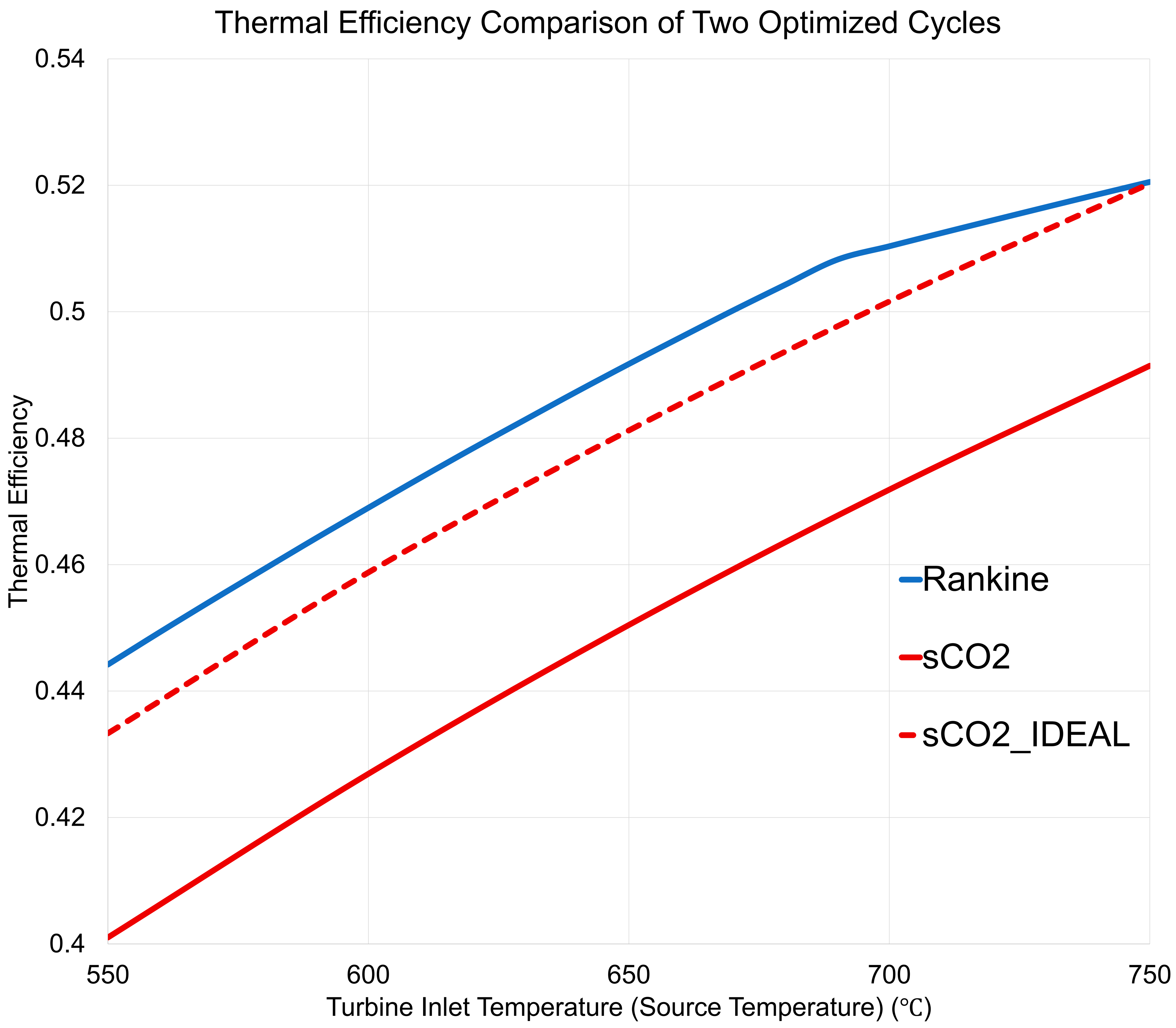




- ♦ Performed as both Monte Carlo and parametric sweeps
- ♦ Pressure ratio near 3 is optimum
- ♦ Very light sensitivity of source temperature on peak PR
- ♦ Optimum SR ~0.7 not sensitive to source temperature

- ♦ High cycle pressure has little sensitivity to max η
- ♦ High cycle pressure changes optimum pressure ratio
- ♦ Suggests that low pressure optimum when it is near critical (cannot condense because sink is at 42C)

Source Temperature Variation



- Note that source temperature assumed to be infinite capacity.
  - This is applicable for primary thermal sources that include nuclear, concentrated solar thermal (CST), geothermal, and primary combustion-based sources (including fossil fuels).
  - It is NOT applicable to most waste heat recovery, where a mass stream of low temperature heat is available.
  - This is the reason waste heat recovery Rankine cycles look a lot different (2 or 3-pressure steam generation without any regenerative feedwater heating)
- Each cycle is operated along optimum found in each individual optimization as source temperature is varied
- All other variables besides source temperature and optimization variables are held constant.
- Sink temperature should be noted to be 42C, which is slightly above CO_2 critical temperature – and cannot condense.

SINK TEMPERATURE VARIATION

Rankine

MS Pressure vs Sink Temperature
MS Pressure (bar)
195
175
155
135
115
95
75
55
5
10
15
20
25
30
35
40
45
50
55
60
65
Sink T (C)

sCO_2

As critical temperature was crossed, optimizations showed significant variation in several cycle parameters. Combination of multiparameter variation and Monte Carlo was used. Table below shows most optimum combination of split ratio, pressure ratio, and low cycle pressure.

Sink temp (℃)	10	20	30	42	50	60
Pressure Ratio	6.65	5.4	3.85	2.95	2.65	2.25
Split Ratio	0.6	0.63	0.66	0.7	0.73	0.77
η_{therm}	0.459	0.446	0.427	0.401	0.384	0.363
POWER (MW)	652	580	495	396	347	288
High cycle pressure (Main compressor out) (bar)	300	300	300	300	300	300
Low cycle pressure (main compressor suction) (bar)	45.1	58.0	77.9	101.7	113.2	133.3
Source temperature (℃)	550	550	550	550	550	550

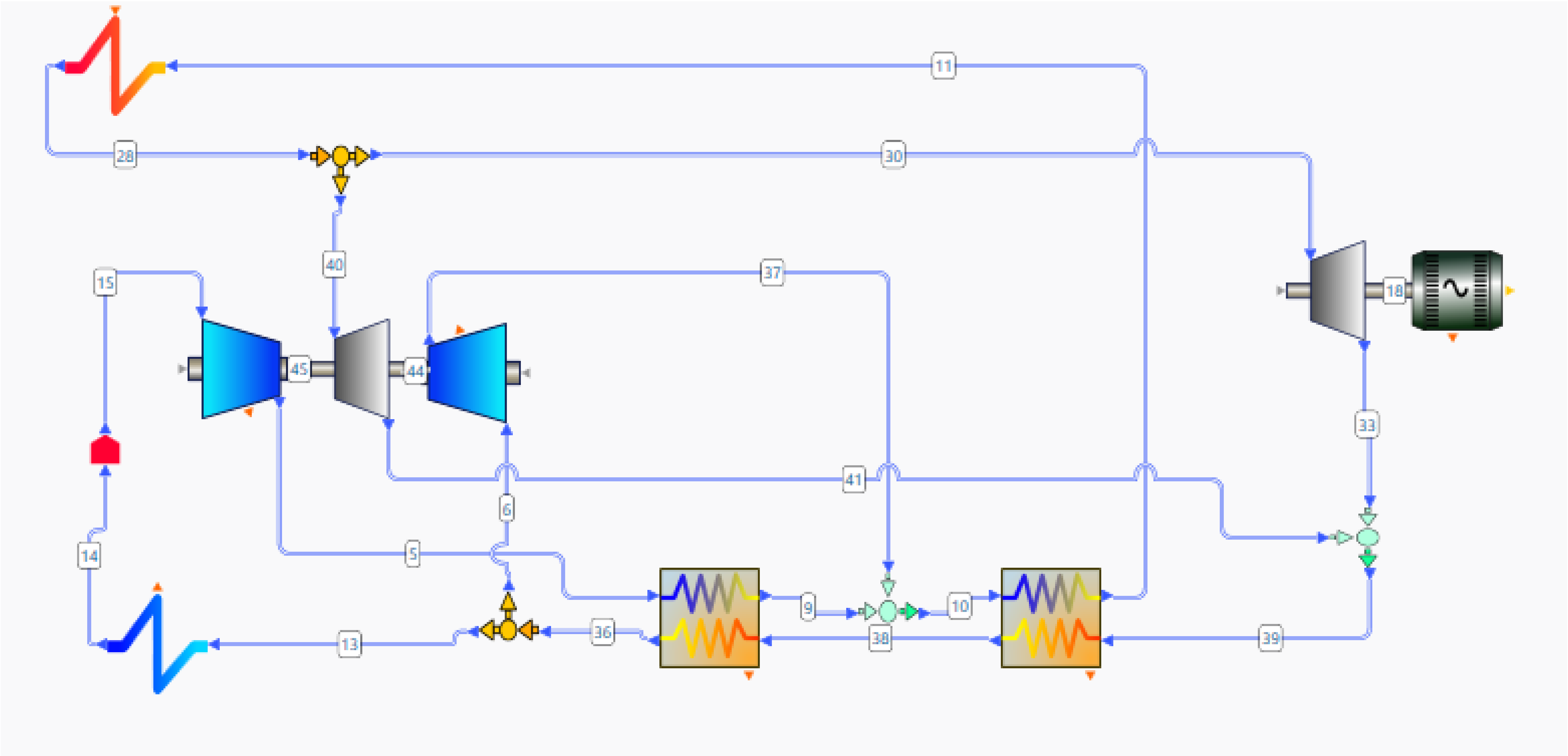
Sink Temperature Variation

Operational Analysis

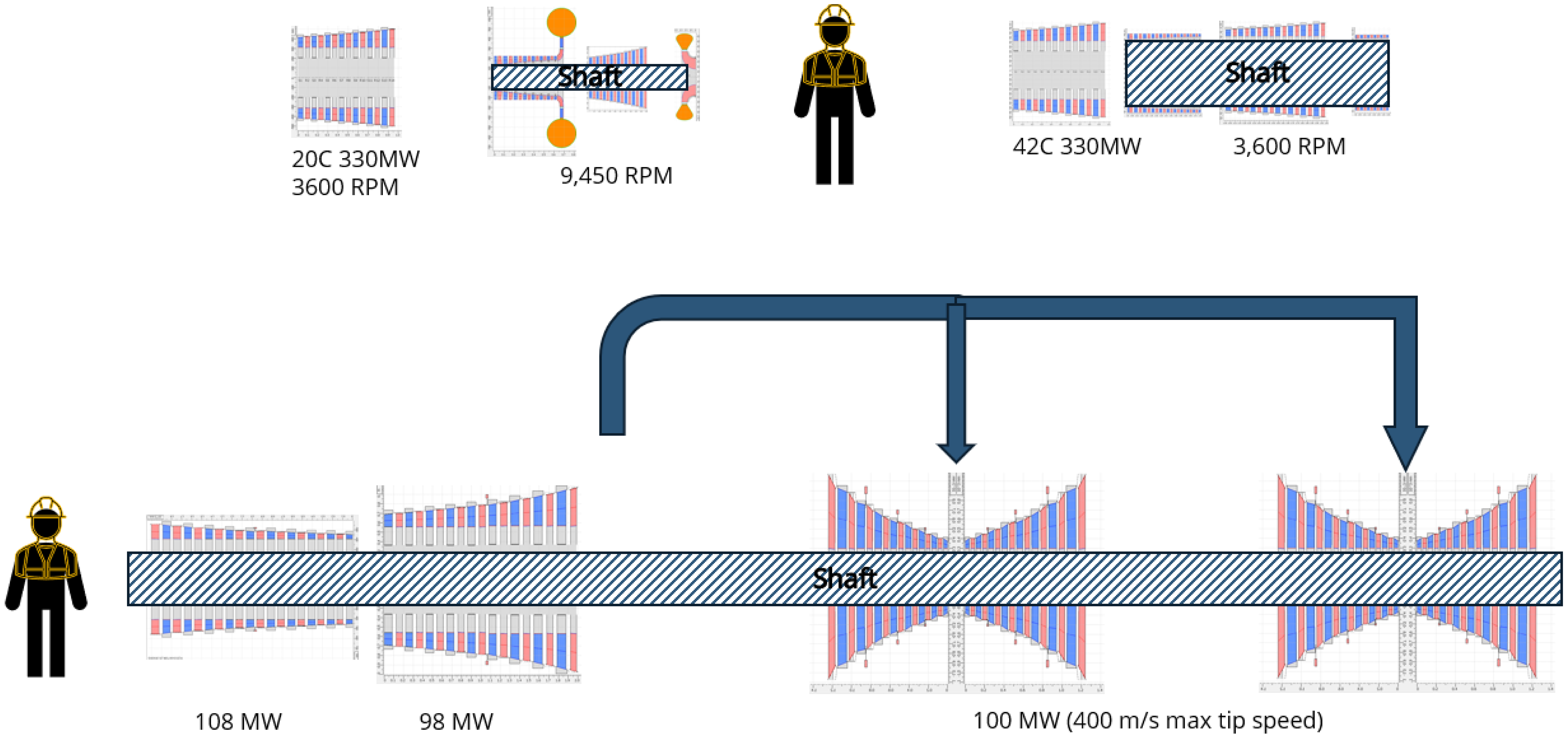
EQUIPMENT SCALE

Controllable Cycle

- In the physical system, it will be a requirement to regulate load (which relates to flow), split ratio, and throttle pressure based on system demands and setpoints.
- Separate power turbine and compressor drive turbine conceptualized to serve the need.
- 20C and 42C sink conditions found to have differentiated solutions.



Turbomachinery



Heat Exchangers

- Heat exchanger parameters for both cycles shown to the right
- Data shown for cycle internal heat exchangers
  - External heat exchangers (source and sink) not explicitly modelled
  - Sink volumetric flow recognized to be drastically different – implying possibility for more compactness for sCO_2
- Larger mismatch in volumetric flows for internal heat exchangers (feedwater heaters) in Rankine cycle – steam to water.

Steam Rankine

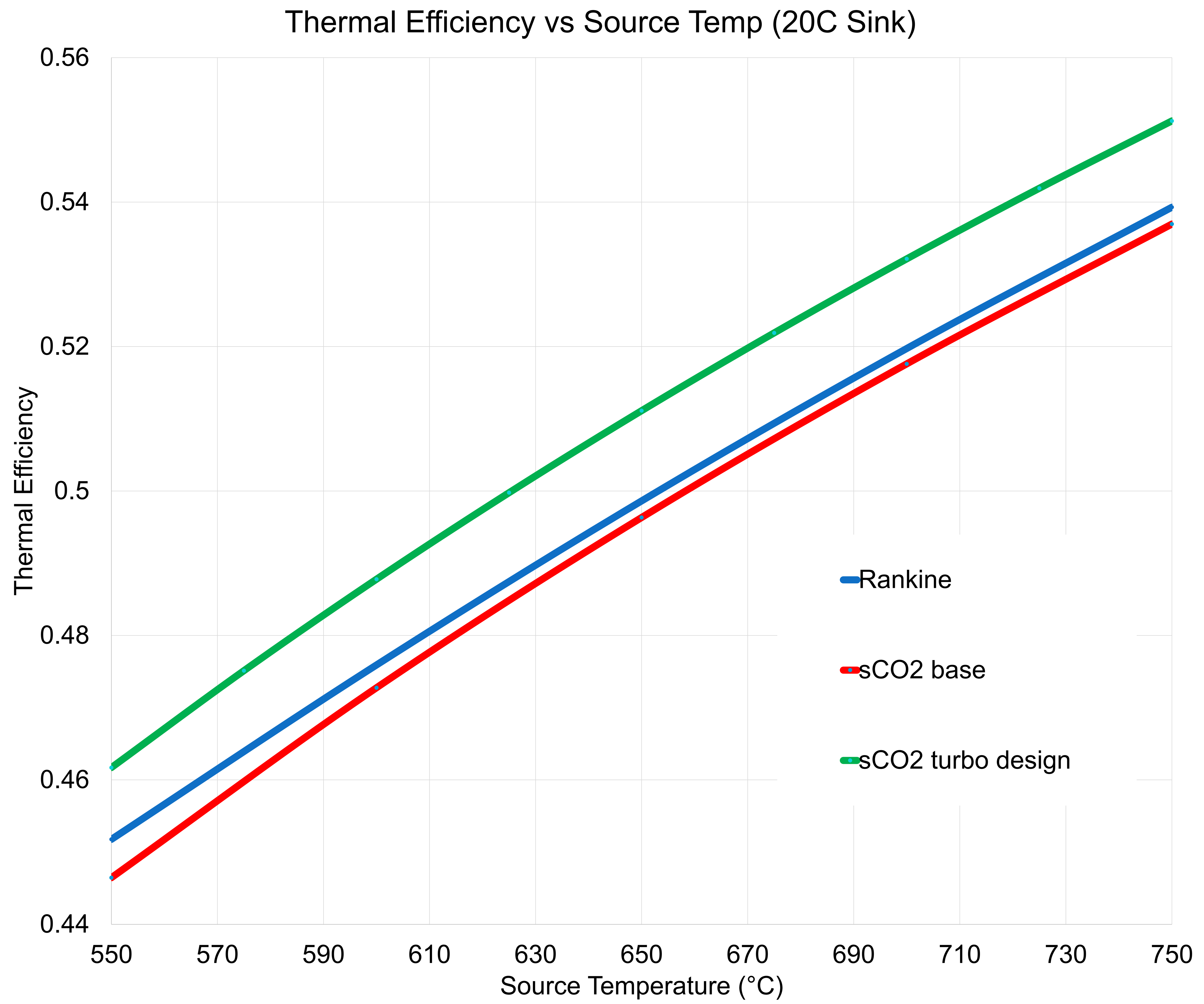
Feedwater Heater	Heat Transfer Rate (MW)	UA - Heat Transfer Conductance (kW/K)
7 (Final heater)	69.3	799
6	53.4	976
5	32.6	302
4 – DA	undefined	undefined
3	19.5	374
2	16.6	682
1	10.8	1,014
Sum Total	202.2	4,147

sCO_2 RCBC

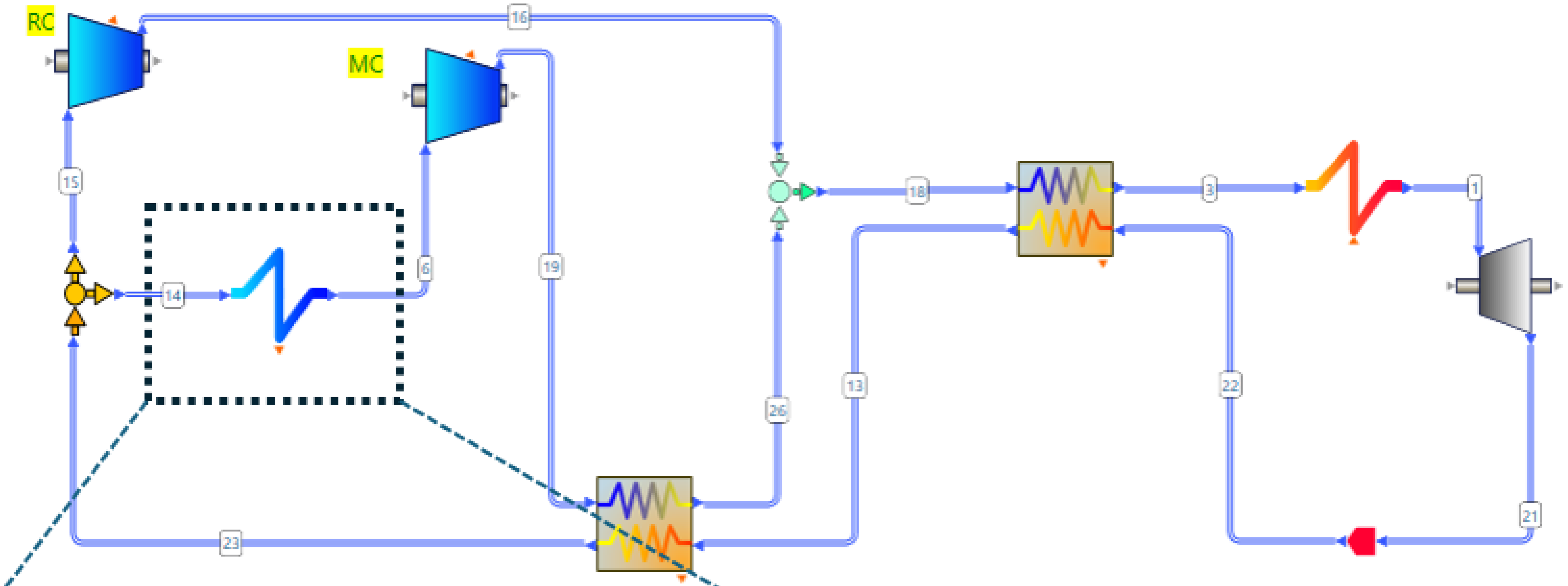
Recuperator	Heat Transfer Rate (MW)	UA – Heat Transfer Conductance (kW/K)
HTR	1,124	69,648
LTR	566	97,788
Sum Total	1,690	167,436

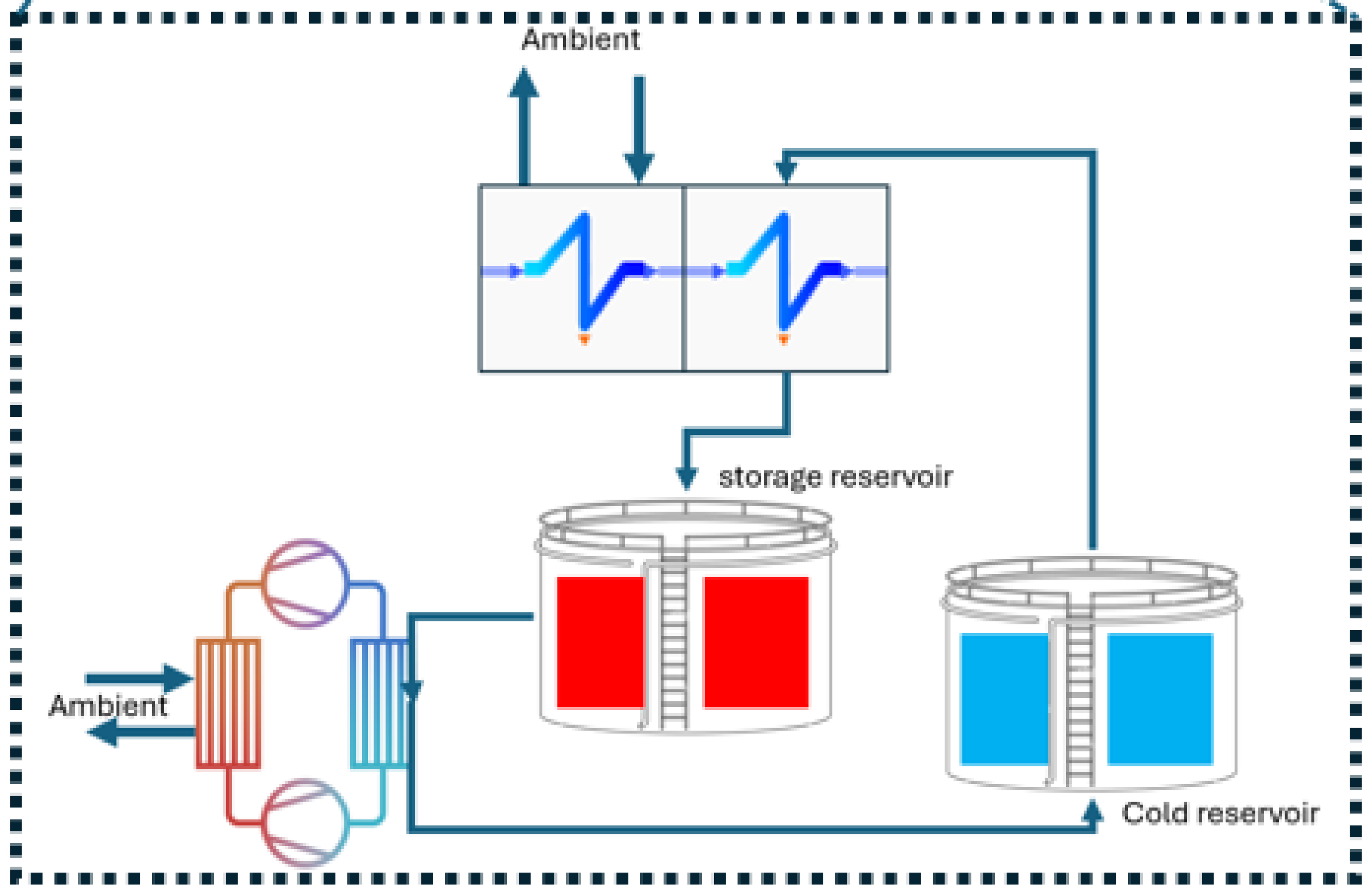
Source Temperature Variation - Revisiting

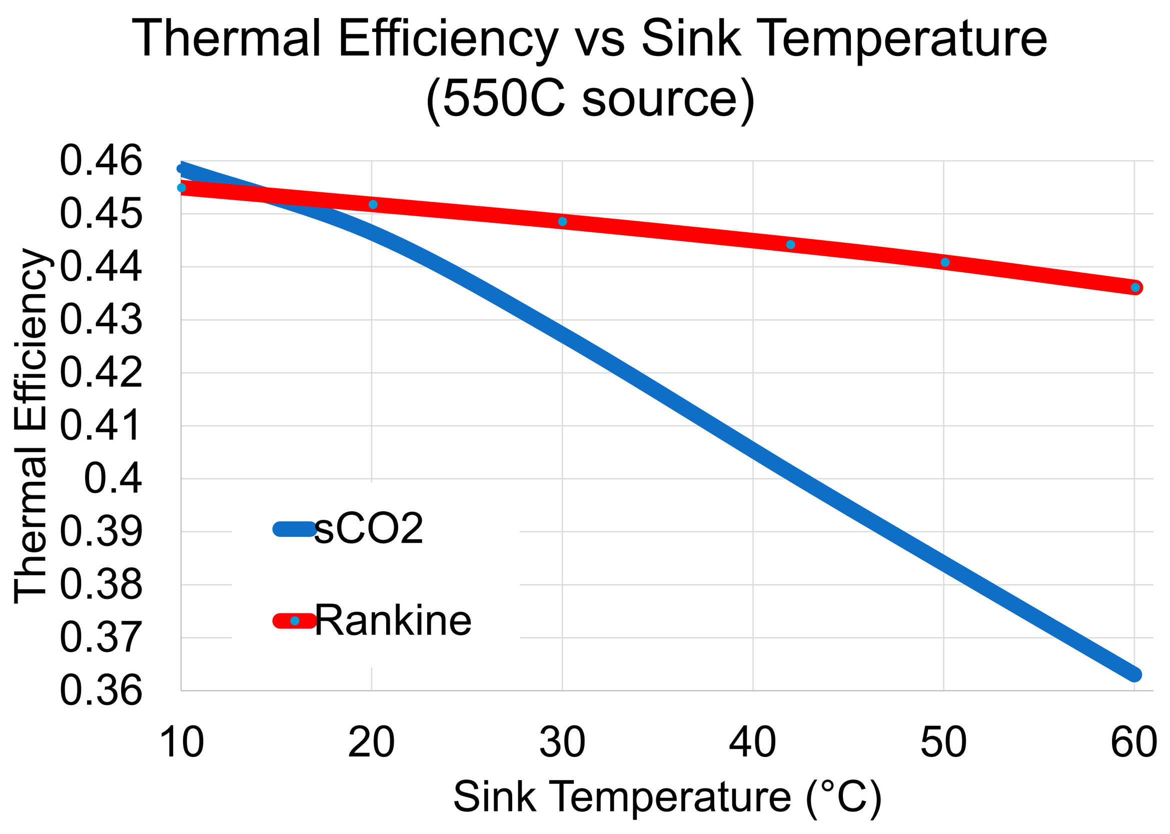
- Perform variation of source temperature using optimized cycles with a sink temperature fixed at 20°C
- One curve shows baseline turbomachinery efficiencies (red)
- Implement turbomachinery conceptually designed specifically for 20°C condensing cycle (green)
  - Main compressor (pump) at 80%, Recompressor at 84%, and Turbines at 90% isentropic efficiencies.
  - Rankine cycle turbomachinery efficiencies unchanged. Throttle pressure curve following optimized schedule
- The condensing sCO_2 cycle shows the benefit gained from reducing compression work

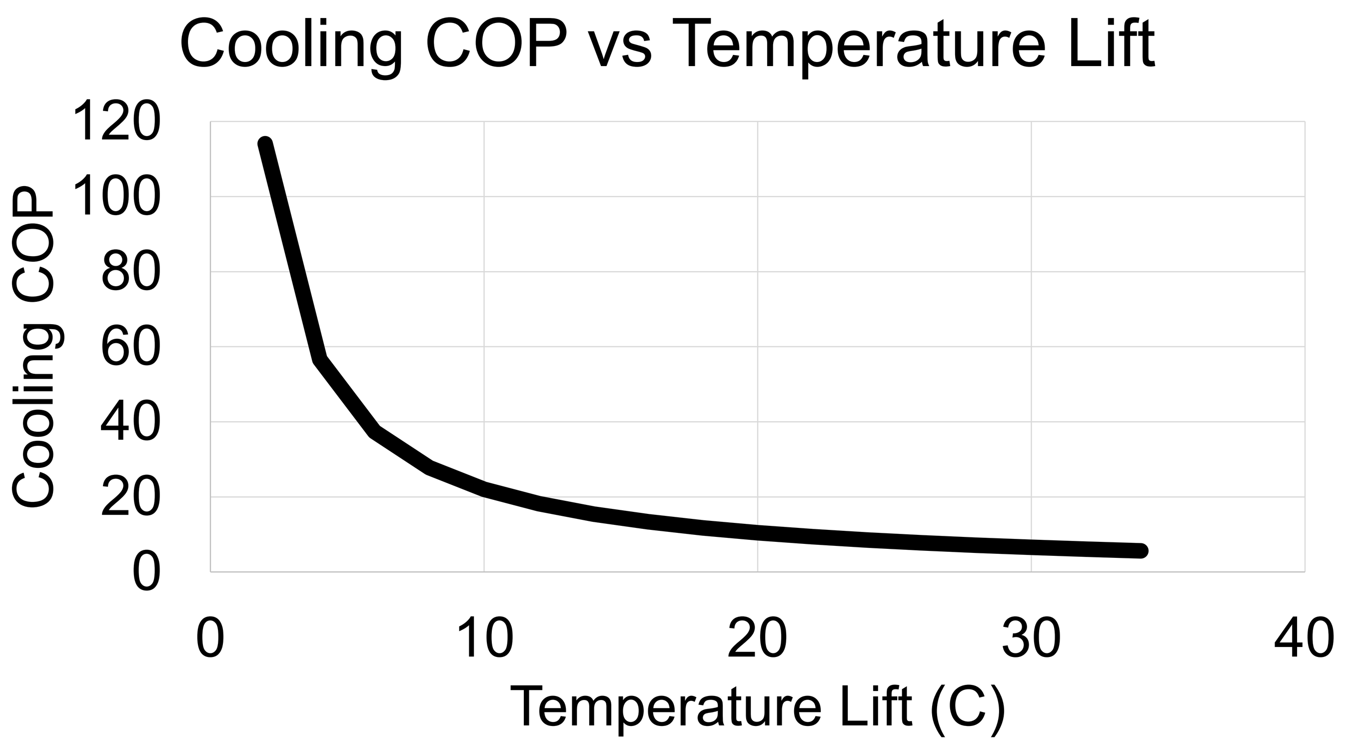


Cold Storage Study









	Scenario 1: July Average	Scenario 2: Yearly Average
Mean Daily Maximum	42℃	30℃
Mean Daily Minimum	30℃	18℃

	Scenario 1	Scenario 2
Relative gain over doing nothing	5%	3%

CONCLUSIONS

Simple Process Comparison

Rankine Process	MS	CRH	HRH	XO	Exh	Cond	FFW
Mass Flow (kg/s)	250	214	193	178	159	178	250
Volume Flow (m^3/s)	5.16	20.7	32.1	126	2695	0.18	0.33
Density (kg/m^3)	48.5	10.4	6.0	1.4	0.06	992	768

RCBC sCO_2 42C Sink	Throttle	Exh	Sink Cooler	Main Comp	Re-Comp
Mass Flow (kg/s)	4000	4000	2800	2800	1200
Volume Flow (m^3/s)	22.4	49	13	4.7	5.7
Density (kg/m^3)	178	81	212	598	212

Conclusions

- For sink conditions typical of hot ambient environments the steam Rankine cycle is more thermodynamically competitive even with higher source temperatures up to 750°C for primary thermal sources – this finding disagrees with commonly held perceptions and some previous research.
- Thermodynamic advantage of sCO_2 is seen where sink conditions can dip below critical temperature, enabling supercritical liquid compression or a condensing cycle.
- Integration of simple cold energy storage shown to provide thermodynamic advantage – a likely implementation for sCO_2 CSP applications.
- There are other tradeoffs in addition to cycle efficiency which may include compactness, volumetric flow matching, and costs. These may favor sCO2 in applications including marine power, nuclear power, aerospace systems, and advanced space power production.

Thank you - Questions

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