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

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ABSTRACT

In this research, thermal cycle numerical simulation models of both steam Rankine and sCO_2 Brayton cycles are developed and used to compare impactful parameters. Care is taken to configure the cycles' layout and assume parameters that are practically achievable in real world applications. This is done through benchmarking based on existing knowledge and experience for the mature steam Rankine cycle, or benchmarking using model fidelity enhancement as with sCO_2 cycles. The Rankine cycle utilizes 7 regenerative feedwater heaters (3 LP, 1 deaerator, and 3 HP), a configuration routinely found in the utility power sector. Extraction pressures along the expansion line were distributed using geometric progression based on main steam and exhaust pressures. The sCO_2 system is primarily configured as a non-reheat recompression cycle. It is shown generally that the "old wisdom" that sCO_2 cycle is fundamentally more efficient does not hold water. We present simulation data showing that practical Rankine cycles are generally more efficient than sCO_2 even at higher source temperature ranges. However, the advantage of sCO_2 power conversion may lie in cases where sink temperature drops below its critical temperature around 31°C.

Both cycles were modeled in AxSTREAM System Simulation with real fluid properties and configured for a nominal power output of approximately 300 MW net electrical production. The comparison was done across turbine inlet temperatures (or source temperatures) ranging from

550°C- 750°C, using consistent boundary conditions and realistic turbomachinery assumptions. A baseline 42°C sink temperature was used as a conservative but realistic assumption. The Rankine cycle power conversion system achieved thermal efficiencies of 44.4% at 550°C, 49.2% at 650°C, and 52.1% at 750°C while the sCO_2 system achieved thermal efficiencies of 40.1% at 550°C, 45.0% at 650°C, and 49.2% at 750°C.

In addition to the thermal efficiency, this study aims to characterize other thermal and fluid features of both power conversion systems to highlight in what applications may maximally benefit from either system. Key among such features are the relative sizes of the turbomachinery and heat exchangers. For each nominally 300MW net power conversion system, a conceptual turbomachinery flow path developed in AxSTREAM Flowpath is presented. Additionally, thermal characteristics of the required heat exchangers for each cycle are compared, thus providing some indication of the relative size and capital cost of the equipment involved.

The results of the sink temperature analysis indicate that there is a strong relationship between the sink temperature and the efficiency of the cycle. The Steam Rankine cycle with a 20C sink achieves an efficiency of 45.17% while the sCO_2 cycle achieves an efficiency of 44.65%, an increase of +1% for Rankine but over +4.5% for sCO_2 compared to nominal 42C sink results. Thus, designing for low sink temperatures may enable efficient power conversion, but only if such temperatures are able to be practically maintained. An interesting application relevant to thermal power conversion systems is CSP, for which typically diurnal temperature cycles of relatively large amplitude are observed.

Using the performance data from the gathered simulations, this work concludes with an analysis of the effects of incorporating cold energy storage over average cycles. This is implemented by running a small heat pump during the night, when even during the summer period of typical places where CSP power is installed (ex. Southwest US) the temperature routinely drops below 20C. This creates cold storage during the hotter daytime to augment the temperature sink and increase power generation efficiency and production during peak periods.

ACRONYMS AND NOTATION

MS	Main Steam: The standard designation of steam line between final steam heating in the steam generator and the highest-pressure turbine inlet in a Rankine cycle power plant.
CRH	Cold Reheat: The standard designation of the steam line between the high pressure turbine exhaust to the reheater in a reheat Rankine cycle power plant.
HRH	Hot Reheat: The standard designation of the steam line between the reheater and the intermediate pressure turbine inlet in a reheat Rankine cycle power plant.
HPT	High Pressure Turbine: Expands main steam
IPT	Intermediate Pressure Turbine: Expands reheated steam
LPT	Low Pressure Turbine: Expands steam that is downstream of the deaerator extraction point.
FWH	Feed Water Heater:
DA	Deaerator: A mixing tank heat exchanger that also enables removal of non-condensable gases from the feedwater stream.
TIT	Turbine Inlet Temperature: Typically the hottest working fluid temperature and representative of the maximum engineering temperature utilization of source heat.

HTR	High Temperature Recuperator: The first recuperator in a recompression Brayton cycle.
LTR	Low Temperature Recuperator: The second recuperator in a recompression Brayton cycle that heats only main compressor discharge.
PR	Pressure Ratio: Specific to this work the pressure ratio refers to the sCO_2 cycle ratio of main compressor discharge to suction.
SR	Split Ratio: Ratio of mass flow rate to main compressor divided by full cycle flow (which is the sum of main compressor plus recompressor flow).
RCBC	Recompression Brayton Cycle
Source	The high temperature heat source
Sink	The low temperature heat reject for the power cycle
CSP	Concentrated Solar Power

1. INTRODUCTION

Supercritical CO_2 (sCO_2) cycles have seen two decades of strong interest since initial consideration during the 1960's with work by Feher (1968) where general supercritical Brayton cycles were conceptualized and evaluated and Angelino (1968) who considered a condensing CO_2 cycle that operates predominantly in the supercritical regime, and 1970's analytical studies by researchers such as Combs (1977) who designed a sCO_2 cycle for a compact and efficient marine power system. At the time, workers produced promising analysis but acknowledged that practical aspects that may inhibit the widespread adoption of CO_2 as the primary working fluid in trans and supercritical Brayton cycles, an inference that history has largely validated. Significant research and development efforts have been invested just in the last decade both experimentally and analytically. Indeed there has been significant progress in maturation of the technology, with a number of demonstration and pilot projects that clearly show the technology is viable and effectively ready for large scale commercial deployment. Nevertheless, a utility scale commercial thermal power plant using an sCO_2 thermal power conversion system has yet to be built.

Several researchers have shown comparisons of sCO_2 power conversion cycles with the common incumbent cycle, the steam Rankine cycle. In most works, the predominant result is that above a certain thermal source temperature (or turbine inlet temperature), approximately around 550°C or so, which varies between different research, that the sCO_2 cycle is superior in terms of its power conversion thermal efficiency. It has also been claimed with significant compelling evidence that a number of other advantages exist for the sCO_2 cycle over the steam Rankine cycle, and chief among these is the compactness of the turbomachinery.

In the present research, we critically examine and compare supercritical CO_2 (sCO_2) cycles with the incumbent steam Rankine cycles using practical limitations and boundary conditions to examine where they can be optimally deployed in commercial applications. Focus in the present work will be on applications of primary heat sources, especially concentrated solar power, and specifically excluding waste heat recovery applications.

2. BASELINE POWER CONVERSION CYCLES MODEL DESCRIPTION

Steam Rankine Cycle

A model of a commercially-typical utility scale Rankine thermal power conversion cycle with seven stages of regenerative feedwater heating was modeled in AxSTREAM System Simulation. The layout of the cycle is shown in Figure 1.

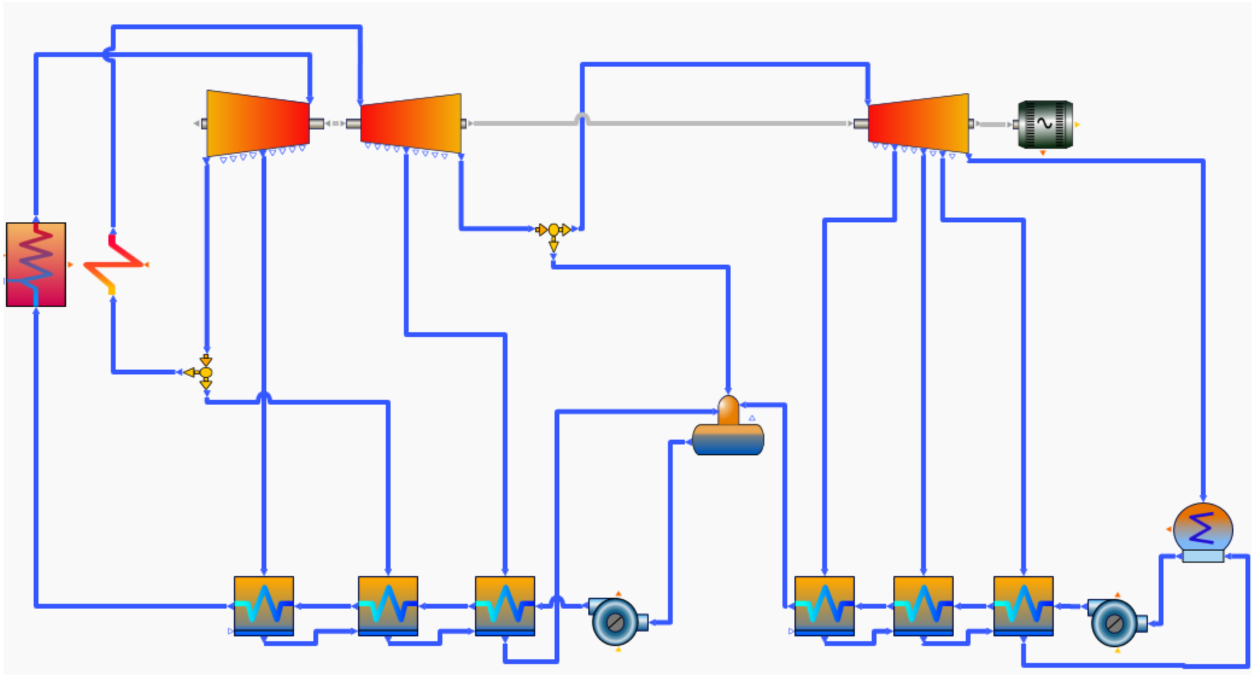

Figure 1: Regenerative Steam Rankine Power Cycle Model Diagram

The cycle includes a heat input for the main steam generator to get the MS line to temperature and includes a reheater after the high pressure turbine (HPT) section for a single reheat cycle with a common main steam and hot reheat temperature. The steam expands in 3 sections: HPT, IPT, and LPT. All those sections have extractions that are fed into high pressure (HP) and low-pressure (LP) feedwater heaters (FWH), as is typical for practical utility Rankine cycle thermal plants. There are 3 LPFWH, 3HPFWH and a deaerator that mixes an IPT extraction with the feedwater stream. The steam condenses and rejects heat in the steam surface condenser at a temperature of 42C, conceptually enabling the heat sink rejection to be at 40C, a small temperature difference to drive the heat transfer. The boundary conditions are shown in Table 1. While the cycle configuration may seem complex, given the minimum of seven extractions and three turbine sections, such a configuration represents a minimum level of complexity that is typically seen in commercial thermal power stations since the 1970's. Modern installations (mostly using coal as the primary energy source but there are other applications) have developed ultra supercritical pressures, double reheat, and as many as 10 feedwater heaters. This is supported by the author's practical experience as well as a number of sources including Cotton (1998) and Xu (2021). The important outcome for the reader to consider is that the analyzed steam-Rankine cycle represents not the state of the art and technology but a practical and comparatively low capital cost implementation as comparison to any sCO_2 technology.

Table 1: 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	Psat(42C) = 0.083 bar
Exhaust Quality	95%
Condensate pump	12 bar
Boiler Pressure Drop	8 bar
Reheater Pressure Drop	2 bar

The mass flow rate of the extractions are determined by the performance parameters of the feedwater heaters. In this case the TTD, temperature difference between the saturation temperature of the extracted steam and feedwater outlet, was chosen as the performance parameter in common with industry practice to specify nominal cycle performance based on such parameters. The FWH parameters and resulting extraction flows are shown in the table below.

Table 2: FWH Performance inputs and nominal extraction

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%)

Table 2: FWH parameters

The low pressure (which is the LP turbine exhaust pressure) is set to 0.083 bar as it is the saturation pressure of 42C. The high pressure, also called Main Steam pressure or HPT inlet pressure, was then set to achieve an exhaust total quality of 95%. At the baseline conditions, the Main team pressure set to result in the given constraints is 124.8 bar as shown in Table 1. A formula was used to replicate realistic turbine expansion steps.

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

Where x represents the pressure ratio between extractions, P_{MS} is the main steam pressure, $P_{exhaust}$ is the exhaust pressure, and *i* being the number of intervals in the expansion between subsequent extractions - 8 in this case.

Table 3: Extraction conditions

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

95% quality was chosen as a realistic number to best replicate the practical capabilities of turbomachinery to extract energy out of a fluid before the effects of liquid particles causing severe erosion damage on last stage blades over the life of a machine.

sCO2 Brayton Cycle

The layout of the recompression Brayton sCO_2 cycle (RCBC) is shown in the Figure 2.

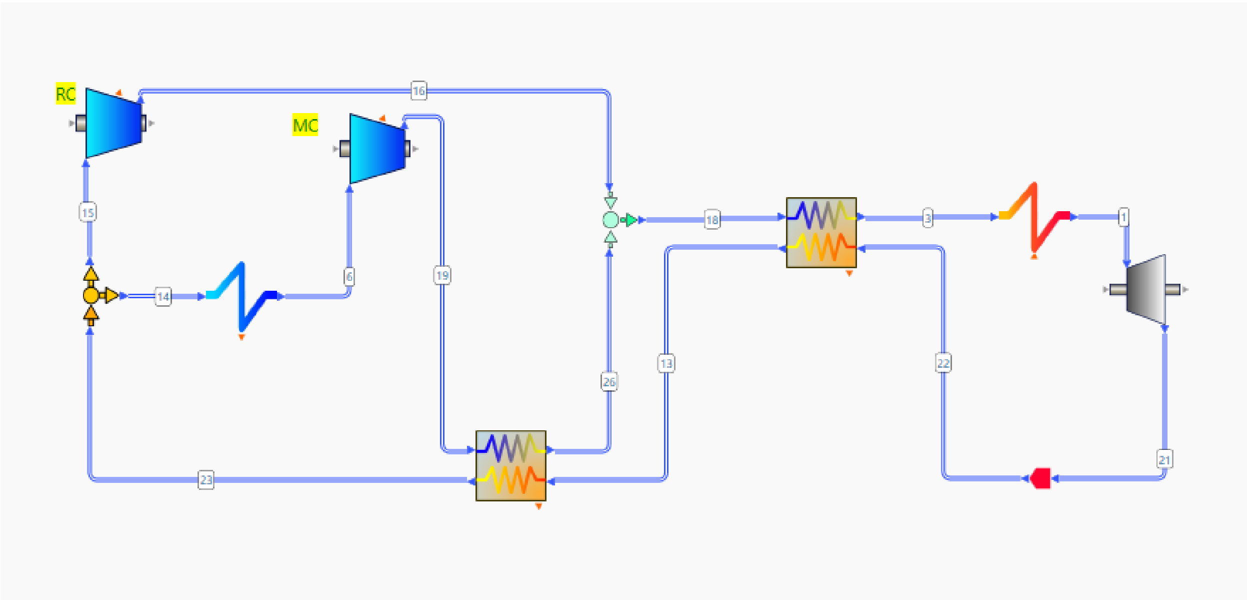

Figure 2: Recompression sCO_2 Thermal Power Conversion Cycle Model Diagram

The cycle includes a gas heater as the primary heat source for the cycle. The CO_2 expands in one turbine section. There are two Recuperators with one operating at high temperature with full stream and the other operating at a low temperature with partial stream. The CO_2 splits and most of the flow goes to the main compressor, MC, where it is cooled prior and rejects heat at a constant T of 42C. The rest of the CO_2 goes to the re-compressor and thus bypassing the cooler/heat rejection. The boundary conditions are shown in Table 4.

Table 4: Boundary Conditions for the Baseline sCO_2 Cycle

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

The Pressure drops used were suggested by Dostal (2004) and are shown in the table below.

Table 5: sCO2 Cycle Equipment Pressure Losses

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

As shown in Table 5, most of the pressure drop take place in the heater and the Hot fluid branch of the LT recuperator. Like the Rankine cycle a parameter was chosen for the Recuperators and in this instance a pinch temperature was chosen. Pinch temperature is defined as the minimum difference between either the hot inlet and cold outlet or hot outlet and cold inlet. The pinch for both the Recuperators was set at 5C, meaning if the minimum temperature difference got to 5C then the heat transfer stops. This is known as a pinch point, and it limits the effectiveness of the recuperators. It is pertinent to also mention that the RCBC was chosen as it has been identified as a sCO_2 cycle that achieves a balance of equipment simplicity and high thermal efficiency.

Turbomachinery Efficiencies

Deep consideration was given to the isentropic efficiencies of turbomachinery for both cycles. Typical turbomachinery efficiencies for steam Rankine cycle plants are unmysterious owing to the vast industry experience gained from more than a century of deployment. The chosen values for the present study are justified by Cotton (1998) and verified by concept design and industry practice. For the turbine and compressor efficiencies of sCO_2, significant quantity of literature assumes efficiencies greater than 90% or even 95% which at best not verified and more likely not practically achievable. More realistic values were chosen based on initial flow path design studies. Of course, there is very little real machine operating experience for sCO_2 turbomachinery. However, we must take into account that sCO_2 turbomachinery is inherently more compact compared to the steam Rankine cycle. This directly implies effects of clearances, tolerances, and seals, which do not scale directly, are exacerbated and therefore that state of the art (SOA) values from steam turbines is likely an upper limit with respect to isentropic efficiency. The values of isentropic efficiencies are supported by the further work described near the end of this paper in developing concept flow paths for all turbomachinery. Due to the fact that the sCO_2 turbomachinery efficiencies are less certain, a practically achievable 85% efficiency and 90% efficiency as an "optimistic" variant is used in certain studies. After more thorough research by the authors, we developed a specific turbomachinery designs for a condensation sCO_2 cycle. The latter is based upon results that will be presented in this paper that show the most optimistic and competitive sCO_2 cycle in one with sink temperature slightly below the critical temperature. These performance predictions agree reasonably well with more thorough concept design work performed by Rahul et. al. (2016). The values of efficiency to be used for the modeling are shown in the Table 6.

Table 6: Turbomachinery Isentropic Efficiencies

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

The comparison was done by comparing thermal efficiencies at each turbine inlet temperature based on previously discussed optimizations. To ensure no doubt an optimistic case was also run for sCO_2 where efficiencies of all turbomachines were bumped up from 85% to 90%, as well as the more specific case of designed turbomachines for the 20°C sink temperature.

A final note about turbomachinery efficiency assumptions needs to be stated with respect to the real long-term efficiency of practical plant equipment. While the turbomachinery efficiencies of steam Rankine plants are based upon measured experience and represent "worn-in" performance long after the new and clean period has long elapsed Cotton (1998), there is insufficient such experience on sCO_2 machinery. Most such machines are one-of-a-kind or experimental. We are not aware of data that suggests how much the performance of typical machinery degrade during what are projected to be normal overhaul periods in Nth of a kind plants and power system.

Source Temperature Entitlement

In the present work, it is necessary to specify and acknowledge that the nature of the source heat is treated essentially as an infinite-capacity hot reservoir. This is a reasonable (which is to say, un-constraining) assumption in some applications but unreasonable in others. It is reasonable to assume an infinite capacity in situations where the heating of the working fluid (either water-stream or CO_2 in the present study) is done either without a secondary hot fluid, or where that secondary hot fluid does not become a waste stream immediately after heating the working fluid. Such secondary fluid does not become a waste stream in cases where (1) the ultimate source of heat is itself not a fluid stream and the secondary fluid is simply a delivery mechanism, as in the case of a molten salt or an oil used in CSP or nuclear applications or others, or (2) where the still-hot secondary fluid transfers useful heat energy elsewhere. In the second situation of the previous statement, the secondary fluid need not be subject to waste heat recovery. An important case of this is a combustion-based heat source. In this situation, the secondary fluid heat source is typically the stream of products of combustion, and after being used to heat the working fluid may still be significantly above the sink temperature. This situation poses no problem as the combustion product stream then can exchange heat with the aspirant air stream to be used for combustion, which is inherently well-matched in stream heat capacity rate to the combustion products. This is typical practice for any boiler-steam generator, and would be equally well implemented in an sCO_2 cycle.

It is not reasonable to assume an infinite capacity heat source primarily when that source is a waste heat stream, which is typically associated with a secondary fluid flow of a temperature relatively close to the TIT of the cycle. In such situations, the temperature that the working fluid enters the source heat exchanger is limited, and this phenomena is already well known and understood in traditional combined cycle power plants where the bottoming cycle, which is most typically steam-Rankine does not employ feedwater heating but rather, in contrast, uses multiple pressure steam generation and induction for the specific reason of extracting the maximum available waste heat and bring the stream to near sink temperature conditions. The effect is explored theoretically and practically by Hofer and Gulen (2006).

In the present work, we specifically exclude waste heat recovery from focus or consideration, and instead focus specifically on primary heat sources that focus on nuclear, combustion, and CSP, among others. Therefore, we henceforth do not worry about the otherwise important consideration to the temperature entitlement of the heat source.

3. SOURCE VARIABILITY OPTIMIZATION

Steam Rankine Cycle

The effect of Source and Sink temperatures on cycle performance was of interest for both cycles. For the Steam Rankine, an iterative study was done to find which high pressures will result in a quality of 95% for HP turbine and IP turbine inlet temperatures in the range of 550-750 in steps of 10C. The high pressure was capped at 350 bar to best represent modern system capabilities in balancing performance with containment of high temperature, high pressure fluid in source heat exchangers and turbomachinery casings. The results of the study are shown in Figure 3.

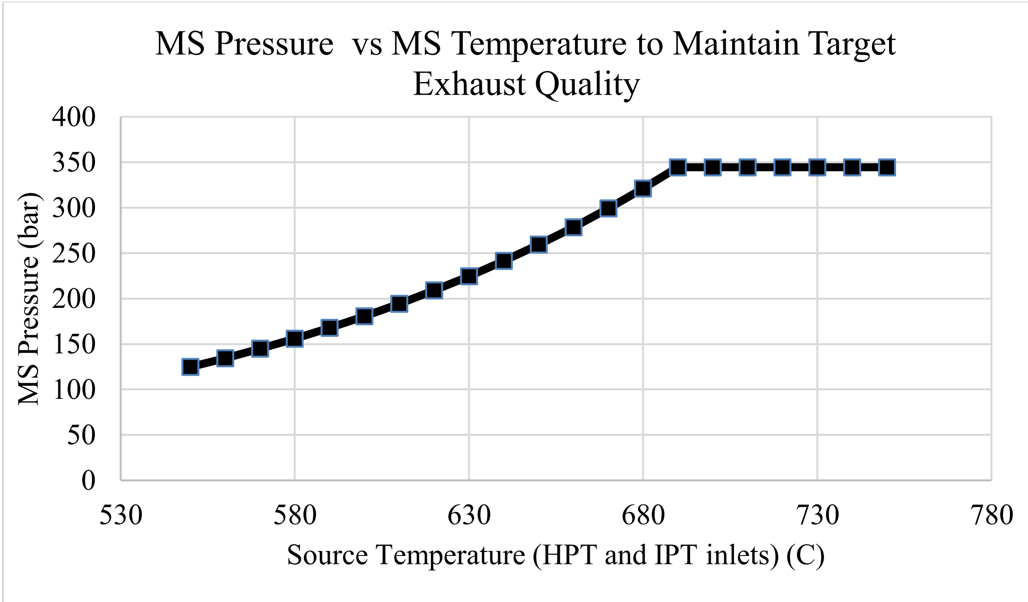

Figure 3: Steam Rankine Cycle Main Steam HPT Inlet Pressure Curve for variation in temperature to maintain 95% LP exhaust quality

The next step was to sweep through all these source temperatures while maintaining a constant baseline sink temperature of 42C. The results will be presented with the sCO_2 cycle.

sCO2 Cycle

For the sCO2 a similar study was done with a couple variations. A combination of monte-carlo and parameter-sweep method simulation models were executed to optimize this cycle. Since there is no implied phase change, there are three fundamental cycle parameters to select: (i) split ratio, (ii) highest pressure, and (iii) lowest pressure. If we fix the high pressure as the outlet of the main compressor, these can be reduced to pressure ratio and split ratio. The first order of business was to determine the optimal split ratio for every pressure ratio. Since the high pressure was set to 300 bar, a map was run that varied the pressure ratio of the main compressor as the major variable and split ratio as the minor variable. The setup of the map is shown in Table 7.

Table 7: sCO_2 Cycle Parameter Optimization Input Variables

Input Variable	Domain	Step
Pressure Ratio (major)	1.75-3.75	0.1
Split Ratio (minor)	0.55-0.95	0.01

The output for this numerical experiment was overall cycle thermal efficiency. This map was run for a turbine inlet temperature of 550-750 in steps of 50C. The results of the simulations were processed to find the split ratio for every combination of pressure ratio and source temperature that maximized thermal cycle efficiency. In this way, a set of curves could be produced such that each curve is of a constant source temperature and shows the optimum recompression fraction as a function of pressure ratio. Furthermore, it was shown from this analysis that the turbine inlet temperature did not impact the optimal split ratio. The data for every temperature inlet was reduced to show the most efficient split ratio and pressure ratio pair. This is shown in Figure 4.

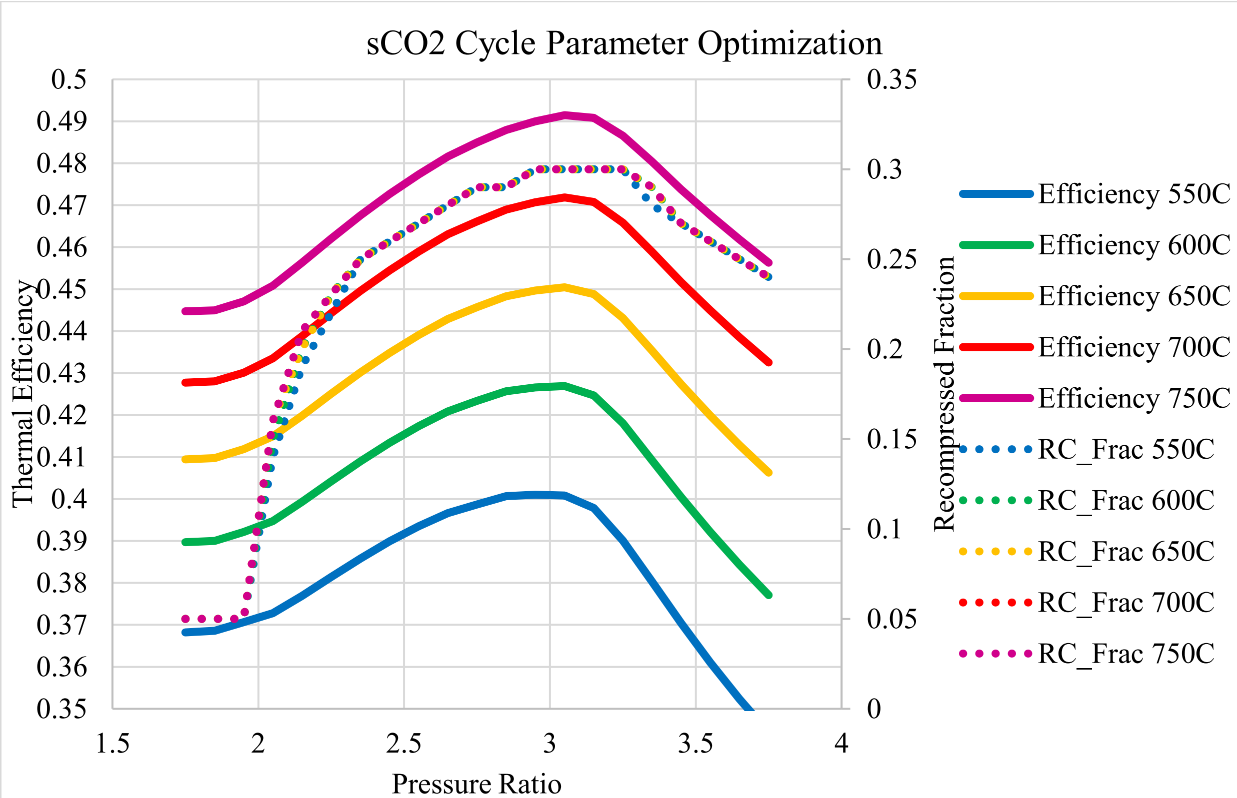

Figure 4: Parameter Optimization for sCO_2 Cycle at 550C source temperature

The next study was done with a variation of the high cycle pressure. It was discovered the high pressure did not impact the peak efficiency for the cycle but rather shifted where it peaked with respect to pressure ratio. This is shown in Figure 5.

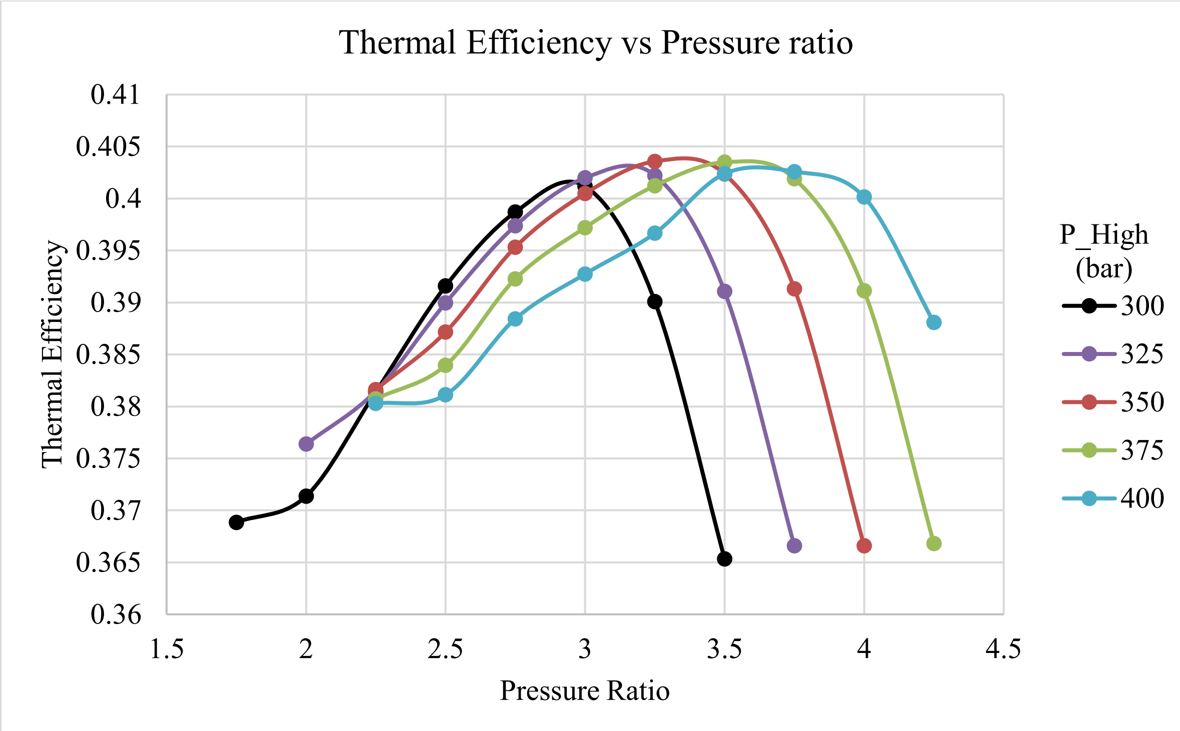

Figure 5: Impact on sCO_2 RCBC thermal efficiency from variation of maximum cycle pressure

Results of Source Temperature Variability

We arrive at a key result in our analysis – the comparison of optimized cycle thermal energy conversion efficiencies between the incumbent steam Rankine cycle and the supercritical CO_2 cycle. This is shown in Figure 6, where as discussed previously, two versions of the recompression Brayton sCO_2 cycle are simulated. One with practically achievable turbomachinery isentropic efficiencies of 85%, and another with turbomachinery efficiencies set to 90%, labeled the "ideal" case. It should be noted that for each source temperature, the cycles were configured to their most efficient practical state. In the case of the steam Rankine cycle, that involves adjusting the main steam pressure to obtain 95% exhaust quality from the expansion line. In the case of the s_{CO2} cycle, that involves adjusting to the most optimum pressure ratio and split ratio as found in the previous section.

It is clearly seen here that the present analysis is showing that even under ideal turbomachinery performance assumptions, the steam Rankine cycle is more efficient in converting thermal to mechanical power. It should be pointed out that this result appears to contradict the conclusions of previous researchers, as shown in Figure 7. The root source of the discrepancy has not been determined. A reasonable conjecture may be that some fundamental equipment performances for the sCO_2 cycle (namely turbomachinery) may have been estimated higher, while at the same time it is possible the performance of the steam Rankine cycles may have been underestimated, or that the configuration was optimized for an application that required lower efficiency. For example, a Rankine cycle operating to recover waste heat energy from a high flow gas stream must intentionally avoid regenerative feedwater heating to maximize heat recovery, thereby sacrificing cycle efficiency (in the technical sense) to enabling more heat to be recovered to lower the temperature of the waste stream.

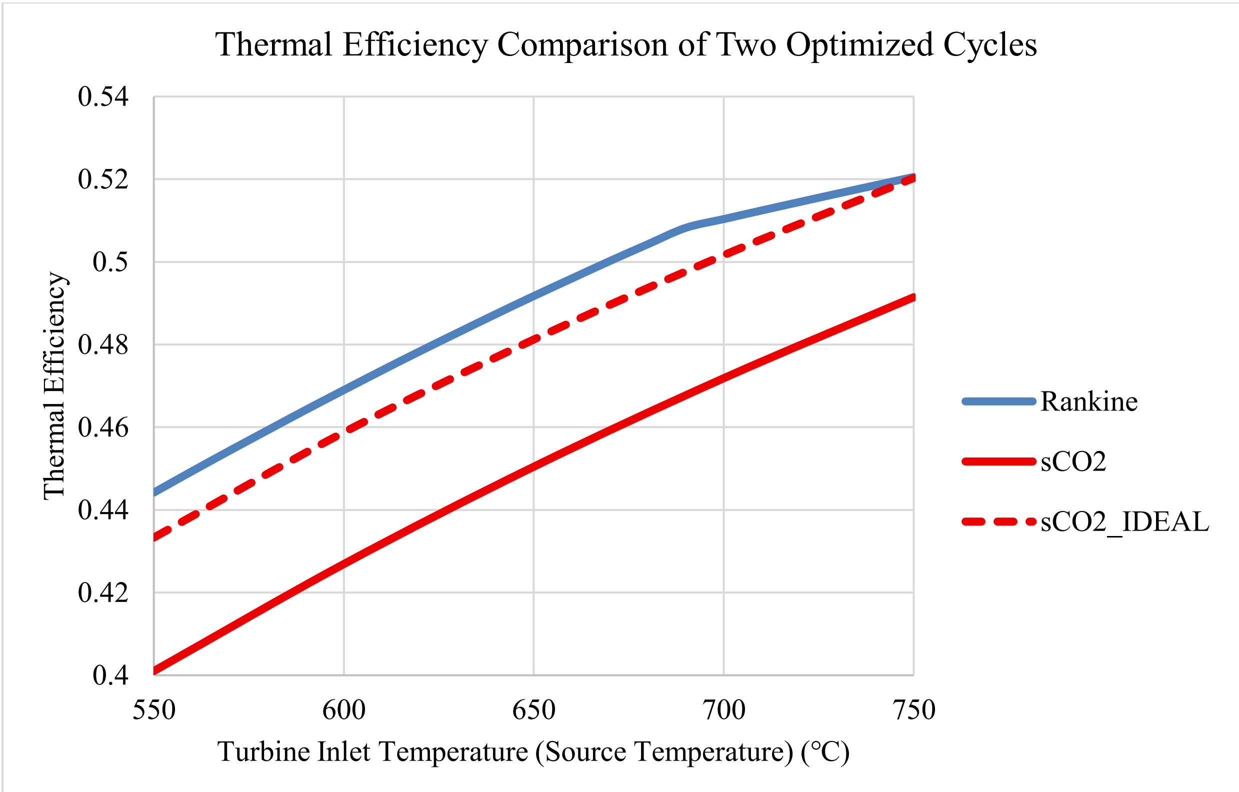

Figure 6: Comparison of thermal efficiency at various source temperatures for the steam Rankine and sCO_2 Brayton cycles. Each cycle is simulated at its optimum cycle parameters. The Ideal curve represents the stretch target of achieving 90% isentropic turbomachinery efficiencies.

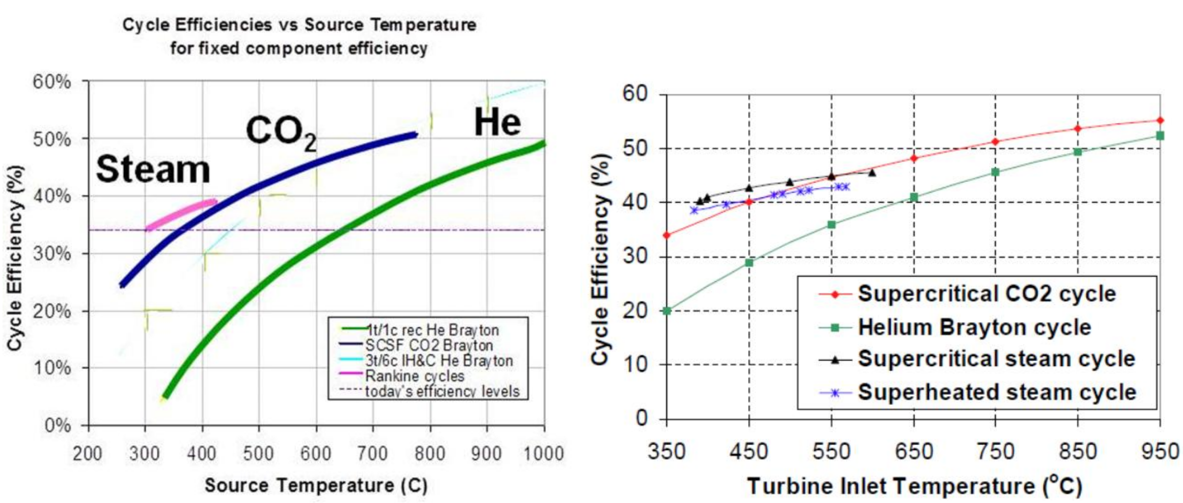

Figure 7: Previous research in cycle efficiency comparison (Flemming, et.al. (2012) and Dostal (2004)

Regardless the root cause, our present results do necessarily lead us to conclude that sCO_2 cycles are not best suitable for power generation. Rather, their application may favor particular niche situations and we furthermore explore their performance.

4. SINK VARIABILITY OPTIMIZATION

After seeing the results of the source study with a constant baseline sink temperature, the next step was to vary the sink temperature while holding the source temperature constant.

Steam Rankine

For the Rankine Cycle once again the condenser quality was held constant at 95% throughout all sink temperatures. An iterative process was completed to determine the MS pressure as a function of sink temperature in the same way that it was done previously for variability of source temperature. This time, the source temperature was maintained at the baseline 550℃ while the sink temperature was adjusted. The resulting curve is shown Figure 8.

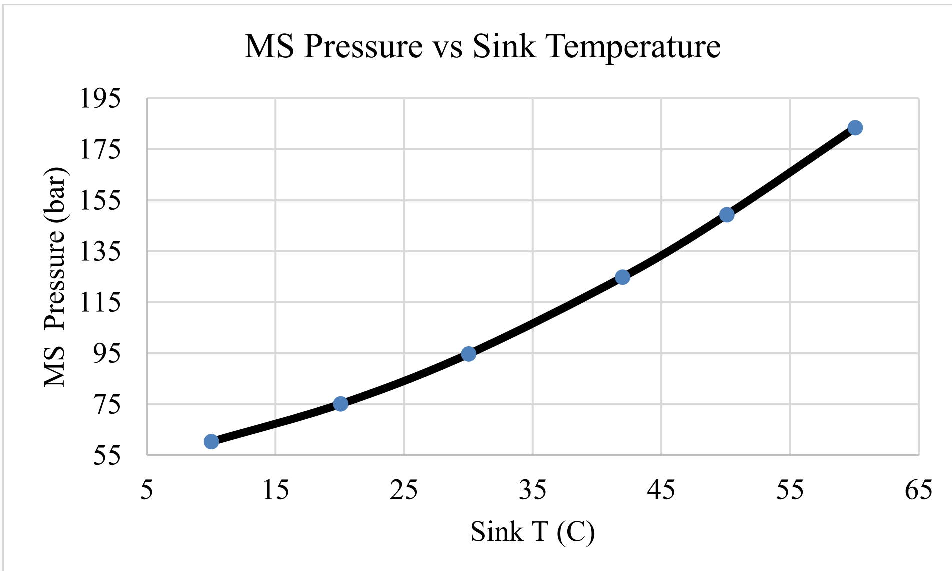

Figure 8: Practical MS pressure in application of sink temperature variation

The thermal efficiency curve was created for the sink temperature variation and will be compared with an optimized sCO_2 cycle.

sCO_2 Cycle

The sink optimization was a little more complex for sCO_2 as the split ratios and pressure ratios had to be reoptimized. For the source variation the optimal split and pressure ratios did not change but for the Sink they changed drastically. For each sink temperature in the set {10℃,20℃,30℃,42℃,50℃,60℃}, a multiparameter sweep was used for the optimization. The baseline cycle was adjusted for the correct sink temperature then the split ratio and pressure ratio were adjusted as the two variable input parameters. The source temperature was held constant at the baseline of 550C. As previously shown, source temperature does not significantly impact optimal pressure and split ratios. Pressure ratio was used as the major variable and was ran for a range of 1.75-7.05 in steps of 0.1, and for each pressure ratio the split ratio was swept from 0.55-0.95 in steps of 0.01. This ensured that the cycle was assessed for each possible combination of the independent design variables. Once the data was gathered for each sink temperature it was consolidated firstly based on pressure ratio. The data was reduced by finding the recompression ratio for each constant pressure ratio which maximized the thermal efficiency for that subset and continuing to sort by increasing pressure ratio as the first criteria then decreasing thermal efficiency as the second criteria. This ensured we had the most efficient split ratio for every pressure ratio. The results for the 50C sink parametric sweep is shown in Figure 9. The relation $RC = 1 - SR$ applies, where RC is recompression ratio, or the fraction of turbine flow bypassing the gas cooler, and SR the split ratio, or the fraction of turbine flow that goes through the gas cooler and main compressor.

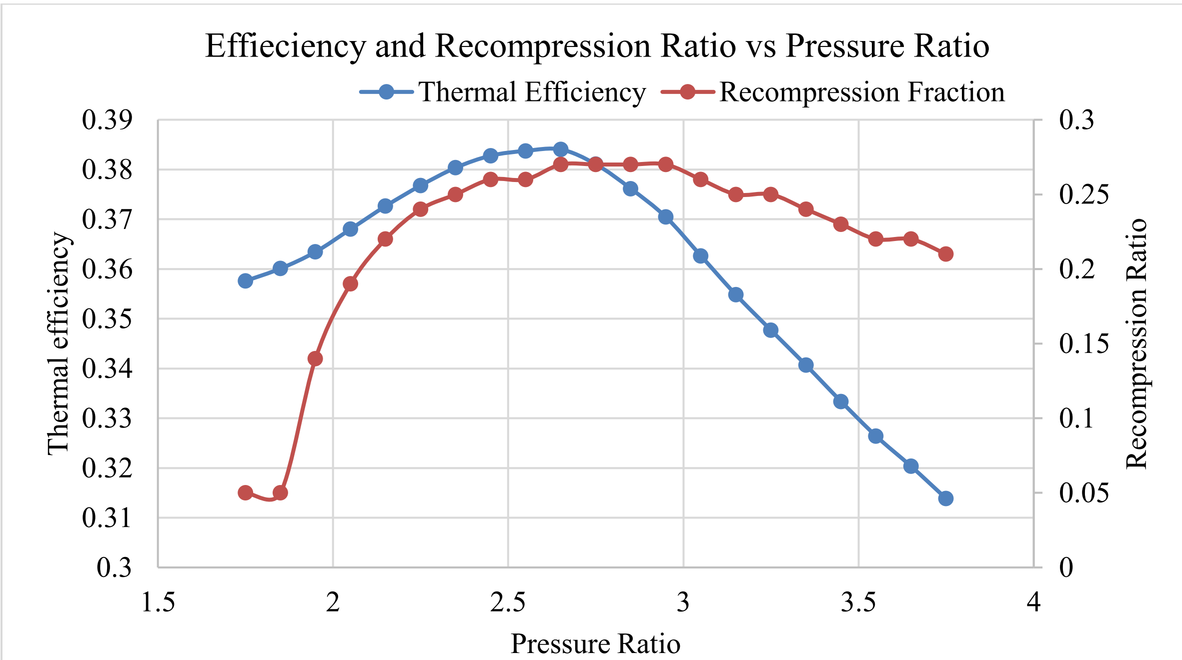

Figure 9: Variable sink sCO_2 cycle parameter optimization

Unlike the case with variation in source temperature, the parametric study of sink temperature had significant variation in the optimum split ratio and pressure ratio. The effect was most pronounced at sink temperatures below the critical temperature of carbon dioxide. This is consistent with an effectively condensing sCO_2 cycle as described by Wright, et. al. (2011). Once this filtered optimization was completed for every sink temperature, the data was consolidated once again for simply the most efficient points for every sink temperature. This resulted in the data presented in Table 8.

Table 8: Optimized parameter conditions for the sink temperature variation

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

The data shows that for sink temperatures below the critical temperature (~31C), a generous split ratio <= 0.7 and a high-pressure ratio is ideal. For above the critical temperature a smaller pressure ratio ≤3.0 and a higher split ratio is ideal. It is also noticed that efficiency drops dramatically when sCO_2 is subcritical at the gas cooler.

Results of Sink Variability

The resulting optimized cycle thermal efficiency over a practical range of sink temperatures at a source temperature of 550°C is shown in Figure 10 for both steam Rankine and sCO_2 Brayton cycles simulations. As can be seen the gap in thermal efficiency closes significantly with reduction of sink temperature, even with the very practical assumption of 85% isentropic efficiencies for sCO_2 turbomachinery.

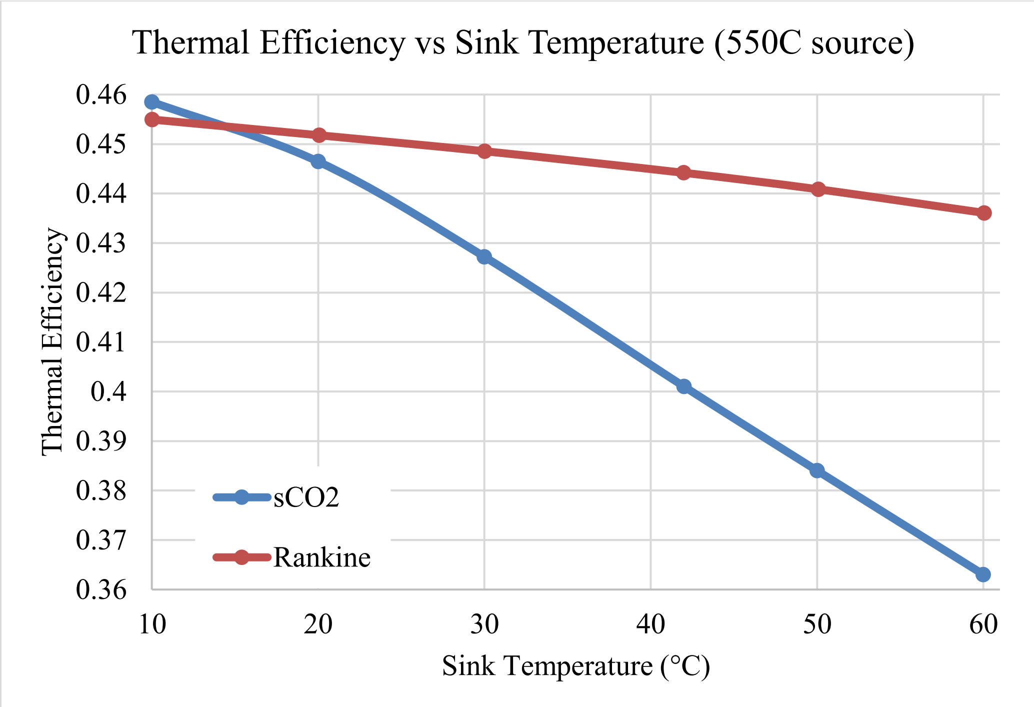

Figure 10: Cycle Efficiency results for variation of sink temperature

It becomes interesting to evaluate the effects of source temperature variation at a lower sink temperature than initially investigated. Although a sink temperature near 40°C is a practical design choice, it is also not unreasonable to consider the case of a 20°C sink that can be practically achieved consistently in cold climates or situations where bodies of water imply abundant heat sinks at steady temperatures.

With evidence from the previous optimizations (Figure 4) of sCO_2 cycles that source temperature has little sensitivity to the optimum pressure ratio or split ratio, a parameter sweep was simulated for both power conversion cycles. The results for both the steam Rankine and sCO_2 Brayton cycles are shown in Figure 11. As can be seen, with reduced sink temperatures, there is almost no gap between the two cycles in terms of efficiency and they are therefore competitive from that perspective. One final study was then conducted after conceptual turbomachinery was designed for the 20°C cycle, which is labeled “turbo design.” It is seen that a condensing sCO_2 cycle may outcompete Rankine cycles with state-of-the-art turbomachinery.

This information learned from this inspired us to consider the question of under what conditions and what specific features of each cycle would favor a particular choice, with specific question as to what may ultimately drive the widespread commercial adoption of sCO_2 cycles as a viable thermal power conversion system. This led us to briefly review the already known benefit of compactness of the equipment due to the working fluid density. It also led us to consider the possibility of cold energy storage. Notwithstanding the potential application of sCO_2 power cycles in many systems that may include air and naval propulsion, we focused attention to concentrated solar power (CSP) applications. Such installations are predominantly in desert environments where solar irradiance is near a maximum, but where unfortunately during peak source power conditions, the sink temperatures are also correspondingly high. However, such environments usually have strong diurnal temperature cycles which prompted us to consider if it is possible to store thermal energy on the cold side, which is particularly less challenging than high temperature thermal energy storage.

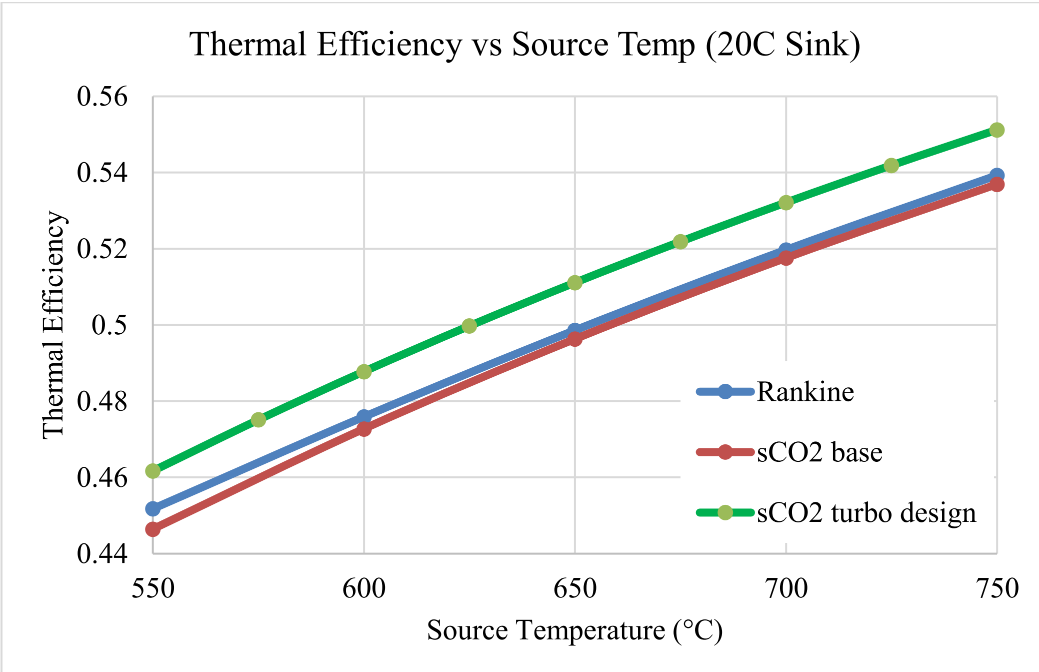

Figure 11: Thermal efficiency for variation of source temperatures at a 20C sink temperature condition for both power conversion cycles. In the green line is shown a specific instantiation where turbomachinery was designed and predicted using AxSTREAM™ meanline and streamline prediction tools.

5. EQUIPMENT SCALE

We first address the more direct comparison of equipment size. Although a number of other works have addressed such comparisons, including for example Dostal (2006) and Moroz et. al. (2014), it must be recognized that particular assumptions and boundary conditions will affect the sizing of the machines and other cycle components. For completeness, we evaluate measures of scale of the major equipment in the cycle with the baseline assumptions we have taken.

Turbomachinery

Turbomachinery trains were conceptually designed based on the baseline parameters of both the steam Rankine PCS and sCO2 PCS using the AxSTREAM™ Flow Path software. The nominal source temperature was 550°C. For variable sink temperatures from 20C to 42C, the steam turbomachinery does not significantly change in size, since the condensate is pressurized via a pump. However, for the sCO_2 turbomachinery, there is a significant effect because the fluid is compressed after the gas cooler, resulting in significant cycle parameter variations and required different compressors/pumps to be designed. Thus, there are two variants of the turbomachinery conceptually designed for the sCO_2 cycles. In both cases, the relative size is similar. The figure below shows the resulting concept designs for the two sCO_2 turbomachinery and the Rankine steam turbine, with a typical human silhouette for size reference.

For the sCO_2 turbomachinery, it was conceptualized that, for control reasons, the compressors would be driven by a dedicated turbine drive. Such a system is shown in Figure 12. For the 42C sink temperature case, it turned out that a synchronous 3600RPM compressor train was viable (but not necessary) for a nominal 330MW net system. However, for the 20C sink temperature case, the main compressor is essentially ingesting liquid (at perhaps supercritical pressure) with very high density and is quite compact, requiring higher rotational speed. A main pump and recompressor on a nominally 9,450RPM shaft was therefore conceptualized and evaluated as feasible.

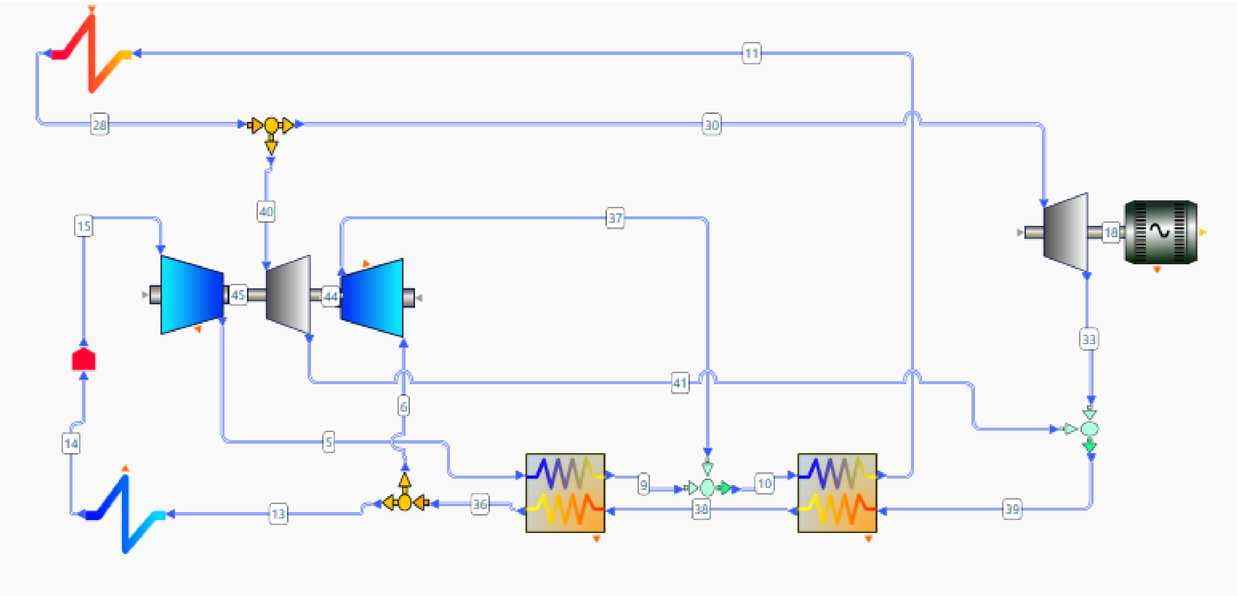

Figure 12: Concept equipment and flow diagram for practical controllable RCBC sCO2 plant

Figure 13 shows the resulting turbomachinery configurations and size comparisons. It should be noted that in these designs, only the flowpath is conceptually considered, with no attention given to structural concerns or the mechanical casings and housings of the machines. More detailed analysis was done by Moroz et al (2014) where both aerodynamic and structural considerations revealed that sCO_2 flow paths would be slightly larger than if only aerodynamics were considered.

As is seen from Figure 13, the sCO_2 turbomachinery is significantly more compact for nominally similar power production capacities. It should be noted however, that the size of the sCO_2 machinery is within the same range as an HP/IP steam turbine cylinder. Unlike steam turbines, the Brayton cycle turbine temperature drop is fairly small in comparison. Thus, with regard to the material requirements for the high temperature part of the flow path where potentially costly and exotic metallurgy would be required, the size is similar between the two cycles.

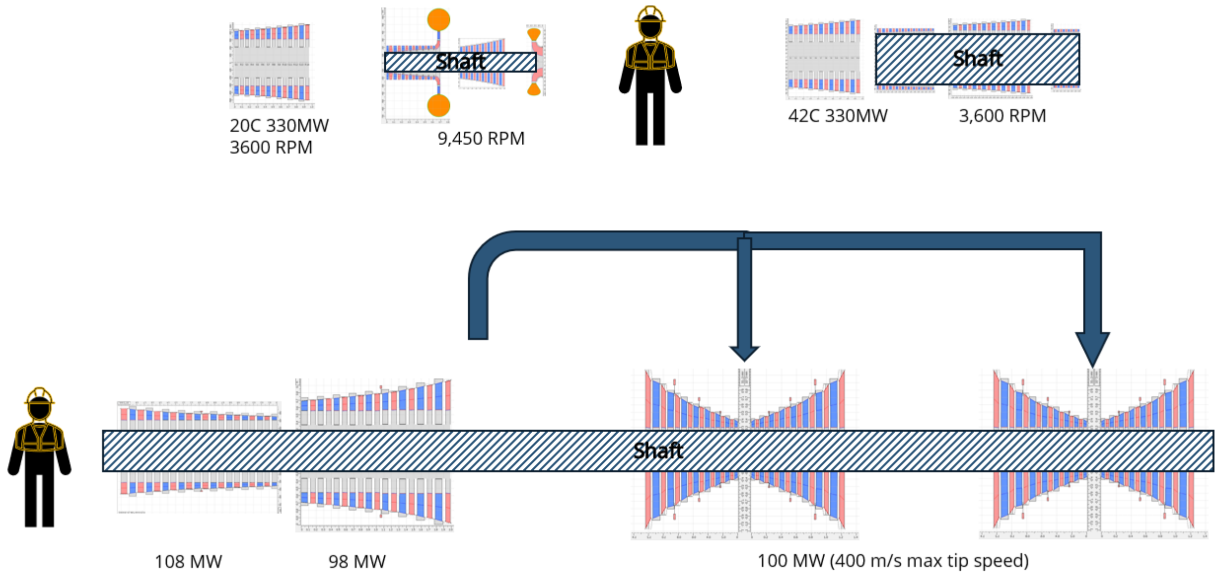

Figure 13: Flow path concept designs for a nominal 305MW-330MW capacity power plant with sCO_2 turbomachinery on top and steam turbomachinery on the bottom. For the sCO_2 cycles, the turbomachinery is distinguished between a 20C sink case (left) and a 42C sink case (right).

Heat Exchange Equipment

We further report the internal heat exchanger total transfer rate and conductance to provide some small insight into the possible sizing aspects of the internal heat exchangers. These internal heat exchangers are the feedwater heaters and recuperators for the steam Rankine and sCO_2 Brayton cycles, respectively.

We recognize that since the thermal efficiency of both cycles are very close within the same order of magnitude, the total thermal power transfers in the sources and sinks are also going to be correspondingly similar. As we did not explicitly model the secondary fluid transfer process, there is no additional thermal sizing information we are able to provide on either the sources or sinks. We also recognize that the data provided is a physical outcome of the 0D thermodynamic modeling and not specific 1D thermofluid flow network or heat exchanger preliminary concept design. Table 9 provides the feedwater heater thermal integral parameters of the nominal 305MW steam Rankine cycle while

Table 10 provides the recuperator thermal integral parameters of a nominal 330MW sCO_2 cycle.

Table 9: Internal cycle heat exchanger parameters for baseline steam Rankine power cycle

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

Table 10: Internal cycle heat exchanger parameters for baseline sCO2 RCBC PCS

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

From the data presented, it is clear that the total internal heat transfer rate in the sCO_2 cycle is about an order magnitude higher compared to the steam Rankine cycle. Furthermore, examination of the total heat conductance shows that the sCO_2 cycle requires a heat transfer conductance between one and two orders larger than a steam Rankine PCS of similar generating capacity. Of course, it should also be recognized that the effective value of U, or the effective overall heat transfer coefficient, may be significantly different due to the fluid density, thermal conductivity, and design of the heat exchanger itself.

6. SINK COLD STORAGE STUDY

The final aspect of the analysis that was conducted was a concept for utilizing the strong improvement in thermal efficiency of the sCO_2 cycle as sink temperature decreases in environments where the ambient temperature fluctuates greatly. This is particularly relevant for CSP applications where there are typical strong diurnal temperature variations. A concept diagram of how to incorporate cold thermal energy storage into the sink of the sCO_2 cycle is diagramed in Figure 14

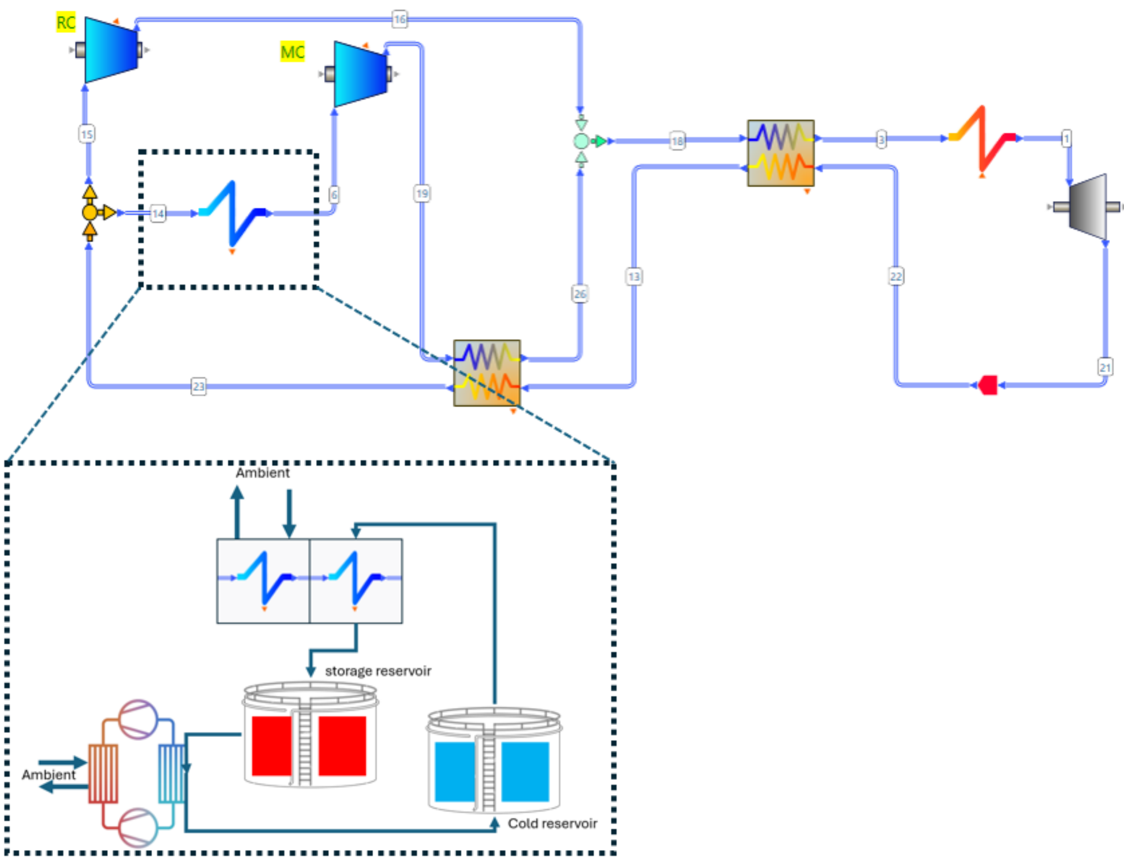

Figure 14: Concept of integrating cold thermal energy storage into the heat sink / gas cooler of the RCBC sCO_2 PCS

A heat pump can be used (if ambient temperatures do not fall sufficiently low) as shown in Figure 14 to cool a reservoir of cold fluid to be used for final cooling in the gas cooler of the sCO_2 cycle. The typical cooling COP of a vapor compression heat pump is shown in Figure 15 based on a basic thermodynamic analysis.

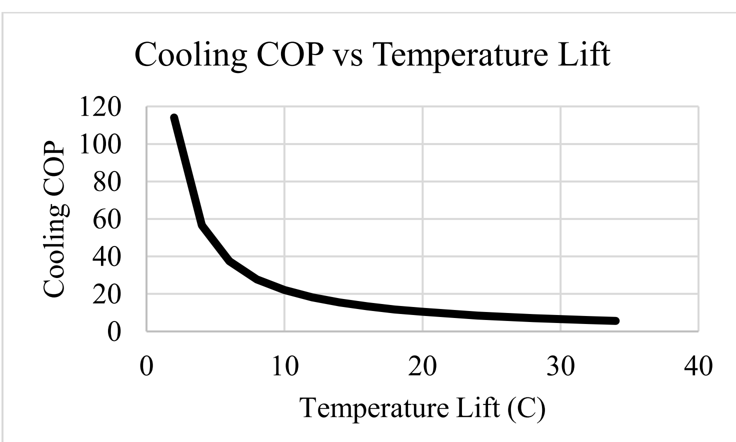

Figure 15: Typical cooling COP as a function of temperature lift for a single stage vapor compression cycle

A fairly simplified analysis is presented here using the combined data of Figure 10 and Figure 15, in combination with a sample ambient temperature data from NOAA for the area of Phoenix, Arizona, a reasonably good representation of a location where CSP would be favorable.

Two scenarios are considered that are shown in Table 11. They are based on historical data and contain a hot climate scenario of July average daily mean minimum and maximum temperatures in Phoenix, Arizona, as well as yearly average daily mean minimum and minimum temperatures as scenario two.

Table 11: Two scenarios of sink conditions based on historical data

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

For the purposes of this sample, it is assumed that the operation of the plan is binary and equally distributed 50/50 between the maximum and minimum conditions and dependent upon the thermal efficiency curve of Figure 10. That is, it is assumed for simplicity that the plant operates half the time in "night mode" in the mean daily minimum ambient condition and the other half in day mode in mean daily maximum condition. The heat pump, if needed, is engaged only during night mode.The results of the analysis are shown in Table 12.

Table 12: Efficiency benefit of cold storage over doing nothing: which means to operate the plant at best possible method without cold storage for the available ambient conditions

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

It should be noted that it is assumed there is a 4℃ temperature difference between the ambient and the sink. While these analysis are quite simplified, it shows that there are scenarios available that may take advantage of the strong thermal efficiency dependence upon the sink temperature for the sCO_2 cycle.

7. CONCLUSION

We showed that with realistic conditions, the incumbent steam Rankine cycle is able to outperform the sCO_2 Brayton cycle with respect to power conversion efficiency, even at very high source temperatures. However, it was shown that at sink temperatures that dropped below the critical temperature of CO_2, the sCO_2 Brayton cycle was potentially competitive. We exampled component scales and explored the possibility of using cold thermal energy storage to enhance the viability of the sCO_2 cycle for CSP applications.

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