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TECHNO-ECONOMIC COMPARISON OF SOLAR-DRIVEN SCO2 BRAYTON CYCLES USING COMPONENT COST MODELS BASELINED WITH VENDOR DATA AND ESTIMATES

Matthew D. Carlson
Sandia National Labs
Albuquerque, NM, USA

Bobby M. Middleton
Sandia National Labs
Albuquerque, NM, USA

Clifford K. Ho
Sandia National Labs
Albuquerque, NM, USA

ABSTRACT

Supercritical carbon dioxide (sCO2) Brayton power cycles have the potential to significantly improve the economic viability of concentrating solar power (CSP) plants by increasing the thermal to electric conversion efficiency from around 35% using high-temperature steam Rankine systems to above 45% depending on the cycle configuration. These systems are the most likely path toward achieving the Department of Energy's (DOE) Office of Energy Efficiency and Renewable Energy (EERE) SunShot targets for CSP tower thermal to electric conversion efficiency above 50% with dry cooling to air at 40 °C and a power block cost of less than 900 \$/kWe. Many studies have been conducted to optimize the performance of various sCO2 Brayton cycle configurations in order to achieve high efficiency, and a few have accounted for drivers of cost such as equipment size in the optimization, but complete techno-economic optimization has not been feasible because there are no validated models relating component performance and cost.

Reasonably accurate component cost models exist from several sources for conventional equipment including turbines, compressors, and heat exchangers for use in rough order of magnitude cost estimates when assembling a system of conventional equipment. However, cost data from fabricated equipment relevant to sCO2 Brayton cycles is very limited in terms of both supplier variety and performance level with most existing data in the range of 1 MWe power cycles or smaller systems, a single completed system around 7 MWe by Echogen Power Systems, and numerous ROM estimates based on preliminary designs of equipment for 10 MWe systems. This data is highly proprietary as the publication of individual data by any single supplier would damage their market position by potentially allowing other vendors to undercut their stated price rather than competing on reduced manufacturing costs.

This paper describes one approach to develop component cost models in order to enable the techno-economic optimization activities needed to guide further research and development while protecting commercially proprietary information from individual vendors. Existing cost models were taken from literature for each major component used in different sCO2 Brayton cycle configurations and adjusted for their magnitude to fit the limited vendor cost data and estimates available. A mean fit curve was developed for each component and used to calculate updated cost comparisons between previously-reviewed sCO2 Brayton cycle configurations for CSP applications including simple recuperated, recompression, cascaded, and mixed-gas combined bifurcation with intercooling cycles. These fitting curves represent an average of the assembled vendor data without revealing any individual vendor cost, and maintain the scaling behavior with performance expected from similar equipment found in literature.

INTRODUCTION

The cost of the power block for a concentrating solar facility using thermal particles and supercritical carbon dioxide (sCO2) power conversion must meet SunShot requirements of 900 \$/kWe in addition to technical requirements of dry cooling and 50% thermal to electric efficiency. This power block cost is a key metric for the Department of Energy (DOE) office of Energy Efficiency and Renewable Energy (EERE) in order to achieve the 75% cost reduction in solar energy targeted in the SunShot vision study [1]. However, determining the cost of a 100 MWe sCO2 power block is difficult because most equipment for the plant has been demonstrated only at a reduced scale in a relevant prototypical engineering environment, equating to a technology readiness level (TRL) of 6 or lower. Due to these uncertainties, we sought to develop a cost modeling methodology that could leverage both publically available and proprietary data in a common baseline for predicting sCO2 equipment costs when performing techno-economic analysis and optimization of cycle configurations.

CYCLES UNDER CONSIDERATION

Many alternative configurations for sCO2 Brayton cycles have been proposed since the first early work by Feher [2], [3] and Angelino [4], [5]. The main goal of this work is to update our cost modeling approach from that used in a previous paper [6], but the four cycles chosen from optimization studies in literature will be discussed again briefly here with key operating conditions and performance metrics shown in Table 1.

The simple closed Brayton cycle (SCBC), either recuperated as shown in Figure 1 or un-recuperated, consists of a single compression process operating near the critical point to take advantage of the high fluid density of sCO2 and resulting low compression work, and a single expansion process operating at high temperatures and pressures where sCO2 acts much like an ideal gas. This high pressure and temperature sCO2 still has about 10 times the density of steam at turbine inlet conditions which allows for smaller turbine impellers. Overall the system has a very low backwork ratio, the ratio between compressor power required and turbine power generated, much like a steam Rankine cycle but without any intermediate phase change. The lack of phase change allows for recuperation between the turbine exhaust and compressor outlet much like a gas Brayton cycle which reduces the required heater power and raises efficiency. Leveraging these two key advantages of low backwork ratio and recuperation otherwise only found separately in steam Rankine and gas Brayton cycles results in cycle efficiencies above 45%. Note that both un-recuperated and recuperated versions of the SCBC are shown in Table 1 with the same column header because both are SCBCs.

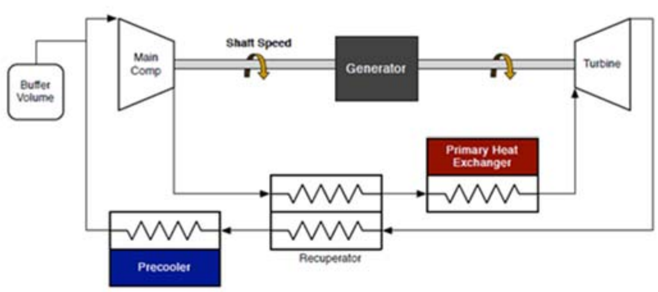

Figure 1. A flow diagram of a SCBC configuration.

While the recuperation process in the SCBC increases efficiency, it is limited by a pinch point occurring in the recuperator. This pinch point occurs because the temperature-enthalpy curves for the high and low pressures levels in the system are non-linear and create a minimum temperature difference within the recuperator rather than at the ends [7].

This pinch point can be mitigated by splitting the recuperation process at the pinch point and adding a second compressor to split the flow and unbalance the heat exchange processes. This recompression closed Brayton cycle, or RCBC, can therefore achieve higher efficiencies than the SCBC under the right conditions but does add increased costs due to the second compressor and higher amount of recuperation.

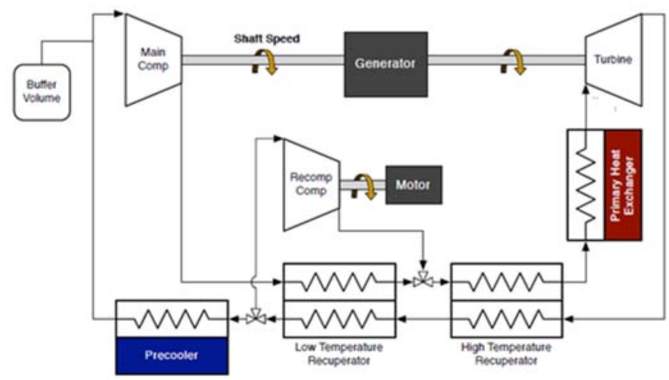

Figure 2. A flow diagram of a supercritical CO_2 RCBC.

Kimzey [8] was one of the first to propose a class of cascaded closed Brayton cycles (CCBCs) as shown in Figure 3. This cycle configuration was created to better match sensible heat sources such as fossil-fired plants and CSP thermal storage systems rather than heat flux sources such as Generation IV nuclear plant designs and direct CSP by trading some efficiency for increased temperature rise in the heater. The components of a CCBC are similar to that of an RCBC, but are arranged as two cascaded SCBCs utilizing the same cooler and compressor. The high-temperature turbine receives heat directly from the sensible heat source, and its high-temperature recuperator acts as the heat source for the low-temperature turbine.

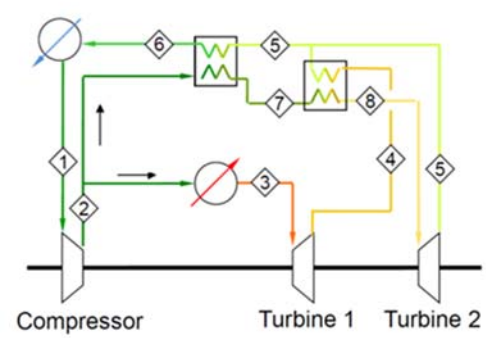

Figure 3. A flow diagram of the first CCBC analysed by Kimzey [8].

The final cycle considered in the study is the Combined Bifurcation with Intercooling or CBI cycle proposed by Garg et al. [9]. This cycle is a variation of another known as the partial cooling cycle [10], but operates with a gas mixture rather than pure sCO2 in order to reduce the critical pressure and increase the critical temperature. Part of the flow from the main compressor is further condensed so that it can be pumped in a state requiring less work to be provided and then brought back up to temperature in a regenerator. This is an alternative approach to managing the pinch point otherwise encountered during the recuperation process. Under some conditions this cycle can achieve higher efficiencies than an RCBC, but the principal advantage present is the reduced operating pressure which could allow for higher turbine inlet temperatures.

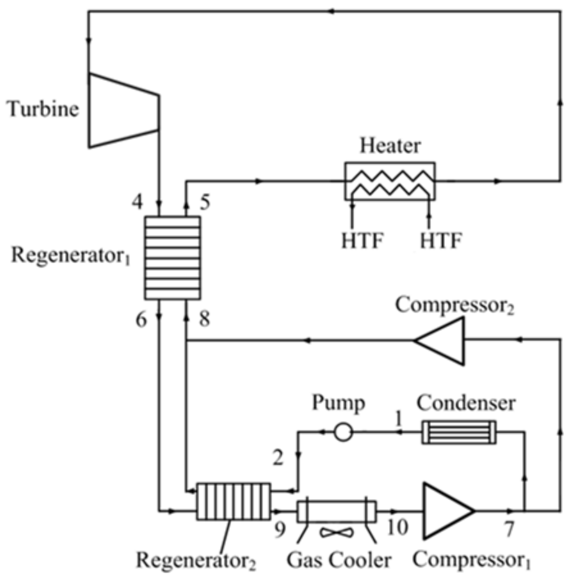

Figure 4. Schematic of a CBI cycle [9].

Table 1. Power cycle design and performance metrics.

	SCBC [11]		**RCBC** [11]	**CCBC** [8]	**CBI** [9]
Net Power (MWe)	100	100	100	133	100
Efficiency (%)	16	46	46	28	51
T_{max}(°C)	700	700	700	600	700
P_{max}(MPa)	20	20	20	27.6	15
P_{min}(MPa)	6.4	8.0	7.3	8.5	2.6
P_{int}(MPa)	N/A	N/A	N/A	N/A	5.0
$T_{comp,min}$(°C)	55	55	55	37	35
η_{comp}(%)	90	90	90	85	90
η_{exp}(%)	90	90	90	90	90
f_{rec} or $f_{cascaded}$(%)	N/A	N/A	11.5	47.5	40
$\Delta T_{HTF,min}$(°C)	25	25	25	25	25
$T_{air,min}$(°C)	30	30	30	30	30
ΔT_{HTR}(°C)	540	172	170	518	159
$\dot{C}_{HTF}$(MW/K)	1.39	1.53	1.27	0.919	1.21
$\dot{q}_{HTF}$(MWth)	623	220	216	449	192

HEAT EXCHANGER COST MODELS

The most accurate method for estimating the cost of heat exchange equipment in a system is to determine the required heat transfer surface area "A" and determine the cost from tables or vendor estimates of cost per unit surface area ($/A). The heat transfer surface area directly scales with the physical size of the equipment with dependence mainly on the type of heat exchanger and certain breakpoints in manufacturing or shipping size. This process requires calculation of the duty "$\dot{Q}$", overall conductance "U", and a mean temperature differential "ΔT_m" for the device as shown in Equation (1), but the highly variable properties of sCO2 cycles make calculation of the latter two parameters unique to every flowsheet configuration and heat exchanger flow arrangement. Note that ΔT_m can be calculated directly for certain heat exchanger configurations, but is used here generally to describe the net driving temperature difference between two fluids.

$$A = \frac{\dot{Q}}{U \Delta T_m} \quad \textbf{(1)}$$

One alternative to this direct cost estimation method is to use the value of cost per unit of thermal duty ($/$\dot{Q}$) from a known heat exchanger to estimate the cost of an unknown heat exchanger based only on the ratio of thermal duties. By observing Equation (1) it can be seen that this method is reliable when the overall conductance and mean temperature differential of the known and unknown heat exchangers are similar because they will cancel out, and applying the duty ratio will be equivalent to scaling the heat transfer surface area. However, small differences in these values can change the physical size of a device for the same duty by factors of 2 to 10 or more [12]. General cost models based on cost per unit thermal duty are therefore impractical, and especially so for sCO2 applications where variability in properties can lead to pinch points in the recuperators which increase the sensitivity of surface area to mean temperature differential.

A cost model based on the conductance-area product (UA) of the heat exchanger solves many of the issues with a thermal duty cost model as proposed by Hewitt and Pugh [13]. The UA of a heat exchanger is essentially it's "thermodynamic size," and depends only on process conditions and the heat transfer configuration between the fluids, such as counterflow, crossflow, parallel flow, or some combination of these three. A pure counterflow configuration is the fundamental performance limit for heat exchanger between any two streams, and therefore a cost determined from a UA value calculated for pure counterflow provides a minimum cost for that equipment.

The Engineering Sciences Data Unit (ESDU) compiled a large dataset [14] on heat exchanger costs using this method for a variety of UA values, fluid combinations, pressures, and heat exchanger types including printed circuit heat exchangers (PCHEs) and tube-fin style air coolers. This dataset does not include any information directly applicable to estimating the cost of equipment for sCO2 service, but the influence of flow properties and pressure on the design of various heat exchangers is similar. In our original paper [6] this dataset was used with recommended scaling factors stainless steel construction and high pressure to calculate component cost values using logarithmic interpolation as shown by Equation (2).

A power-law cost curve was recommended for general scaling, however this approach is undesirable as the behavior of the original cost model and the suggested power-law fit curves were not well correlated, leading to differences between the two of as much as 25%.

As part of the baselining exercise performed for this paper an alternative approach is suggested with normalized behavior curves for the primary heat exchanger, recuperator, and air coolers and condensers shown in Table 2 rather than the use of power-law fits to the same data. These curves are normalized by their UA values at 1e6 W/K because as highlighted in Figure 5 the specific cost or "C^*" value generally asymptotes at large UA values. This behavior also provides reasonable justification for extrapolation above the existing dataset as heat exchanger original equipment manufacturer (OEM) margins reduce to their minimum and cost becomes a linear function of UA. These curves can be scaled by the C^* value at large values of UA collected in Table 5 to calculate equipment costs.

$$C^* = \exp\left[ln(C_1) + \frac{ln\left(\frac{C_1}{C_2}\right) ln\left(\frac{UA}{UA_1}\right)}{ln\left(\frac{UA_1}{UA_2}\right)}\right] \quad \textbf{(2)}$$

Table 2. Suggested normalized cost scaling behaviour for primary heat exchangers, recuperators, and air coolers or condensers in $/(W/K) for different levels of UA in (W/K). Adapted from the ESDU [14].

UA (W/K)	$5x10^3$	$3x10^4$	$1x10^5$	$3x10^5$	$1x10^6$
Primary Heat Exchanger ($/(W/K))	1.9	1.3	1.1	1.0	1.0
Recuperator ($/(W/K))	6.3	1.4	1.3	1.1	1.0
Air Coolers / Condensers ($/(W/K))	7.6	2.4	1.3	1.1	1.0

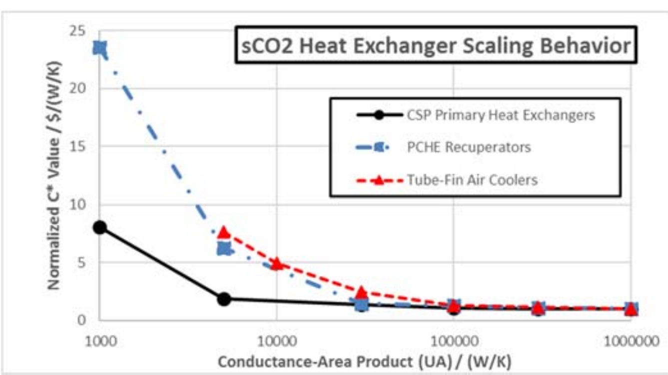

Figure 5: Suggested cost scaling behaviour for primary heat exchangers, recuperators, and air coolers or condensers in $/(W/K) for different levels of UA in (W/K). Adapted from the ESDU [14].

The viability of this approach is demonstrated by the results in Table 3 comparing the original cost estimates using the ESDU [14] dataset directly and the "naïve" fitting values from Table 5. Increases of 3% or less can be seen which is well within the uncertainty bands for this cost estimation approach.

Table 3. Comparison of heat exchanger cost using the ESDU [14] dataset directly and the proposed approach.

Category	Model	SCBC		RCBC	CCBC	CBI
Recuperation ($/kWe)	[14]	0	243	244	122	259
	Naïve	0	250	251	125	267
	Change	0%	2.9%	2.9%	2.5%	3.1%
Cooling ($/kWe)	[14]	545	85	154	574	350
	Naïve	547	86	155	576	351
	Change	0.4%	1.2%	0.6%	0.3%	0.3%

TURBOMACHINERY COST MODELS

Unlike cost scaling for heat exchangers, turbomachinery generally exhibits power-law scaling behavior with capacity reflecting the underlying scaling of both impeller speed and diameter with power. The curves from Peters et al. [15] summarized below in Table 4 were modified with recommended factors of 2.5 to account for compressor construction from stainless steel rather than carbon steel, and a factor of 3 for turbines to account for nickel alloy construction rather than carbon steel construction. Because the cost curves from literature were compiled for steam, gas, and other service rather than for sCO2 power cycles the original compressor cost models were further modified with an approximate density ratio factor of 0.2 to account for the increased power density of CO2 necessary when assuming identical volumetric flow rate and head rise of the CO2 compressor to the air compressor. This value is adjusted from 0.2 up to 0.8 with decreasing density as the compressor inlet pressure is lowered for the same compressor inlet temperature when evaluating the SCBC cycle.

Table 4. Turbomachinery cost models from [15].

	Power-law Cost Scaling
*Motor-Driven Compressor ($)**	$461.91\left(\dot{W}/kW\right)^{0.9339}$
*Turbine-Driven Compressor ($)**	$643.15\left(\dot{W}/kW\right)^{0.9142}$
*Radial Expander ($)***	$4001.4\left(\dot{W}/kW\right)^{0.6897}$
*Axial Gas Turbine ($)***	$9923.7\left(\dot{W}/kW\right)^{0.5886}$
*Centrifugal Pump ($)****	$124427\left(\dot{V}/\frac{m^3}{s}\right)^{0.3895}$

*Includes factors of 2.5 and 0.2 for stainless steel construction and density ratio of air and CO2 at 8 MPa.
**Includes factor of 3 for nickel alloy construction.
***Includes factors of 2.4 for stainless steel construction and 2.8 for elevator operating pressure.

In our previous paper [6] the two different compressor and turbine models were both calculated and averaged to account for uncertainties in the ultimate configuration of each sCO2 Brayton cycle and the ultimate scaling exponent for sCO2 equipment. Data for centrifugal pumps were also used to calculate the one liquid pump present in the Combination Bifurcation with Intercooling (CBI) cycle based on scaling with volumetric flow rate. The original cost model was multiplied by factors of 2.4 and 2.8 to account for stainless steel rather than carbon steel construction and the elevated pressure of the CBI cycle, respectively, as recommended by Peters et al. [15].

BASELINING COMPONENT COST MODELS

The literature cost models so far described were originally developed to cover a range of conventional heat exchanger and rotating equipment, but did not included any data on sCO2 equipment. In order to transform the available cost models into a relevant context for sCO2 cycle optimization studies and to track progress on reducing equipment costs these models were baselined against the limited data available for actual manufactured equipment costs and vendor cost estimates. The term "baselining" was chosen because the set of available cost data is severely limited and generally clustered around 1 MWe- and 10 MWe system scales. This data is not sufficient to perform a true fitting operation, but does allow the magnitudes of existing cost models to be adjusted to pass through the available data. The entire dataset, both adjusted cost models and direct cost data, is then fit as appropriate to either the logarithmic heat exchanger curves or turbomachinery power law fits. None of the actual vendor data is shown in this paper as the release of any individual price information whether it is protected as proprietary or not has the potential to damage the market position of the responsible vendor.

A total of 18 vendor cost estimates ranging in UA values from 0.15 to 1.25 MW/K and costs from 0.2 to 5 M$ were used to baseline the normalized cost or "C" values from the original ESDU dataset [14]. Table 5 summarizes the best fits for each heat exchanger type based on the asymptotic value at large values of UA. The "naïve" model value is derived from the ESDU dataset and does not benefit from any vendor data, and the baselined model value does benefit from scaling to the available vendor data.

Table 5. Heat exchanger equipment cost scaling based on the original ESDU [14] dataset and baselining vendor data.

	Naïve	Baselined
Primary Heat Exchanger	*3.5*	*-*
Recuperator	*1.25*	*1.1 - 4.0*
Air Coolers / Condensers	*2.75*	*~2.3*

No reliable cost data is currently available for the primary heat exchanger between a CSP tower thermal storage material and the sCO2 power cycle so the original naïve fit is still used. There is still a large variance in recuperator costs from different vendors for the same design conditions as shown by the baselined value of 1.1 to 4.0 in Table 5, with some of the most promising quotes droping just below the naïve estimates based only on the ESDU dataset. Finally, the baseline value for air coolers and condensers was determined from a much smaller set of cost estimates representative of 5 MWe and 10 MWe RCBC systems with less variation observed than with the recuperator data. Results using the baseline recuperator value of 1.1 and air cooler value of 2.3 are shown later in Table 7.

The baselining process for turbomachinery is most easily shown visually in Figure 6 through Figure 9. Figure 6 and Figure 7 are plots of the naïve and baselined compressor cost estimates using adjusted literature models versus the compressor power consumption. The compressor cost model curves were multiplied by factors to pass through 8 data points ranging from 1.5×10^3 to 2.5×10^4 kW and costs from 2 to 20 M$, and ended up requiring the same factor for all models due to the small amount of variation between them. The resulting power-law fit, which includes both the baselined literature cost model data and the vendor data is shown in Figure 7 with the resulting coefficient of determination value of 0.9263.

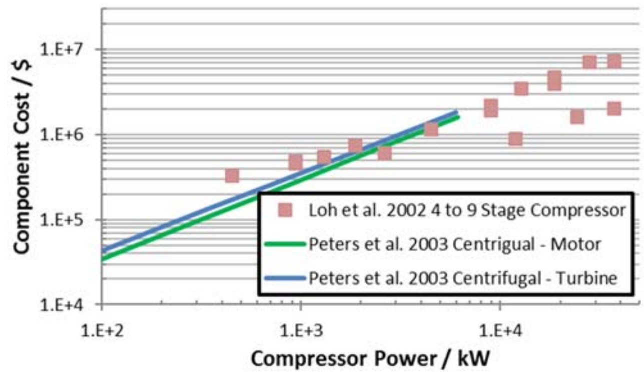

Figure 6: Naïve literature cost models from [15] and [16] after adjusting for stainless steel construction and density.

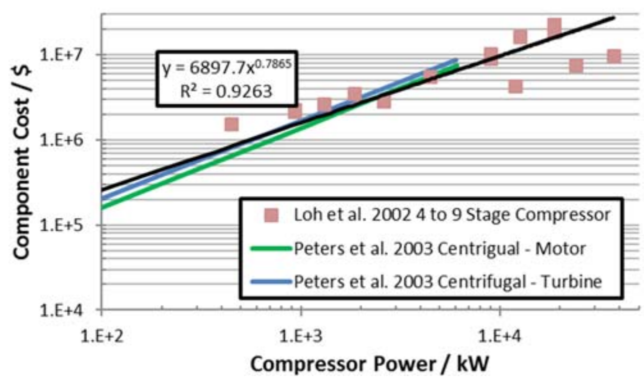

Figure 7: Baselined literature cost models with the resulting power law fit for all data. Vendor data not shown.

The same process is carried out for the turbine data as shown in Figure 8 and Figure 9. In this case the different cost models required different multiplying factors to pass through the available cost data but a higher coefficient of determination was achieved of 0.9712. Only 6 data points were available ranging from approximately 1×10^4 to 1×10^5 kW and 2 to 10 M$.

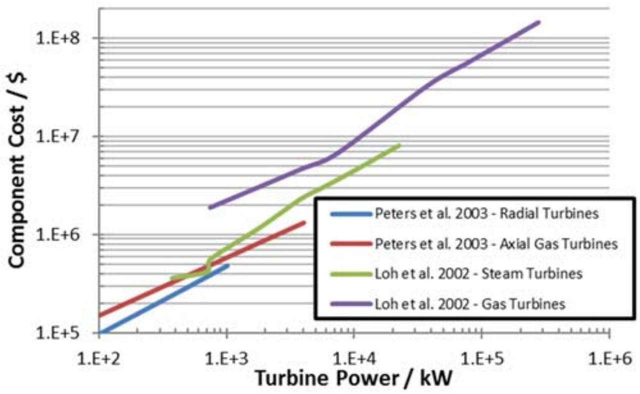

Figure 8: Naïve literature cost models from [15] and [16] after adjusting for nickel alloy construction.

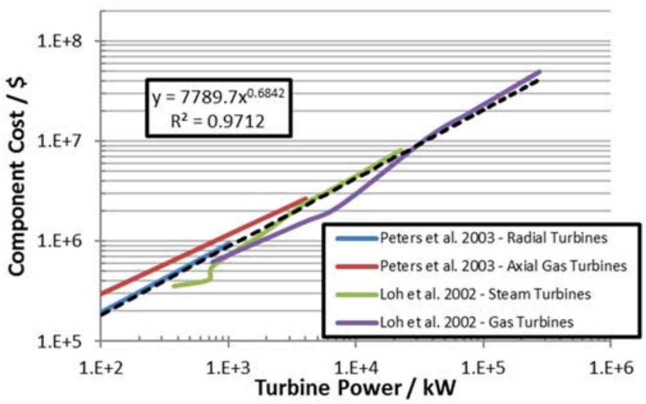

Figure 9: Baselined literature cost models with the resulting power law fit for all data. Vendor data not shown.

The numerous cost models used for turbomachinery were condensed into two single models summarized in Table 6 based on the available vendor data. The turbine cost model trends very similarly to the original literature models but with a magnitude closer to the high end of the literature cost model magnitudes. Estimates made using this model should therefore trend higher in cost as the two cost models from Peters et al. [15] were previously averaged together. The compressor cost model however is very different with a more favorable scaling exponent but the magnitude increasing by approximately a factor of 10. Compressor costs are therefore expected to be much higher than previous estimates.

It should be noted that this approach neglects any difference between compressors and pumps which results in an approximate 45 k$ increase in the cost for compression for the CBI cycle in Table 7 between the baselined compressor model and the naïve pump model. Due to the range of possible compressor and pump operational conditions cost models for different ranges may be necessary to improve accuracy.

Table 6. Baselined turbomachinery cost models.

	Power-law Cost Scaling
Compressor ($)	$6898(\dot{W}/kW)^{0.7865}$
Turbine ($)	$7790(\dot{W}/kW)^{0.6842}$

UPDATED CYCLE COST COMPARISON

These baselined cost models can be used to update the cycle cost comparisons calculated previously [6] as shown in Table 7. While the costs for recuperation and cooling generally go down slightly for all cycles, the costs for turbomachinery dramatically increase by a factor of 2 to 3.

This is particularly notable as the CCBC and CBI cycle flowsheets explored were not optimized for the dry cooling conditions required for CSP. Table 1 highlights that the ambient heat sink temperatures were raised as close as reasonable to the optimized compressor inlet temperature but still remain at least 15 °C below the SunShot target ambient temperature of 55 °C.

Table 7. Comparison of power cycle costs using both adjusted literature and baselined models.

Category	Model	SCBC	RCBC	CCBC	CBI	
Heater ($/kWe)	*Naïve*	375	209	318	277	288
Recuperation ($/kWe)	*Naïve*	0	250	251	125	267
	Baselined	0	220	221	110	235
Cooling ($/kWe)	*Naïve*	547	86	155	576	351
	Baselined	458	72	130	482	294
Compression ($/kWe)	[6]	243	114	147	80	74
	Baselined	625	328	440	233	311
Expansion ($/kWe)	[6]	160	128	135	138	120
	Baselined	338	268	283	284	250
Total ($/kWe)	*Naïve*	1325	787	1006	1196	1100
	Baselined	1796	1097	1392	1386	1378

CONCLUSION

This paper presents an updated approach to estimate the cost of sCO2 equipment using logarithmic interpolation and asymptotic extrapolation of normalized heat exchanger cost models and turbomachinery cost models from literature baselined to the limited set of available cost data. This approach allows the general trend in component cost models developed from experience with conventional equipment to be used in lieu of sCO2 component cost data. The common, open approach can further allow the research community to agree on cost trends in different equipment types without revealing proprietary price data from individual vendors.

The updated cycle cost results show promise that the heat exchanger cost models originally used were reasonably accurate to sCO2 components with baselining suggesting they over-predicted costs by 10% to 20%. The turbomachinery cost models from literature however were quite far off from relevant sCO2 cost data. Turbines were being under predicted by approximately 50% while compressors were under predicted by between 60% and 75%. While divergence from the literature cost models was expected as they weren't derived from any data for sCO2 equipment, the compressor cost models were the only ones to land outside of the rough order of magnitude guidelines of -30%/+50%.

The trends between cycles remain consistent with the recuperated SCBC offering the most promise at achieving the SunShot goal of 900 $/kWe and the RCBC, CCBC, and CBI cycles all holding promise but still far from this target mainly due to the anticipated costs for compression. Further work is needed to develop bottom-up cost estimates where possible and to collect additional vendor estimates so that the uncertainty in component costs can be reduced and reliable techno-economic optimization can be performed to identify the power cycle layout most likely to achieve SunShot goals.

NOMENCLATURE

A	Heat Transfer Surface Area
°C	degrees Celsius
C^*	Heat Exchanger Specific Cost
CCBC	Cascaded Closed Brayton Cycles
CBI	Combined Bifurcation with Intercooling
CSP	Concentrating solar power
DOE	Department of Energy
EERE	Energy Efficiency and Renewable Energy
ESDU	Engineering Sciences Data Unit
OEM	Original Equipment Manufacturer
PCHE	Printed Circuit Heat Exchanger
$\dot{Q}$	Heat Exchanger Duty
RCBC	Recompression Closed Brayton Cycle
ROM	Rough Order of Magnitude
SCBC	Simple Closed Brayton Cycle
sCO2	supercritical carbon dioxide
TRL	technology readiness level
U	Overall Conductance
UA	Conductance-Area Product
ΔT_m	Mean Temperature Differential

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