Thermo-Economic Analysis of Four sCO2 Waste Heat Recovery Power Systems

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ABSTRACT

The results of a thermo-economic analysis of four SCO_2 Waste Heat Recovery (WHR) power systems are summarized in this report. The "Simple Recuperated Brayton Cycle" (SRBC) was used as a reference power cycle. Two patented power cycles that were developed for Waste Heat Recovery (WHR) applications are also examined. These include the "Cascade Cycle" and the "Dual Recuperated Cycle". The fourth cycle is a well-known cycle that is often used in WHR Organic Rankine cycles that uses split flow and preheating. In this report it is called the "Preheating Cycle". The three non-reference cycles use multiple turbines, recuperators, heaters and split flow to increase the total electrical power (and thus the system economics).

The results of the analysis indicate that as a group, all three of the WHR power cycles produce substantially more power and have larger annual revenue capabilities than the simple recuperated Brayton cycle (SRBC). They all produce at least 1.2-1.6 MWe more power/ or up to 22% more electrical power than the SRBC. Generally these improvements are due to the ability of the process flow diagram to make better use of the available waste heat. In addition, the thermo-economic analysis optimizes the operating conditions to maximize the annual net revenue by selecting the heat exchanger approach temperatures, (higher effectiveness), split flow fractions, and turbine inlet temperature. One surprising observation was that, as a group, all non-reference cycles perform similarly both in terms of power performance and economic performance. Still, even though the WHR systems tend to produce more power and annual revenue, than the reference case, the increased capital costs lowers the rate of

return, from 18% for the SRBC to 14-15% for the WHR cycles, indicating that simpler sCO2 bottoming cycles may offer the best economic benefit.

Finally, in terms of the sCO2 based combined cycle power plant, there is sufficient economic return (15-20%), plus lower Levelized Cost of Electricity (LCOE) than is typically available on the grid, and less sensitivity to the cost of fuel due to the lower heat rates (~7000 versus 9611 BTU/kWh typical for a LM2500PE gas turbine) to warrant their development for use as primary power in distributed generation applications.

Introduction

One of the first commercial applications for supercritical CO2 (sCO2) power systems is likely to be 5-20 MWe Waste Heat Recovery (WHR) power systems for medium scale gas turbines (15-60 MWe) and for industrial applications (particularly steel mills, cement plants). The value proposition is illustrated by the fact that, when used as a bottoming cycle on the medium sized gas turbines, the sCO2 plant produces approximately 33% the gas turbine power. This therefore increases the net efficiency from 32%-35% to 45%-47% for the combined cycle. Capital costs are expected to increase to about $1000/kWe depending on the size of the gas turbine. Also, the sCO2 bottoming cycle is expected to be small and modular as required by most distributed power applications needed for medium sized gas turbine combined cycle. In addition, this is a size where bottoming cycles are not well served by steam.

Because sCO2 WHR power systems use several types of heat exchangers (primary, heat rejection, and recuperators) to make effective use of the heat source, the cost of the heat exchangers are expected to be a significant fraction of sCO2 bottoming cycle. For this reason, this report describes a thermo-economic analysis that determines the optimum sCO2 power system type and operating conditions to maximize the annual net revenue while taking into account the cost of all the heat exchangers. This analysis was performed on four types of sCO2 power systems.

The four sCO2 power cycles that were analyzed are illustrated in the process flow diagrams shown in Figures 1-4. The first power cycle is the Simple Recuperate Brayton Cycle (SRBC) (Figure 1) and is used as a reference case. The second and third power cycles are the "Cascaded cycle" and the "Dual Recuperated cycle" that were specifically developed for Waste Heat Recovery systems and are shown in Figures 2-3 (1,2). The fourth power cycle is a well-known and documented cycle that is widely used for WHR is the preheating cycle as shown in Figure 4 (3, 4, and 5).

The primary goal of a WHR power cycle is to produce as much power as possible from the available waste heat. This generally requires that the power cycle maximize the product of the waste heat recovery efficiency and the sCO2 thermal cycle efficiency, not just the sCO2 thermal cycle efficiency. The method to increase performance varies among the different types of sCO2 power cycles. However, as shown in Figures 2-4, each of the WHR sCO2 process flow diagrams includes the use of multiple turbines, multiple-recuperators, or multiple heaters to have a power cycle that works well with waste heat. Plus, each of the WHR power cycles use some form of split flow to increase performance.

Because the WHR power cycles increase the complexity and number of components within the power cycle compared to the SRB cycle, it will also increase cost. The thermo-economic analysis maximizes the net revenue benefit achieved by the power cycle while taking into account the additional cost of the added components. Thus, the thermo-economic analysis includes the performance gains and added costs of each sCO2 cycle type, as well as selecting the operating points which generally impact the effectiveness or size/cost of the heat exchangers and recuperators to maximize the economic benefit (net annual revenue).

The thermal analysis for the sCO2 power cycles assumed that all the power cycles were operated using the LM-2500PE gas turbine, a 25 MWe General Electric Gas turbine. The performance of this gas turbine is described in the product brochure (6), which is summarized in Table 1. In summary this gas turbine produces a waste heat source of approximately 40.7 MW_{th} at 822.1 K/549 C.

The sCO2 thermal performance analysis was based on standard thermodynamic cycle analysis using the Refprop properties for CO2 (7), and using Microsoft Excel (8) solver. Other important operating conditions that were assumed are shown in Table 2. They include the isentropic efficiency of the turbine and compressors, as well as a peak pressure of 3500 psia (24.8 MPa), with a compressor inlet pressure of 7.7 MPa and 32.3 C. All cycles were assumed to be single phase supercritical power cycles (no condensation).

WHR Performance Benefit

The WHR performance benefit that these cycles provides is illustrated in Figures 1-4 by showing the heat source "glide" temperature super-imposed on the T-S curve. The glide temperature shows the temperature reduction of the waste heat combustion gas in the primary heat exchangers. It plots the gas turbine exhaust gas temperature as a function of the CO2 entropy within the primary heat exchangers or the preheater. To make effective use of the waste heat, each process flow diagram should reduce the combustion-gas waste-heat to as low a temperature as practical, while always assuring that the combustion gas temperature is greater than the incoming CO2 temperature. Generally, temperatures below 90 C are avoided to avoid condensation.

For the **SRB cycle** (Figure 1), the combustion-gas exit temperature of the primary heat exchanger must be larger than the high pressure exit temperature from the recuperator (typically at least 20 C-40 C). For the SRB cycle the recuperator exit temperature is relatively high (479.1 K / 206 C); it therefore limits the amount of heat that can be transferred from the waste heat combustion gases to the CO2. For this analysis, about 61.2% of the available waste heat is transferred to the CO2.

In contrast the primary heat exchanger in the **"Cascade Cycle"** heats the CO2 from the compressor exit temperature (~343 K/70 C) to its turbine inlet temperature; thus this cycle can lower the waste heat combustion gases to a temperature that is only a little above the temperature of compression. This results in a much higher waste-heat-recovery-efficiency (85.6%). This is clearly illustrated in the glide temperature curve of Figure 2 that shows the combustion-gas temperature decreasing linearly to a low value. The remainder of the "Cascade Cycle" uses two recuperators and two turbines. This additional complexity is required to keep the power cycle efficiency high while converting thermal power to electrical power. Also note that the flow rate through the primary heat exchanger is reduced compared to the SRBC, which may result in some other benefits such as smaller piping and lower pressure drop.

The primary heat exchanger in the **"Dual Recuperated Cycle"** (Figure 3) is similar to the SRB cycle because the CO2 provided to the primary heat exchanger is limited to Low-Temperature recuperator high pressure exit temperature. However, this temperature is much lower than that of the SRBC because this cycle uses two recuperators, so its waste-heat-recovery-efficiency is 78.3% making it more effective than in the SRB cycle. Like the Cascade cycle, the Dual-Recuperated cycle also uses two turbines and two recuperators to keep the thermal conversion efficiency of the power system high. This cycle also has reduced flow rate through the primary heat exchangers.

The **"Preheating Cycle"** is essentially the SRB power cycle except that a fraction of the flow from the compressor is sent through a preheater, while the remainder of the CO2 flows through the recuperated Brayton cycle. Though not a requirement, the flow fraction and amount of preheating is arranged to assure that the enthalpy leaving the preheater equals the enthalpy leaving the high pressure leg of the recuperator where the two flows are combined. Thus, the combined primary heater and preheater allow the waste-heat combustion gases to be lowered to a temperature that is a little above the main compressor exit temperature because of the two-step heating process as shown in the glide-temperature curve in Figure 4. The two-step heating process allows for efficient waste-heat-recovery efficiency (82.1%), while allowing for high thermal power conversion efficiency.

For each power cycle, the thermo-economic analysis adjusted the sCO2 operating conditions for the turbine inlet temperature, the split flow fraction, (all WHR power cycles use split flow), and the heat exchanger approach temperatures. The maximum system pressure was limited to 24 MPa (~3500 psia) and the compressor inlet pressure was set to 7.7 MPa. These sCO2 performance assumptions plus other assumptions are summarized in

Table 1: Operating assumptions for the sCO2 Waste Heat Recovery cycles.

sCO2 Assumptions	Value
Turbine Isentropic Efficiency	**85%**
Compressor Isentropic Efficiency	**82%**
Losses Shaft to Electrical (Bearings, Gears, Generator, etc.)	**7%**
Water Inlet Temp	**292.2 K = 19 C**
Compressor Inlet Pressure	**7700 kPa**
Compressor Inlet Temp	**305.4 K**
Compressor Pressure Ratio	**3.12**
Operating Conditions Varied to Optimize Net Annual Revenue ***Variables were: Turbine Inlet T + All Heat Exchanger Approach Temperatures, Split Flow Fraction and Compressor Mass Flow Rate***	**-**

Table 2: Operating conditions for the waste heat source are based on the LM-2500PE gas turbine (ref 1).

Primary Gas Turbine	LM-2500PE
Mass Flow Rate Gas Turbine	**68.8 kg/s**

Temp of Gas Turbine Exhaust	**822.1 K**
Efficiency (at Gen Terminals at 15 C ambient)	**35.5%**
Thermal Combustion Power (@ 10146 kW/ BTU/hr)	**63,131 kW_{th}**
Thermal Exhaust (Waste Heat) Power kW (@ 15 C)	**40,731 kW_{th}**

Economic Analysis

The thermo-economic analysis was performed by selecting operating conditions to maximize the annual revenue generated by the power cycle while taking into account the capital costs of the turbo-machinery, the recuperators, the CO_2 primary heaters, the waste heat rejection system costs, and the auxiliary equipment. To determine the effective annual capital expenditure costs, a 20 year system lifetime was assumed along with a 5% annual discount rate. The plant utilization factor was assumed to be 85%, and the revenue generated from the sale of electricity is $0.06/kW-hr. These values are summarized in Table 3.

Table 3: Economic Cost Assumptions used to determine the annual cost of purchasing the heat exchanger and other balance of plant components. Also provided are the assumptions used for estimating the revenue generated including the discount rate, capacity factor, plant lifetime, and sales price of electricity.

Economic Assumptions	Value
Plant Lifetime	**20 yr**
Plant Utilization Factor	**85%**
Discount Rate	**5%**
Sale Price of Electricity	**$0.06/$kWh_e$**
Thermal Exhaust (Waste Heat) Power kW (@ 15 C)	**40,731 kW_{th}**

The net revenue was calculated as the difference in the annual revenue from the sale of electricity minus the annual discounted cost of the heat exchangers and the turbo-machinery plus balance of plant subsystem costs. This allows the economic analysis to determine if the additional hardware or larger more expensive heat exchangers provide a net revenue benefit.

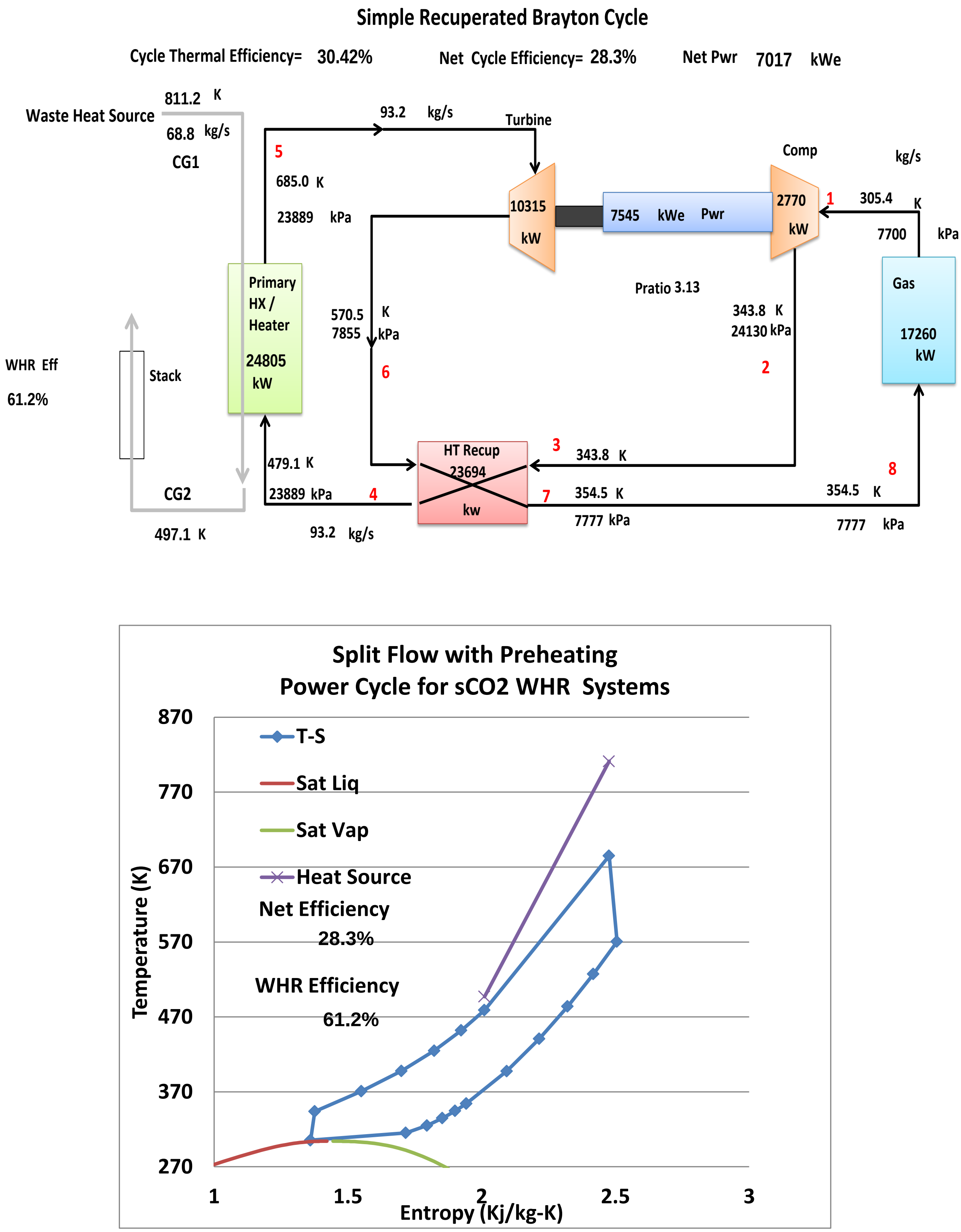

Figure 1: Process Flow Diagram for the Simple Recuperated Brayton Cycle used for Waste Heat Recovery is shown in the image. The T-S curve for the power cycle and the heat source glide temperature is shown in the lower image.

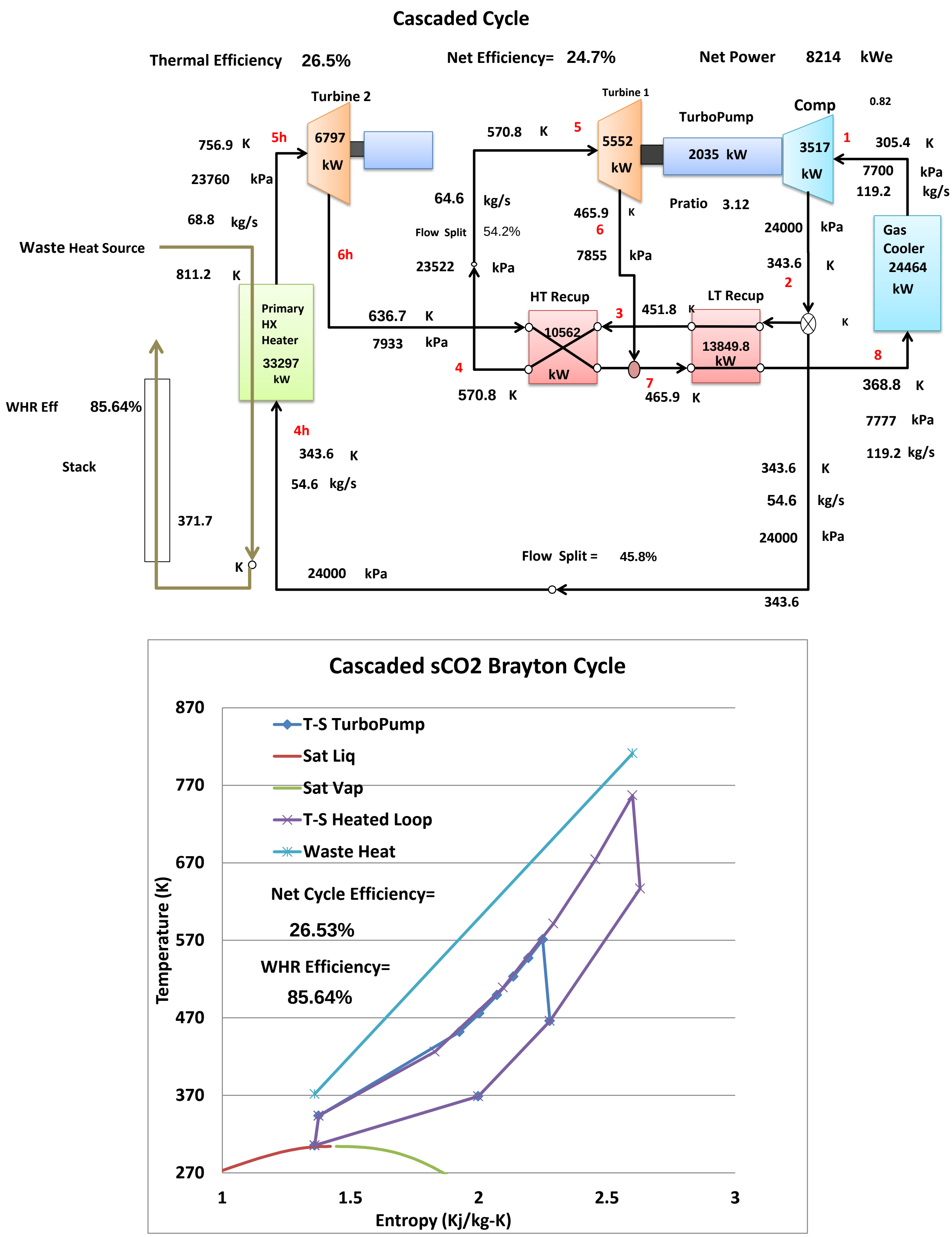

Figure 2: Process Flow Diagram for the Cascaded Brayton Cycle used for Waste Heat Recovery. The T-S curve for the power cycle and the heat source glide temperature is shown in the lower image.

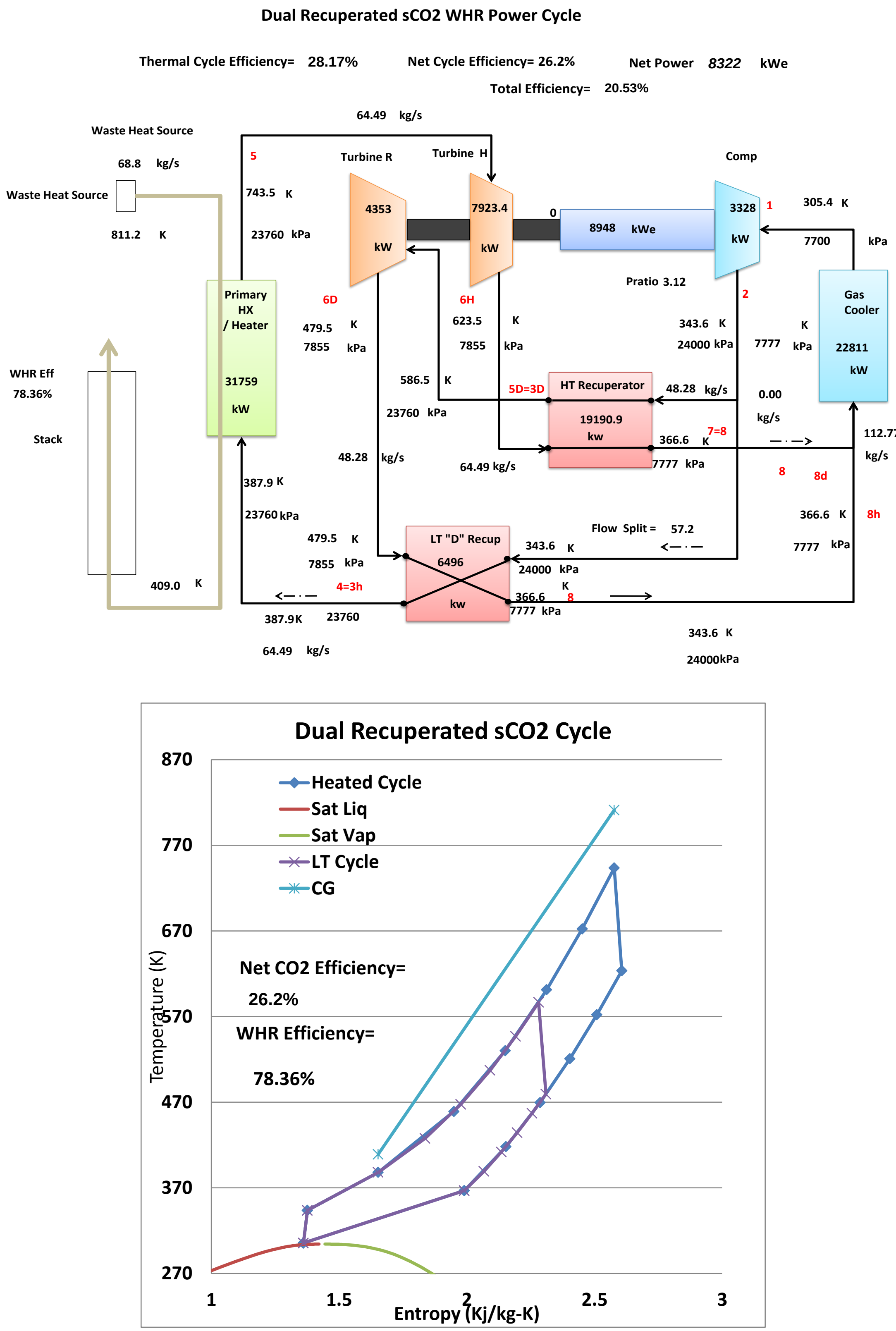

Figure 3: Process Flow Diagram for the Dual Recuperated sCO2 Waste Heat Recovery Power Cycle. Bottom image shows the heat source glide temperature and the T-S curve for this WHR power cycle.

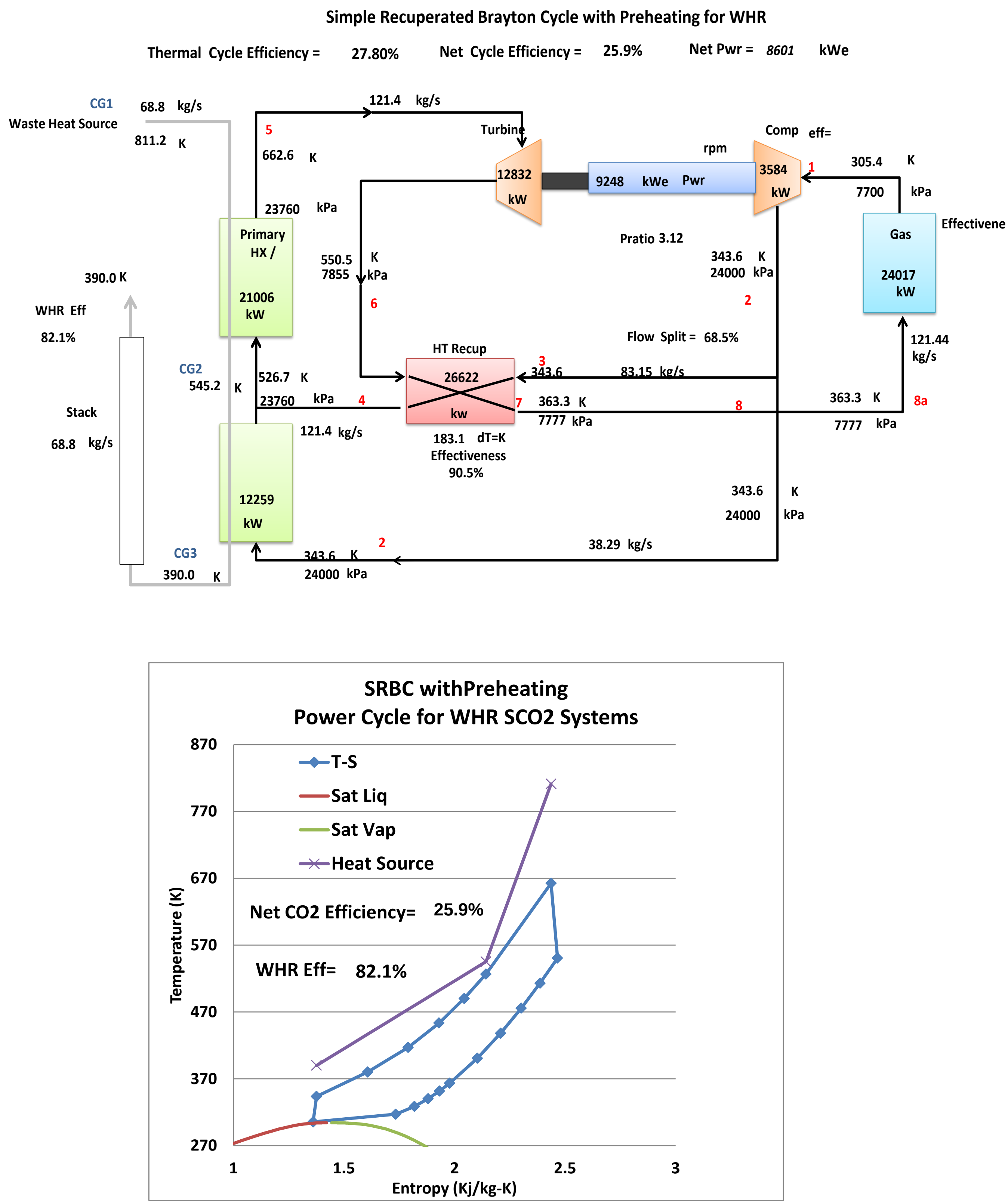

Figure 4: Process Flow diagram for the SRBC with Preheating sCO2 Waste Heat Recovery Power Cycle. Bottom image shows the heat source glide temperature and the cycle T-S curve illustrating the improvement on WHR efficiency for this cycle.

Heat Exchanger and Turbomachinery plus BOP Cost Basis

Because the primary goal of the economic analysis is to maximize the revenue of the sCO2 power plant by optimizing size of the heat exchangers the analysis grouped the costs into two categories:

1) the heat exchanger costs, and

2) the turbomachinery cost plus auxiliary and balance of plant hardware costs.

The heat exchangers include the primary heater, the preheater, both the high temperature and low temperature recuperators, and the CO2 chiller. The report specifically identifies the impact of costs to each of these heat exchangers. The heat exchanger costs uses a simple model that assumes the costs are proportional to U*A where U is the universal heat transfer coefficient (W/m^2-C) and A is the area (m^2). This is a convenient measure of the cost because U*A=Q/LMDT where Q is the heat transferred by the heat exchanger and LMDT is the log-mean-delta-Temperature difference across the heat exchanger which is predicted by the thermal energy balance analysis for each sCO2 power cycle. In this report we use the end-point temperatures to determine the LMDT, because the known costs for the heat exchangers are also based on the end-point LMDT. An alternative approach that would provide a better value for U*A would correct for the temperature distribution through the heat exchanger.

The turbomachinery plus auxiliary BOP components costs include the turbines, compressors, seals and bearings, gear box systems, generator, motors, variable frequency drives, piping, skids, Instrumentation and Control Systems, oil lubrication , oil cooling, and purge gas management systems, CO2 makeup systems, and a chill water cooling system. These costs are grouped into one cost among all of this turbomachinery and BOP hardware, and there is no break out of costs nor is there a penalty associated with systems requiring two turbines as in some of the sCO2 WHR power cycles. Thus, the two patented WHR systems that use two turbines will likely have less beneficial economics than presented here. Also, developer non-recoverable engineering costs are not included in these estimates, but the NRE costs from the OEM supplier are included. So the costs are more appropriate for a first-of-a -kind (FOAK) sCO2 system and not the n^{th} of a kind plant.

The cost basis for the various sCO2 power system components are listed in Table 4 and described below.

Recuperator

The recuperator specific cost are based on dollar cost per unit of U*A which has units of $/($kW_{th}$/K). The cost values used in in this report are $2500/($kW_{th}$/K) as listed in Table 4. This value is consistent with budget quotes that the authors have received for advanced high pressure recuperators of similar size. It is estimated that the uncertainty is +/- 30%. This estimate does not take into account cost reductions based on the purchase of multiple recuperators, economies of scale, or spreading the recuperator NRE over multiple recuperators. It also does not include NRE and profit from the supplier company.

sCO2 gas Chiller

The gas chiller heat exchanger is similarly based on tube and shell costs for water cooled CO2. It has specific cost units of $1700/($kW_{th}$/K), and is based on budget quotes for similarly sized heat exchangers. Again its uncertainty is +/- 30%. This cost value is also listed in Table 4.

Waste Heat Recovery Unit (WHRU or HRU)

The waste heat recovery heat exchanger specific costs are based on gas fired heaters that use API Standard 560 technology and can operate at the design pressures and temperatures. This technology is similar to that used for the Waste Heat Recovery heat exchanger Unit (WHRU), but has differences. For the WHR sCO2 power systems the WHRU will operate at a lower temperature differences between the waste-heat combustion gas and the CO2 working fluid than in direct fired gas heaters. This means that the piping material temperatures in the WHRU can operate at substantially lower temperatures. But, because of the lower dT in the WHRU a larger heat transfer area is expected. The larger area will tend to increase cost; but in contrast the lower material temperatures can use thinner wall tubing and less expensive steels, that will lower costs. Because these two effects counteract each other, it is reasonable to use a specific cost of \$5000/(kW_{th}/K) which is consistent with vendor quotes provided for the gas fired heaters in this size range. However, the uncertainty is larger. We estimated cost uncertainty to be approximately -50%/+30% depending largely on the maximum material temperature and pressure required in the Waste Heat Recovery sCO2 power system. This value is listed along with other specific costs in Table 4.

Turbomachinery plus other component BOP costs

The turbomachinery plus auxiliary BOP components costs include the turbines, compressors, seals and bearings, gear box systems, generator, motors, variable frequency drives, piping, skids, Instrumentation and Control Systems, oil lubrication , oil cooling and purge gas management systems, CO2 makeup systems, and a chill water cooling system. These costs do not delineate specific cost among this BOP hardware but instead are treated as a group. For a first-of-a-kind (FOAK) system the turbomachinery and auxiliary systems costs were estimated assuming that the turbomachinery costs were proportional to the net power produced. This value was selected to be 1000 \$/kWe, as shown in Table 4. It is expected that this value can be substantially reduced over time as a production line is established.

Table 4: Estimates of the component specific costs includes non-recoverable-engineering for FOAK systems. The uncertainty is estimated to be +/- 30%.

Component Description	Cost Units	Component Specific Costs
Recuperators (cost/UA)	\$/(kW.th/K)	2500
Fin Tube Primary Heater (cost/UA)	\$/(kW.th/K)	5000
Tube and Shell CO2-Chiller (cost/UA)	\$/(kW.th/K)	1700
Turbomachinery+Gen+Mtr+Gear+Piping+Skid+I&C+Aux.BOP	\$/kWe	1000

Summary of Performance and Economics

A summary of the thermo-economic analysis results are provided in Table 5 for the reference power cycle and for the three WHR power cycles. This table provides a substantial amount of detailed information that the reader may find useful, but the point of this report is to focus on the MAJOR conclusions not on the relative ranking in performance of one system over another. This is important because small changes in the analysis assumptions will easily switch the relative ranking of the sCO2 power types. The major results are discussed in two categories. The first category addresses the performance benefit achieved by using optimized operating conditions for the heat exchangers among the four sCO2 power systems. The second category examines the overall system performance and economics values. The most important summary information is provided in the bottom rows of Table 5, and in Figures 5 A-D in the form of charts illustrating the power improvements, the net annual revenue, the specific capital costs, and waste heat recovery electrical efficiency.

Figure 5A compares the total power produced by the four sCO2 systems. It is very clear that the three power systems designed specifically for WHR increase the power substantially from about 7 MWe to 8.2-8.6 MWe. This additional power (1.2-1.6 MWe) is able to increase the net annual revenue taking into account the additional costs of the heat exchangers. The annual revenue increases from $2.17 M$/year for the reference SRBC to $2.34-2.47 M$/year for the WHR power cycles (Figure 5B). To achieve this additional revenue the specific capital costs increases from $1.7/We to the $1.9-$2.0 $/We range (Figure 5 C).

In terms of efficiency, WHR efficiency is in the 80% range for the WHR power cycles and 61% for the reference case, while the net CO2 conversion efficiency is near 25% - 28% respectively. Thus, the sCO2 bottoming cycle converts the Gas Turbine waste heat to electricity at a value of 17.3% for the reference case and 20.5-22.2% for the WHR power systems (Figure 5D). The total combined cycle power system the net efficiency (at the generator) is estimated to 46.6% for the SRBC, but in the range of 48.5-49.1% for the WHR sCO2 power cycles as shown in Figure 6. The gas turbine alone has a net efficiency of 35.5%, therefore the increase to 46%-49% is very substantial and illustrates the economic improvement for these medium scale combined cycle power plants.

If the capital cost of the gas turbine is estimated to be 0.75$/We, while the bottoming cycle has a specific cost of 2$/We (see Table 5), the net specific cost of the combined cycle is $1.05/We for a system capable of producing a net 33 MWe with a net efficiency of 46- 49% which clearly illustrates the benefit of the sCO2 bottoming cycle when used on medium scale gas turbines.

In terms of heat exchanger performance Table 5 shows that the maximum annual net revenue is achieved by using primary heat exchangers with very high effectiveness values (up to 95%), in recuperators with effectiveness values that are generally in the low 90% values, and with chillers having lower effectiveness in the 74-82% ranges. The significance of this is that net revenue can be maximized using high effectiveness heat exchangers to provide the additional performance when amortized over the 20 year lifetime of the system.

Still, as mentioned before, the WHR systems produce 1.2-1.6 MWe of additional electric power compared to the reference case but at a total system capital cost increase of 3.7-5 M$ (see last row of Table 5). This means the rate of return is on the order of 18% for the SRBC, and 14-15.6% for the more complicated WHR power cycles. This indicates that the SRBC provides the greatest marginal performance increase for the least cost, compared to the specifically designed WHR sCO2 power cycles.

Table 5: Summary of the thermo-economic analysis results comparing the costs and performance of the four sCO2 WHR power systems. These include 1) the SRBC (Simple Recuperated Brayton Cycle, 2) the Cascaded cycle, 3) the Dual Recuperated cycle, and the SRBC Preheating cycle.

		SRBC	Cascaded	Dual Recup	Preheating
LM-2500PE	**Units**				
Waste Heat (Brochure) LM2500	kW.th	40,731	40,731	40,731	40,731
Waste Heat Combustion Model	kW.th	40530	40530	40530	40530
Mass flow rate thru Comp	kg/s	93.2	124.6	112.8	121.4
Max flow rate in Heater	kg/s	93.18	56	64.49	121.4
Efficiency of WHR		**61.2%**	**85.6%**	**78.4%**	**82.1%**
Net SCO2 Cycle Efficiency		**28.3%**	**24.7%**	**26.2%**	**25.9%**
Total Efficiency		**17.31%**	**21.13%**	**20.53%**	**21.22%**
Max Turbine Inlet T	K	685.0	756.9	743.5	662.6
Max Turbine Inlet T (C)	**C**	**411.8**	**483.8**	**470.3**	**389.5**
Stack Exit Temp (K)	K	497.1	372	409.0	390.0
Stack Exit Temp (C)	**C**	**223.9**	**99**	**135.9**	**116.8**
Total UA	**kW/K**	**1795**	**2807.98**	**2447**	**2966**
Recup UA	kW/K	630.5	1036.96	782.8	1226.76
Heater UA	kW/K	446.6	837.58	794.1	740.71
Chiller UA	kW/K	718.2	933.45	870.4	998.97
Recup Costs	$	1,576,314	2,592,403	1,957,006	3,066,890
Heater Costs	$	2,232,836	4,187,885	3,970,675	3,703,565
Chiller Costs	$	1,221,021	1,586,858	1,479,629	1,698,246
Total HX Costs		**5,030,171**	**8,367,146**	**7,407,310**	**8,468,702**
HEAT EXCHANGER EFFECTIVENESS					
Prim HT HX	%	94.6%	94.0%	95.0%	93.5%
Preheater HX	%				90.8%
HT Recup	%	94.0%	92.3%	83.1%	90.5%
LT Recup	%		88.4%	91.8%	-
CO2 Chiller	%	73.8%	82.3%	81.8%	81.2%
Closest Approach Temperature (K)	**(K)**	**10.7**	**18.0**	**21.1**	**18.0**
CC Heat Rate (GT only=9611 BTU/kWh)	**BTU/kWh**	**7323**	**7037**	**7012**	**6949**
Effective Revenue from Elect Sales	**$M/year**	**2.168**	**2.339**	**2.456**	**2.473**
Approx $/kWe Net	**$/kWe**	**1717**	**2019**	**1890**	**1985**
Net Elect. Power	**kWe**	**7017**	**8214**	**8322**	**8601**
Combined Cycle Total Efficiency	**%**	**46.6%**	**48.5%**	**48.7%**	**49.1%**
Total Capital Costs (FOAK)	**M$**	**12.047**	**16.581**	**15.729**	**17.070**
Rate of Return	**%**	**18.0%**	**14.1%**	**15.6%**	**14.5%**

Table 6: Summary of Combined Cycle Levelized Cost of Electricity (LCOE), using the four WHR bottoming cycles, and for the gas turbine alone.

	SRBC	Cascaded	Dual Recup	Preheating	Gas Turbine Only
LCOE at Fuel Costs of 3$/MMBTU	**$0.0294**	**$0.0291**	**$0.0289**	**$0.0289**	**$0.0342**
LCOE at Fuel 5$/MMBTU	**$0.0416**	**$0.0409**	**$0.0406**	**$0.0405**	**$0.0502**

Another, important point is that to achieve the increased annual revenue benefits, the system costs are sensitive to the primary heat exchanger costs having the highest specific costs and the highest required effectiveness (93.5-95%) for all of the systems. Thus, the cost of this WHR unit is very important, and more work is required to more accurately determine the size and cost of the primary WHR heaters.

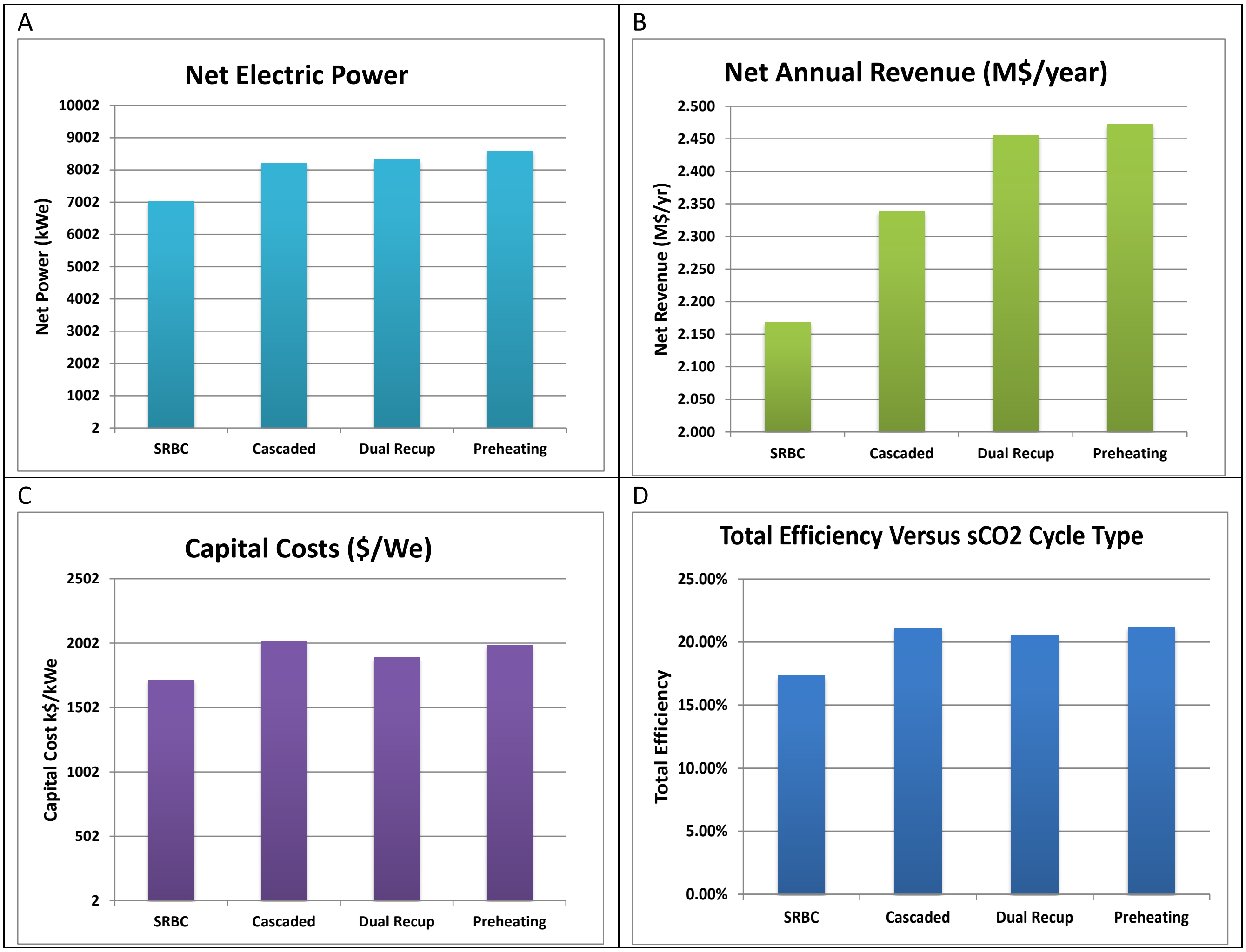

Figure 5: Bar charts showing the Net Electric Power, the Net Annual Revenue Generated, specific Capital Costs, and Total Efficiency (ratio of electrical power produced divided by total amount of waste heat) for the four sCO2 power system types.

Conclusions

A thermo-economic analysis was performed on four types of sCO2 power systems, including the simple recuperated Brayton cycle, the “Cascade” and “Dual Recuperated” cycles and the “Preheating” cycle. The last three cycles were specifically designed for use with WHR power applications. The WHR power systems maximize the net annual revenue produced by appropriately selecting the operating conditions of the sCO2 power system type, and by optimizing the amount of waste heat recovery that can be obtained. The waste heat recovery is illustrated the glide temperature curves for each of the four sCO2 power systems.

The major conclusions of this thermo-economic analysis are that the net cycle efficiency increases from 35.5% in the gas turbine to approximately 46.5% - 49% in the combined cycle, while the levelized cost of electricity (LCOE) in the combined cycle compared to the gas turbine alone is reduced by approximately $.01/kWhe for natural gas fuel costs of $5/MMBTU for all of the WHR bottoming cycles.

These values are illustrated in Table 2, Table 6, and in Figures 6A and 6B. These results are applicable when the sCO2 bottoming cycle is added to a medium sized gas turbine such as the LM-2500PE (see Table 2). To achieve this performance improvement the specific capital cost increases the gas turbine costs to about $1.05 /We assuming $2/We for the sCO2 bottoming cycle and $0.75/We for the gas turbine.

Other important conclusions are:

- The sCO2 power systems designed for WHR all increase the net power produced by 1.2-1.6 MWe compared to the simple recuperated Brayton cycle.
- The WHR sCO2 power systems as a group all increase net annual revenue by about the same amount, about 12-15% above the reference SRBC case.
- The cost of all the heat exchangers (recuperators, primary heat exchangers, gas chillers, and preheaters are about 40-50% of the total system costs.
- In terms of maximizing net annual revenue, it makes economic sense to pay for the additional cost for heat exchangers having high effectiveness (93% -95%), provided one is willing to accept an internal rate of return of 8-10%.
- The waste heat recovery efficiency is lowest for the SRBC (61.2%) while it is highest for the WHR sCO2 power systems 78.4%-85.6%.
- The net conversion efficiency of electricity from the heat in the CO2 is 28.2% for the SRBC and near 25% for the WHR sCO2 power cycles.
- The marginal rate of return is maximized by using the SRBC which has the fewest number of components among all the sCO2 cycles analyzed (~18% versus ~15%). Note that conclusion requires a very effective primary HX in the SRBC, and offsetting the loss in WHR efficiency by using a close approach temperature of 10 C for the recuperator to keep the net CO2 conversion efficiency near 28%.

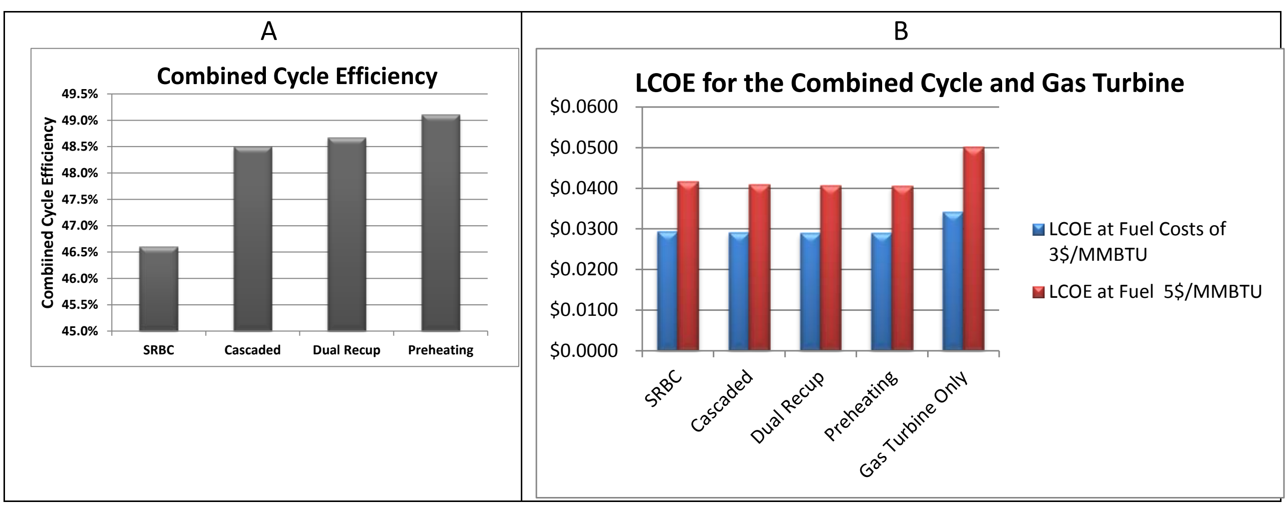

Figure 6: Bar chart A shows the approximate combined cycle efficiency for a LM-2500PE gas turbine using four types of sCO2 bottoming cycles. Bar chart B show the levelized cost of electricity at $3/MMBTU and $5/MMBTU fuel costs for the combined cycle plants and the gas turbine plants alone.

In conclusion, there is sufficient economic justification to warrant the development of sCO2 bottoming cycles, for the intermediate scale gas turbines. It is likely that ultimate decision will be based on the type of business that the potential sCO2 power plant customer has (utility, Original Equipment Manufacturer (OEM), end-user, military, or investor). In addition the market competition and the ability of the sCO2 combined cycle power plant to expand the market segment, and offer energy efficiency with lower environmental consequences will also play strong roles in the development/investment decision.

REFERENCES

1) Grant Kimzey, "Development of a Brayton Bottoming Cycle Using Supercritical Carbon Dioxide as the Working Fluid, Electric Power Research Institute Report, Palo Alto (CA), 2012.
2) Grant Kimzey, http://www.swri.org/utsr/presentations/kimzey-presentation.pdf
3) .David Montgomery, Caterpillar ARES Program Manager, "An Otto Rankine Combined Cycle for High Efficiency Distributed Power Generation", https://www.erc.wisc.edu/documents/symp09-Montgomery.pdf, June 10, 2009.
4) Campanari S., Macchi E. "Thermodynamic analysis of advanced power cycles based upon solid oxide fuel cells, gas turbines and Rankine bottoming cycles", ASME paper n. 98-GT-585, International Gas Turbine & Aeroengine Congress & Exhibition, Stoccolma, Giugno 1998.
5) Campanari S., Lozza G., Macchi E. "Analisi comparativa di cicli termodinamici ad acqua, aria ed ammoniaca per il recupero di calore da correnti gassose", Atti del VIII convegno nazionale "Tecnologie e Sistemi Energetici Complessi – Sergio Stecco", pp. 49-68, Bologna, giugno 1995.
6) "Fast, Flexible Power" Brochure, Aeroderivative Product and Service Solutions, GE Power & Water. www.ge-energy.com/aero , 2015.
7) Lemmon, E.W., Huber, M.L., McLinden, M.O. NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.1, National Institute of Standards and Technology, Standard Reference Data Program, Gaithersburg, 2013.
8) Microsoft Excel (2010), Version: 14.0.7166.5000 (32-bit) Microsoft Corporation, 2010.