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A Consideration for Trading Regenerator Size with Turbine Improved Efficiency in SCO_2 Systems to Enable a More Economical SCO_2 System

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ABSTRACT

There is considerable enthusiasm in the utility power industry over the very high efficiencies that are attainable with supercritical CO_2 systems. The high efficiencies are achieved with turbocompressor units that have been identified to be almost an order of magnitude smaller in size compared to their comparably-powered steam turbine systems. However, the size of the high and low temperature recuperator heat exchangers in a recompression-type SCO_2 cycle, with an advancement of a reheat turbine, shown in Figure 1a, requires effectiveness to be as high as 95-98%, to achieve the high efficiencies that have been promoted for the SCO_2 cycle. It would seem contrary to the stated goals of the SCO_2 power plant to achieve the desired reduction in size of the turbine-compressor but still require very large and expensive, high pressure recuperator heat exchangers. This paper presents some analyses to support the hypothesis that the goal of high cycle efficiency and reduced size for SCO_2 System Power Generation Systems can be aided by “trading” improved turbomachinery efficiency for regenerator effectiveness. It is further asserted that turbomachinery efficiency can be improved by utilizing aero designs that utilize the Additive Manufacturing’s Direct Metal Laser Sintering capabilities to form homogeneous, high strength metals into aerodynamic, more efficient designs.

INTRODUCTION

A supercritical CO_2 system operates on the same principle as a Brayton cycle but operates above the critical point of the working fluid. As the name implies, the working fluid is CO_2. Thus the operating pressures must be above 1070 psi (~72 bar,a) and 88°F: the critical pressure and temperature for CO_2. A primary benefit of the SCO_2 system is the ability to operate at extremely high thermal efficiencies. To attain these high efficiencies, the SCO_2 system must operate at very high energy source temperatures that are typically above 700°C. Such source temperature is expected to be derived from nuclear or concentrated solar energy power plants. Another major benefit of a supercritical system is the compactness of the CO_2 turbine and compressor, compared to steam turbines and natural gas-fired, gas turbine engines at utility-size power ratings of 250 MW or higher. Fortunately, an increase in the efficiency of the turbomachinery does not disproportionately increase the size of the compressor or turbine as it does for the recuperators.

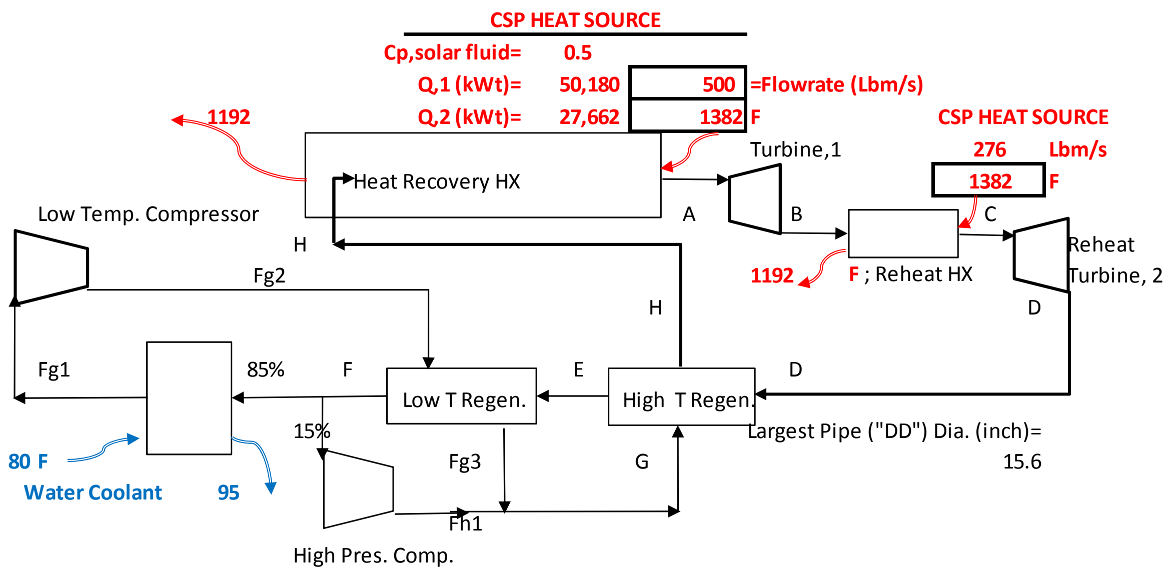


Figure 1a. Recompression SCO_2 Cycle but with a Reheat Turbine Sub-System

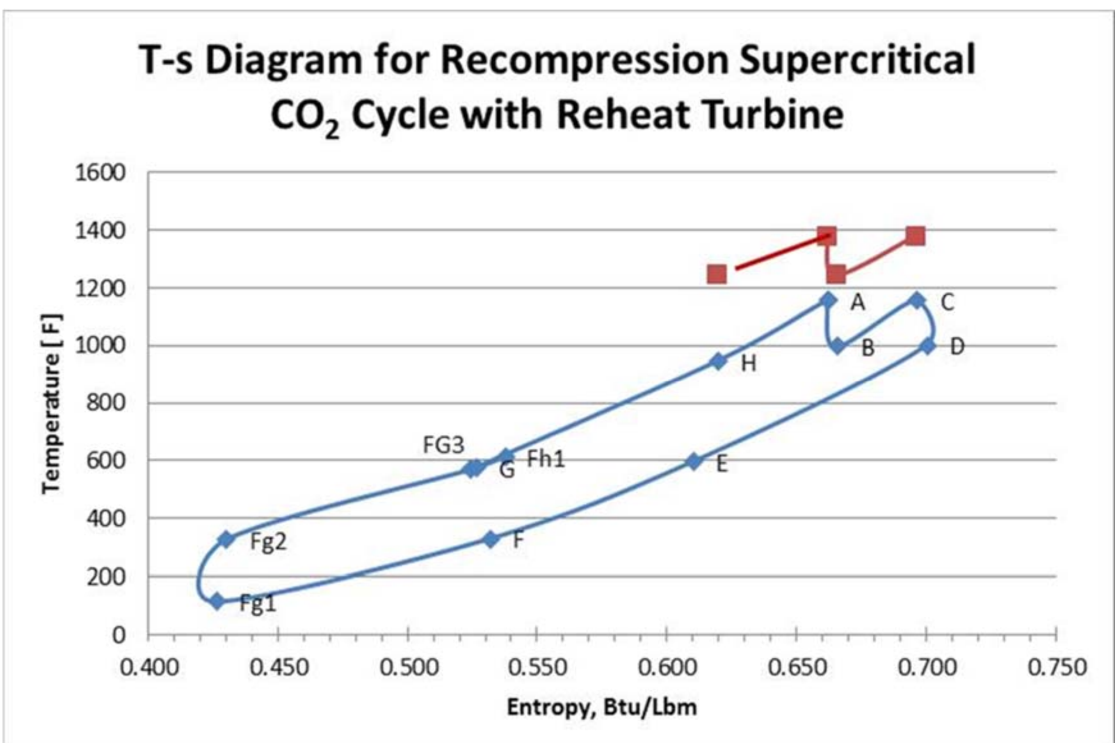


Figure 1b. T-s Diagram for Figure 1 Recompression System with Reheat Turbine

The T-s diagram for the cycle shown in Figure 1a is provided in Figure 1b. The complete property table for a recompression SCO_2 cycle case study, with an advancement of a reheat turbine system is shown in Figure 2. The T-s diagram includes the temperature profiles of the heat source that are superimposed on the same T-s diagram. The recompression cycle uses two compressors, two recuperators, and, usually, just one turbine. The two recuperators help to improve the efficiency of the cycle by preheating the working fluid to a very high temperature before it enters the primary SCO_2 heater. The function of a recompressor that operates in parallel with the main compressor, but before the cooling of the SCO_2, is to utilize the work of compression required of the recompressor to increase the temperature of the SCO_2 as it enters the high temperature recuperator. This heating increases the temperature of the SCO_2 fluid as it enters the SCO_2 heater. The consequence is a higher cycle efficiency. However, the increase in the cycle efficiency requires that the flow rate of the heat source be increased to make available the same amount of heat energy into the heater. Without increasing the flow rate of the heat source, the discharge temperature of the heat source, as it passes through the heater, would be reduced and, ultimately, be limited by the inlet temperature to the heater.

It may be observed from Figure 3 that the heat source temperature profile is constrained by the temperature of the working fluid that enters the CO_2 heater. It may be shown that to achieve the highest cycle efficiency, the temperature of the CO_2 entering the heater must be as high as possible. This high temperature imposes a constraint on the discharge temperature of the heat source. Thus, to generate the maximum amount of power, it is necessary to have a small change in temperature of the heat source and to have a very high fluid flow rate to produce large power. This constraint limits the type of heat sources that can be candidates for providing the heat input to the SCO_2 cycle via the heater. The need for the heat source of an SCO_2 system to have high temperature and only a small change in temperature in which to recover the heat energy and a considerable amount of flow rate, is found to be only available with a nuclear and a concentrated solar power system. For these heat sources, the engineer has control of the magnitude of heat source flow rate and the temperature difference through which this heat is recovered. Thus, the case study shown in Figure 2 applies an SCO_2 system to a concentrated solar collector application using molten sodium salt heat transfer fluid.

As may be observed in Figure 2, the flow rate of the molten salt, its temperature, and specific heat are entered into the computer (Excel) model of the SCO_2 recompression system. The reheat turbine is assumed to be able to utilize a flow rate through the reheater that has the same temperature and a slightly different flow rate than used in the primary SCO_2 heater. The system can generate a net 27,330 kW from 838 Lbm/s of 600°C molten salt heat transfer fluid. The cycle efficiency is observed to be 43.6%. But, note the very high effectiveness and larger sizes of the high and low temperature recuperators: 96% and 99% effective with UA's equal to 11.5 x 106 and 6.2 x 106 Btu/Hr/°F, respectively.

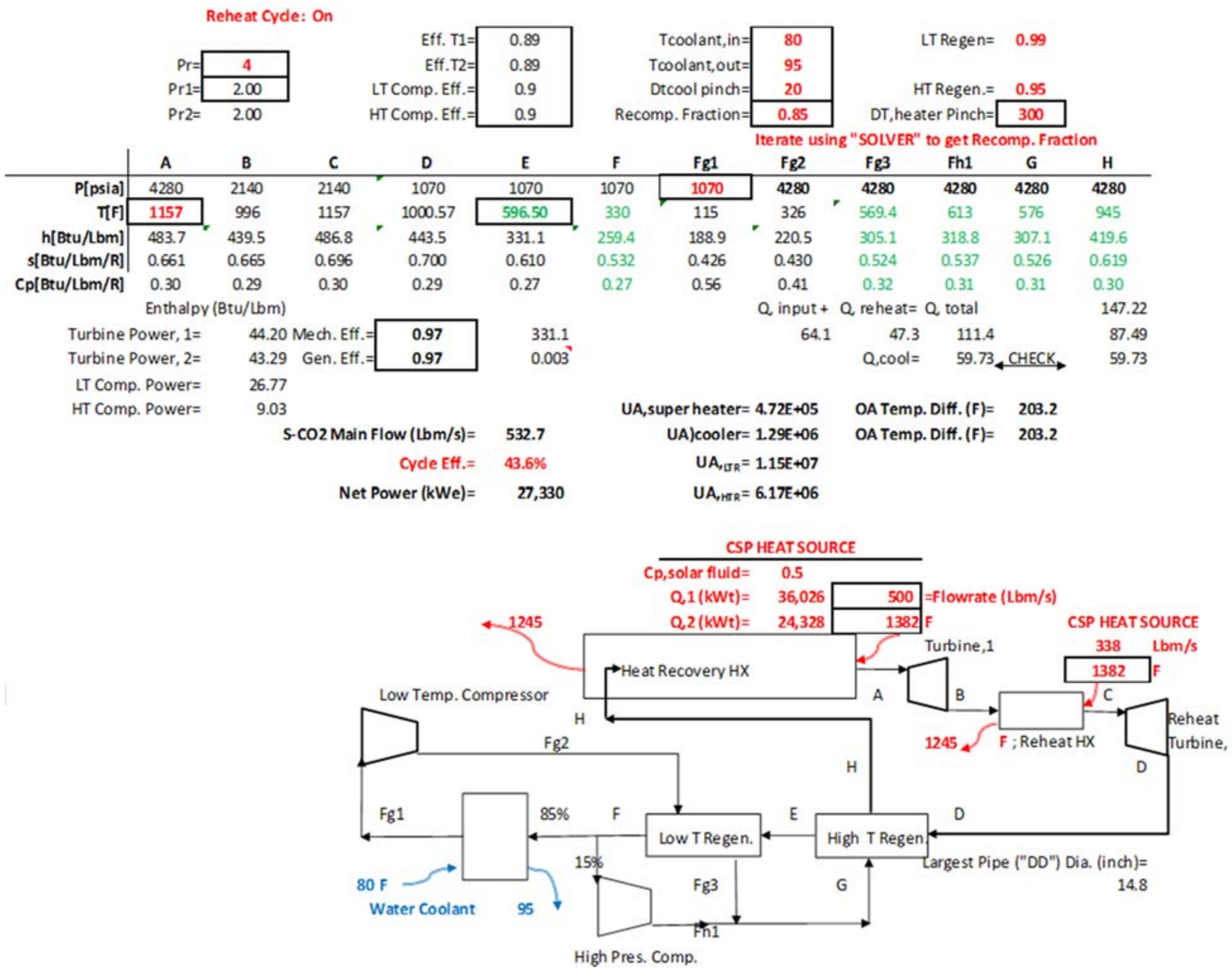
It is interesting to determine the performance of the SCO_2 system with recuperators that have been reduced in size by 30%. Figures 3a and 3b present the state points for the same cycle shown in Figure 2. However, in Figure 3a, the recuperator effectiveness has been reduced to result in the surface area to be reduced by 30%. With a reduction in the recuperator sizes, the cycle efficiency is reduced by only 10 percent and the power output is even higher at 29,000 kWe. The power is increased, due to a larger heater! Clearly, the cycle efficiency and the power are related to the sizes of the heat exchangers. Figure 3b is identical to Figure 3a, except the recuperator size is reduced by an additional 17%, but the efficiency of the compressor and turbine are increased by only 2%, to keep the overall cycle efficiency the same as before, at 39.3%.

In summary, comparing regenerator sizes and compressor-turbine efficiencies for Case 1 and 2 in Figures 3a and 3b:

UA, low temp. regen: 2.8 MBtu/hr/R vs. 2.33 MBtu/hr/R

UA, high temp. regen: 2.16 MBtu/hr/R vs. 1.67 MBtu/hr/R

A reduction in regenerator size of 17.3%, by improving turbine and compressor efficiency by only 2.2%.



	A	B	C	D	E	F	Fg1	Fg2	Fg3	Fh1	G	H
P[psia]	4280	2140	2140	1070	1070	1070	1070	4280	4280	4280	4280	4280
T[F]	1157	996	1157	1000.57	596.50	330	115	326	569.4	613	576	945
h[Btu/Lbm]	483.7	439.5	486.8	443.5	331.1	259.4	188.9	220.5	305.1	318.8	307.1	419.6
s[Btu/Lbm/R]	0.661	0.665	0.696	0.700	0.610	0.532	0.426	0.430	0.524	0.537	0.526	0.619
Cp[Btu/Lbm/R]	0.30	0.29	0.30	0.29	0.27	0.27	0.56	0.41	0.32	0.31	0.31	0.30

Figure 2. BASELINE Cycle State Points using Concepts NREC's Computer-Validated Cycle Model of Recompression SCO_2 System with Reheat Turbine. Note: 1. Turbine and Compressor Efficiencies: 89% and 90%. 2. High and Low Temp. Regenerator Effectiveness: 99% and 95%, and 3. Cycle Efficiency = 43.6%

Reheat Cycle: On

Pr=	4	Eff. T1=	0.89	Tcoolant,in=	80	LT Regen=	0.86
Pr1=	2.00	Eff.T2=	0.89	Tcoolant,out=	95		
Pr2=	2.00	LT Comp. Eff.=	0.9	Dtcool pinch=	20	HT Regen.=	0.81
		HT Comp. Eff.=	0.9	Recomp. Fraction=	0.85	DT,heater Pinch=	300

Iterate using "SOLVER" to get Recomp. Fraction

	A	B	C	D	E	F	Fg1	Fg2	Fg3	Fh1	G	H
P[psia]	4280	2140	2140	1070	1070	1070	1070	4280	4280	4280	4280	4280
T[F]	1157	996	1157	1000.57	705.00	381	115	326	629.2	674	636	909
h[Btu/Lbm]	483.7	439.5	486.8	443.5	360.6	273.1	188.9	220.5	323.7	337.6	325.8	408.8
s[Btu/Lbm/R]	0.661	0.665	0.696	0.700	0.636	0.548	0.426	0.430	0.542	0.554	0.544	0.611
Cp[Btu/Lbm/R]	0.30	0.29	0.30	0.29	0.27	0.27	0.56	0.41	0.31	0.31	0.31	0.30

Enthalpy (Btu/Lbm)

Q, input + Q, reheat= Q, total 158.83

75.0 47.3 122.3 87.49

Q,cool= 71.38 CHECK 71.34

Turbine Power, 1=	44.20	Mech. Eff.=	0.97	360.6
Turbine Power, 2=	43.29	Gen. Eff.=	0.97	-0.042
LT Comp. Power=	26.77			
HT Comp. Power=	9.80			

S-CO2 Main Flow (Lbm/s)= 575.6

Cycle Eff.= 39.2%

Net Power (kWe)= 29,090

UA,super heater= 5.96E+05 OA Temp. Diff. (F)= 216.3

UA)cooler= 1.46E+06 OA Temp. Diff. (F)= 216.3

$UA_{,LTR}$= 2.80E+06

$UA_{,HTR}$= 2.16E+06

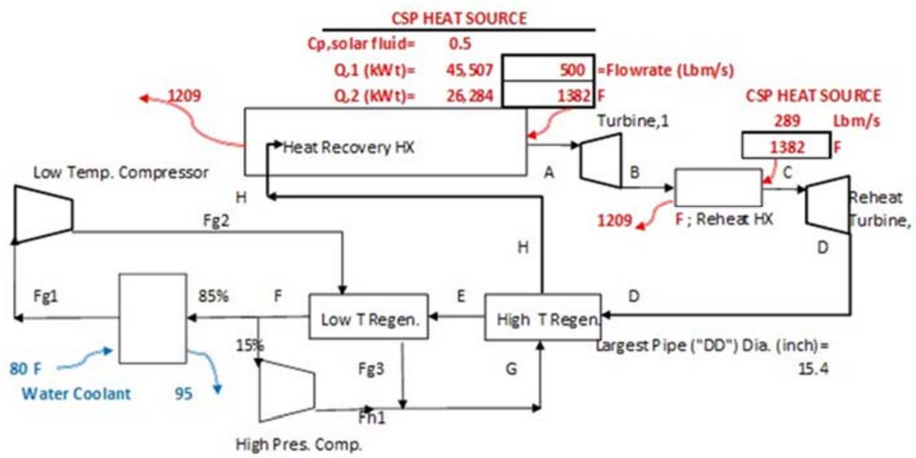


Figure 3a. CASE 1: SCO_2 Cycle with the Same Turbomachinery Efficiencies as shown in Figure 2 BUT with 78% Smaller Regenerator Results in only a 10% drop in Cycle Efficiency to 39.2%

Reheat Cycle: On

	Eff. T1=	0.91	Tcoolant,in=	80	LT Regen= 0.82
Pr= 4	Eff.T2=	0.91	Tcoolant,out=	95	
Pr1= 2.00	LT Comp. Eff.=	0.92	Dtcool pinch=	20	HT Regen.= 0.76
Pr2= 2.00	HT Comp. Eff.=	0.92	Recomp. Fraction=	0.85	DT,heater Pinch= 300

Iterate using "SOLVER" to get Recomp. Fraction

	A	B	C	D	E	F	Fg1	Fg2	Fg3	Fh1	G	H
P[psia]	4280	2140	2140	1070	1070	1070	1070	4280	4280	4280	4280	4280
T[F]	1157	993	1157	997.18	731.50	397	115	324	637.8	689	646	892
h[Btu/Lbm]	483.7	438.5	486.8	442.5	367.9	277.5	188.9	219.8	326.4	342.2	328.8	403.4
s[Btu/Lbm/R]	0.661	0.665	0.696	0.699	0.643	0.554	0.426	0.429	0.544	0.558	0.546	0.607
Cp[Btu/Lbm/R]	0.30	0.29	0.30	0.29	0.28	0.27	0.56	0.41	0.31	0.31	0.31	0.30

Enthalpy (Btu/Lbm)

					Q, input +	Q, reheat=	Q, total		164.61
Turbine Power, 1=	45.19	Mech. Eff.=	0.97	367.9	80.3	48.3	128.6		89.46
Turbine Power, 2=	44.26	Gen. Eff.=	0.97	0.006		Q,cool=	75.15	CHECK	75.15
LT Comp. Power=	26.19								
HT Comp. Power=	9.82								

		UA,super heater=	6.57E+05	OA Temp. Diff. (F)=	233.7
S-CO2 Main Flow (Lbm/s)=	592.4	UA)cooler=	1.53E+06	OA Temp. Diff. (F)=	233.7
Cycle Eff.=	39.1%	$UA_{,LTR}$=	2.33E+06		
Net Power (kWe)=	31,427	$UA_{,HTR}$=	1.67E+06		

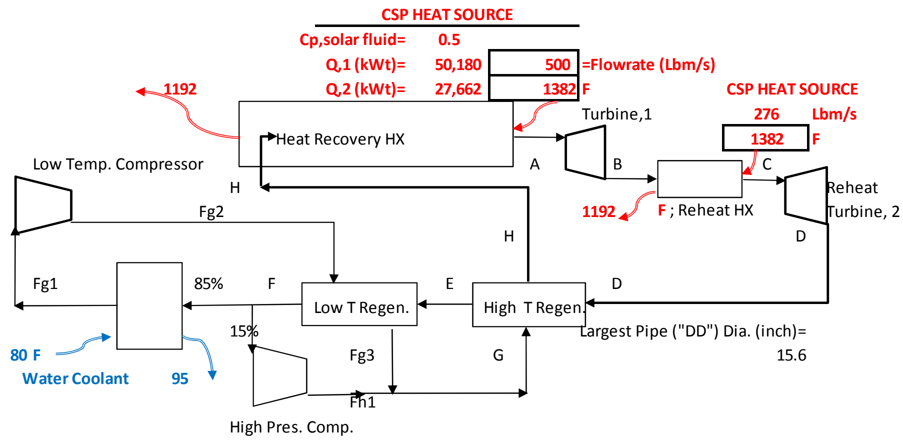


Figure 3b. CASE 2. Optimize Case 1 by Increasing Compressor and Turbine Efficiencies to 91% and 92% but Reducing Regenerator Effectiveness until Cycle Efficiency is the same as Case 1

A more general parametric study has been completed, using an alternative SCO_2 computer model developed by James Pasch, PhD (Sandia National Laboratories). The parametric study varied the turbine efficiency and the effectiveness of the high temperature recuperator (HTR) and calculated the net effect on the cycle efficiency in the more traditional recompression SCO_2 cycle. The Recompression Cycle (without the reheat turbine sub-system) is shown in Figure 4.

Figures 5a and 5b present the result of the parametric study that demonstrates the effect of changes in the effectiveness of the HTR and the turbine on the cycle efficiency. For this study, the HTR effectiveness was varied from 90% to 97% and the turbine efficiency was varied from 88% to 92%. An HTR effectiveness of 90% and a turbine efficiency of 80% was used as the data basis for the results shown in Figures 5a and

5b. The data output is given in Table I in the Appendix of this Study. It may be concluded from a study of Figures 5a and 5b, that a large increase in the recuperator size only provides a disproportionate, small increase in the efficiency of the cycle. For example, from Figure 5a, it may be observed that a 50% increase in the HTR recuperator only provides an increase in the cycle efficiency of 8%. However, from Figure 5b, a projection of the 90% HTR effectiveness curve, indicates that only an 11% increase in the turbine efficiency or an individual increase of only 4% for the turbine and the compressor can achieve the same 8% cycle efficiency.

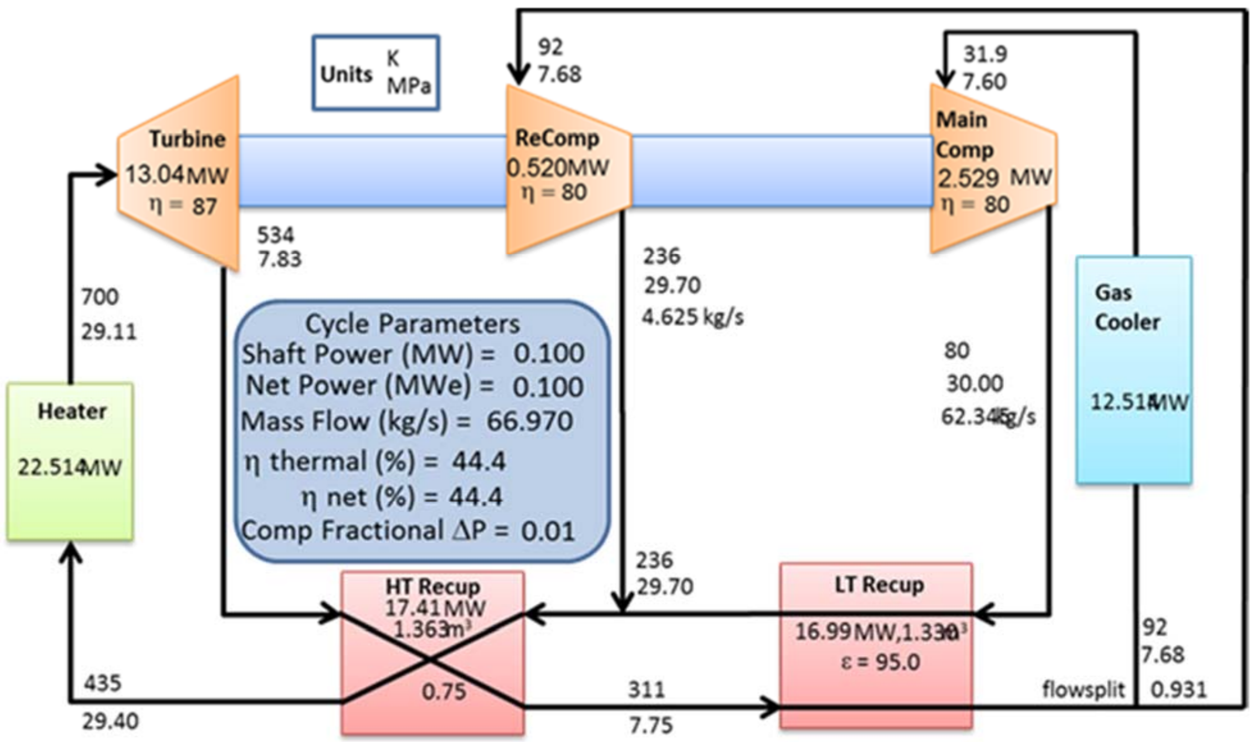


Figure 4. Sandia National Laboratories' Computer Output for Traditional Recompression SCO_2 Cycle.

It can be shown that the cost to increase the HTR recuperator size by 50% is considerably more than improving the efficiency of the compressor and turbine by only 4%. The later improvements can be aided by the advent of using Additive Manufacturing to improve the aerodynamic blade efficiencies of the compressor and turbine impellers. This is due to the 3-dimensional metal printing of the blade contours that may be difficult for the 5-axis machining manufacturing process.

RESULTS AND DISCUSSION

A thermodynamic cycle analysis of the SCO_2 system suggests that it may be more economically rational to improve the efficiencies of turbomachines rather than improving the efficiency of the HTR recuperator.

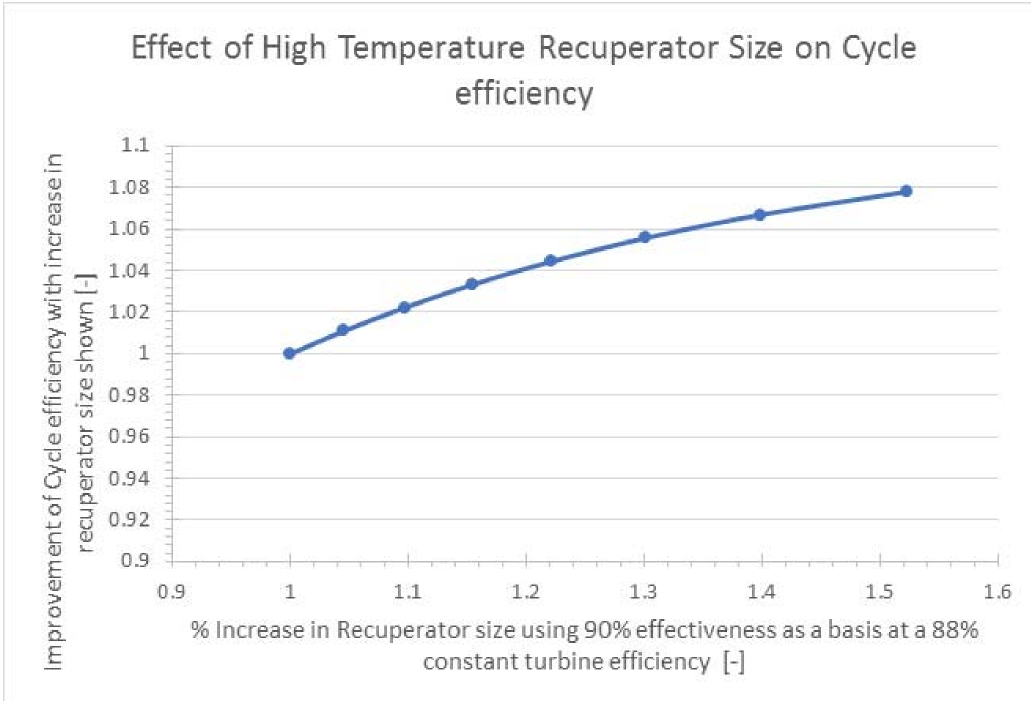


Figure 5a. Effect of HTR Size on Cycle Efficiency

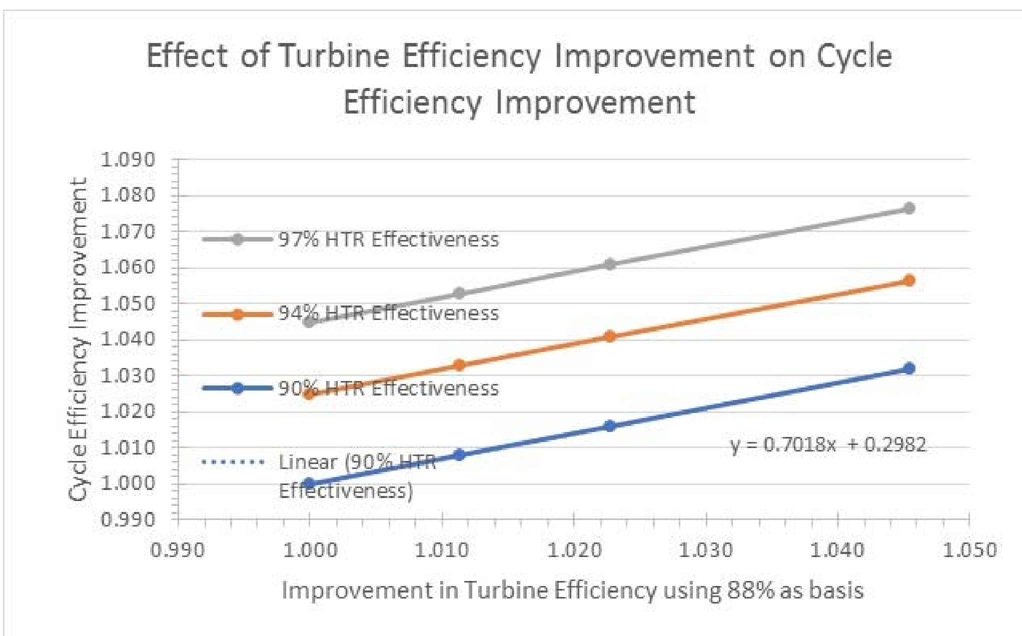


Figure 5b. Effect of Turbine Efficiency Improvement on Cycle Efficiency Improvement

Table I. Results from Parametric Study using Sandia National Lab. SCO_2 Computer Model: RETS© (Recompression Closed Brayton Cycle Engineering and Trade Studies) developed by Dr. James Pasch. The highlighted area in this Table was used to calculate Figures 5a and 5b.

HTR,effect.\tubine eff.	0.75	0.76	0.77	0.78	0.79	0.8	0.81	0.82	0.83	0.84	0.85	0.86	0.87	0.88	0.89	0.9	0.91	0.92	0.93	0.94
0.8	41.5235	41.9858	42.4425	42.8938	43.3364	43.777	44.2057	44.6393	45.0611	45.485	45.897	46.3078	46.7139	47.1155	47.5088	47.9015	48.2897	48.6699	49.05	49.4254
0.81	41.7286	42.1981	42.6556	43.1106	43.5569	43.9977	44.4365	44.8597	45.2917	45.7116	46.1302	46.5402	46.9492	47.3497	47.7456	48.1369	48.5238	48.9101	49.288	49.6661
0.82	41.948	42.4154	42.8768	43.3257	43.7824	44.2267	44.6656	45.0991	45.5201	45.9466	46.3645	46.7774	47.1854	47.5848	47.9871	48.3809	48.7702	49.1549	49.531	49.9073
0.83	42.1695	42.6312	43.1032	43.5593	44.013	44.4574	44.8997	45.3329	45.7643	46.1867	46.6076	47.0196	47.4266	47.8324	48.2334	48.6218	49.0174	49.4004	49.775	50.1571
0.84	42.3932	42.8656	43.3316	43.7915	44.249	44.697	45.1392	45.5757	46.0066	46.432	46.8521	47.263	47.6805	48.0852	48.4808	48.8796	49.2696	49.6549	50.036	50.4117
0.85	42.6258	43.0989	43.5655	44.0328	44.4905	44.9384	45.3841	45.8239	46.2581	46.6827	47.1058	47.5235	47.9319	48.3351	48.7414	49.1386	49.5268	49.9059	50.297	50.6712
0.86	42.8642	43.3415	43.8122	44.2726	44.7378	45.1893	45.6384	46.0739	46.5113	46.939	47.3651	47.7816	48.1928	48.6027	49.0033	49.3988	49.7849	50.179	50.56	50.9357
0.87	43.1051	43.5867	44.0614	44.5295	44.9872	45.4461	45.8949	46.3376	46.7743	47.2011	47.6302	48.0496	48.4592	48.8676	49.2707	49.6686	50.0613	50.4491	50.832	51.21
0.88	43.3559	43.8418	44.3169	44.7889	45.2543	45.7091	46.1535	46.6035	47.0435	47.4732	47.9013	48.3193	48.7316	49.1426	49.5482	49.9441	50.3392	50.7247	51.11	51.4807
0.89	43.6095	44.1035	44.5828	45.0549	45.5202	45.9786	46.4304	46.8758	47.3148	47.7433	48.1786	48.5994	49.0143	49.4236	49.8273	50.2256	50.6184	51.0061	51.384	51.766
0.9	43.8774	44.3643	44.8436	45.3278	45.7968	46.2548	46.7101	47.1588	47.5968	48.0326	48.4623	48.8859	49.2991	49.7109	50.1125	50.5085	50.8989	51.298	51.678	52.0573
0.91	44.1527	44.644	45.1275	45.6035	46.0722	46.5337	46.9881	47.4358	47.8856	48.3245	48.7527	49.1746	49.5949	50.0048	50.4087	50.807	51.2043	51.5915	51.973	52.3547
0.92	44.4316	44.9272	45.4149	45.895	46.3675	46.8327	47.2907	47.7418	48.186	48.6236	49.05	49.4746	49.893	50.3053	50.7116	51.1122	51.507	51.8962	52.275	52.6535
0.93	44.7224	45.2182	45.7102	46.1899	46.6663	47.1307	47.5922	48.042	48.4894	48.9253	49.3592	49.7818	50.2029	50.6128	51.0117	51.4145	51.8114	52.2027	52.588	52.9636
0.94	45.0173	45.5174	46.0137	46.4974	46.9731	47.4411	47.9015	48.3451	48.7957	49.2393	49.6712	50.0965	50.5202	50.9325	51.3387	51.7387	52.1328	52.521	52.893	53.2753
0.95	45.3161	45.8206	46.3165	46.7995	47.2883	47.7504	48.219	48.6754	49.1243	49.5612	49.996	50.4239	50.8402	51.2498	51.6633	52.0603	52.4564	52.8466	53.226	53.6044
0.96	45.6326	46.137	46.6372	47.1241	47.6074	48.0778	48.5401	48.9947	49.4467	49.8864	50.3239	50.7545	51.1731	51.5903	51.9957	52.3947	52.7876	53.1796	53.56	53.9355
0.97	45.9587	46.4675	46.9672	47.458	47.9403	48.4142	48.8749	49.3326	49.7878	50.2251	50.6602	51.0882	51.5091	51.9286	52.3254	52.7372	53.132	53.5095	53.897	54.2741

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