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AN ASSESSMENT OF SUPERCRITICAL CO2 POWER CYCLES INTEGRATED WITH GENERIC HEAT SOURCES

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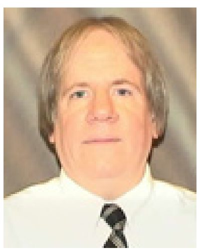

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Wally Shelton is an energy analyst supporting research/process development projects as a member of the NETL/DOE's Office of Project Performance and Benefits (OPPB). Focused on the analysis and development of processes based on fossil fuels spanning both existing and emerging technologies that produce electrical power or a combination of power and chemicals. Key aspects involve performing system process studies that determine the efficiency, economic, regulatory and environmental impacts. The process studies are targeted to contribute to formulating and determining government/public policy. Wally has a BS (SUNYAB) and MS (U. of Minnesota) in Chemical Engineering.

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ABSTRACT

This paper documents a detailed analysis of the supercritical CO_2 (sCO_2) recompression Brayton cycle to better understand at a fundamental level the dependence of the cycle performance on operating conditions. The focus of this study is on cycle performance, particularly efficiency. No cost estimates were performed. However, a number of indirect measures of cycle cost were examined to provide an indication of whether the operating conditions were in a region of great cost sensitivity.

In this study, a sCO_2 recompression cycle is described and a set of baseline operating conditions proposed. The heat source for the study was considered generic or agnostic. No temperature value, temperature range, or heat flux was assumed to limit cycle design. Rather the heat source was assumed to be at a sufficient temperature to provide the heat input specified.

The cycle was modeled in Aspen Plus® to determine stream conditions and heat and power duties. These were used to estimate the cycle efficiency. To better understand the dependence of the cycle performance on the operating conditions, a series of sensitivity analyses were performed to quantify the impact of perturbations on the overall cycle performance.

Sensitivity analyses were performed on the CO_2 cooler bypass fraction, the cycle pressure ratio, the turbine exit pressure, the compressor efficiency, the turbine efficiency, the cycle pressure drop, the minimum temperature approach, and the CO_2 cooler temperature. The sensitivity analyses served several purposes. To some degree, they provided greater insight into the underlying cause for the cycle's performance and the range under which the cycle might be considered advantageous, at least from a process efficiency standpoint. The sensitivity analyses also helped determine if any of the operating parameters were especially controlling of the cycle performance. Finally, the sensitivity analyses provided some quantitative basis for adjusting the operating conditions to achieve some cost and/or performance optimum.

Since a reliable cost estimate of the system was unavailable, a series of sensitivity analyses were performed on process variables and derived quantities that offer at least an indirect indication of process cost. The primary purpose of this analysis was to determine whether the analyses used to establish a high efficiency baseline configuration drove the operating conditions to a point where practical operation was infeasible or that the driving forces were unacceptably low. The study examined the volumetric flow rates into the turbine and main compressor, the recuperator duty and driving force, the hot source duty, the specific cycle power, and the total CO_2 flow rate. All of the sensitivity analyses showed that the baseline configuration fell on a reasonably flat and stable portion of the sensitivity curve to pressure ratio. This suggests that the baseline operating conditions are probably feasible and may not be too far removed from an economically optimum cycle configuration.

Introduction

DOE has recently shown an increased interest in the potential application of supercritical CO_2 (sCO_2) cycles for power generation. At the direction of NETL, a detailed thermodynamic analysis of the sCO_2 Brayton cycle was performed to better understand and quantify the potential advantages of such a power cycle. As part of that study, a series of sensitivity analyses were performed to help identify the range of applicability for the cycle. The major sensitivity variables were turbine inlet temperature, cycle pressure ratio, and turbo-machinery efficiencies.

This paper extends the earlier work for one candidate Brayton cycle configuration: namely the recompression Brayton cycle. In addition to the previously noted sensitivity variables, this study examines the impact of variations in the assumed cycle pressure drop, minimum approach temperature, and CO_2 cooler temperature. The primary performance metric for this study is overall cycle efficiency.

Of course, cost is a very important consideration in trying to identify an optimal configuration for a power cycle. However, limited design information is available for the key components of this system. In lieu of detailed design information, sensitivity analyses were conducted on a number of indirect measures of system cost to provide some indication of whether a highly efficiency cycle could also be cost effective.

sCO_2 recompression Brayton cycle

Figure 1 shows a simplified block flow diagram for a sCO_2 recompression Brayton cycle. In this cycle, high pressure CO_2 is heated to a specified temperature at the hot source denoted by point D. For this study, the hot source is generic and the heat input, Q_H, is specified. After heating, the CO_2 is expanded in a turbine to generate power. The lower pressure CO_2 then enters two stages of recuperation where it exchanges heat with the cold compressed CO_2. On exiting the recuperator, the low pressure CO_2 stream is split with a portion passing through a cooler. The cooled and bypass streams are both compressed before entering the cold side of the two stage recuperator. The bypass stream passes through the first stage only of the recuperator while the cooled stream passes through both stages.

For all configurations examined in this study, the CO_2 was supercritical. There were no heat losses other than the heat loss in the CO_2 cooler. The calculated efficiency does not include auxiliary power losses, mechanical drive losses, or generator losses.

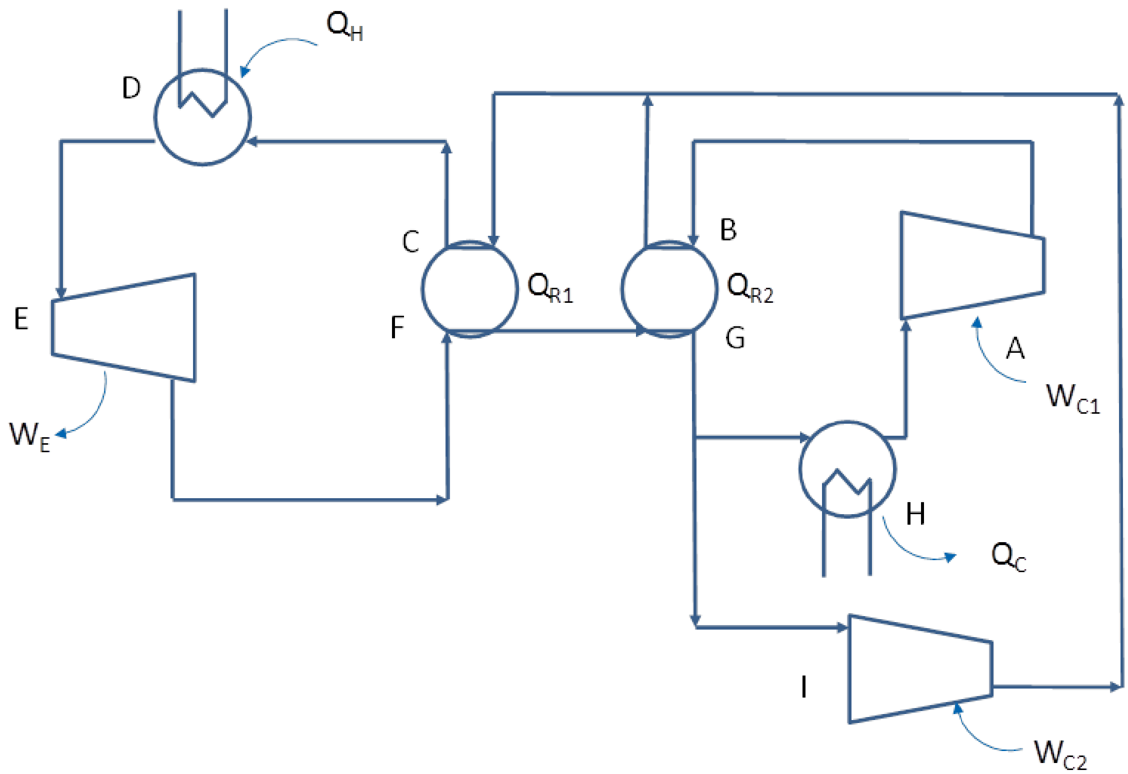


Figure 1 sCO_2 recompression Brayton cycle

Table 1 shows the parameters for the baseline cycle configuration. For the most part these baseline parameters are arbitrary and represent a reasonable starting point for the sensitivity analyses that examine how the cycle performance changes with changes to the parameter values.

The sCO_2 cycle was modeled using Aspen Plus®. CO_2 was the only species assumed in the system. The flow rate of CO_2 was determined in a Design Specification such that the specified turbine inlet temperature was attained. The distribution of heat duties between the two recuperator stages was calculated so as to attain the minimum approach temperature specification. The approach for determining the CO_2 cooler bypass fraction will be described in the following section. The pressure drop was divided evenly between the high pressure and low pressure portions of the cycle.

Table 1 Baseline parameters used in Aspen Plus® simulations

Parameter	**Value**
Heat source	Generic
Nominal thermal input	64 MMBtu/hr (18.8 MW)

Turbine exit pressure	1350 psia (93.1 bar)
Cooler exit temperature	35 °C (95 °F)
Turbine inlet temperature	700 °C (1292 °F)
Compressor isentropic efficiency	0.927
Turbine isentropic efficiency	0.85
Cycle pressure drop	60 psia (4.1 bar)
Minimum temperature approach	5.6 °C (10 °F)
Nominal compressor pressure	5100 psia (351.6 bar)
Nominal compressor pressure ratio	3.9
Nominal CO_2 cooler bypass fraction	0.283

Figure 2 shows a modified block flow diagram for the sCO_2 recompression Brayton cycle with stream labels included. Table 2 is a stream table with the results from the Aspen Plus® simulation of the baseline configuration.

Figure 2 Block flow diagram for the sCO_2 recompression Brayton cycle

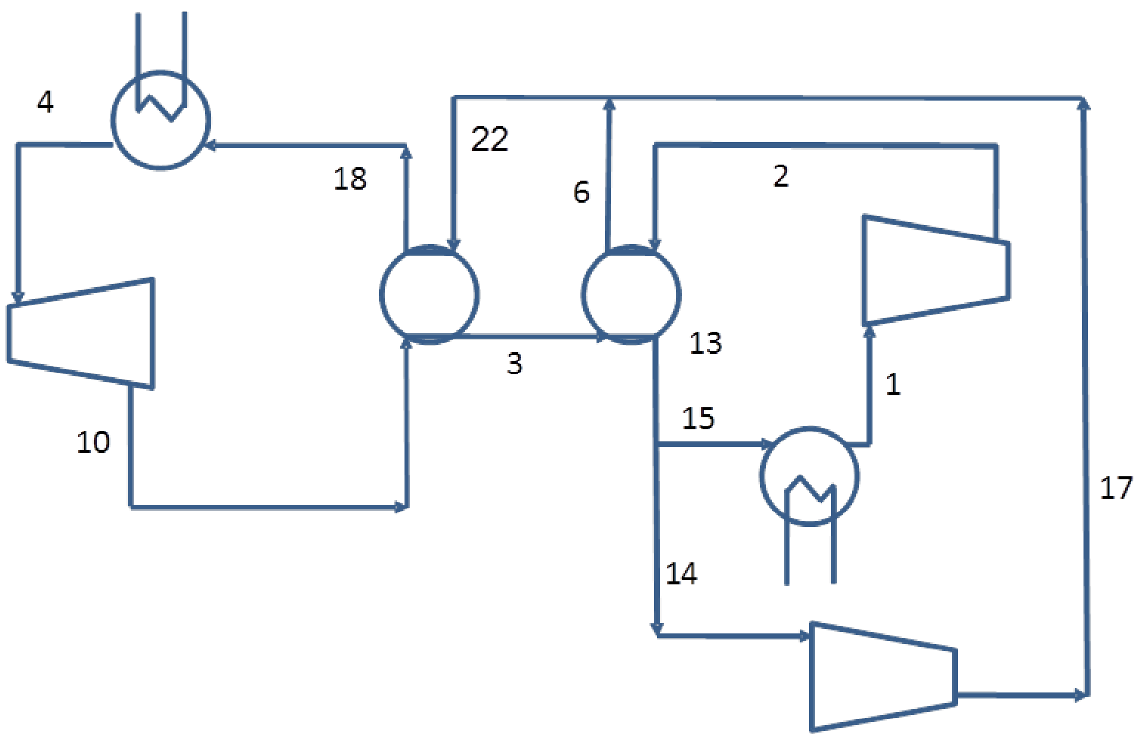


Table 2 Aspen Plus® model results for the baseline sCO_2 recompression Brayton cycle

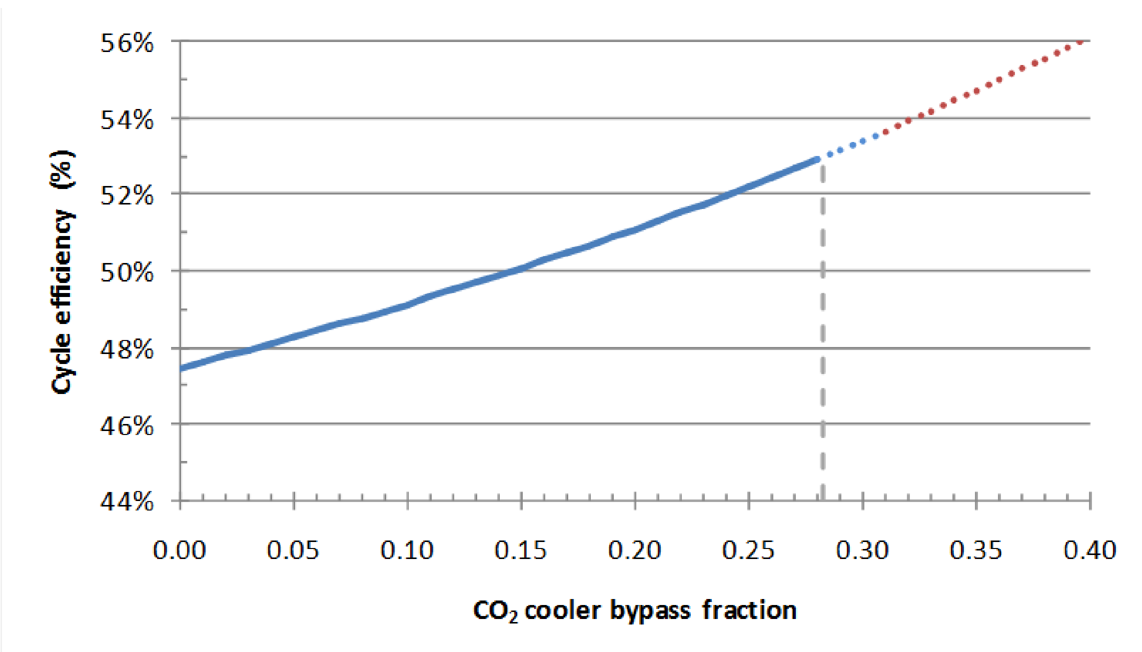
Stream label	1	2	3	4	5	6	7	8	9	10	11	12
Temperature (°F)	95	192.9	452.3	452.3	897.9	1292	978.9	462.3	202.9	202.9	202.9	452.3
Pressure (psia)	1320	5100	5100	5100	5070	5070	1350	1320	1320	1320	1320	5100
Vapor Frac	1	1	1	1	1	1	1	1	1	1	1	1
Mole Flow (lbmol/hr)	8,559	8,559	8,559	11,931	11,931	11,931	11,931	11,931	11,931	8,559	3,372	3,372
Mass Flow (lb/hr)	376,696	376,696	376,696	525,087	525,087	525,087	525,087	525,087	525,087	376,696	148,391	148,391
Volume Flow (ft^3/hr)	10,067	8,066	15,716	21,907	37,210	48,463	139,074	85,320	48,324	34,668	13,657	6,191
Enthalpy (MMBtu/hr)	-1479.74	-1472.52	-1432.04	-1996.16	-1919.66	-1855.25	-1903.45	-1979.95	-2020.43	-1449.45	-570.978	-564.119
Mole Flow (lbmol/hr)												
CO_2	8,559	8,559	8,559	11,931	11,931	11,931	11,931	11,931	11,931	8,559	3,372	3,372

A series of sensitivity analyses was performed in which each of the parameters in Table 1 except heat input was adjusted to determine the impact on cycle efficiency. The results of these sensitivity analyses are presented and discussed in the following sections.

Sensitivity to CO_2 cooler bypass fraction

Figure 3 shows a plot of cycle efficiency versus the CO_2 cooler bypass fraction. All other cycle configuration parameters except bypass fraction were the same as shown in Table 1.

Figure 3 Efficiency versus CO_2 cooler bypass fraction

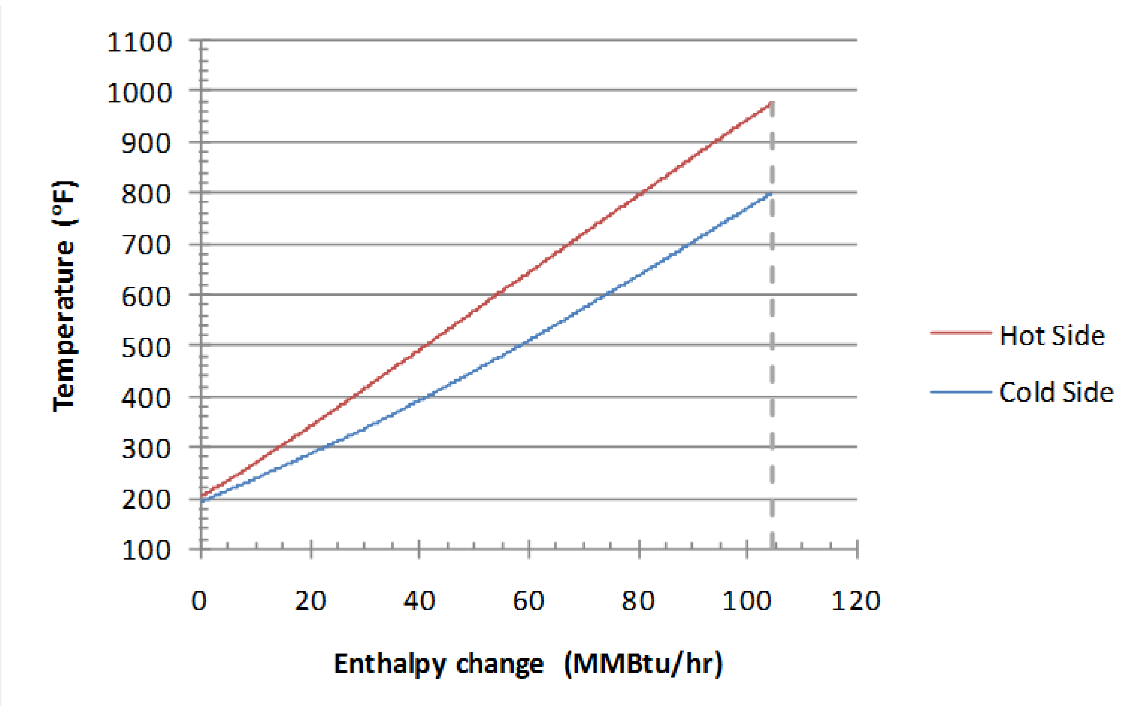


The bypass fraction was varied between 0.0 and 0.4. The plot shows that the cycle efficiency increases monotonically with an increase in the CO_2 cooler bypass fraction. That the cycle efficiency should increase with the bypass fraction is not an intuitive result. The bypassed CO_2 must be compressed to the same exit pressure as the cooled CO_2 but since its temperature is higher, the compression power will be greater on a per unit mass flow basis. This will act to lower the net cycle output and decrease efficiency. The reason for the increase in cycle efficiency is due to an increase in the recuperator effectiveness as the bypass fraction increases.

Although not shown on the plot, as the bypass fraction increases from zero, the hot side temperature approach in the second stage of the recuperator decreases. At a bypass fraction of 0.283 (depicted with a vertical line in Figure 3) the temperature approach reaches the target baseline minimum of 10 °F (5.6 °C). Further increases in the bypass fraction are feasible but result in temperature approaches below 10 °F (5.6 °C) (blue dotted line in Figure 3). At a bypass fraction of 0.31, the temperature approach is zero hence bypass fractions beyond this point are infeasible.

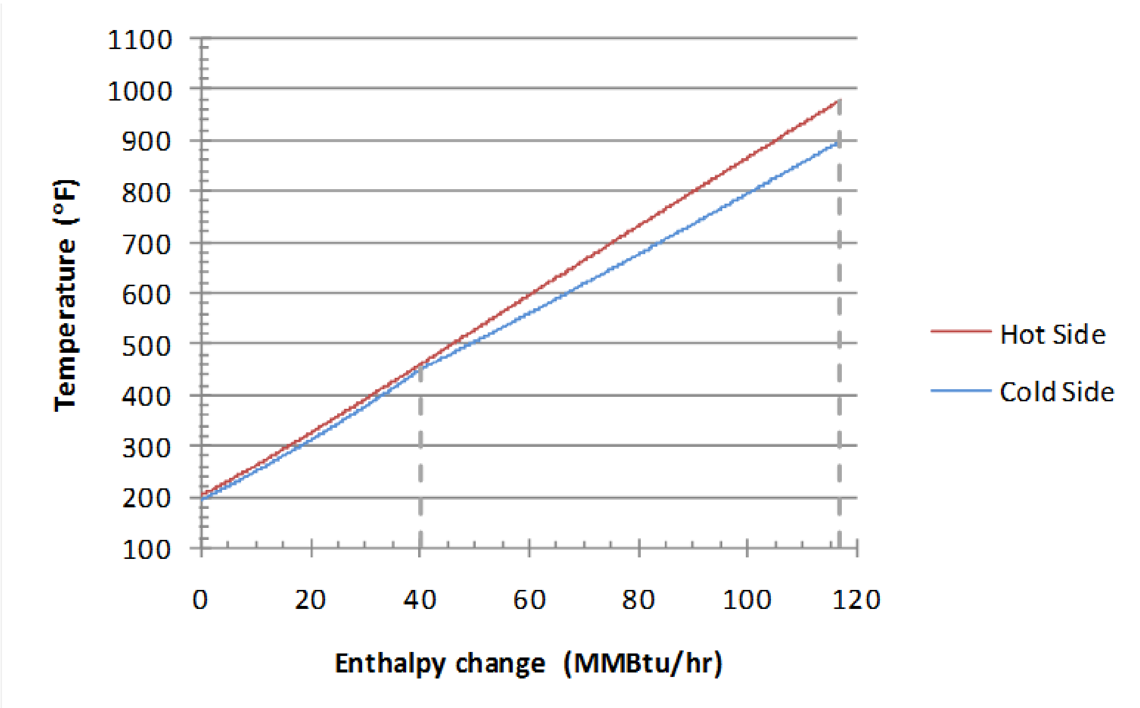
The increase in recuperator effectiveness results from the fact that for the baseline configuration, the thermal capacitance of the cold side CO_2 is significantly greater than the thermal capacitance for the hot side CO_2. This in turn is due to a significant increase in the heat capacity of CO_2 in the critical region. Figure 4 shows the temperature-enthalpy diagram for the recuperator at baseline conditions but with zero CO_2 cooler bypass.

Figure 4 Temperature-enthalpy diagram for the recompression Brayton cycle without bypass



With zero bypass, the recuperator is essentially single stage. The temperature difference between the hot side and cold side increases from the minimum approach value at the cold end to approximately 178 °F (98.9 °C) at the hot end. Figure 5 shows the temperature-enthalpy curve for the recompression Brayton cycle with a CO_2 cooler bypass fraction of 0.283. At this maximum value, the second stage recuperator has attained the minimum temperature approach at both the cold end and hot end. In addition, the overall recuperator effectiveness has increased significantly (maximum temperature approach of 81 °F, 45 °C), the recuperator duty has increased almost 12 percent, and the cold side exit temperature has increased from 801 °F (427 °C) to 898 °F (481 °C). This in turn increases the amount of CO_2 in the cycle and hence the net power output and cycle efficiency.

Figure 5 Temperature-enthalpy diagram for the recompression Brayton cycle, bypass = 0.283



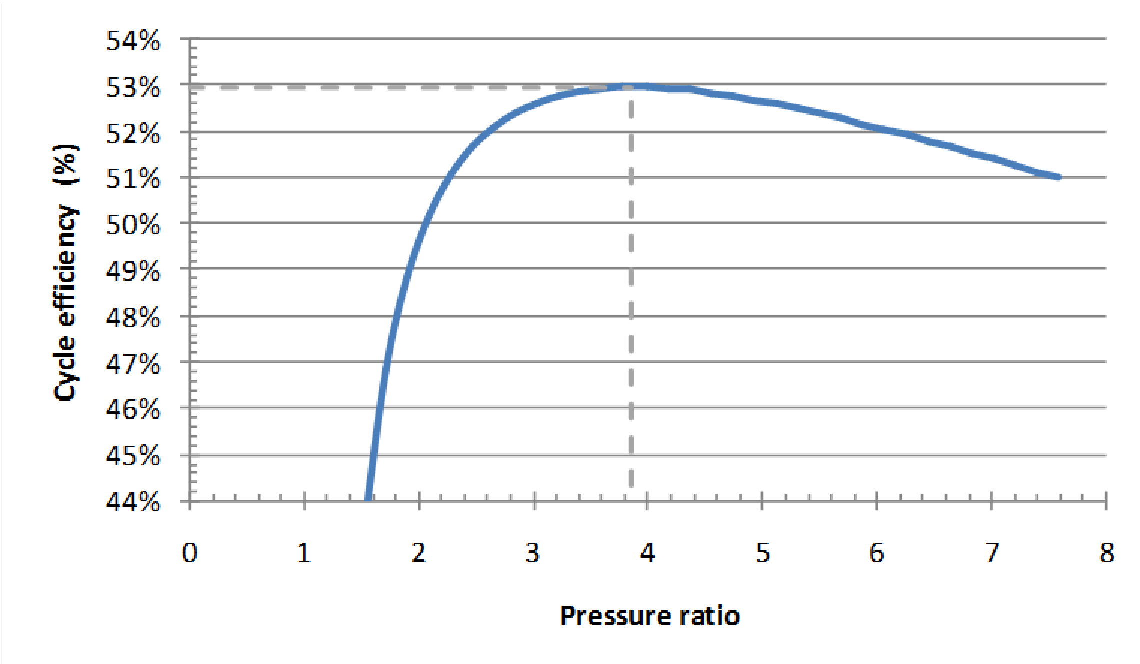
Sensitivity to pressure ratio

Figure 6 shows a plot of cycle efficiency as a function of pressure ratio, i.e., the compressor exit pressure divided by the compressor inlet pressure. As with all other sensitivity analyses, the cycle configuration parameters have the same values shown in Table 1 with the exception of the sensitivity variable itself.

The cycle efficiency shows a maximum at a pressure of approximately 3.9. Also, the shape of the curve in Figure 6 is asymmetric. The cycle efficiency shows a much greater dependence on pressure ratio on the low pressure side of the peak efficiency than on the high pressure side.

The character of the efficiency versus pressure ratio dependence is a generic result for the sCO_2 recompression Brayton cycle. At a pressure ratio of zero there is no output from the turbine and the cycle efficiency is zero. As the pressure ratio increases above zero, the turbine output increases faster than the compressor duty, primarily because of the large temperature difference in the CO_2 at these points. At some point, however, the pressure ratio becomes large enough that the increase in the compressor duty exceeds the increase in turbine output and the efficiency falls. Another factor contributing to the fall in cycle efficiency is that as the pressure ratio increases, the recuperator duty falls until at some point, neither recompression nor recuperation is feasible.

Figure 6 Cycle efficiency versus pressure ratio

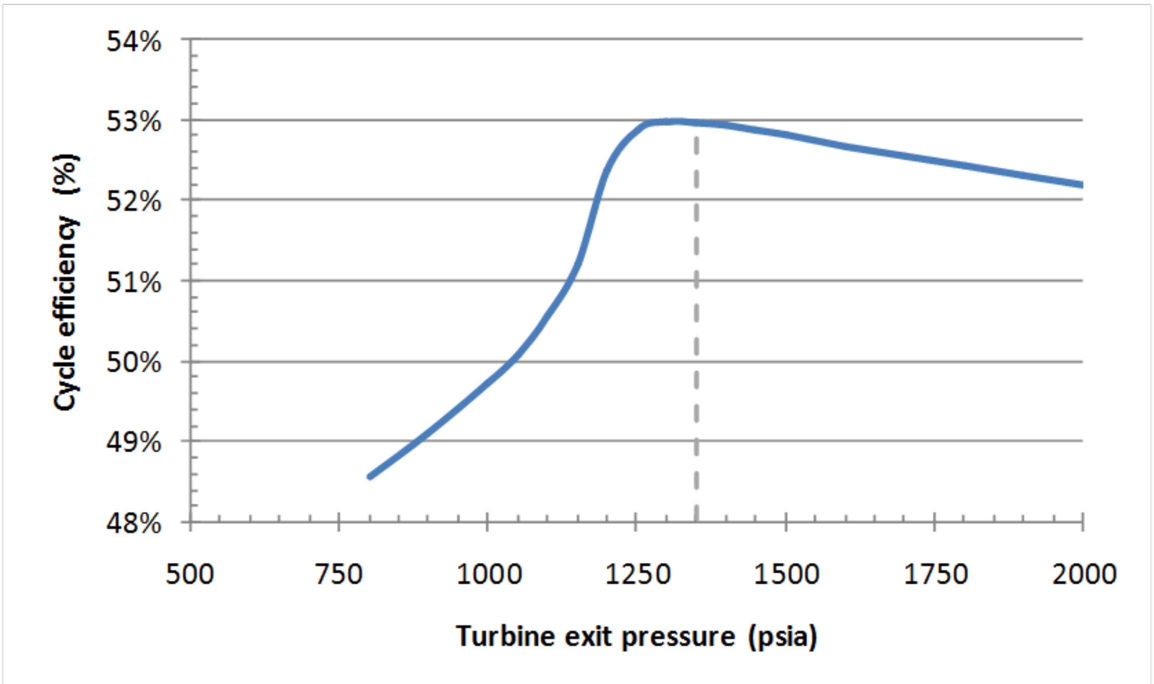


Sensitivity to pressure

In an ideal Brayton cycle, the cycle efficiency does not depend on the absolute pressures but only on the cycle pressure ratio. With the highly non-ideal CO_2, however, the absolute pressure becomes a significant cycle parameter impacting cycle efficiency. The sensitivity analysis performed to generate the plot in Figure 6 was repeated with varying values for the turbine exit pressure, ranging from 800 psia (55.2 bar) to 2000 psia (137.9 bar). The maximum in the cycle efficiency versus pressure ratio plot was calculated and the results plotted against the turbine exit pressure. These results are shown in Figure 7.

The efficiency dependence on pressure is quite complex. From a turbine exit pressure of 800 psia (55.2 bar), the cycle efficiency rises rapidly with the curve showing an inflection point at approximately 1150 psia (79.3 bar) followed by a maximum at 1314 psia (90.6 bar). As the pressure increases further, the cycle efficiency drops at a slow rate. The cycle efficiency shows a near plateau at turbine exit pressures between 1270 psia (87.6 bar) and 1380 psia (95.1 bar). To dampen the impact of pressure drop in the system, it was decided to use a turbine exit pressure of 1350 psia (93.1 bar) for the nominal baseline configuration.

Figure 7 Cycle efficiency versus turbine exit pressure



Sensitivity to turbine inlet temperature

Figure 8 shows the cycle efficiency versus pressure ratio curve for the sCO_2 recompression cycle using the baseline system rerun at five different values for the turbine inlet temperature. These temperatures represent different potential hot sources and hence different potential applications for the cycle. Figure 9 shows the peak efficiency for the plots in Figure 8 plotted against turbine inlet temperature.

The increase in cycle efficiency with turbine inlet temperature is an expected result. In the temperature range studied, on the average the cycle efficiency increases one percentage point for every 25 °C (45 °F) increase in the turbine inlet temperature. Of additional note is the slight drop in the pressure ratio at peak efficiency as the turbine inlet temperature decreases.

Figure 8 Cycle efficiency as a function of pressure ratio for different TIT

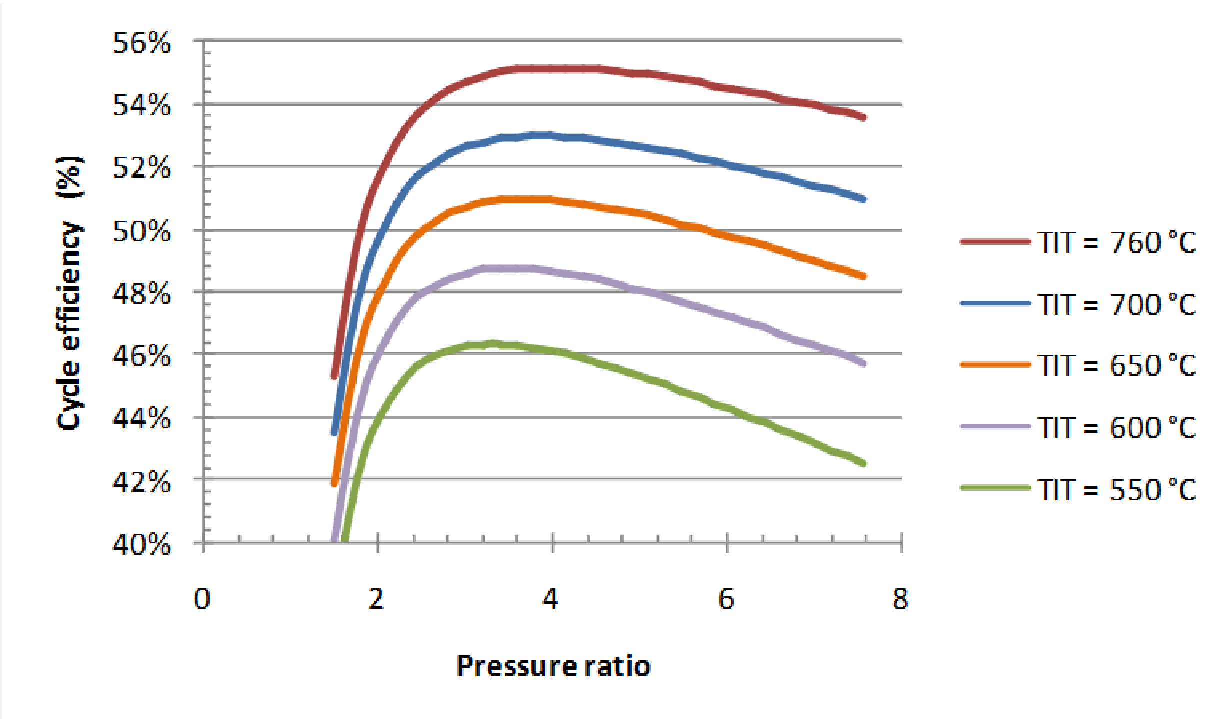
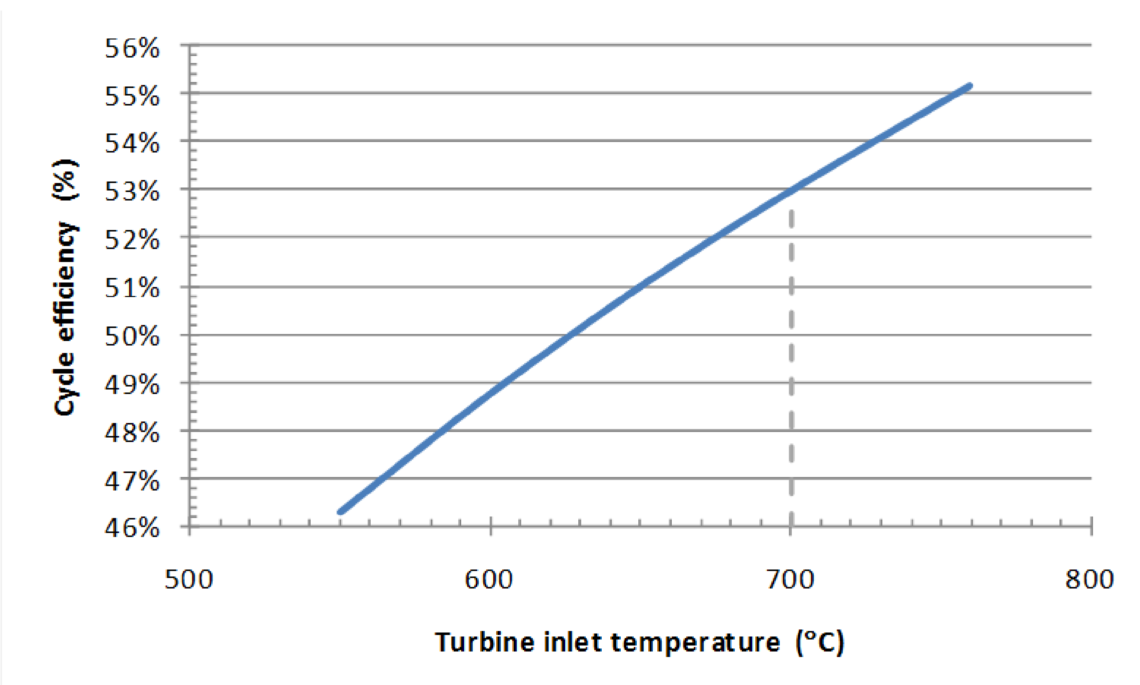


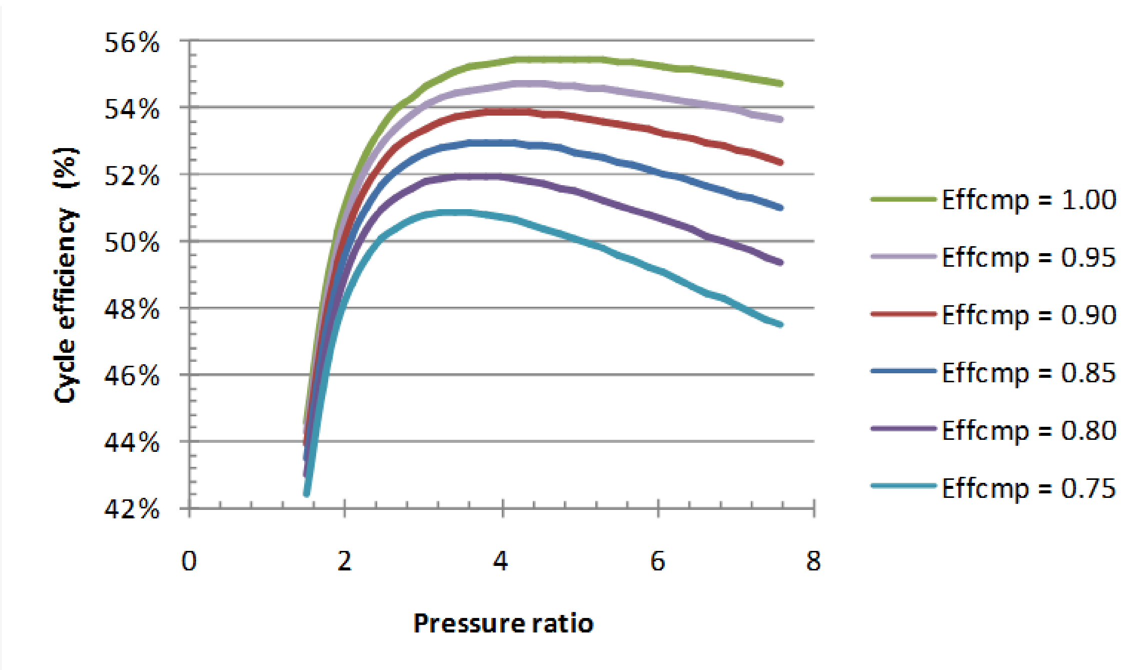
Figure 9 Cycle efficiency as a function of turbine inlet temperature



Sensitivity to compressor efficiency

Figure 10 shows the cycle efficiency versus pressure ratio curve for the sCO_2 recompression cycle using the baseline system rerun at six different values for the compressor isentropic efficiency ranging from 0.75 to 1.0. The nominal compressor isentropic efficiency of 0.85 is shown on this plot. Over the range of compressor efficiencies examined, the average cycle efficiency increases a little less than one percentage point for every five percentage point increase in compressor efficiency.

Figure 10 Cycle efficiency versus pressure ratio at different compressor isentropic efficiencies



Sensitivity to turbine efficiency

Figure 11 shows the cycle efficiency versus pressure ratio curve for the sCO_2 recompression cycle using the baseline system rerun at seven different values for the turbine isentropic efficiency ranging from 0.75 to 1.0. The nominal turbine isentropic efficiency of 0.927 is shown on this plot. Figure 12 plots the cycle efficiency versus isentropic efficiency for both the turbine and the compressor.

Over the range of compressor efficiencies examined, the average cycle efficiency increases a little more than two percentage points for every five percentage point increase in compressor efficiency. This sensitivity is more than double that observed for the compressor efficiency. The reason for the relatively muted impact of the compressor efficiency is that inefficiency in the compressor results in extra heat product that raises the temperature of the CO_2. This energy spares the heat requirement from the hot source, mitigating the extra work required by the compressor.

Figure 11 Cycle efficiency versus pressure ratio at different turbine isentropic efficiencies

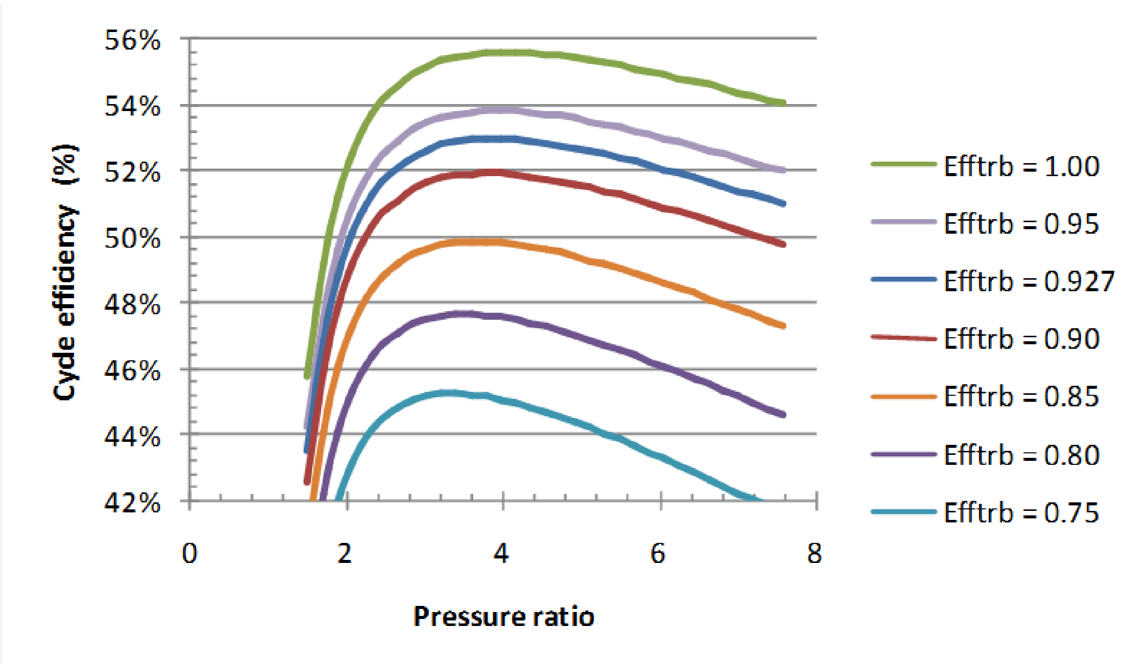
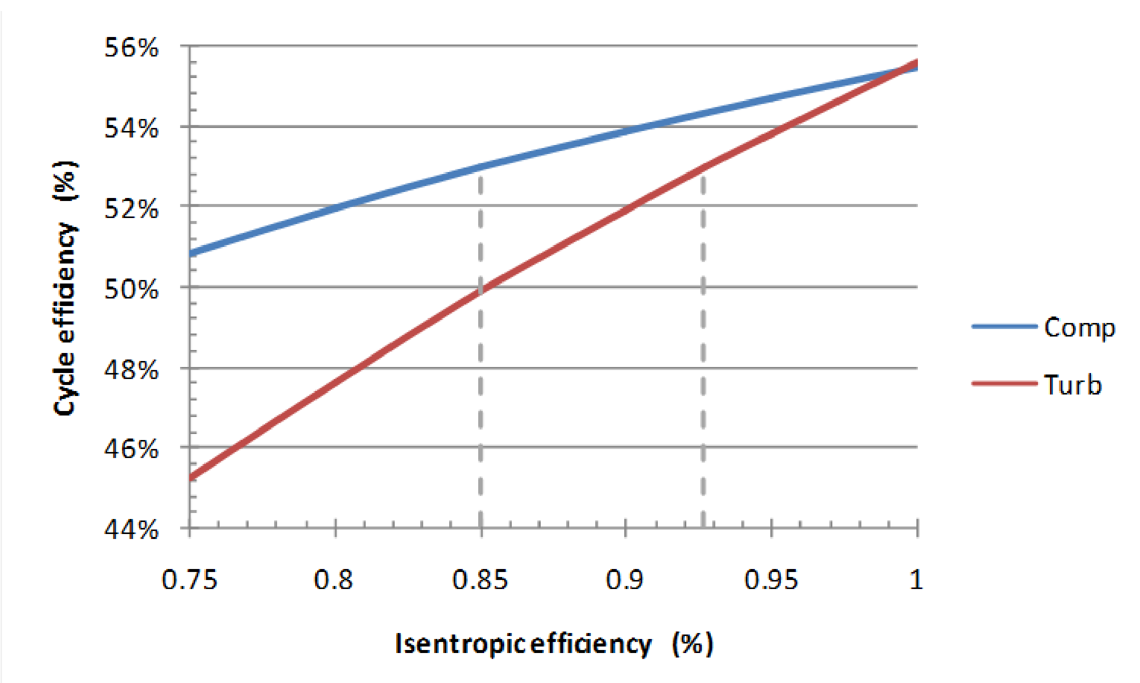


Figure 12 Cycle efficiency versus compressor and turbine isentropic efficiency



Sensitivity to cycle pressure drop

The assumed pressure drop of 60 psia (4.1 bar) for the recompression Brayton cycle is a crude estimate and not based on actual equipment designs or system optimization. To help determine the impact of this parameter on overall system performance, a sensitivity analysis was performed in which the cycle pressure drop was varied between zero and 180 psia (12.4 bar). The resulting efficiency versus pressure ratio curves are shown in Figure 13. Figure 14 shows the peak efficiency for the plots in Figure 13 plotted against cycle pressure drop. The vertical line on Figure 14 corresponds to the baseline configuration.

Figure 13

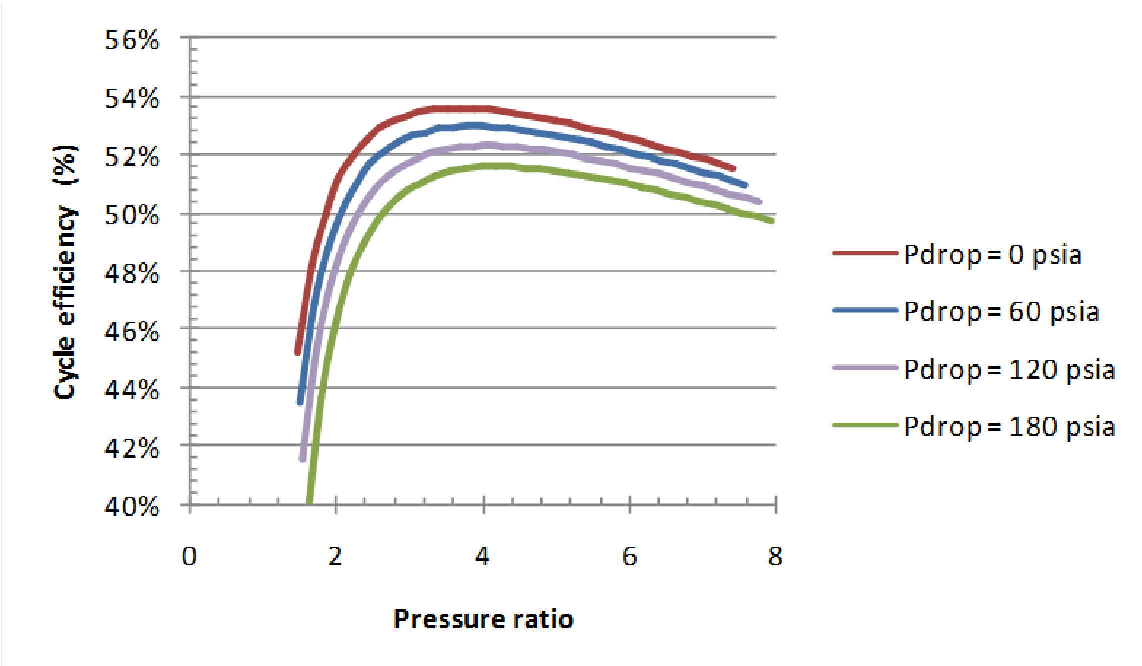
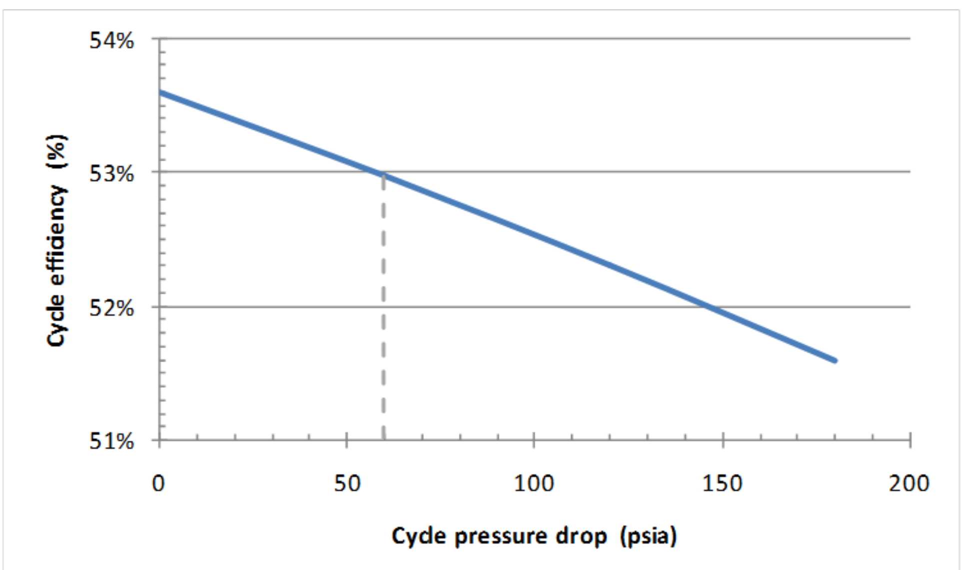


Figure 14 Cycle efficiency versus cycle pressure drop



As expected, the cycle efficiency decreases as the cycle pressure drop increases. However, the sensitivity of the efficiency to pressure drop is less pronounced than with other operating parameters. On average, the cycle efficiency drops approximately 0.1 percentage points for every 10 psia (0.7 bar) increase in the pressure drop.

Sensitivity to minimum approach temperature

The minimum temperature approach of 10 °F (5.6 °C) used for the baseline configuration was an arbitrary and somewhat aggressive target. A larger temperature approach would decrease cycle efficiency but may be worthwhile if it results in a substantial cost savings due to larger driving forces and smaller recuperators. Figure 15 shows the results of an analysis in which the sensitivity of the efficiency to pressure ratio was rerun at varying values for the minimum temperature approach between zero and 40 °F (22.2 °C). Figure 16 shows the peak efficiency for the plots in Figure 15 plotted against minimum approach temperature. The vertical line on Figure 16 corresponds to the baseline configuration.

The cycle efficiency shows an intermediate sensitivity to the minimum temperature approach dropping a little more than one percentage point for every 10 °F (5.6 °C) increase in the minimum approach temperature.

Figure 15 Cycle efficiency versus pressure ratio at different minimum approach temperatures

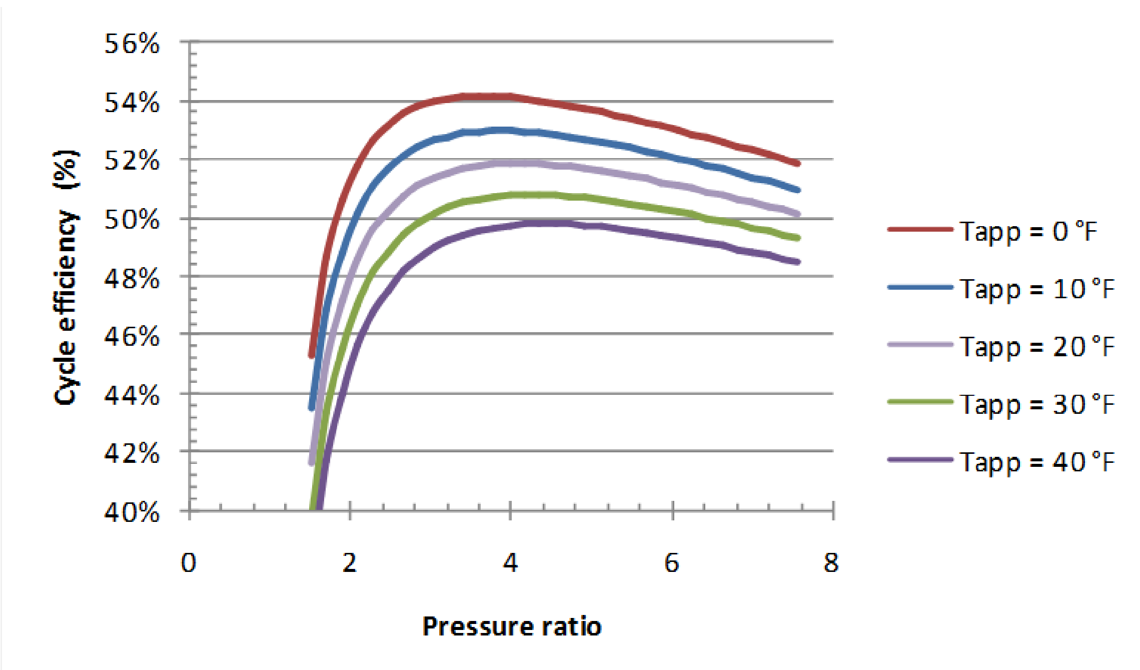
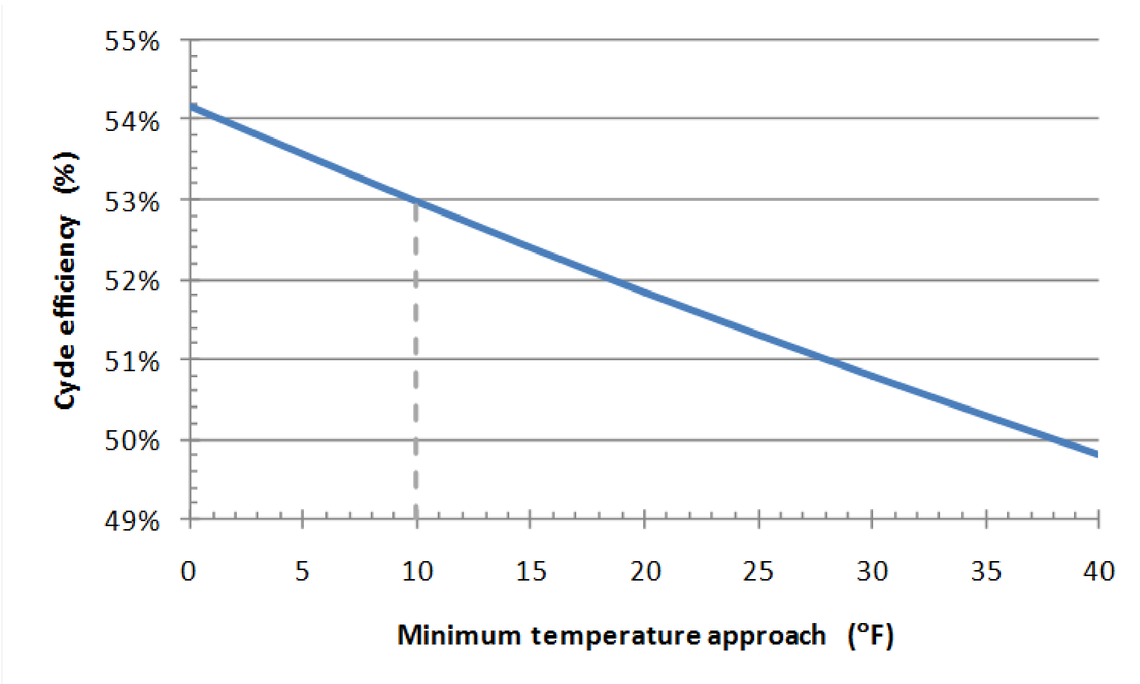


Figure 16 Cycle efficiency versus minimum approach temperature



Sensitivity to CO_2 cooler temperature

Figure 17 shows the results on an analysis in which the sensitivity of the efficiency to pressure ratio was rerun at varying values for the CO_2 cooler temperature between 32 °C (89.6 °F) and 45 °C (113 °F). Figure 18 shows the peak efficiency for the plots in Figure 17 plotted against CO_2 cooler temperature. The vertical line on Figure 18 corresponds to the baseline configuration.

The CO_2 cooler temperature was arbitrarily selected as several degrees above the CO_2 critical temperature. This was done to assure computational stability for the sensitivity runs. The calculated sensitivity to CO_2 cooler approach temperature is moderate. In the range of 32 °C (89.6 °F) to 38 °C (100.4 °F), the cycle efficiency drops on the average 0.19 percentage points for every 1 °C (1.8 °F) increase in the CO_2 cooler temperature. In the range of 38 °C (100.4 °F) to 45 °C, (113 °F) the cycle efficiency is more sensitive to CO_2 cooler temperature dropping by 0.33 percentage points for every 1 °C (1.8 °F) increase in the CO_2 cooler temperature.

Figure 17 Cycle efficiency versus pressure ratio at different CO_2 cooler temperatures

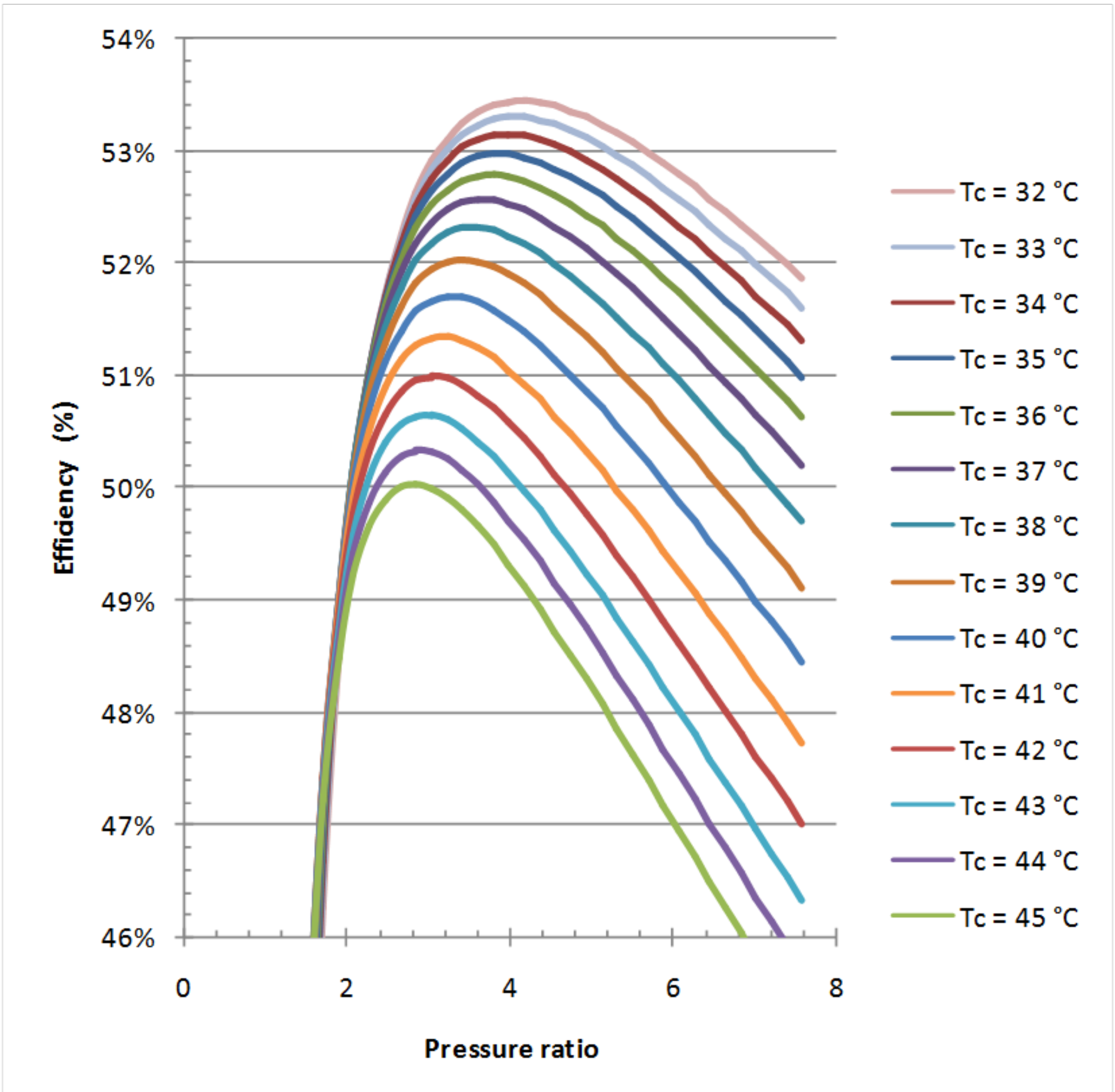
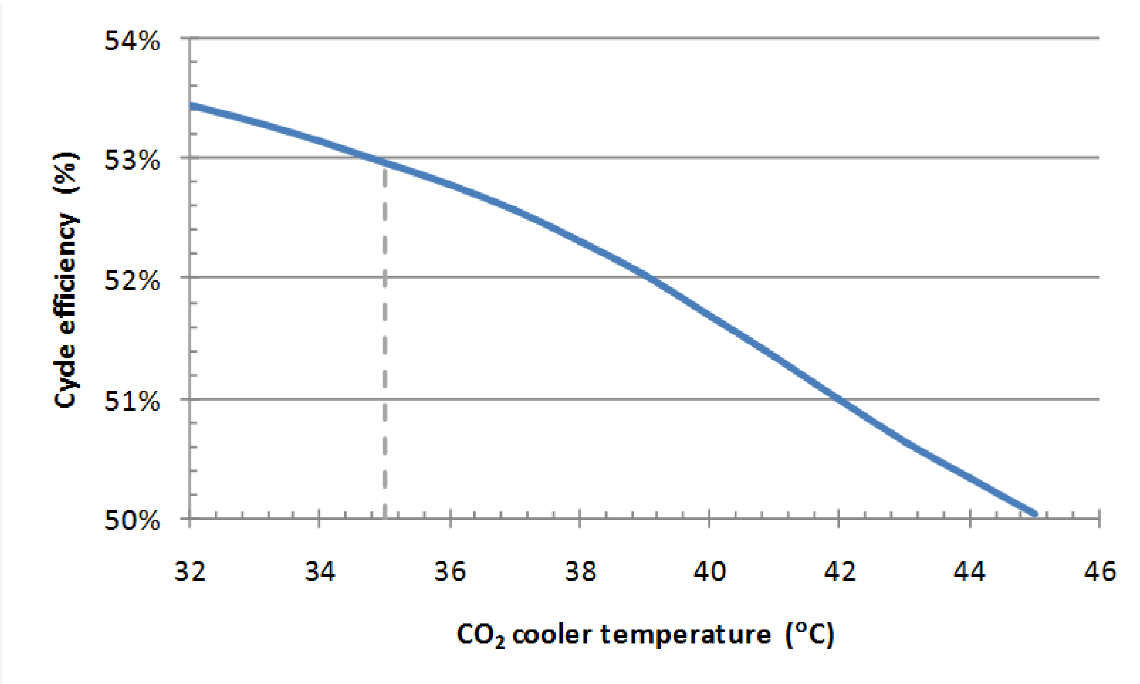


Figure 18 Cycle efficiency versus CO_2 cooler approach temperature



Indirect Cost Indicators

Although the supercritical CO_2 power cycle is not a new concept, there have been no large scale or commercial power plants built using this cycle. The public literature does not contain detailed design information for the major pieces of equipment required in the cycle so estimates for the cost of such a process remain highly uncertain.

Although a reliable plant cost estimate cannot be made, there is still useful information that can be gleaned by examining process variables and other derived quantities that offer an indirect measure of cost. While the variables cannot generate a cost estimate, if these quantities appear to have a steep sensitivity curve in the region of candidate operation, it could be inferred that the candidate design is problematic from either a cost effectiveness standpoint or an operability standpoint. This can potentially result from a design that attempts to maximize efficiency without regard to the impact on driving force.

Table 3 lists the major pieces of equipment in a sCO_2 power cycle and for each, lists a number of quantities that would expect to influence plant cost.

Table 3 Indirect cost metrics for sCO_2 recompression Brayton cycle

sCO_2 power cycle component	Indirect cost metric	Comment
Overall Cycle	Temperature	Materials selection
	Pressure	May suggest higher cost (vessel thickness, seals, …) or lower cost (reduced sizes)
	Cycle efficiency	Inversely related to operating costs
	Mass flow rate, specific power	Indicator of overall plant size
Compressor/Turbine	Power, Mass flow	Indicator of unit size
	Pressure ratio	Related to number of stages required
	Inlet volumetric flow	Indicator of inlet size
Recuperator	Total heat duty, LMTD (UA)	Indicator of unit size
Heat source	Total heat duty, LMTD (UA), CO_2 thermal capacitance	Indicator of unit size

The following Figures depict the results of sensitivity analyses on the key indirect cost variables. Figure 19 shows the main compressor inlet volumetric flow rate as a function of pressure ratio at the cycle conditions listed in Table 1. The vertical line indicates the operating state for the baseline configuration.

Figure 19 suggests that the baseline operating point is in a relatively flat portion of the sensitivity curve for the main compressor inlet volumetric flow rate.

Figure 19 Main compressor inlet volumetric flow rate versus pressure ratio

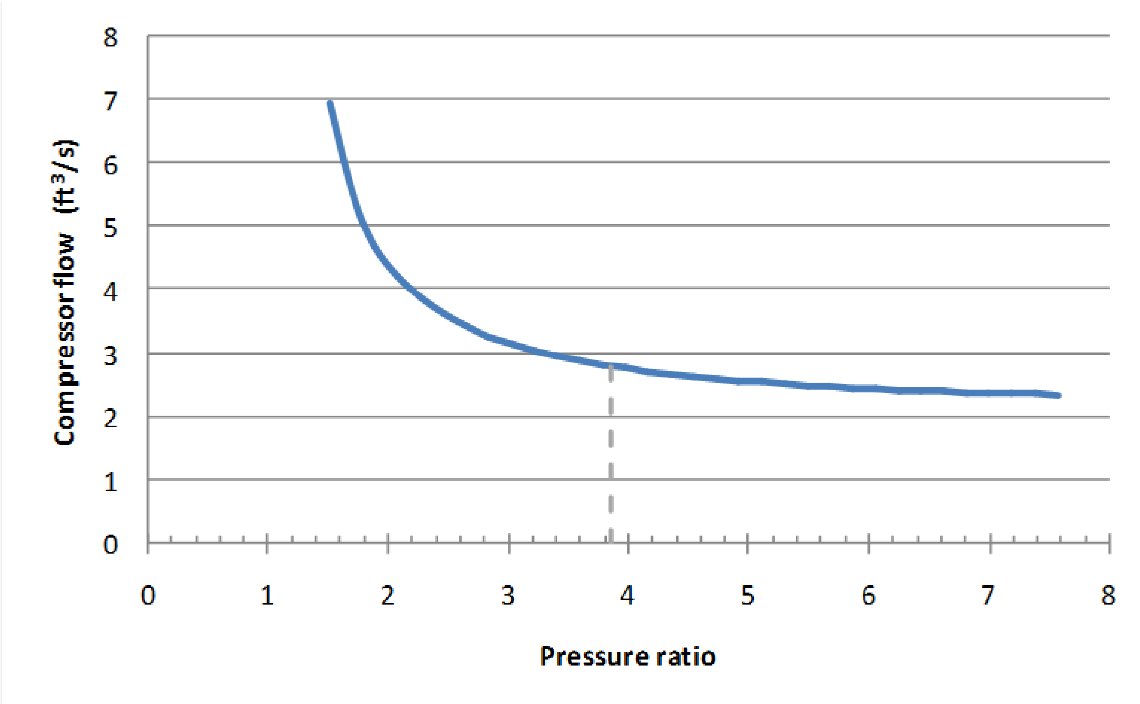


Figure 20 through Figure 25 show analogous plots for six additional indirect cost variables: turbine inlet volumetric flow rate, recuperator duty, recuperator driving force, hot source duty, specific power, and CO_2 flow rate. In all cases the results are similar to the results for main compressor inlet volumetric flow rate. While the baseline operating point is not a minimum cost configuration, it lies on a relatively flat region of the sensitivity curve suggesting that the operating point probably does not represent an economically infeasible state.

Figure 20 Turbine inlet volumetric flow rate versus pressure ratio

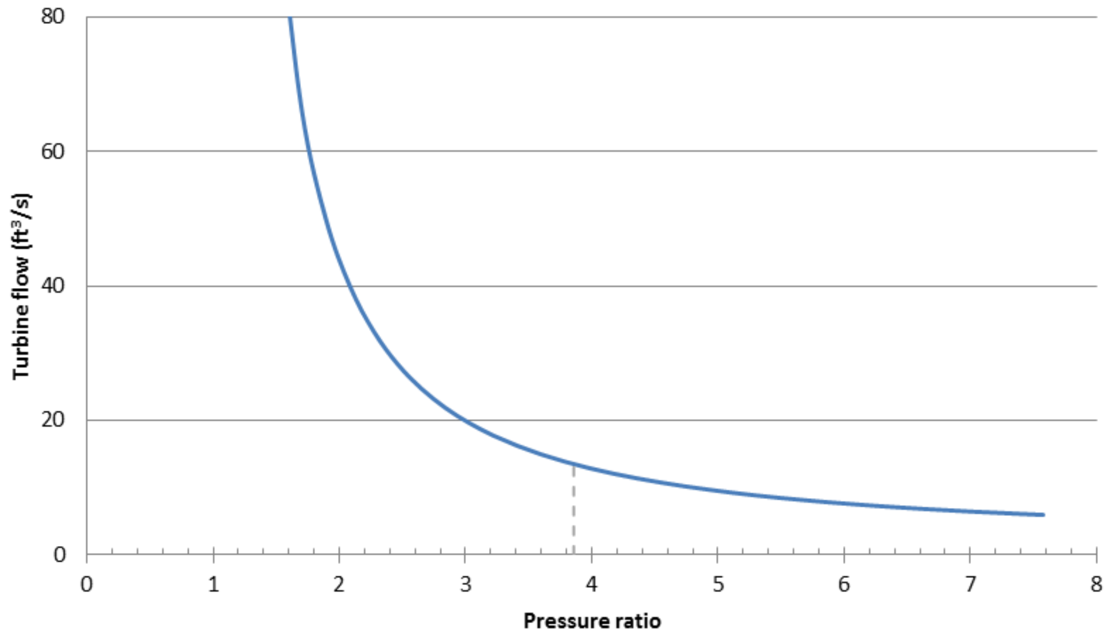


Figure 21 Total recuperator duty versus pressure ratio

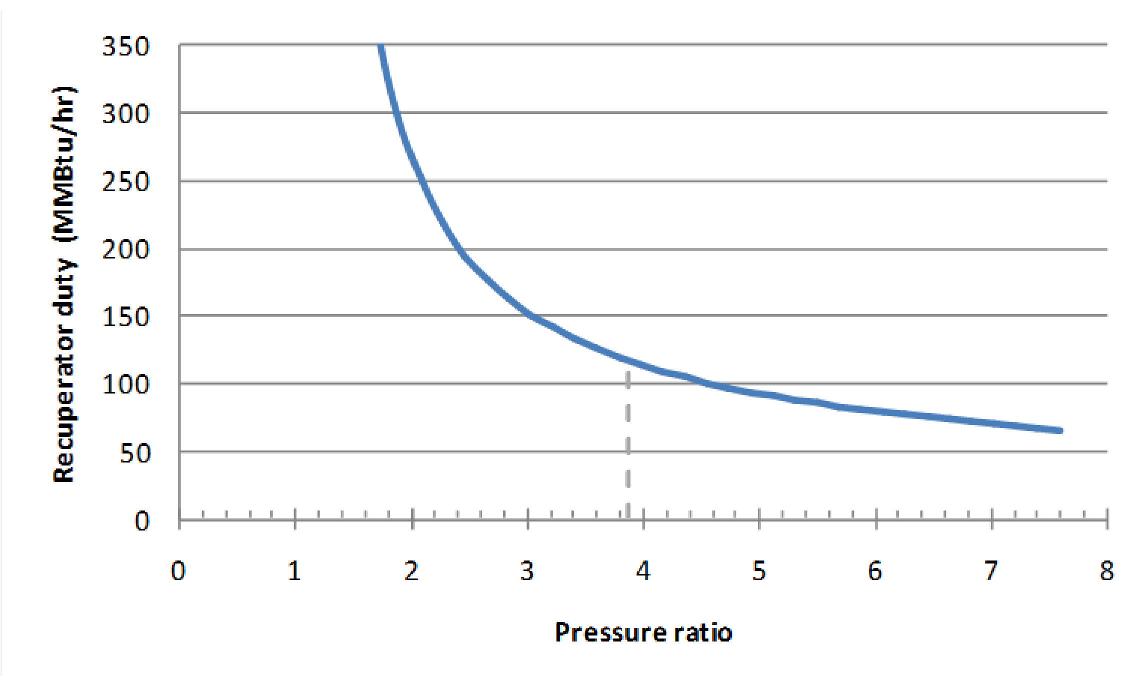


Figure 22 Total recuperator UA versus pressure ratio

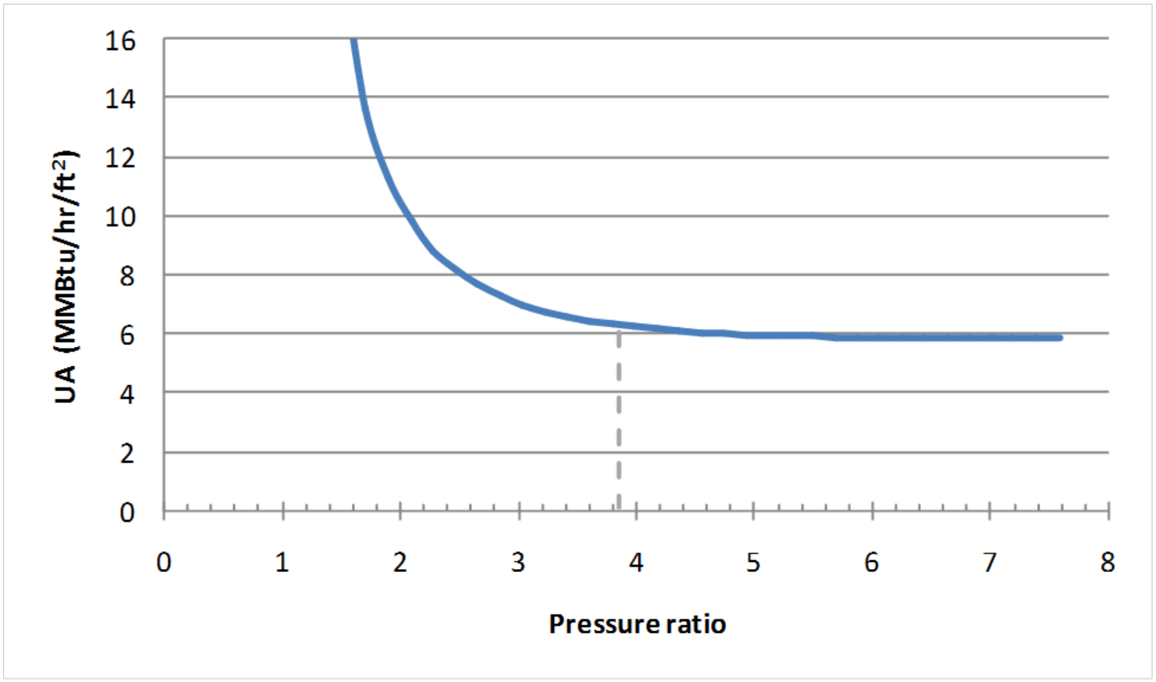


Figure 23 Hot source duty versus pressure ratio

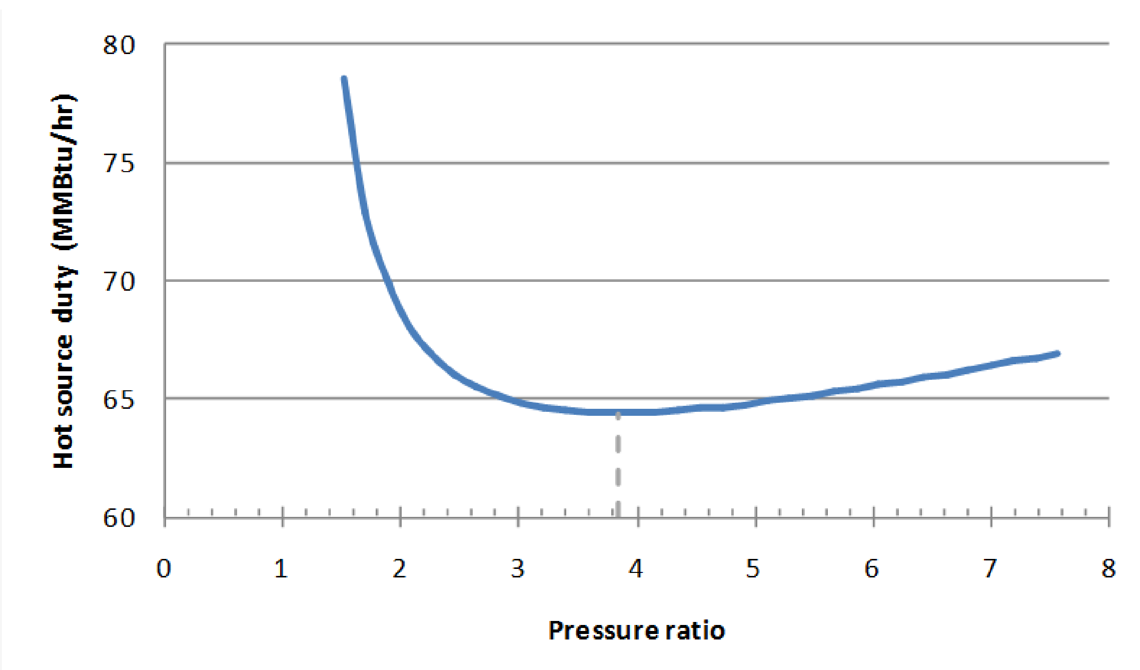


Figure 24 Specific power versus pressure ratio

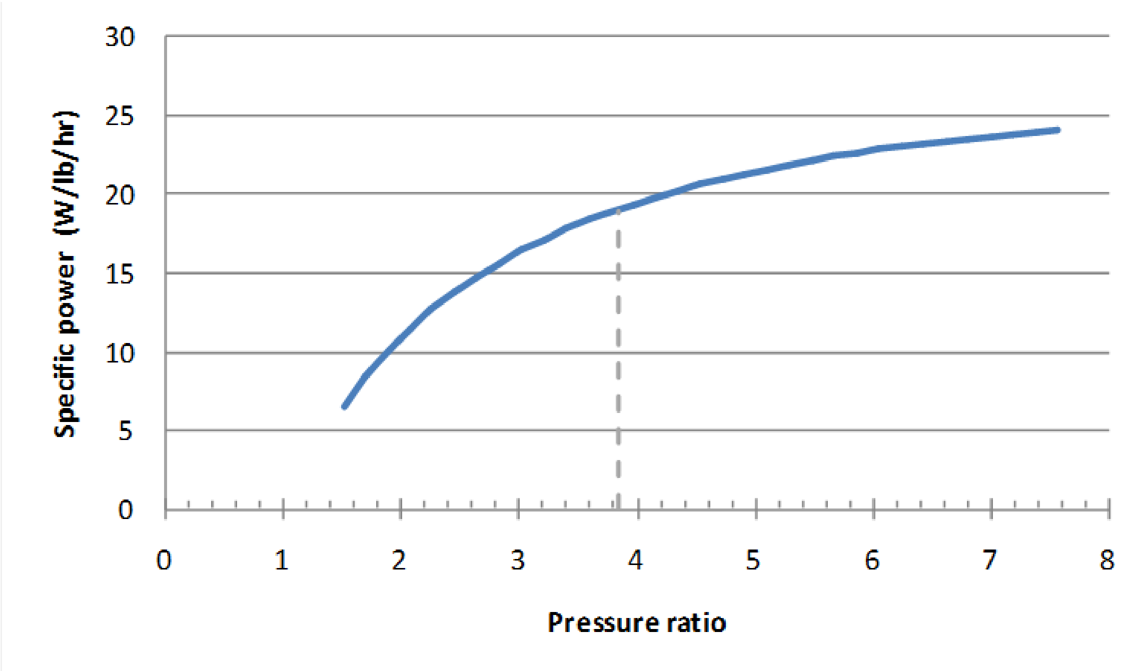
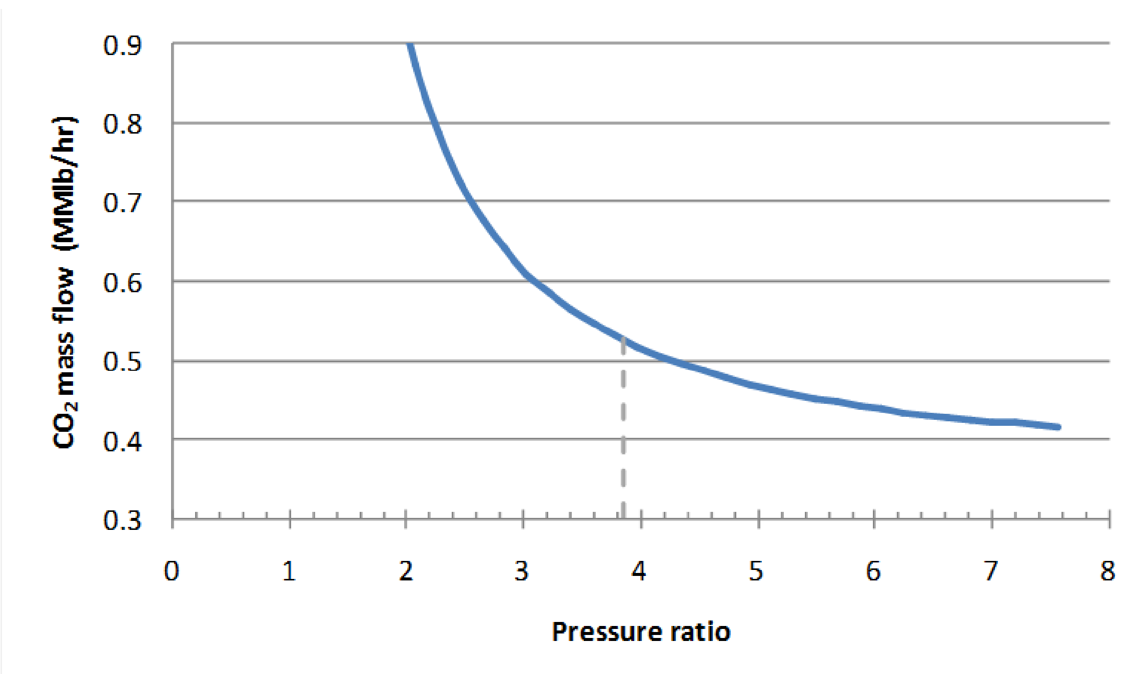


Figure 25 Total CO_2 flow rate versus pressure ratio



Summary and Conclusions

This paper has presented a detailed sensitivity analysis of the sCO_2 recompression Brayton cycle around a baseline configuration. The cycle has a conventional configuration with a two-stage recuperator and bypass stream around the CO_2 cooler. In this study, the heat source was treated as generic, *i.e.*, there was no temperature or temperature profile associated with the heat source and the heat input to the cycle was specified.

The first sensitivity analysis examined the impact of varying the fraction of CO_2 that bypassed the CO_2 cooler. It was determined that the bypass fraction should be as high as possible subject to the allowed minimum temperature approach in the recuperator. It was shown that this bypass stream acts to increase the effectiveness of the recuperator allowing a greater amount of heat exchange between the high pressure and low pressure portions of the cycle. The next sensitivity examined the impact of pressure ratio on the cycle efficiency. It was shown that the cycle efficiency versus pressure ratio curve passes through a maximum and that this maximum occurs at a relatively low pressure ratio in the range of 3-4. For the baseline configuration, the maximum occurred at a pressure ratio of 3.9. The cause of the maximum in cycle efficiency was also explored.

The next sensitivity analysis examined the impact of the absolute cycle pressure on the cycle efficiency. This was examined by adjusting the turbine exit pressure. Although for an ideal Brayton cycle, efficiency does not depend on the absolute pressure, this was found not to be the case for the sCO_2 recompression Brayton cycle. As with the pressure ratio, the cycle efficiency passes through a maximum at some value for the turbine exit pressure and for turbine exit pressures lower than the maximum, the cycle efficiency is very sensitive to even small changes in pressure ratio. The baseline configuration setting of 1350 psia (93.1 bar) was found to be just above the point of maximum efficiency and at a point in the curve that was relatively flat and hence relatively stable to system perturbations.

The next sensitivity analysis examined the impact of the turbine inlet temperature on cycle performance. This in effect helped assess the range of applicability of the cycle for different potential heat sources. The results were consistent with the expectation that cycle efficiency increases with increasing turbine inlet temperature when the CO_2 cooler temperature remains constant. It was found that as a rough approximation, the cycle efficiency increases one percentage point for every 25 °C (45 °F) increase in the turbine inlet temperature. It was also noted that as the turbine inlet temperature increases, the pressure ratio at maximum cycle efficiency also increases and that the cycle efficiency becomes less sensitive to perturbations in the pressure ratio.

The next two sensitivity analyses examined the impact of the turbo-machinery efficiencies on the cycle performance. These studies could be considered to provide some guidance in setting possible efficiency targets for an R&D program to develop this equipment. As expected, both the compressor and turbine efficiencies exert a significant influence on cycle performance. However, the turbine efficiency has a little more than twice the impact of the compressor efficiency, at least in the range of values examined. An increase of 0.05 in the isentropic efficiency for the turbine results in a little more than a two percentage point increase in cycle efficiency whereas the same increase in the compressor isentropic efficiency results in less than a one percentage point increase in cycle efficiency.

The next two sensitivity analyses examined the cycle pressure drop and minimum temperature approach on the cycle efficiency. Although the qualitative dependencies are easy to anticipate, the quantitative assessment provides useful guidance in the design compromises needed to develop a cost effective power cycle. It was found that the cycle efficiency had a significant dependence on the minimum temperature approach and only a relatively minor dependence on the pressure drop. For every 10 psia (0.7 bar) increase in pressure drop the cycle efficiency dropped approximately 0.1 percentage points. For every 10 °F (5.6 °C) increase in the minimum temperature approach the cycle efficiency decreases a little more than one percentage point.

The final sensitivity analysis examined the impact of the CO_2 cooler temperature. At lower cooler temperatures near the critical temperature of CO_2, the cycle efficiency decreases by 0.19 percentage points for every 1 °C (1.8 °F) increase in the cooler temperature. At higher temperatures above 38 °C (100.4 °F), the cycle efficiency decreases by 0.33 percentage points for every 1 °C (1.8 °F) increase in the cooler temperature. The results indicate that an air cooled CO_2 cooler may be feasible.

Obviously a determination of the full benefits of the sCO_2 cycle depends on a reliable estimate of the capital and operating costs of the process. This is a challenge given the relatively low technology readiness level of some of the cycle components. In the absence of a cost estimate, a series of sensitivity analyses were performed on process variables and derived quantities that offer at least an indirect indication of process cost. The primary purpose of this analysis was to determine whether the analyses used to establish a high efficiency baseline configuration drove the operating conditions to a point where practical operation was infeasible or that the driving forces were unacceptably low. The study examined the volumetric flow rates into the turbine and main compressor, the recuperator duty and driving force, the hot source duty, the specific cycle power, and the total CO_2 flow rate. All of the sensitivity analyses showed that the baseline configuration fell on a reasonably flat and stable portion of the sensitivity curve to pressure ratio. This suggests that the baseline operating conditions are probably feasible and may not be too far removed from an economically optimum cycle configuration.

NOMENCLATURE

Ar	=	Argon
atm	=	Atmosphere (14.696 psi)
BFD	=	Block flow diagram
Btu	=	British thermal unit
Btu/hr	=	British thermal units per hour
CO_2	=	Carbon dioxide
cuft	=	Cubit foot
DOE	=	Department of Energy
Effcmp	=	Compressor isentropic efficiency
Efftrb	=	Turbine isentropic efficiency
ESPA	=	Energy Sector Planning and Analysis
FE	=	Fossil energy
h, hr	=	Hour
lb	=	Pound
lb/hr	=	Pounds per hour
lbmol	=	Pound mole
lbmole	=	Pound mole
MM	=	Million
MMBtu	=	Million British thermal units
NETL	=	National Energy Technology Laboratory
Pex	=	Turbine exit pressure
PR	=	Pressure ratio
psia	=	Pound per square inch absolute
Q_C	=	Heat duty for cooler
Q_H	=	Heat input from hot source
Q_R	=	Heat duty for recuperator
R&D	=	Research and development
SC	=	Supercritical
sCO_2	=	Supercritical carbon dioxide
T	=	Temperature
TIT	=	Turbine inlet temperature
W	=	Watt
Wc	=	Compressor power
We	=	Expander power
°C	=	Degrees Celsius
°F	=	Degrees Fahrenheit
°R	=	Degrees Rankine