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# THERMODYNAMICS OF CONVENTIONAL AND NON-CONVENTIONAL SCO$_2$ RECOMPRESSION BRAYTON CYCLES WITH DIRECT AND INDIRECT HEATING

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## ABSTRACT

Aerojet Rocketdyne (AR), in cooperation with the Department of Energy (DOE), has been evaluating supercritical $CO_2$ ($sCO_2$) recompression Brayton cycles and associated turbomachinery concepts for electric power production using fossil, nuclear, solar, and other lower temperature heat sources. Present work summarizes the modeled performance of $sCO_2$-based power cycles in the field over a broad range of temperatures (285-3,500°F) (141-1,927°C), such as ~285°F (~141°C) for geothermal, ~1,000°F (538°C) for Nuclear, 1,100-1,400°F (593-760°C) for Solar, ~1,300°F (704°C) for Coal fired, and ~2,000-3,500°F (1,093-1,927°C) for natural gas fired power plants. Simulated performance is compared with conventional steam Rankine power cycles showing the potential benefits of these advanced $sCO_2$ cycles. Thermodynamic analysis using TS diagrams is provided to explain the differences in efficiencies.

## INTRODUCTION

The $sCO_2$-based Brayton cycle was primarily investigated for nuclear applications, first during the 1960's and then since the early 2000's [2, 3]. Although the Brayton cycle is not expected to offer efficiency benefits at lower turbine inlet temperatures <1,000°F (<538°C) over the steam Rankine cycle, the higher density of $sCO_2$ and its liquid-like behavior in the supercritical region reduces turbomachinery size and compression costs. Combining this with potential safety enhancements and/or lower operating and maintenance costs could lead to lower electricity costs. At higher turbine inlet temperatures of >1000°F

(>538°C), the efficiency advantage of the $sCO_2$ Brayton cycle is expected to increase over that of the steam Rankine cycle. Combining this increased efficiency and lower turbomachinery cost will likely lead to a reduction in electricity costs. Furthermore, issues such as corrosion and hydrogen embrittlement are known to limit higher steam turbine inlet conditions; while for $sCO_2$ cycles, the potential exists to operate at higher temperatures and pressures without these limitations. The $sCO_2$ cycle can also take advantage of multiple reheats for additional efficiency benefits.

This paper presents the efficiencies of several supercritical $sCO_2$ recompression Brayton cycles (RCBC) with various direct and indirect heat sources derived from computer simulated analysis. The efficiencies of these cycles are evaluated against other conventional systems, such as the steam Rankine or Organic Rankine cycles. Also provided is an assessment of the benefits of using $CO_2$ versus steam. System performance results modeled from varying turbine inlet conditions, turbine outlet pressure, and main compressor inlet temperature for a double recuperated RCBC system are also provided.

## EFFICIENCY COMPARISON OF VARIOUS SYSTEMS

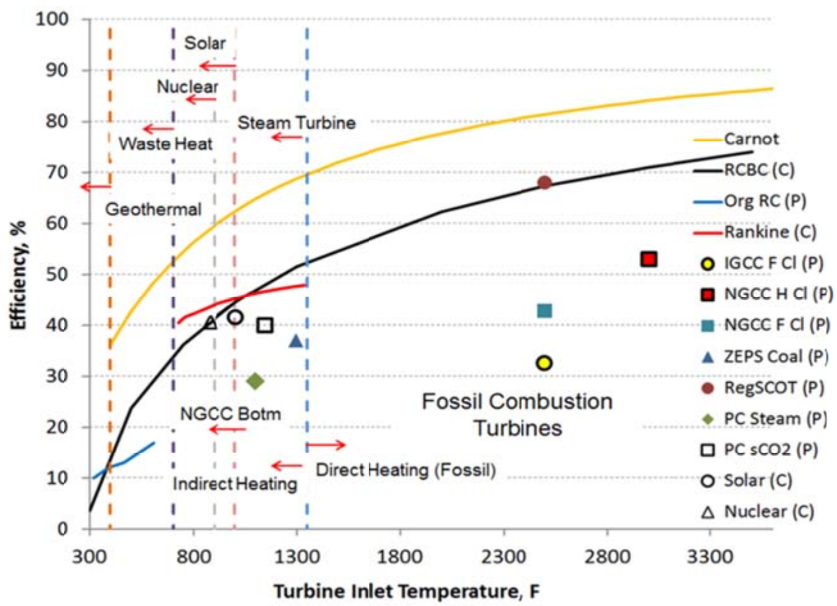


**Figure 1. Turbine Inlet Temperature vs Efficiency of Various Systems (C for cycle efficiency, P for plant efficiency)**

Figure 1 shows the efficiency of various systems as a function of turbine inlet temperature. The Carnot cycle represents the maximum theoretical efficiency at each inlet temperature and is a good measure of the deviation from idealization of all other systems. A key temperature threshold is 1,300°F (704°C), as it divides the incorporation of heat for power systems: indirect heat versus direct heat. Below 1,300°F (704°C), indirect heat sources are used in which an external heat source supplies heat to a closed loop cycle of various fluids. Above 1,300°F (704°C), direct heat is applied to a system in which heat from the combustion gases mix directly with the working fluid to create an open loop cycle. Cycle efficiency is displayed for indirectly heated, closed loop systems, while plant efficiency, based on higher heating values (HHV), is depicted for directly heated, open loop systems.

The organic Rankine (Org RC) and steam Rankine cycles are indirectly heated. Several $sCO_2$ Brayton Cycles are also indirectly heated: Nuclear, Solar, Pulverized Coal (PC) $sCO_2$, PC Steam, and Zero Emission Power and Steam (ZEPS™). The Integrated Gasification Combined Cycle (IGCC), Natural Gas Combined Cycle (NGCC), and Regen SCOT (RegSCOT) are directly heated. The recompression Brayton cycle can be either indirectly or directly heated. As seen in Figure 1, the indirectly heated cycles typically have lower efficiency than the directly heated cycles due to the lower turbine inlet temperature.

*Steam Rankine Cycle (herein denoted as the Rankine Cycle)*

The majority of current coal fired power plants use a Rankine cycle for power generation. The Supercritical plants are viable in the temperature range of 800-1,100°F (427-593°C) and have corresponding cycle efficiencies ranging from 40-45%, as seen in Figure 1. With the Advanced Ultra Supercritical plants, recent advancements have enabled temperatures up to 1,300°F (704°C) and roughly 49% cycle efficiency.

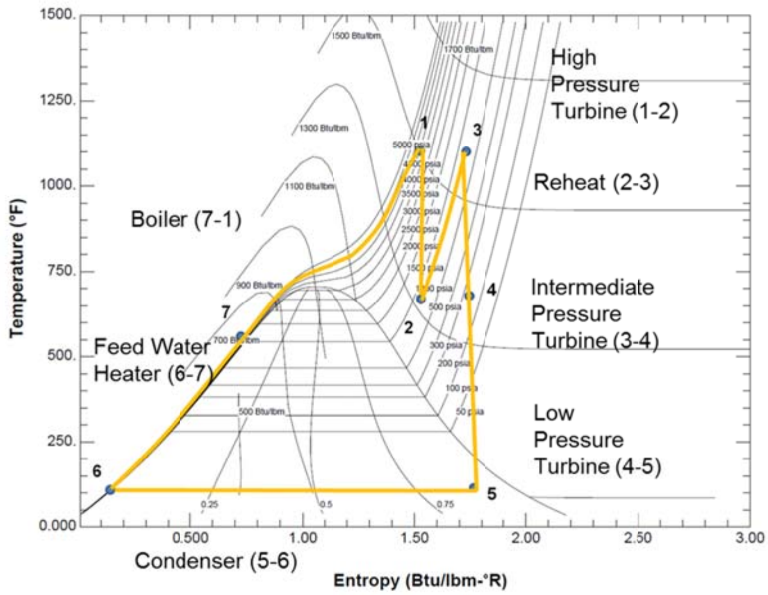


**Figure 2. Temperature-Entropy Diagram of a Rankine Cycle with Single Reheat**

The Temperature-Entropy (TS) diagram for the Rankine cycle with a single reheat is shown in Figure 2. As with all indirectly heated systems, an external heat source is supplied to a working fluid, in this case steam, which is in a closed loop cycle. The heat source vaporizes the water in a boiler (points 7-1). The steam is then expanded in three stages: through a high pressure turbine (points 1-2), an intermediate pressure turbine (points 3-4), and a low pressure turbine (points 4-5) with a reheat between the high and intermediate pressure turbines (2-3). The reheat is beneficial because supplying little additional heat allows for a greater amount of power extraction from the intermediate pressure turbine. The fluid is then condensed (points 5-6) and heated back up to 1,100°F (593°C) (points 6-7-1). As seen from points 5 to 6, a great amount of heat is lost in the condenser. The large pressure ratio across the turbines from 2,500 psi to 1 psi (17.2 MPa to 0.007 MPa) leads to a large temperature range from 1,100°F to 100°F (593 - 38°C). Thus, a large amount of heat is required to raise the system fluid up 1,000°F (538°C). This lowers the efficiency of the cycle. Also, there is a decrease in efficiency of the work produced by the low pressure turbine since the inlet temperature is only ~650°F (343°C). One benefit, however, of the Rankine cycle is the low amount of compression work due to pumping a liquid phase fluid.

*Natural Gas Combined Cycle (NGCC)*

Natural gas based power plants use an open loop Gas Turbine Brayton cycle in which natural gas combustion products are used as the working fluid for power generation. Often a Rankine bottoming cycle is incorporated, which uses the heat of the exhaust from the Gas Turbine to generate steam. The open loop gas turbine cycles typically operate at lower pressures (around 300 psia (2.1 MPa)) and higher temperatures (up to 2,750°F (1,510°C)) for F class turbines, which display a plant efficiency of ~43% [1]. More advanced and efficient gas turbines, such as the H or J class turbines, operate at higher temperatures and pressures and raise the overall plant efficiency up to 57%.

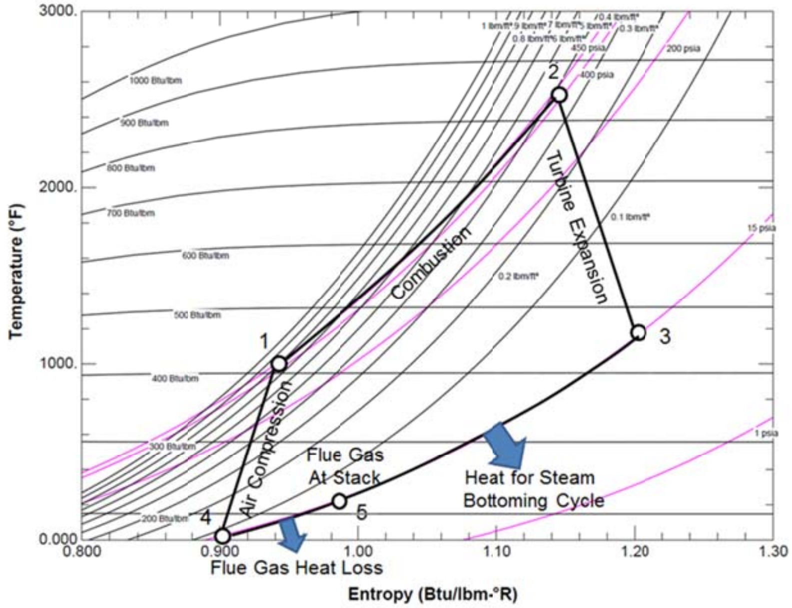


**Figure 3. Temperature-Entropy Diagram of an Air Fired F Class Gas Turbine**

Figure 3 shows the TS diagram for an NGCC cycle with an F class gas turbine. Natural gas is combusted with air and mixed with the working fluid (points 1-2). The gases are then expanded through a turbine (points 2-3) and cooled (points 3-5-4), where some of the heat is recovered in a Heat Recovery Steam Generator (HRSG) (points 3-5) and the remaining is lost (points 5-4). Points (4-1) show the compression work for air prior to combustion. The pressure ratio across the turbine of this Brayton cycle from 450 psi to 14.7 psi (3.1 MPa to 0.1 MPa) leads to a temperature range of the hot gas from ~2,550 to ~1,200°F (~1399 to ~649°C). The lower pressure of the gases equates to a lower density of fluid which requires larger equipment units.

*$sCO_2$ Brayton Cycle*

The $sCO_2$ Brayton cycle utilizes supercritical $CO_2$ as the working fluid. It can be either an indirectly heated, closed loop cycle or a directly heated, open loop cycle with recycled $CO_2$. Heat sources for indirectly heated cycles include geothermal, waste heat, nuclear, solar, pulverized coal (PC), and natural gas. The ZEPS™ cycle is another example. It is an indirectly heated coal power plant. The range of turbine inlet temperatures achievable with these heat sources is shown in Figure 1. In the temperature range of 800-1,100°F (427-593°C), these cycles are less efficient than the Rankine cycle; however, they are much more efficient than the Organic Rankine cycle, albeit the inlet temperatures for the Brayton cycle are much higher than for the Organic Rankine cycle. The directly heated $sCO_2$ Brayton cycles include the recompression Brayton Cycle (RCBC) of which the Regeneratively Cooled Supercritical $CO_2$ Turbomachinery (RegSCOT) is an example. This cycle is expected to have a much higher plant efficiency than current NGCC or IGCC Cycles.

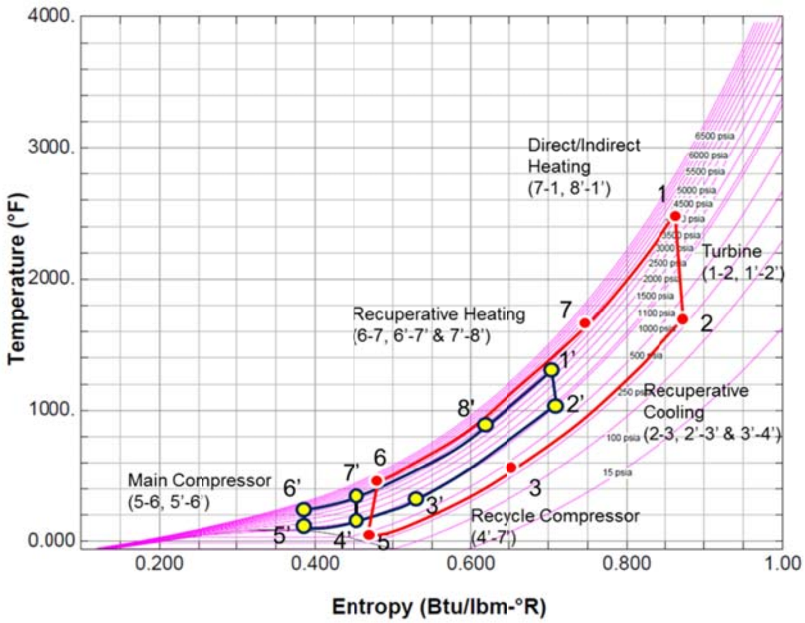


**Figure 4. Temperature-Entropy Diagram of a Recompression Brayton Cycle**

Figure 4 shows the TS diagram for a low temperature (depicted in black) and high temperature (depicted in red) $sCO_2$ Brayton cycle. The low temperature is a double recuperation cycle while the high temperature is a single recuperation cycle. The $sCO_2$ is expanded in a turbine (points 1-2, 1'-2'). Heat from the fluid is then recuperated in a high temperature recuperator (points 2-3, 2'-3') and a low temperature recuperator (points 3'-4'). It is then compressed back up to system pressure through a main compressor (points 5-6, 5'-6') or a recycle compressor (points 4'-7'), heated through recuperative heating (points 6-7, 6'-7' & 7'-8'), and then heated back up to system temperature (points 7-1) either directly or indirectly. The high turbine inlet temperature leads to a higher turbine power production. The recuperated heat is also used more efficiently when incorporated back into the system directly as opposed to when used to generate power in an HRSG. This lowers the amount of heat needed to raise the system back up to system temperature from an external source. Cooling down to a lower temperature and pressure also lowers the compression work required. The combination of these increases the overall efficiency of the system. Another added benefit is that the higher pressure of the system also increases the density of the fluid, allowing the turbomachinery and ducting to be more compact based on analysis.

*Benefits of $CO_2$ vs $H_2O$*

There are many benefits of using $CO_2$ as the working fluid in place of steam. A comparison of the density and specific heat for $CO_2$ and $H_2O$ is shown for typical turbine inlet conditions for a RCBC in Table 1. The density of $CO_2$ is twice that of $H_2O$ at 1,100°F (593°C) and 3,514 psia (24.2 MPa) which allows the turbines to be more compact. The specific heat of $CO_2$ is also half that of $H_2O$ so less heat is lost.

| Fluid | T (°F) | P (psia) | ρ ($lb_m/ft^3$) | $C_p$(Btu/lbm°R) |
|---|---|---|---|---|
| $H_2O$ | 1,100 | 3,514 | 4.3 | 0.71 |
| $CO_2$ | 1,100 | 3,514 | 8.8 | 0.30 |
| $CO_2$ | 1,300 | 3,000 | 6.7 | 0.31 |

**Table 1. Density and Specific Heat of $CO_2$ and $H_2O$ at Turbine and Pump Inlet Conditions**

Open loop gas turbine cycles operate at lower pressures (around 300 psia (2.1 MPa)) and higher temperatures (up to 2750°F (1510°C)). Higher temperatures increase cycle efficiency but also require special material and design considerations such as turbine cooling and single crystal superalloy blades with coatings to meet life requirements. Environments that include $H_2O$, such as those in steam cycles and cycles using combustion products as the working fluid, increase this challenge. The $sCO_2$ Brayton cycle has the potential to reduce the risk of operating at higher temperatures since the working fluid does not contain corrosive steam/water (with indirectly heated cycles) or has a smaller fraction of corrosive steam/water (with directly heated cycles). In addition, the $sCO_2$ Brayton cycle has a higher efficiency potential at a given temperature.

**Recompression Brayton Cycle System Analysis**

System studies were conducted on a closed loop, double recuperation recompression Brayton Cycle. All simulations were performed in Aspen Plus. The fluid properties required were obtained using the included NIST REFPROP database. Elements evaluated were turbine inlet temperature and pressure, turbine exit pressure, and main compressor inlet temperature effects on cycle efficiency.

*Turbine Inlet Temperature and Pressure Effect*

Turbine inlet temperature and pressure conditions were varied to determine the effect on system performance. Figure 5 shows the effect of turbine inlet conditions on cycle efficiency, along with the Carnot efficiency. The general trend for cycle efficiency is similar to that of the Carnot efficiency, with each increment of 500°F (260°C) yielding an efficiency benefit of ~5%. It can be seen that temperature is a stronger driver than pressure. The performance gain diminishes with increasing pressure. For lower pressures of 2,000-4,000 psia (13.8-27.6 MPa), there is roughly a 2-3% efficiency gain per 1,000 psia (6.9 MPa) increase. However, for pressures above 4,000 psia (27.6 MPa), there is little efficiency benefit.

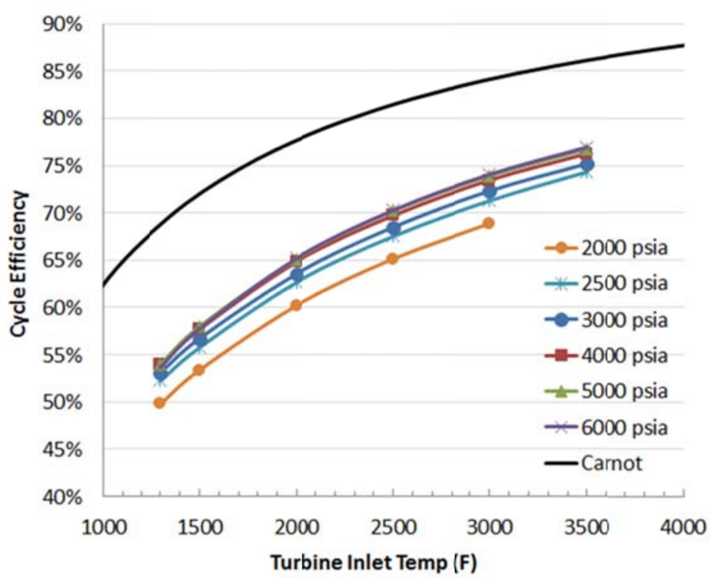


**Figure 5. Turbine Inlet Temperature and Pressure Effect on Cycle Efficiency**

*Turbine Outlet Pressure Effect*

The effect of turbine exhaust pressures was evaluated. The results are shown in Figure 6. Cycle efficiency was found to increase with increasing pressure from 200 psi to 1,100 psia (1.4 to 7.6 MPa), with the highest efficiency at 1,100 psia (7.6 MPa). The normalized mass flow rate, also shown in Figure 6, indicates that increasing exit pressures results in an increase in system mass flow. While the maximum efficiency is at 1,100 psia (7.6 MPa), the efficiency penalty for slightly subcritical conditions is small. Going slightly subcritical also offers a significant reduction in mass flow, which could allow for smaller ducts, turbomachinery, and heat exchangers; thus, reducing the footprint and capital cost. Reducing the turbine outlet pressure also offers the advantage of reducing the inlet temperature to the recuperator

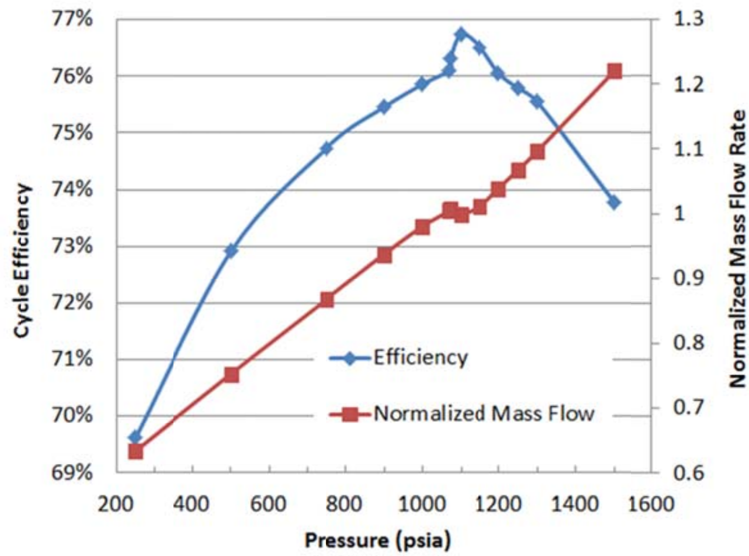


**Figure 6. Turbine Outlet Pressure Effect on Cycle Efficiency**

*Main Compressor Inlet Temperature Effect*

The effect of main compressor inlet temperature on cycle efficiency was also evaluated. Figure 7 shows the results of compressor inlet temperature versus cycle efficiency and normalized mass flow rate. Cycle efficiency achieves its maximum value at an inlet temperature of 90°F (32°C), which is near the critical temperature of $CO_2$. It can be seen that cycle efficiency drops more rapidly for inlet temperatures greater than 90°F (32°C) than for lower temperatures. This is due to a drastic increase in the main compressor work above 90°F (32°C), as seen in Figure 8. The increase in compressor work is consistent with the gas-like behavior of the fluid. For temperatures below the critical value, the compression work decreases. However, the amount of heat rejected increases. Since the increase in the amount of heat rejected is smaller than the decrease in compression work, there is still an efficiency drop but not as drastic as above 90°F (32°C). In addition, an examination of the behavior of the normalized system mass flow rate in Figure 7, indicates a significant decrease in this quantity allowing for smaller equipment.

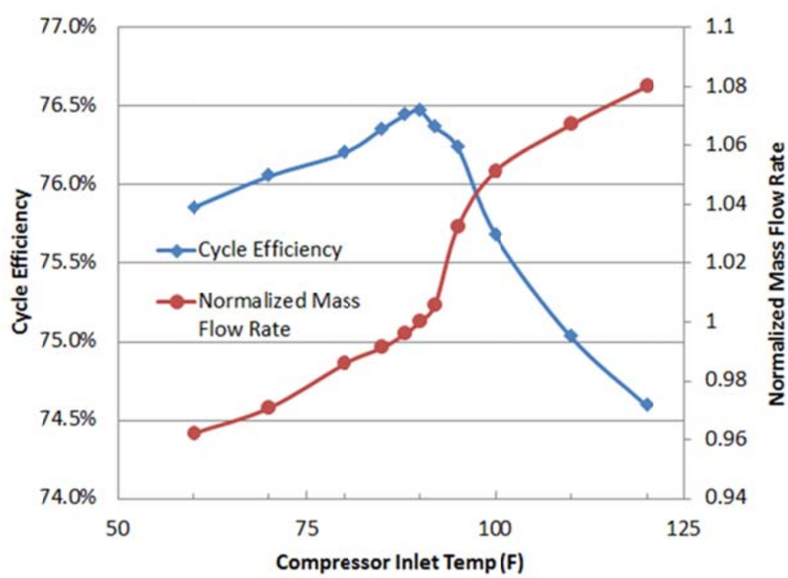


**Figure 7. Main Compressor Inlet Temperature on Cycle Efficiency**

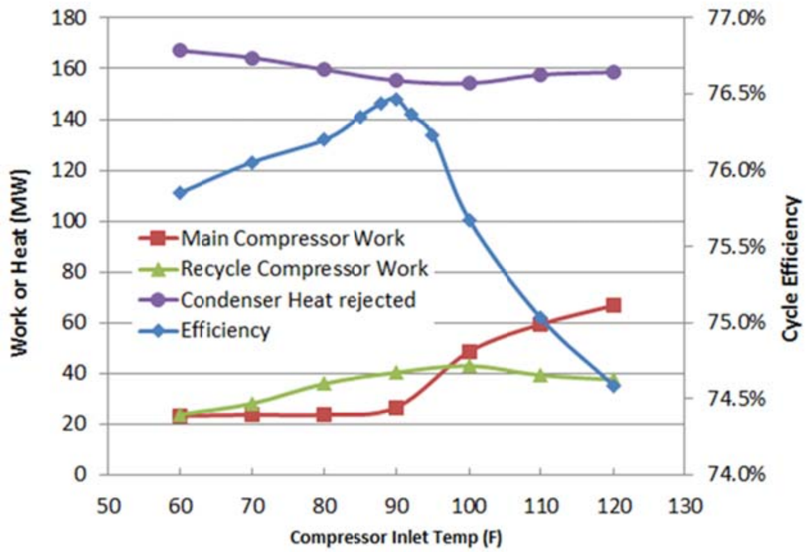


**Figure 8. Heat Duty or Work Load for Various Compressor Inlet Temperatures and Cycle Efficiency**

## CONCLUSION

The efficiencies of various systems, including the Rankine cycle and $sCO_2$ Brayton cycle, have been evaluated. Higher turbine inlet temperatures lead to higher efficiency. Significant efficiency gains have been cited for the Brayton cycle at higher inlet temperatures. Although the Brayton cycle does not offer efficiency benefits at lower turbine inlet temperatures, <1,000°F (<537°C), over the Rankine cycle, the higher density of $sCO_2$ and its liquid-like behavior in the supercritical region reduces turbomachinery size and compression costs. Combining this with potential safety enhancements, and/or lower operating and maintenance costs, is expected to lead to a lower cost of electricity compared to current technology.

Modeled analysis results of a double recuperation recompression Brayton cycle, simulated in Aspen Plus, have led to the following conclusions: 1) turbine inlet temperature is a stronger driver of cycle performance than turbine inlet pressure; 2) cycle efficiency is optimal for turbine exit pressures near the critical pressure of $CO_2$, although slightly subcritical conditions may still be considered due to other system benefits such as mass flow rate: and 3) a main compressor inlet temperature around the critical temperature is optimal for system performance.

## NOMENCLATURE

AR = Aerojet Rocketdyne
DOE = Department of Energy
HHV = Higher Heating Value
HRSG = Heat Recovery Steam Generator
IGCC = Integrated Gasification Combined Cycle
LCOE = Levelized Cost of Electricity
NGCC = Natural Gas Combined Cycle
Org RC = Organic Rankine
PC = Pulverized Coal
RegSCOT = Regeneratively Cooled Supercritical $CO_2$ Turbomachinery
RCBC = recompression Brayton Cycle
$SCO_2$ = Supercritical $CO_2$
SCOT = Supercritical $CO_2$ Turbomachinery
TS = Temperature-Entropy
ZEPS™ = Zero Emission Power and Steam

## REFERENCES

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Last name, Initial(s), Last Name, Initial, Year, “Title,” Paper Number or other source.


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