SAND2011-2690C

Experimental Investigation of the S-CO_2 Condensing Cycle

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Abstract

The S-CO_2 recompression Brayton cycle has shown promise as an efficient power conversion system for advanced high temperature nuclear reactors and other heat sources. In a typical proposed cycle, the compressor inlet operates at a pressure comfortably above the critical point to avoid two-phase conditions at any point within the cycle. However, it has also been proposed to operate the main compressor at cooler, single-phase liquid conditions below the critical pressure. For an operating CO_2 Brayton cycle, this would result in gas-phase CO_2 at the gas cooler inlet, and condensation within the primary waste heat rejection system. Exit conditions of the cooler would be a saturated or subcooled liquid. The net effect of expanding the operating envelope as described, creating a „condensing" Brayton or Rankine cycle, is greater efficiency for a given turbine inlet temperature.

The downside of this approach is that it requires lower heat rejection temperatures that may not always be available. Consequently, it may not be desirable to design a plant specifically for „condensing" operation, but instead to design a „standard" supercritical CO_2 cycle which has the flexibility to take advantage of lower environmental temperatures when possible. This is currently standard practice in steam-Rankine plants, which are known to operate at higher efficiencies in cooler weather. At many locations within the US, environmental temperatures in winter or at night are sufficiently low that CO_2 condensation below 88°F (31°C) is feasible.

Sandia performed a series of experiments to demonstrate the feasibility of operating the condensing cycle using hardware designed for operation above the critical point. Initial testing used the Sandia S-CO_2 Compression Loop. This is a high-pressure S-CO_2 loop (up to 2500 psi) with a compressor, heater, and a downstream expansion valve, but no turbine. Tests were performed to show that the radial compressor could effectively compress liquid CO_2. This also proved that the gas foil bearings, shaft seals, and other turbomachinery internals can operate with liquid CO_2 at the compressor inlet. Other tests expanded on this to show that the compressor could also be operated with gas-phase CO_2, and even two-phase mixtures (above 1000 psi). In addition this research loop was modified by adding a 50kW heater to warm the CO_2 following compression. With this upgrade, it was then possible in a single experiment to compress liquid CO_2, heat it in the heater, and then expand it in a valve, providing single phase gas to the gas chiller or a two-phase mixture to the gas chiller to demonstrate its performance as a condenser. The gas cooler used in initial tests was a spiral heat exchanger.

Following these results, similar tests were run on Sandia‟s larger S-CO_2 full Brayton cycle. The main compressor was run with subcooled liquid and with saturated liquid CO_2 inlet conditions while generating electricity. This loop is configured with Heatric printer-circuit heat exchangers (PCHEs), widely considered a critical component of the compact supercritical CO_2 power conversion cycle. Therefore these tests also confirmed the ability of these advanced geometry heat exchangers to operate as high pressure CO_2 condensers (again without modification).

The fundamental outcome of these tests is the flexibility implied for operating characteristics of a future CO_2 power plant: a full-scale multi-megawatt plant may be designed for „standard" S-CO_2 Brayton operation, but used in the condensing cycle mode as cooler temperatures are available, without significant losses or damage to hardware. Further, gains in efficiency allowed by use of the condensing cycle increase the competitiveness of the CO_2 cycle for application to lower temperature heat sources in cool climates, such as fossil, geothermal, solar thermal and light-water reactor (LWR) systems.

1. Introduction

Advanced power conversion systems that optimally couple to the thermal characteristics of next generation advanced reactors have the potential to provide higher efficiency nuclear electricity at greatly reduced costs. Improvements in plant efficiency can increase electrical output directly and have the same financial impact as direct reductions in construction and operating costs. Therefore, there is significant motivation to investigate power conversion system approaches that can maximize the power output of advanced reactors.

Supercritical Brayton cycles and other advanced supercritical cycles are one of the most promising approaches to achieving high efficiency and cost effective power conversion. These cycles have the potential to achieve higher efficiencies across the range of advanced reactor outlet temperatures. Also, the high power densities and liquid-like working-fluid densities throughout the system allow for the use of extremely compact power conversion machinery. Large-scale supercritical CO_2 systems are estimated to be about 1/10th the size of a comparable steam Rankine cycle, and have the potential to reduce power conversion system capital costs. These systems would also have strategic value because of their small mass, transportability, and their efficient coupling to almost any heat source including solar, geo-thermal, fossil, and nuclear systems. The Department of Energy (DOE) and a number of small and large industrial companies have active programs exploring supercritical CO_2 power systems for all the above mentioned applications.

This CO_2 condensing re-compression cycle was originally described in reports by Angelino [1, 2]. The key feature that differentiates the proposed cycle in this work from other S-CO_2 power systems is the condensation that occurs in the waste heat rejection unit. The lower compressor inlet temperature required for condensation also reduces the inlet pressure, and therefore increases the fluid density, resulting in a larger compression ratio and allowing for multiple stages of turbine reheating. This means that it operates essentially as a recuperated Rankine cycle. An important benefit of this research is in demonstrating that this power system can take advantage of lower heat rejection temperatures when they are available by allowing the CO_2 to condense in the gas chiller. This will increase the net power cycle efficiency, increase generated power, and improve economics without hardware modifications to the plant.

2. Benefits of the Condensing Cycle

A schematic of a typical re-compression Brayton cycle connected directly to the reactor coolant through a primary heat exchanger is illustrated in Figure 1. In the supercritical Brayton cycle, turbomachinery can be small due to the high power density of the fluid and because fewer stages (1-3) of turbine and compressor are needed. In addition the efficiency can be very high because compressor work is low, and because the re-compression cycle allows the recuperators to transfer 3-4 times more heat than is provided by the primary heat exchanger. In a highly recuperated loop, heat is transferred to the CO_2 fluid over a very limited temperature range (~150 K) which increases the cycle efficiency.

Typical state points for this power cycle, when connected to a LMR (liquid metal reactor) operating at a mixed mean exit temperature near 823 K (550 C), are illustrated in Figure 2 for a ~100 MWe power plant and a power cycle efficiency of 45.5%. This cycle analysis assumes 5% pressure drop in loop components and uses efficiencies of 85%, 87%, and 90% for the main-compressor, re-compressor, and turbine respectively.

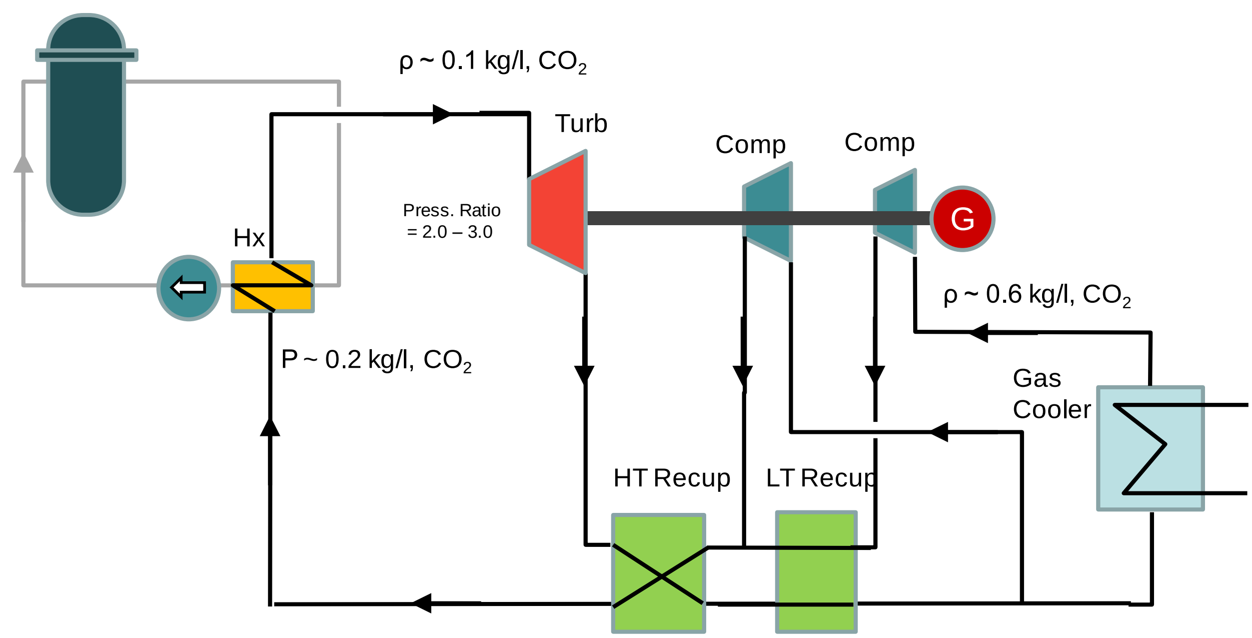

Figure 1: Schematic of supercritical CO2 re-compression Brayton cycle. The cycle is shown connect to a reactor through a heat exchanger.

The temperature-entropy (T-S) diagram for the Brayton cycle is shown by the red curve in Figure 2. The "condensing" power cycle is the same except that the gas cooler now condenses CO_2 to provide the compressor inlet with a saturated liquid. The cycle diagram for each is presented in Figure 2, which illustrates the efficiency gains that can be obtained by operating this cycle in a condensing mode, as shown by the blue lines. The smaller area below the heat rejection portion of the T-S diagram indicates greater power generation and greater efficiency, which is the reason for the interest in this condensing cycle. In this analysis the „condensing‟ cycle increases the efficiency to 48.3% for the same peak turbine inlet temperature [3].

For the "condensing" power cycle, the power conversion efficiency will increase as the heat rejection temperature is lowered. Calculations of power cycle efficiency were made based on models developed in Microsoft Visual Basic and Excel, dynamically linked to RefProp [4], and evaluated using Excel‟s native solver feature to optimize on various parameters.

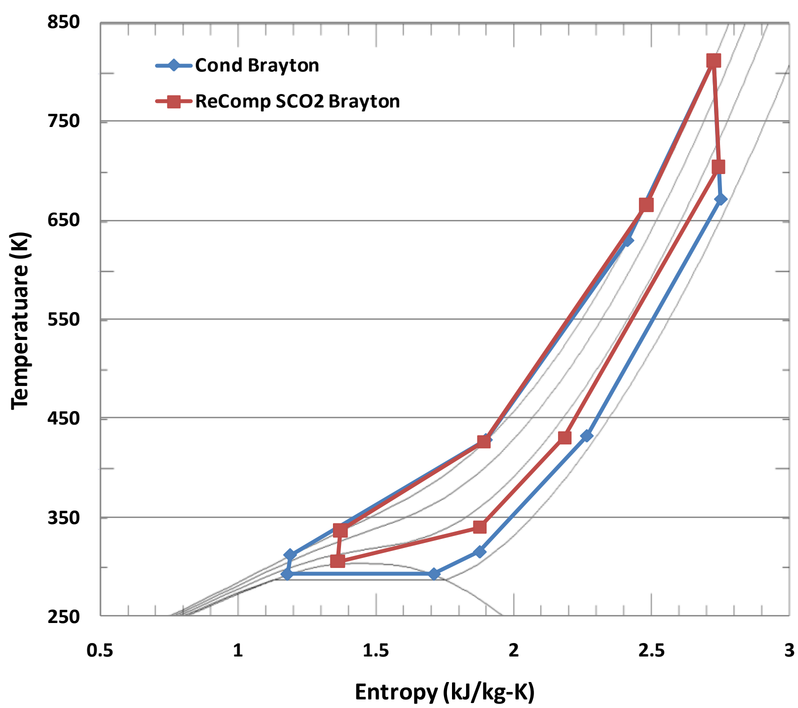

Figure 2: T-S diagram illustrating the re-compression S- CO_2 Brayton cycle (red) and condensing re-compression power cycle (blue). Both systems have a turbine inlet T of 810 K and a peak pressure of 20 MPa. The lines of constant pressure are shown at 5, 10, 15 and 20 MPa.

As can be seen, there is a marked improvement when using the condensation cycle at lower heat rejection temperatures. Lowering the compressor inlet temperature from 305K to 300K (80.3°F) increases the cycle efficiency by over 1.5%, and lowering it to 295 K (71.3°F) increases the cycle efficiency by 2.7%. For large power systems these are large increases in efficiency and they can have a strong impact on the economics of the power plant.

3. Experimental Work

The preceding discussion shows that the condensing re-compression power cycle can increase the cycle efficiency for power systems operating at a turbine inlet temperatures of 450 – 750°C by about 2.5- 3 percentage points for a 10 K reduction compressor inlet temperature (to about 71°F). To achieve these efficiency gains, it also requires a greater pressure rise in the pump/compressor, and requires that the waste heat exchanger provide liquid CO_2 to the compressor/pump inlet. Experimentally verifying these capabilities was the goal of this research project. These tests were carried out in the Sandia Compression Loop; schematic diagrams of this facility are provided in Figure 3. Temperature and pressure state-points are measured at the entrance and exit of each major component, as indicated on the figures as (1) compressor inlet, (2) compressor outlet, (3) heater outlet, and (4) expansion valve outlet. Mass flow and density were measured at compressor inlet using a Coriolis flow meter. A remote-controlled expansion valve is used to vary the mass flow around the loop, enabling the evaluation of compressor performance characteristics.

The loop (Figure 4) was fabricated by Barber Nichols Inc. [5] (Arvada, CO) under contract to Sandia. It is constructed primarily of 304 and 316L stainless steels, and has a maximum allowable working pressure of 18MPa (2600 psi) at 820K. The compression loop uses a 50kWe motor driven compressor, capable of spinning the compressor at design speeds up to 75,000 rpm with a pressure ratio of 1.8 and a flow rate of 3.53kg/s for a compressor inlet condition of 305.3K(89.87°F) and 7.690MPa (1115 psi). More details regarding this facility can be found in Wright, et al [6].

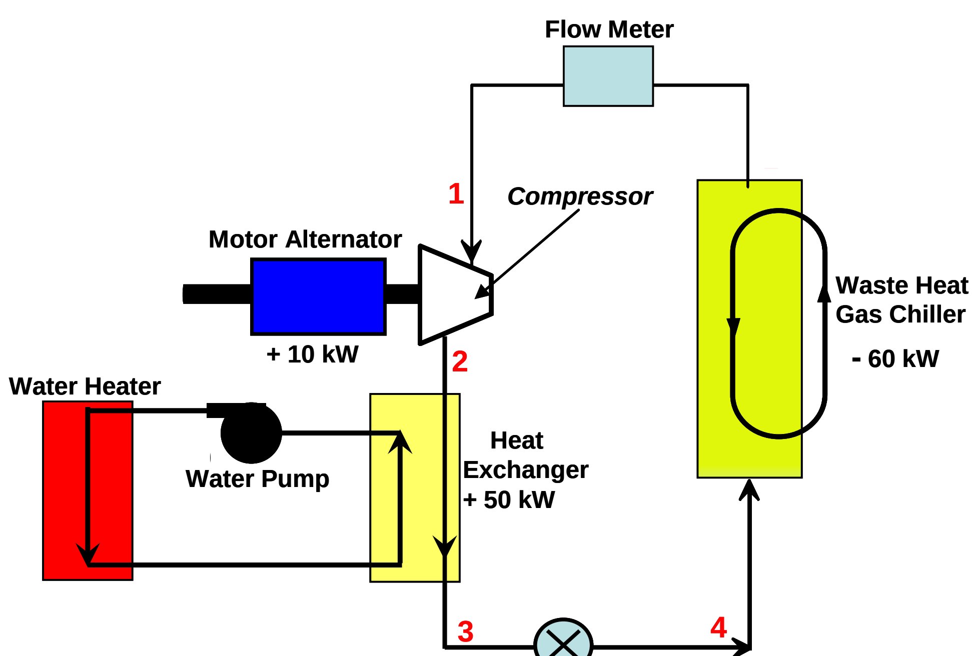

Figure 3: Schematic of the Sandia Turbocompressor Test Loop.

Figure 4: Photograph of the Sandia Turbocompressor Test Loop.

A condensing supercritical-CO_2 power cycle must have hardware that can accomplish three things on the cold leg of the power cycle. First the compressor must be able to "pump" or compress liquid-like fluid densities; second, the pump/compressor must also be able to provide sufficient pressure rise when pumping liquids rather than gases; and finally the waste heat rejection exchanger (gas cooler) must be able to condense the CO_2 and thereby provide a continuous stream of liquid CO_2 when operated at the lower temperatures and pressures than exist at the critical point.

Three types of experiments were performed to explore the behavior of the Sandia compression loop when operated as a condensing power system.

1. The first set of experiments were used to compress/pump liquid CO_2 using the S-CO_2 compressor.
2. The second series of experiments operated the compressor over a wide range of compressor inlet conditions that varied from pure vapor to pure liquid. These experiments also compressed two phase CO_2.
3. The third series of experiments required hardware modifications to the compression loop to add heating power to the CO_2 after compression. The upgraded heater added sufficient enthalpy to the fluid (50 kW) so that after expanding the CO_2 through the static valve, the fluid would either be a saturated gas or a two phase fluid well under the critical temperature and pressure.

Each of these three tests is summarized briefly below.

1) In a condensing supercritical-CO_2 power cycle, the compressor inlet properties consist of a fluid with liquid-like densities and with temperature and pressure below the critical point. The as-designed compressor inlet temperature and pressure are 305.4 K and 7.7 MPa, with a density

of 0.579 kg/liter. At Sandia the tests were performed by filling the loop to the appropriate fill mass and operating the gas chiller with sufficiently cold water so that the compressor inlet had liquid-like densities. In these tests the CO_2 fluid density at the compressor inlet was 0.793 kg/liter at 295K. This was chosen to match inlet conditions of the ~3% efficiency increase predicted in Section 2.

The turbocompressor was then operated at speeds of 35, 40, 45, and 50 krpm. For each new speed, the expansion valve setting was varied in order to vary pressure ratio and mass flow around the loop. This was used to create a compressor map for this turbocompressor operating in the liquid CO_2 regime. Data for the corrected mass flow is plotted vs. corrected ideal specific enthalpy rise, for a given corrected compressor speed. This was overlaid onto curves drawn for design conditions shown in Figure 5. The term „corrected" refers to similarity factors developed by Barber-Nichols Inc to account for differences between the design point and actual operating conditions [5].

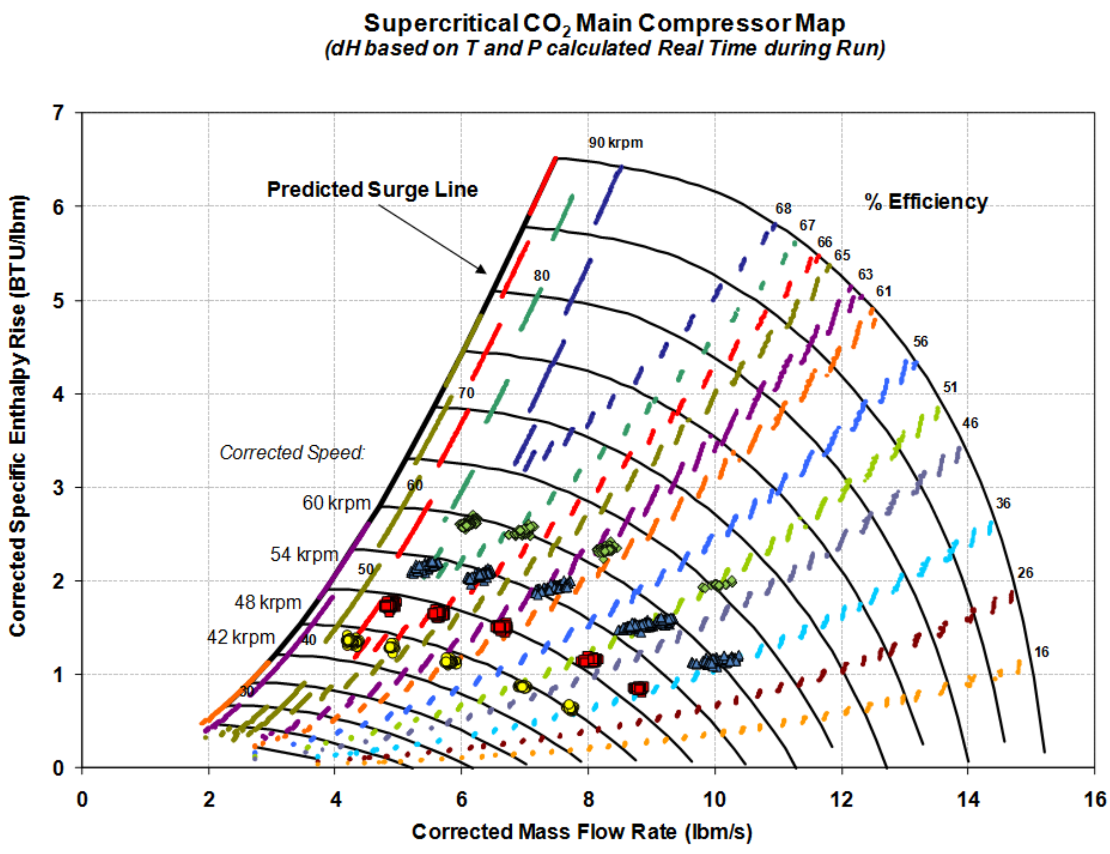

Figure 5: Compressor Map drawn at conditions of 0.793kg/L, 295K.

As seen in the figure, the corrected factors work fairly well in placing the liquid data on the design curves. Some deviations are seen towards the higher speed, higher mass flow cases where it can be seen that the curves implied by the data are not as steep.

Another illuminating way to view this data is in the form of efficiency vs. flow coefficient, and head coefficient vs. flow coefficient curves. An example can be seen in Figure 6 below.

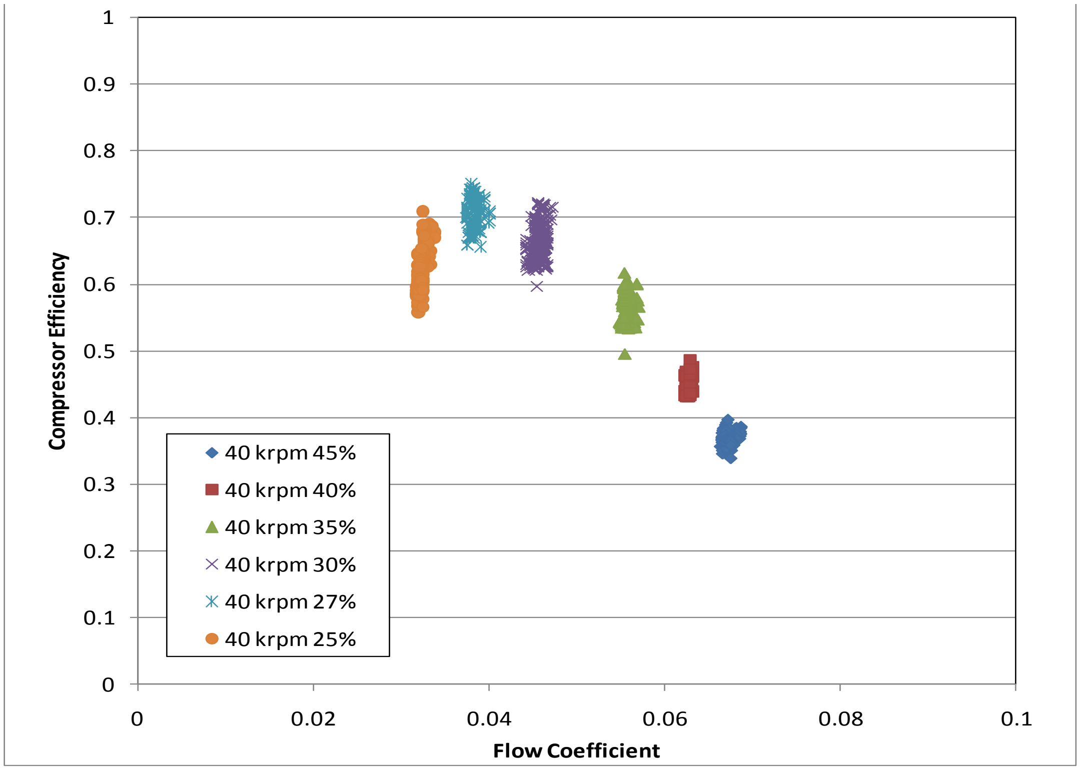

Figure 6: Compressor efficiency data at conditions of 0.793kg/L, 295K.

In this case all data are from a single test run at 40,000 rpm. The percentage indicated in the legend is the fractional opening of the expansion valve. As the valve is closed, pressure ratio increases, and mass flow drops off. Initially this results in increasingly larger compressor efficiencies. Finally a peak is reached at the 27% open valve position. This and other more detailed analysis has demonstrated that the compressor operating in the liquid regime could achieve efficiencies on par with the design point (~70%).

2) It is also of interest in advanced power cycles to demonstrate that the supercritical fluid compressor can operate over a wide range of other conditions, aside from this liquid data point at 295K. Tests in single phase liquid, gas, and two-phase were carried out to simultaneously investigate CO_2 equation of state and compressor operation. A sample of these results is shown in Figure 7: These tests began with the compressor operated at a steady 25,000 rpm, and the loop filled to a dense liquid state. While adding heat, and continually running the compressor to ensure good mixing around the loop, the CO_2 inventory was slowly decreased to maintain constant pressure as temperature increased. This was accomplished for pressures of 1000, 1060, 1080, and 1100 psi.

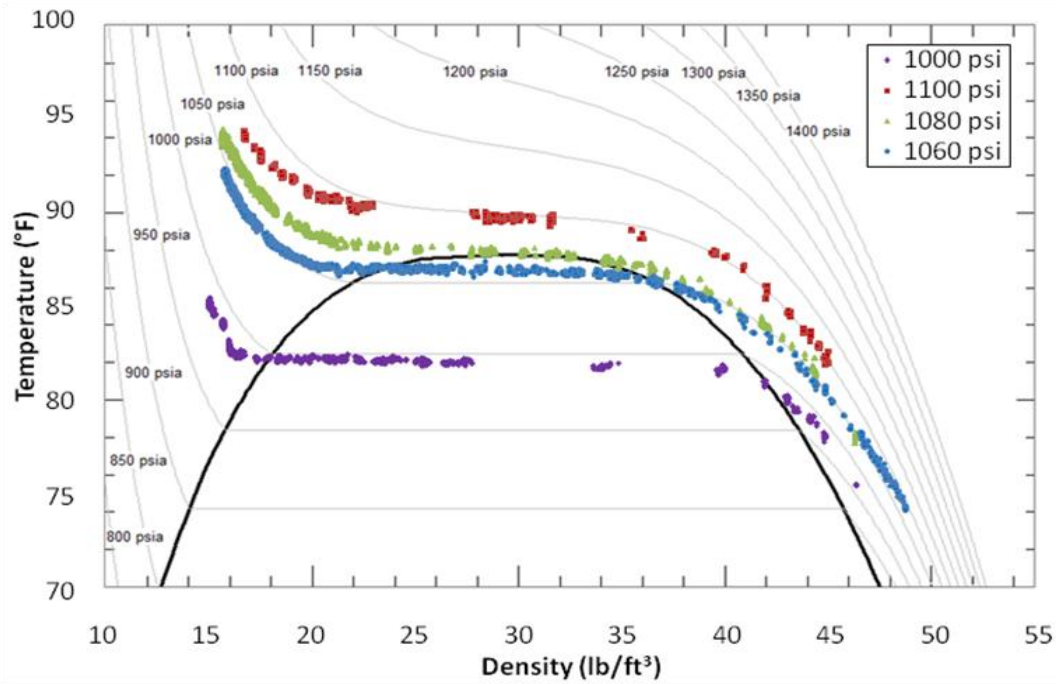

Figure 7: Various compressor inlet conditions that have been tested within the facility.

The compressor operated stably over the inlet conditions shown in Figure 7, as evidenced by steady mass flow rates and pressure ratios. As the saturation curve was approached from the liquid or vapor side, mass flow measured by the Coriolis flow meter began to oscillate about its average value by ~ 5%, indicative of the onset of two-phase flow. (In fact, this phenomenon was later developed into a technique for roughly measuring the saturation curve of supercritical fluid mixtures.) Still, two-phase operation did not cause cavitation or otherwise hamper the ability of the compressor to operate smoothly. This is likely due to the small difference in properties between the liquid and vapor phases at pressures approaching the critical point. The amplitude of these two-phase oscillations has been seen in other test cases to increase with decreasing pressure, and presumably there exists a pressure below which operation in two-phase will cause severe issues at the compressor.

3) The next set of tests aimed to test the ability of the waste heat unit to condense CO_2, without need for modifications or auxiliary equipment such as a separator. In a steam Rankine plant, condensers must use gravity to separate the liquid from the vapor prior to the feedwater pump inlet. In an S-CO_2 power system, waste heat rejection would take place at high pressure (> 6 MPa ~900 psia), not much below the critical point. At this pressure, the liquid/vapor density ratio is on the order of 3:1, not 1000:1 as in steam system condensers, this may decrease complexity of hardware required to run the CO_2 condensing cycle.

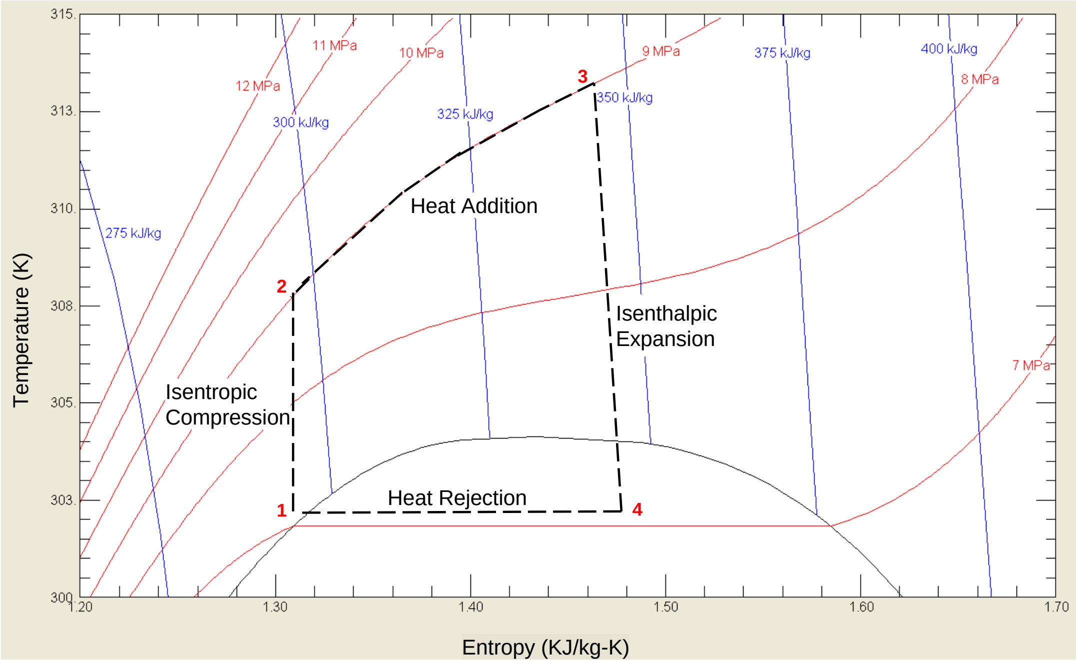

Figure 8: Predicted T-S diagram of the SNL CO_2 compression loop in the condensing configuration.

In order to test condensing, the compression loop cycle was run as conceptualized in Figure 8. The T-S points illustrate the paths of compression, heating, expansion and condensation and are shown by the state-points located at the exit of the compressor (2), heater (3), expansion valve (4), and gas cooler (1). To accomplish this, the research loop was upgraded with a 50kW indirect heating loop. Tests were run such that the loop operated at steady state, with heat added following compression, and an equal amount exhausted to the gas chiller.

A selection of data from these runs is shown in Figure 9. The top figure shows loop operation near the design point for all hardware. Then, the compressor inlet temperature is reduced and inventory added to achieve to eventually achieve the conditions pictured in the bottom figure. For these tests, entropy state points are calculated using RefProp and two state properties. At the compressor inlet, entropy is evaluated from measured pressure and density values; at other locations, entropy is calculated using measured temperature and pressure values. This is due to the fact that points near or within the saturation curve cannot be accurately captured from the temperature/pressure relationship since these curves are flat in the saturation region. This also causes an issue with the “Valve Out” data in the middle and bottom panels of Figure 9. This state point is within the two-phase region; therefore the value found using RefProp (in purple) is incorrect. No density meter exists at this location, therefore an assumption of isenthalpic expansion through the valve can be assumed for estimation, shown as “Valve Out-Corrected”. The top panel of Figure 9, as well as Figure 10, confirm that the isenthalpic assumption is appropriate (in these cases, no approximations were used, and expansion is seen to follow a constant enthalpy line).

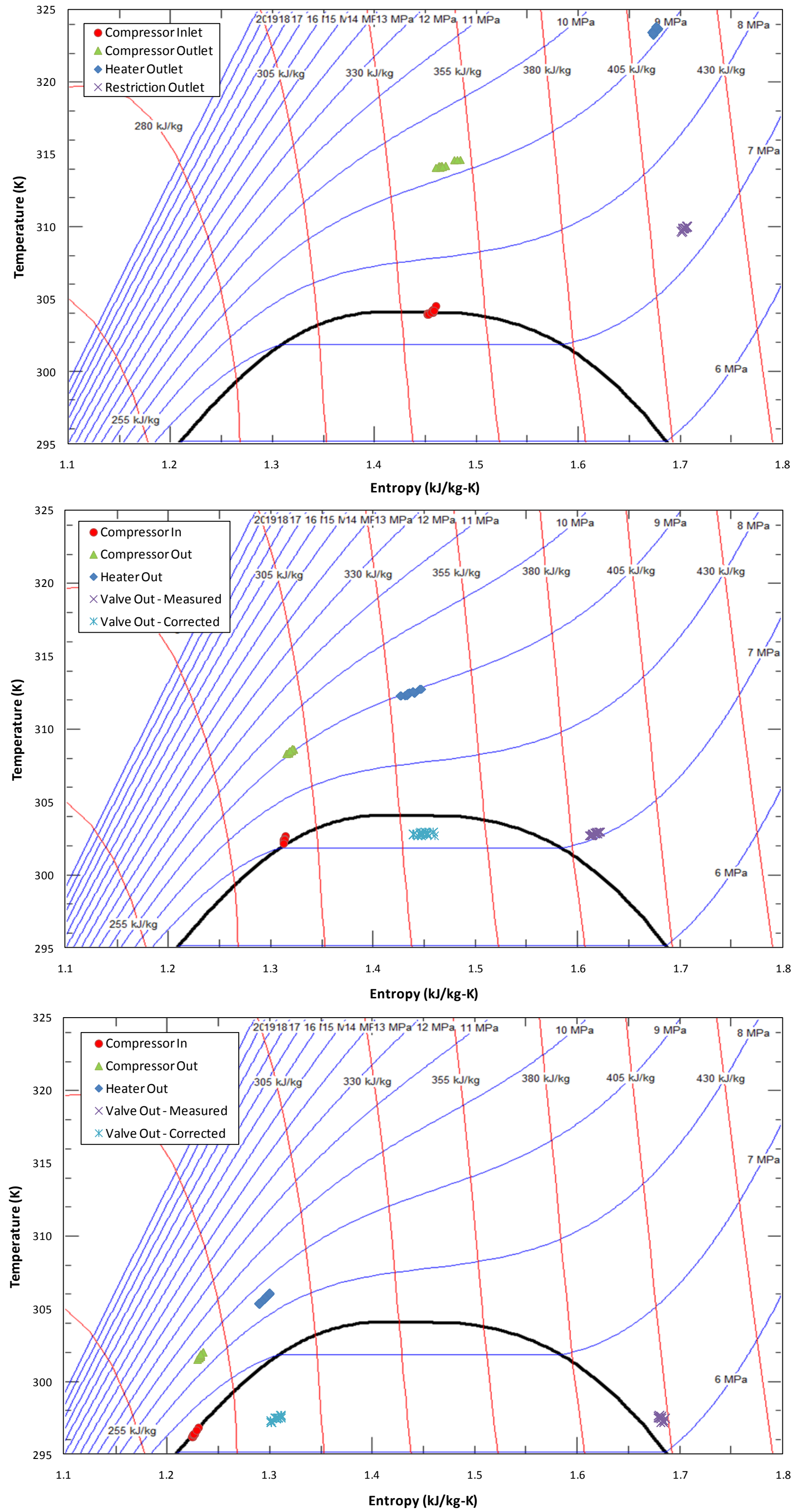

Figure 9: T-S diagram of S-CO_2 compression loop with: (Top) supercritical CO_2 at compressor inlet. (Middle) T-S diagram of S-CO_2 compression loop with liquid CO2 (303 K) at compressor inlet. (Bottom) liquid CO_2 (296 K) at compressor inlet.

In all probability, depending on cooler design, pressure, and degree of sub-cooling below the critical point it won‟t always be possible to fully condense the CO_2 without use of a vapor separator or other machinery. To date this has not been observed. Figure 10 shows another case similar to Figure 9 in which the compressor is operated in the two-phase region. A larger scatter of data at the compressor outlet and heater outlet is seen, indicative of minor two-phase oscillations in the system.

Here, the compressor inlet temperature was cooled below the critical temperature, but not enough CO_2 mass was present in the system to reach the liquid state. This helps to illustrate that in a future CO_2 power plant, as the compressor inlet is cooled below the critical point towards a condensing cycle, it is also necessary to increase the fill mass of the system to assure that the exit conditions from the gas cooler are liquid and not two-phase. Thus some form of inventory control is required to increase or decrease the loop fill mass in any system that wishes to intermittently use the condensing S-CO_2 power cycle.

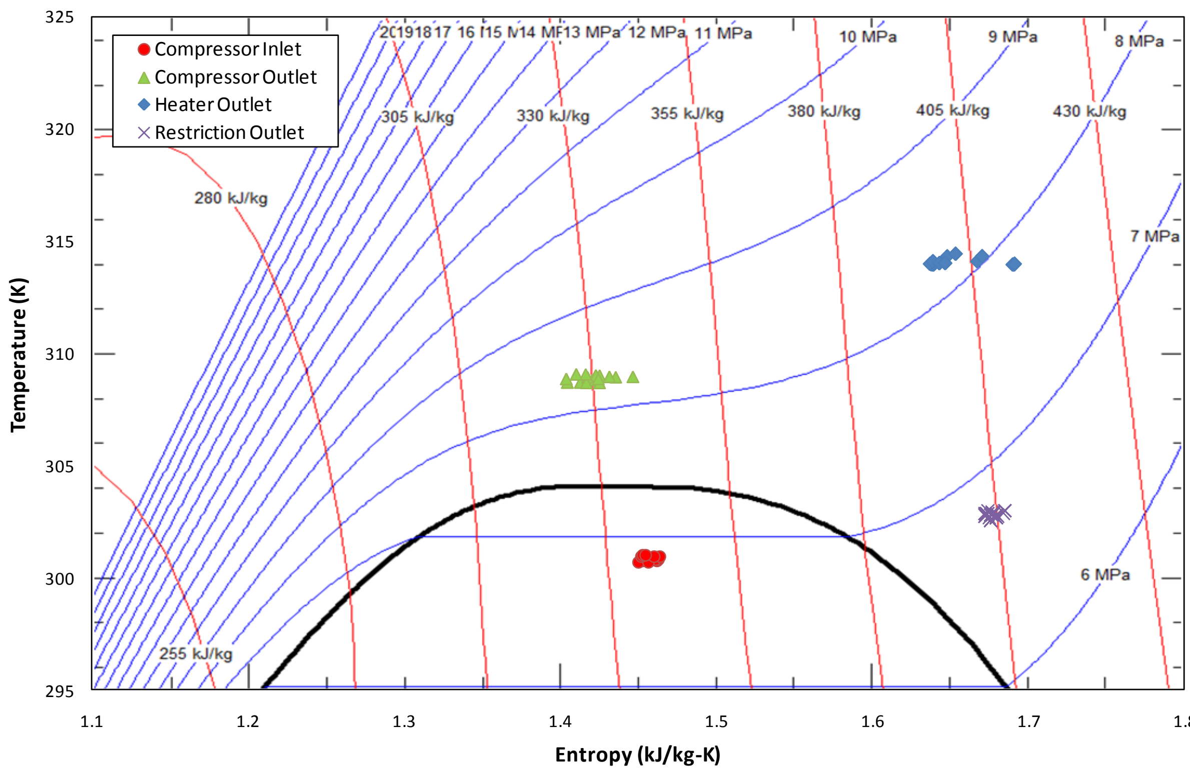

Figure 10: T-S diagram of S-CO2 compression loop with two-phase liquid CO2 (296 K) at compressor inlet.

Following the successful results demonstrated in this work, a fourth and final series of experiments was run on the Sandia recuperated split-flow Brayton loop during which electric power was produced, while the main compressor operated in the saturated liquid region. These tests are not detailed in this report but included in a companion paper [7].

Overall, this combination of tests show that the supercritical compressor can "pump" liquid-phase CO_2; that a spiral or PCHE gas chiller heat exchanger, when fed two-phase CO_2, can produce liquid CO_2 if sufficient cooling is provided; that the main compressor is capable of compressing liquid, vapor, and two-phase CO_2; and that a simple heated recuperated Brayton cycle can produce net electric power while the compressor inlet is in the two phase regime. At this stage, these tests provide strong evidence that condensing S-CO_2 power cycles are indeed possible and the theoretically predicted improvement in efficiency will not be limited by hardware issues. More testing of this type is warranted and will be performed on the S-CO_2 Compression Loop and on the simple heated recuperated Brayton cycle.

4. Summary and Conclusions

The condensing cycle described in this work can achieve larger thermodynamic efficiency that the „standard" S-CO_2 cycle by lowering compressor inlet temperature, pumping liquid CO_2, and allowing condensation to occur in the heat rejection unit. Because of the greater pressure ratio required in the compressor and turbine, it is possible to add multiple stages of reheat which also may increase efficiency further. These characteristics describe a cycle which is, on the cold side, more reminiscent of the Rankine cycle used at steam plants. Furthermore, the cost savings associated with the much smaller CO_2 power conversion system is another factor which favors this cycle over the larger power conversion cycle used by steam plants. Large-scale supercritical CO_2 systems are estimated to be about 1/10th the size of a comparable steam plant, and therefore have the potential to dramatically reduce capital costs.

An experimental program was conducted to demonstrate the feasibility of operating closed Brayton cycle turbomachinery and other loop components in the range of conditions required for the condensing cycle. Tests were carried out to run the compressor using a high density liquid CO_2 at inlet, which yielded compressor maps at several rotation speeds showing high efficiency and large compression ratios. Another series of tests involved operating the compressor at low speed (25krpm) while tracing constant pressure lines of CO_2, from a liquid, through the two-phase zone, and into the gas phase. The stable operation of the compressor, all other turbomachinery internal components, and consistent readings of the instrumentation during these scenarios instilled confidence that the loop was capable of operating at all necessary points.

Finally, a 50kW heater was added to the loop in order to simulate a heated Brayton cycle. Again the compressor was operated with single-phase liquid inlet conditions; the fluid was heated, and then passed through a restriction and into the two-phase saturation region. Here, the gas chiller acted as a condenser, providing a saturated liquid to the compressor. This demonstrated that the spiral chiller could be used as-designed for the purpose of condensing, requiring no separator or hardware modifications. These results imply flexibility in operating conditions that would allow a full-scale plant to be designed for "standard" S-CO_2 Brayton operation, but used for condensing when cooler temperatures are available. From the work described here and that being performed in concurrent studies, the supercritical CO_2 cycle is emerging as a system which the picture of stable operation over a variety of conditions and applications, in contrast to early concerns that these systems would be hostage to a narrow, unstable band.

5. Acknowledgements

The authors thank the DOE-NE and the Sandia Laboratory Directed Research and Development program for funding this work.

Sandia National Laboratories is a multi-program laboratory managed and operated by Sandia Corporation, a wholly owned subsidiary of Lockheed Martin Corporation, for the U.S. Department of Energy's National Nuclear Security Administration under contract DE-AC04-94AL85000.

6. References

[1] Angelino G., "Perspectives for the Liquid Phase Compression Gas Turbine", Journal of Engineering for Power, Trans. ASME, Vol. 89, No. 2, pp. 229-237, (1967).

[2] Angelino G., "Carbon Dioxide Condensation Cycles for Power Production", ASME Paper No. 68-GT-23, (1968).

[3] Wright, S., Radel, R., Conboy, T., "Modeling and Experimental Results for Condensing Supercritical CO_2 Power Cycles", SAND2010-8840, (2010).

[4] Lemmon, E.W., Huber, H.L., McLinden, M.O., REFPROP – A Computational Tool for Reference Fluid Thermodynamic and Transport Properties, NIST Standard Reference Database 23, Version 8.0, (2007).

[5] Barber-Nichols Incorporated, Arvada, CO, USA (2011).

[6] Wright, S., Radel, R., Vernon, M., Rochau, G., and Pickard, P., "Operation and Analysis of a Supercritical CO_2 Brayton Cycle", SAND2010-0171, (September 2010).

[7] Wright, S., Conboy, T., Fuller, R., "Break-Even Power Production in a Supercritical CO_2 Brayton Cycle" (in Proceedings of the Supercritical CO_2 Power Cycle Symposium, Boulder, Colorado, May 24-25, (2011).