DRAFT

SAND2019-10299C

# Guidelines For The Design And Operation Of Supercritical Carbon Dioxide R&D Systems

Matthew D. Carlson,[1, a)]

[1]Concentrating Solar Technologies, Sandia National Laboratories, P.O. Box 5800, MS-1127, Albuquerque, NM 87185-1127, USA

[a)]Corresponding author: mcarlso@sandia.gov

**Abstract.** This paper captures guidelines for the design and operation of $sCO_2$ systems for research and development applications with specific emphasis on single-pressure pumped loops for thermal-hydraulic experiments and implications toward larger $sCO_2$ Brayton power cycles. Direct experience with R&D systems at the kilowatt (kW), 50 kW, 200 kW, and 1 megawatt thermal scale has resulted in a recommended work flow to move a design from a thermodynamic flowsheet to a set of detailed build plans that account for industrial standards, engineering analysis, and operating experience. Analyses of operational considerations including $CO_2$ storage, filling, pressurization, inventory management, and sensitivity to pump inlet conditions were conducted and validated during shakedown and operation of a 200 kilowatt-scale $sCO_2$ system.

## INTRODUCTION

Research in to Supercritical carbon dioxide ($sCO_2$) Brayton power cycles has flourished over the past decade based on their potential to improve on the thermal efficiency, capital and operating expense, and size of conventional and high-temperature steam Rankine power systems. Some individual components and operational strategies are familiar to those in the power generation industries and refrigeration industries, yet the unique overlap of both low-temperature and high-temperature considerations in high pressure $sCO_2$ systems along with operation of sensitive turbomachinery near the critical point of the working fluid has required significant iteration with limited publication of best practices.

Lessons learned from this experience must be understood by the Generation 3 concentrating solar power (Gen3CSP) gas, liquid, and particle pathway teams as well as the broader community of CSP system integrators to safely and confidently operate $sCO_2$ power cycles and achieve levelized costs of electricity (LCOE) less than 6 ¢/$kWh_{electric(el)}$. [1] This work summarizes key elements related to the design work flow and operational considerations of $sCO_2$ R&D systems with emphasis on the type of single-pressure pumped loop planned for cooling the Gen3CSP primary heat exchanger.

## DESIGN WORK FLOW

The design work flow for $sCO_2$ R&D systems can proceed along the same lines of any systems engineering effort as shown along the left-hand branch of Figure 1 with tight feedback loops for design reviews at each stage of increasing detail and definition, and a corresponding validation and verification (V&V) process shown along the right-hand branch during the implementation phase

A set of R&D goals defined by the customer or developed through discussions with the customer provide the highest-level direction on the overall system design and derived requirements throughout the process and are used to evaluate the delivered system at the end of the R&D lifecycle.

A conservative set of interfacing and system requirements should then be developed to define a relatively stable set of boundary conditions for the R&D system and for use in final acceptance testing. For $sCO_2$ systems the greatest

DRAFT

challenge at this stage is defining maximum allowable working pressures (MAWPs), mean-design metal temperatures (MDMTs), and non-welded connections due to the high operating pressures and potentially high operating temperatures involved compared with standard commercial equipment.

System configurations and component design options can then be explored while also working with vendors on specific component design requirements to evaluate the feasibility of different configurations and components given the cost, schedule, and facility limitations of the system. At this stage must tradeoffs between commercial readiness and component performance must often be made due to requirements for high operating pressure and the complexities of $sCO_2$ fluid behavior at design, off-design, and operationally necessary conditions ranging from supercritical, two-phase, sonic gas, cryogenic liquid, and even solid phases. The configuration and component requirements resulting from these iterations are then used for procurement, inspection, and acceptance testing during the implementation phase.

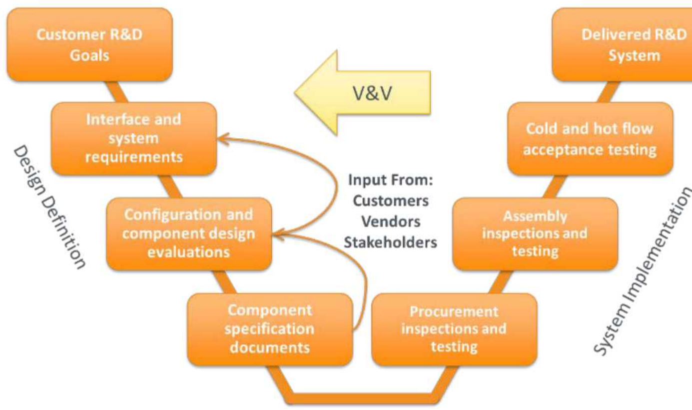
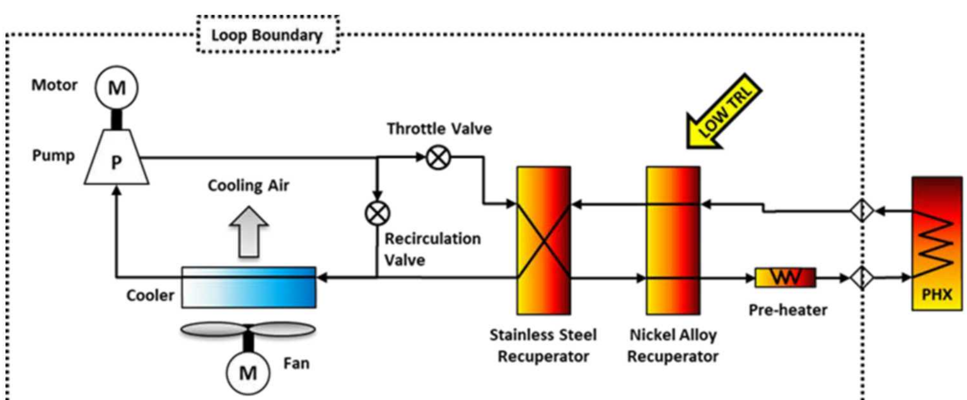



Figure 1. A diagram of the V model for an R&D system lifecycle.

## Design Point Thermodynamic Modeling

Thermodynamic modeling of the desired process flow diagrams for various system configurations can provide rough order of magnitude estimates of component requirements assuming reasonable efficiencies for compression processes and component pressure drops as shown in Figure 2, Figure 3, and Table 1. This design seeks to provide $sCO_2$ cooling flow to a PHX under test by the Gen3CSP gas, liquid, and particle pathway teams with a nominal temperature rise of 150 °C and outlet conditions of 715 °C at 250 bar with at least 1 $MW_{th}$ heat removal.

These results can be used to estimate individual component performance and assess tradeoffs among design requirements with implications toward cost and performance. Iteration on these requirements with original equipment manufacturers from the longest lead-time component to commercial off-the-shelf (COTS) products can refine the system design and set up V&V criteria during implementation. The recuperated design shown is one of several system configurations modeled to understand the behavior of component sizes and identify the option with the lowest potential cost and highest expected technology readiness level.



DRAFT

Figure 2. A process flow diagram of a recuperated single-pressure primary heat exchanger (PHX) test system.

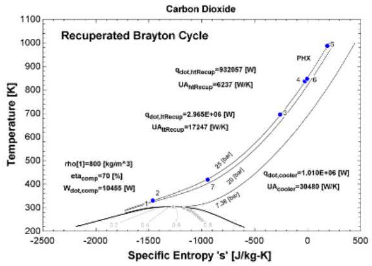



Figure 3. A temperature vs. specific entropy diagram of the recuperated PHX test system.

| State | T [°C] | P [bar] | ρ [kg/m³] |
|---|---|---|---|
| 1 | 56.7 | 243 | 800 |
| 2 | 57.6 | 259 | 807 |
| 3 | 423 | 256 | 190 |
| 4 | 565 | 254 | 153 |
| 5 | 715 | 250 | 127 |
| 6 | 575 | 248 | 147 |
| 7 | 147 | 245 | 416 |

Table 1. Temperature, pressure, and density for each state point.

## Designing for Flexible Operating Conditions

Careful selection of the system design point can provide significant flexibility to adapt to changing customer requirements and future operation. The mass flow rate required to achieve a range of design points for the $sCO_2$-side of the PHX described previously is shown in Figure 4 over a range of design temperature rises or inlet temperatures. Selecting a system design point with the lowest expected temperature rise, corresponding to the highest required flow rate, results in a system that can still achieve a wide variety of other design conditions with different levels of turn-down without any modification to the system configuration or equipment through a combination of flow rate control and recuperator bypass control.

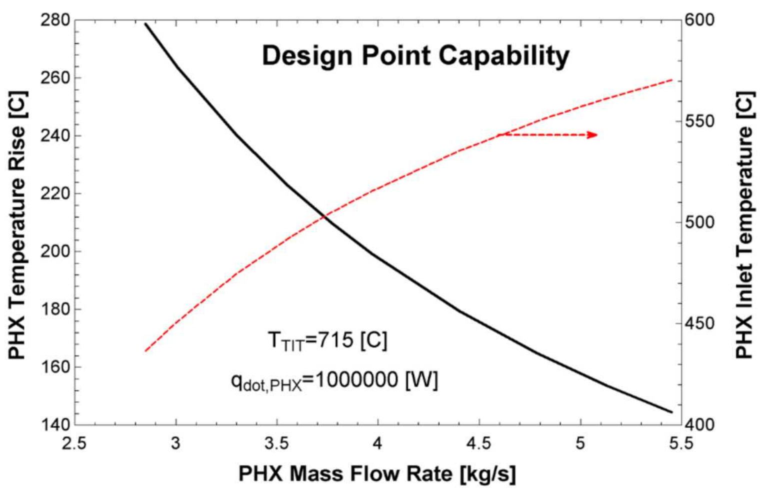



Figure 4. Design point capability for a 1 $MW_{th}$ PHX delivering $sCO_2$ at 715 °C for a range of different $sCO_2$-side design temperature rises or inlet temperatures given various mass flow rates.

## Overpressure Protection

Requirements and guidelines for overpressure protection of different equipment are provided in American Society of Mechanical Engineers (ASME) boiler and pressure vessel (BPV) code and piping code standards, American Petroleum Institute (API) standards and recommended practices, and local site-specific requirements which may be more stringent than industry standards. Failure to provide sufficient overpressure protection could ultimately result in death or injury of on-site personnel in addition to unrecoverable damage to equipment.

Equipment used in an $sCO_2$ system must be designed to accommodate the highest possible combinations of pressure (MAWP) and temperature (MDMT) expected under operating conditions with allowance for elevation of system pressure during a pressure relief event as shown in Figure 5. For single-pressure systems the pressure is nearly uniform throughout the system and the most extreme operating condition must be considered, but for power cycles operating at more than a single pressure this may include elevated settle-out pressures seen by equipment normally operating at lower pressures.

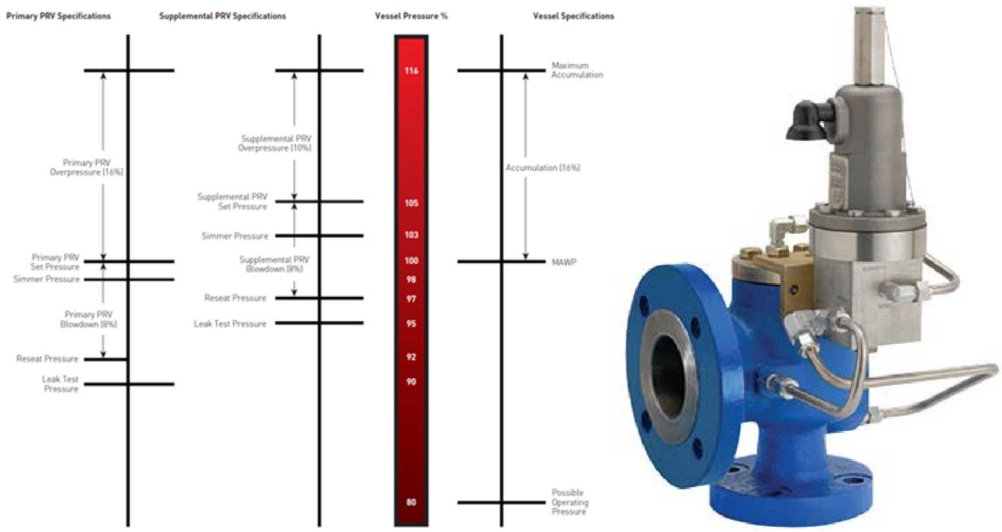



Figure 5. A diagram of allowable set, reseat, and overpressure points relative to system MAWP for an ASME BPVC Section VIII installation with multiple relief devices under overpressure conditions not involving external fire [2] and an example pilot-operated pressure relief valve used to protect a system from overpressurization.

## Carbon Dioxide Storage

$CO_2$ for industrial uses is generally stored in pressurized vessels at saturation conditions providing potential to draw liquid, gas, or either phase from the storage tanks. The main difference between these storage approaches is the amount and type of insulation required depending on the saturation temperature and pressure of the storage system. Storage of $CO_2$ as a liquid is the most compact method, but the liquid density quickly rises with temperature requiring highly effective insulation or active cooling for long-term storage.

Dewar systems provide the longest storage time of saturated $CO_2$ using vacuum insulation and typically have a maximum saturation pressure of 300 psig as shown in Figure 6. The inter-wall volume of a double-walled pressure vessel is evacuated to form a vacuum-insulated pressure chamber with additional multi-layered insulation to maintain the $CO_2$ at its relatively low saturation temperature close to 0 °F. These dewars can be equipped with caster wheels for mobility or moved on dewar carts by a single person and are often delivered to a site by a local gas supplier. Larger installations require built-in dewar systems resupplied from $CO_2$ delivery trucks which use the pressure of the vapor space above the $CO_2$ liquid to transfer liquid from the delivery truck to the on-site dewar as vapor in the on-side dewar is vented.

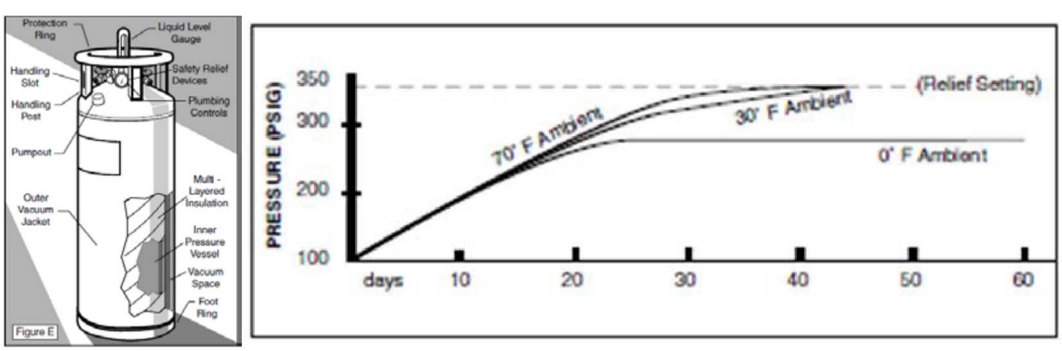



Figure 6. A diagram and cross-section of a typical lab-scale dewar system and holding times for saturated $CO_2$ in a vacuum-insulated Chart Dura-Cyl $CO_2$ dewar at various ambient temperatures.

## DESIGN WORK FLOW

An analysis of operational modes provides guidance for design requirements of R&D system components and control needs for stable operation.[3]–[5] At each stage of operation unique combinations of thermodynamic and thermal hydraulic conditions can occur that must be considered when designing and operating a system in order to avoid adverse conditions. sonic flow and solid $CO_2$ freeze plugs within fill and vent lines can reduce or block flow unexpectedly. Cryogenic cold shock from liquid $sCO_2$ injection and system-wide heating and cooling from pressurization and depressurization may exceed thermal ramp rate limits for individual components or piping networks. Finally, inventory management must be designed to maintain appropriate $sCO_2$ thermodynamic conditions within each segment of the loop from startup through shutdown without excessive delays due to the larger thermal mass within high pressure Brayton cycle recuperators.

### Filling to Supercritical Conditions

The filling process should occur in three stages in order to avoid local or system thermal shock and to fill the system as quickly as possible. A vacuum hold is first conducted to remove contaminating gasses and water before $CO_2$ gas is backfilled into the system. Next, the system should be equalized with the vapor pressure of the $CO_2$ storage system, and at least 100 psig, to avoid the potential for dry ice formation in the system by expanding liquid $CO_2$ flow below the triple-point of $CO_2$. Finally, liquid can be transferred into the system to increase the average system density and eventually pressurize the system to startup conditions where the density at the pump is high enough to provide proper bearing lubrication and motor cooling.

Trace heating should be used on the hot-side equipment downstream from the recuperator to keep this equipment at a temperature well above the pseudocritical temperature $T_{pc}$ shown in Figure 7. This ensures that the density of the $CO_2$ on the hot-side of the system during filling and at supercritical conditions remains low so that a minimal amount of pressure trim venting or inventory recovery is required as the system heats to operating conditions.

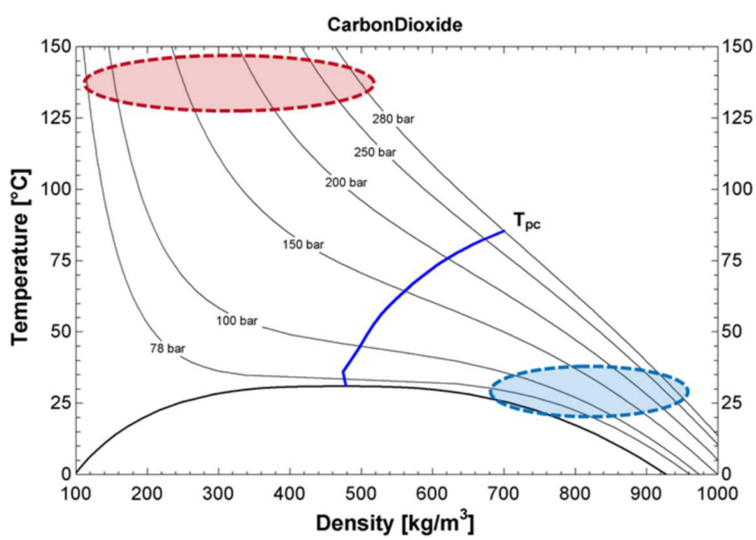



Figure 7. Temperature-density curves for $CO_2$ with lines of constant pressure and the pseudocritical curve ($T_{pc}$) shown.

### Pressurization and Depressurization

Changes in pressure for a $CO_2$ system will produce corresponding changes in temperature throughout the entire gas volume due to the Joule-Thomson effect exhibited by all real gasses. The Joule-Thomson coefficient, shown below in Equation (1) and Figure 8, represents the ratio of the change in gas temperature to its change in pressure for relatively short time scales that can be approximated as adiabatic. As shown in Figure 6, the Joule-Thomson coefficient remains positive for all relevant pressures and temperatures in the system leading to the gas warming as it is compressed and cooling as it expands (when the system depressurizes) as is expected from most gasses. The magnitude of his behavior can lead to unexpected heating and cooling or even hot or cold shock in an $sCO_2$ system as it is filled and vented and require a limited maximum filling and venting rate to maintain the system below a required thermal ramp rate. For example, if the thermal ramp rate must be limited to 10 °F per minute then assuming an average Joule-Thomson coefficient of 0.1 °F/psig the pressurization and depressurization rate must be limited to 100 psig/minute.

$$\mu_{JT} = \left(\frac{\partial T}{\partial P}\right)_H = \frac{V}{C_p}(\alpha T - 1) \quad \textbf{(1)}$$

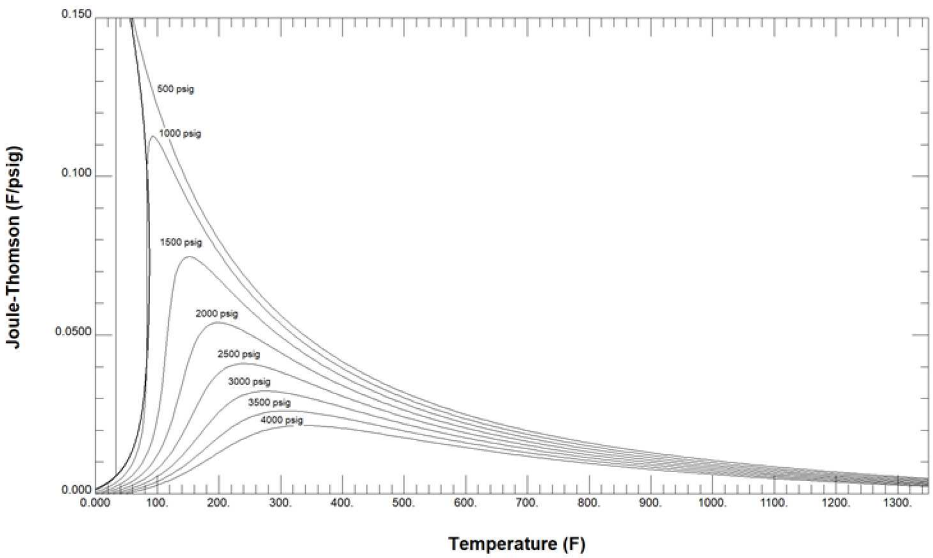



Figure 8. Joule-Thomson Coefficient of $CO_2$ over the range of expected temperatures and pressures.

## Inventory Control

Inventory control, consisting of both addition of $CO_2$ to increase pressure and removal of $sCO_2$ to decrease pressure, can be accomplished in several ways. The addition of $CO_2$ involves straightforward operation of the same pressurization system used to fill the system provided the compressor is designed to provide dead-head conditions up to the maximum operating pressure of the system. Removal of $sCO_2$ can be done simply by venting, but this releases significant volumes of $CO_2$ which eventually must be made up by delivery of $CO_2$ to the site. Instead, and especially for larger systems, a partial or complete recovery of $CO_2$ can be performed to limit the quantity of make-up $CO_2$ required over time.

Complete inventory recovery of $CO_2$ in the system would require cooling the fluid down to the saturation temperature of the storage system before expanding it through a valve to reach saturated liquid conditions. However, the saturation temperature for low pressure storage is well below ambient temperature and would require a dedicate refrigeration system to reach the appropriate temperature. A much simpler and easier approach is to cool the $CO_2$ to an achievable temperature for water chilling systems such as 50 °F as shown in Figure 9 before expanding it to storage pressures to achieve approximately 80% inventory recovery. This approach allows for high recovery rates for a relatively small power penalty of less than 25 kW or 7.1 tons of refrigeration for typical vent flow rates.

# DRAFT

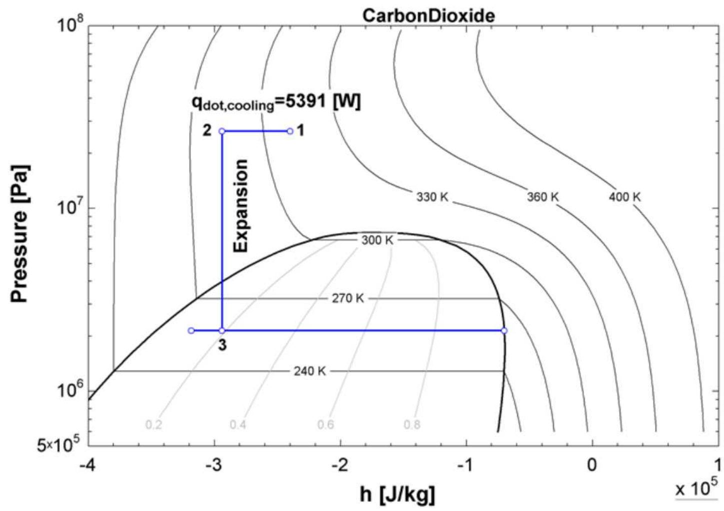
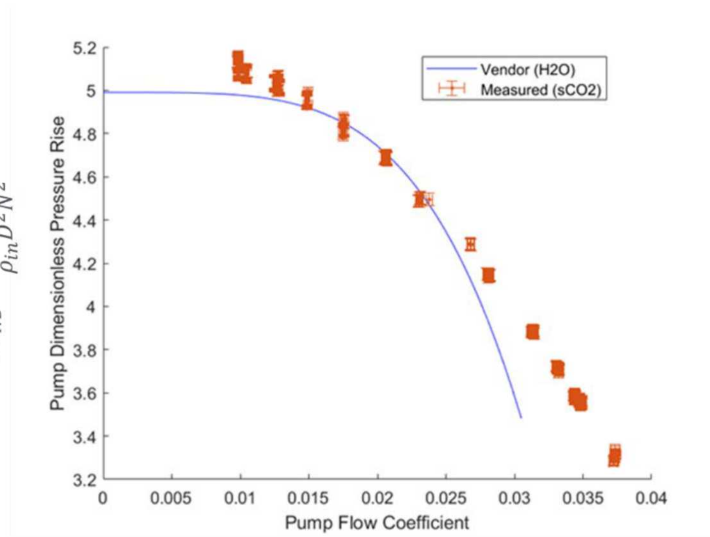



Figure 9. A pressure-enthalpy diagram for $CO_2$ with states and processes traced for the isobaric cooling and isenthalpic expansion of $CO_2$ initially at pump inlet conditions for injection back into a 300 psig dewar.

## Single-Pressure Pumped Operation

Sandia has experience operating three different centrifugal $sCO_2$ circulation pumps ranging from 1.5 to 7.5 hp in a variety of systems. Overall these pumps have closely matched the performance expected based on vendor data as shown in Figure 10 with noticeably better performance at flow rates above and below the best efficiency point suggesting that loss mechanisms related to shock and flow friction are reduced with $sCO_2$. Others at the University of Wisconsin at Madison and Bechtel (formerly Knolls Atomic Power Lab) have also accumulated significant experience with this style of pump without any noticeable issues.

$$DP_{ND} = \frac{DP}{\rho_{in} D^2 N^2}$$




$$\varphi = \frac{\frac{D}{D_{hub}} \dot{Vol}}{D^3 N}$$

Figure 10. A summary of 7.5 hp Parker Autoclave MagnePump operating data in $sCO_2$ compared with vendor curves.

One caveat is that care must be taken to account for differences in density when using a pump designed originally for water or another fluid with a different density. If vendor curves are provided based on data for water Equations 2

# DRAFT

and 3 should be used to determine non-dimensionalized pressure and flow coefficients "$DP_{ND}$" and "$\varphi$" which can be scaled to the new operating conditions for expected density based on the impeller diameter "D," hub diameter "$D_{hub}$," volumetric flow rate "$\dot{Vol}$," rotation rate "N," head rise "DP," and inlet density "$\rho_{in}$."

$$\varphi = \frac{\frac{D}{D_{hub}}\dot{Vol}}{D^3 N} \quad \textbf{(2)}$$

$$DP_{ND} = \frac{DP}{\rho_{in} D^2 N^2} \quad \textbf{(3)}$$

## CONCLUSIONS

The design and operation of $sCO_2$ systems poses an unconventional combination of thermodynamic and thermal hydraulic challenges ranging from cryogenic conditions through the highest service temperatures encountered in conventional pressurized systems. Thanks to the work of numerous research institutions these challenges have been met and overcome at scales up to 1 $MW_{th}$, with operations planned at 10 and 50 $MW_{th}$ scales soon. The discussion of design processes and analysis results of operational modes presented in this paper along with those expected from future work will allow CSP researchers and systems integrators to understand and safely operate $sCO_2$ systems as the solar industry progresses to LCOEs less than 6 ¢/$kWh_{el}$.

## ACKNOWLEDGMENTS

The work presented in this publication was supported in part by the DOE Solar Energy Technologies Office (SuNLaMP-0000000-1507 and DE-FOA-0001697-34151). Sandia National Laboratories is a multimission laboratory managed and operated by National Technology and Engineering Solutions of Sandia, LLC., a wholly owned subsidiary of Honeywell International, Inc., for the U.S. Department of Energy's National Nuclear Security Administration under contract DE-NA-0003525. SAND2019-XXXXX A.

## REFERENCES

[1] M. Mehos, C. Turchi, J. Jorgenson, P. Denholm, C. Ho, and K. Armijo, "On the Path to SunShot. Advancing Concentrating Solar Power Technology, Performance, and Dispatchability," NREL/TP--5500-65688, SAND2016--2237 R, 1256863, May 2016.

[2] "Commercial CO2 Refrigeration Systems Guide for Subcritical and Transcritical CO2 Applications." Emerson Climate Technologies Inc., 2015.

[3] Jim Pasch and David Stapp, "Testing of a New Turbocompressor for Supercritical Carbon Dioxide Closed Brayton Cycles," in *Proceedings of ASME Turbo Expo 2018 Turbomachinery Technical Conference and Exposition*, Oslo, Norway, 2018, pp. GT2018–77044.

[4] Timothy C. Allison, J. Jeffrey Moore, Doug Hofer, Meera Day Towler, and Joseph Thorp, "Planning for Successful Transients and Trips in a 1 MWe-Scale High-Temperature Sco2 Test Loop," in *Proceedings of ASME Turbo Expo 2018 Turbomachinery Technical Conference and Exposition*, Oslo, Norway, 2018, pp. GT2018–75873.

[5] E. M. Clementoni, T. L. Cox, and C. P. Sprague, "Startup and Operation of a Supercritical Carbon Dioxide Brayton Cycle," in *Proceedings of ASME Turbo Expo 2013*, San Antonio, Texas, USA, 2013, p. 8.