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Operational Transients in Supercritical CO_2 Systems: A Case Study on a 10 MWe Cycle

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ABSTRACT

Supercritical carbon dioxide power cycles can offer higher thermal efficiencies, and compact component sizes for their power size due to high densities. However, the transient phases of plant operation, such as those in startup and shutdown, can pose significant technical and operational challenges that impact system reliability, component lifespan, and operability. This paper examines the dynamic behavior of a 10 MWe Net sCO2 power system during these phases, focusing on potential thermal stresses, and control system responsiveness. Startup procedures involving temperature and pressure ramping, grid synchronization, or load bank synchronization must be considered to prevent material fatigue and control instability. Similarly, shutdown sequences require precise thermal and pressure controls, such as those utilized within mass control, to ensure a smooth system shutdown that is not hard on the system. This study outlines best practices and identifies key areas for component development and plant operations, efficient, and repeatable transitions between operational states in future commercial deployments.

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INTRODUCTION

Supercritical carbon dioxide (sCO_2) is a fluid state achieved when carbon dioxide is maintained at or above its critical pressure and temperature, which occurs at 73.8 bar and 31.1°C [1]. At this critical point, sCO_2 exhibits unique and highly dynamic fluid properties that differ significantly from those in its gaseous or liquid states. One of the most notable characteristics of sCO_2 is its combination of high density, which resembles that of a liquid, and low viscosities, comparable to that of a gas. This combination results in a fluid with high power densities and low head losses, making it particularly attractive for power generation systems. These thermophysical properties allow for the design of compact and efficient turbomachinery, which can be significantly smaller than those used in traditional air-based Brayton or steam-based Rankine cycles. This can reduce turbomachinery size by up to 75 percent compared to those counterparts. These attributes make sCO_2 power cycles especially attractive for a wide range of energy applications, including concentrated solar power (CSP), nuclear energy, small modular reactors, waste heat recovery, and advanced fossil fuel systems [2], [3], [4].

Cycle analysis and design generally involves designing machinery to meet performance at its "design" point, with higher efficiencies taking place at a specific thermodynamic point. Nonetheless, machinery can experience transient conditions that are outside of its specified operating point during events such as startups, shutdowns, load changes, and even emergency shutdowns. The conditions during those scenarios can present unique challenges, with fluid properties changing rapidly in ways that were otherwise unplanned. Furthermore, mismatches in design specifications, ramp rates, and control strategies between different subsystems, such as turbines, heat exchangers, and compressors, can lead to integration challenges. Equipment procured from different manufacturers may not always align in terms of operational flexibility, further complicating control during dynamic operation. Understanding and modeling transient behavior is essential for developing reliable control strategies and ensuring the stable operation of sCO_2 systems in real-world applications. This paper aims to showcase transient behavior observed in a 10 MWe sCO_2 power cycle under various operating conditions and elaborate on those lessons learned.

The Simple Cycle configuration utilizes three heat exchangers: a heater coil for adding heat to the cycle, a High Temperature Recuperator (HTR) for recuperating heat from the turbine exhaust and the main compressor discharge, and a Main Process Cooler (MPC) for rejecting heat from the cycle. When configured for RCBC, an additional recuperator, Low Temperature Recuperator (LTR), will be added to further recuperate heat from the HTR low-pressure discharge and the main compressor discharge. The heater coil will be discussed further in subsequent sections.

Overview of sCO_2 Power Cycles

Supercritical carbon dioxide exhibits thermodynamic properties that are within those of a gas and a liquid, especially near and above its critical point at 73.8 bar and 31.1°C [1], [5], [6], [7]. Unlike ideal gases, the fluid's density, compressibility, and other properties can vary significantly with small changes in temperature and pressure around this region. The high density of sCO_2 contributes to turbomachinery's smaller component sizes, giving a higher mass flow rate for a volumetric flow rate of a similarly sized air Brayton counterpart. Additionally, its low viscosity results in reduced frictional losses within piping and machinery. These properties combined lead to higher thermodynamic efficiency in power cycles and compact machinery and heat transfer equipment. For example, comparing CO_2 at its supercritical point versus air: CO_2 exhibits densities 100 times higher, specific heats 6 times higher, and viscosities 0.9 times those of air [1]. In other words, CO_2 can store much more heat, is much denser, and similarly viscous. Now, when comparing it to water, at the supercritical point the density is 0.4 times that of water, at least 11.6 times less viscous than water, and specific heats are highly similar; nonetheless, at 106 bar, where CO_2 is much denser, it is 0.76 times the density of water; this showcases the properties of CO_2.

What makes CO_2 highly favorable at its supercritical point is the fact that it achieves supercritical state at room temperatures (~31-32°C). Other fluids require high pressures and temperatures for them to achieve supercritical properties; for example, air requires 373.6 bar and 132.63°C, and water requires 220.6 bar and 374.1°C [1]. Steam cycles typically operate at high pressures, around 200-300 bar, and use water vapor and water as the working fluid [8], [9]. This demands large heat exchangers and condensers due to the phase change involved. This often leads to bulky and heavy equipment, along with complex water treatment requirements. The decreased pressure and temperature to achieve supercritical point of CO_2 is what makes it favorable for power cycles, dynamic properties, and great heat transfer properties within favorable pressure and temperature state points.

A typical sCO_2 power cycle includes the same core components as it's air or steam counterparts: a compressor, turbine, heat exchangers (including the primary heat source and sink), recuperators, and control systems. The compressor increases the pressure of the CO_2 after it has been cooled to a predetermined temperature, taking advantage of the fluid's high density to minimize compression work. The turbine extracts mechanical power by expanding high-temperature, high-pressure CO_2, converting thermal energy within, into and out of the cycle. This indirectly fired cycle being heat source agnostic, can utilize nuclear, coal, solar thermal, natural gas, or waste heat. Recuperators are crucial components that recover heat from the turbine exhaust to preheat the CO_2 before it goes into the heat source, significantly improving overall cycle efficiency by reducing the heat input required.

TRANSIENT OPERATION CHALLENGES

At the STEP facility, the heater consists of a natural gas heater with a coil embedded in a natural gas fired duct burner. The heater tubes, which must withstand temperatures up to 725°C sCO_2, are made from Inconel 740H tubes wrapped in stainless steel fins. The combustion air entering the duct burner is fed from a fixed speed blower, and temperature is controlled by modulating a natural gas control valve.

The heater typically starts up after the main compressor starts up but before the turbine is allowed to spin. At heater startup, a minimum sCO_2 flow rate must be established in the heater coil to prevent burning out of the heater coils. Once the initial sCO_2 flow is established, per NFPA requirements, the combustion air blower will turn on and conduct a purge of the duct burner prior to burner firing. At the completion of the purge sequence, a pilot flame, followed by the main flame, will ignite, and flame stability will be monitored. Once the heater temperatures at the minimum firing position are stabilized, the operator can start the heater ramp. Due to concerns about thermal stresses on the heater bundle, the heater vendor put a temperature ramp up limit of 55°C/hr up to 260°C and 110°C/hr above 260°C. This limitation on heater ramp rate limits the rate at which the cycle can get to full power.

During heater shut down, the natural gas flow is cut off to the burner, and the combustion air blower will continue running to purge the heater of any natural gas that could potentially still reside in the cavity. The burner can be brought back to its minimum firing position before natural gas shutoff, but there are no limitations on how fast the burner may shut off; nonetheless, equipment might see fast transients during subsequent restarts due to the lack of heat input during the transient.

During Simple Cycle testing, it was observed that hot restarts can result in slugs of cold sCO_2 leaving the heater and impacting the TSV/TCV and the turbine. This is caused by the purge sequence both after shutting off and before relighting the burner. During these purge sequences, the full combustion airflow runs through the heater and rapidly cools the heater coils. Because the heater requires a minimum flow of sCO_2 through the coils prior to startup, this results in the sCO_2 being cooled in the coils and sending a cold slug into components downstream of the heater. The team is working on several options to minimize the amount of cold sCO_2 that could cold shock the hot soaked components downstream of the heater. For example, Figure 1 shows the thermal shock caused during a trip event downstream of the heater prior to relighting.

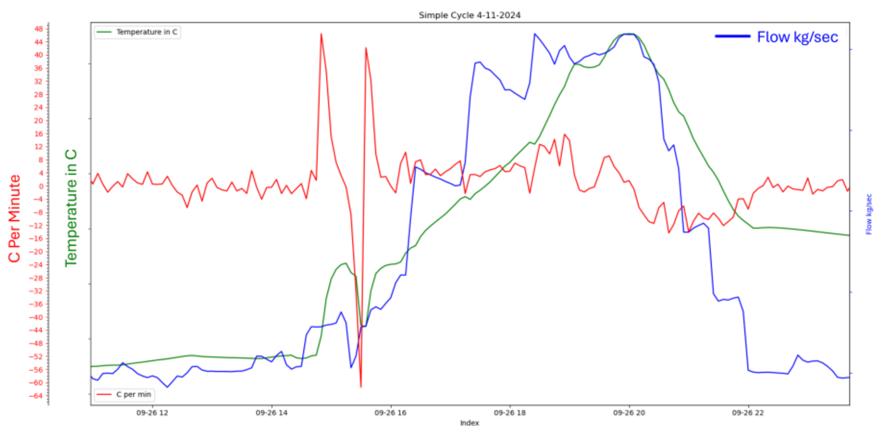

Figure 1. Degree per minute immediately after a trip.

STARTUP DYNAMICS

The plant control system must have a means to prevent the formation of liquid CO_2 during operation. When the plant is shut down or tripped while still pressurized, however, it is possible, and even unavoidable in certain conditions, to form liquid CO_2 in certain sections of the system. Thus, when the plant restarts, there is a risk of running liquid CO_2 droplets or slugs through the high-speed rotating machinery.

Figure 2 shows compressor discharge state points alongside a startup instance. This figure shows that when the compressor is nonoperational, the dry gas seal supply temperature is at roughly 80°C, warming the surrounding areas of the compressor, but by the time operation starts and the compressor reaches liquid mode at 9,000 rpm, the compressor is then cooled to the average of the loop's CO_2 flow state. During this subsequent startup period, oscillations have become visible and lasted for about 45 seconds and approximately six cycles [10].

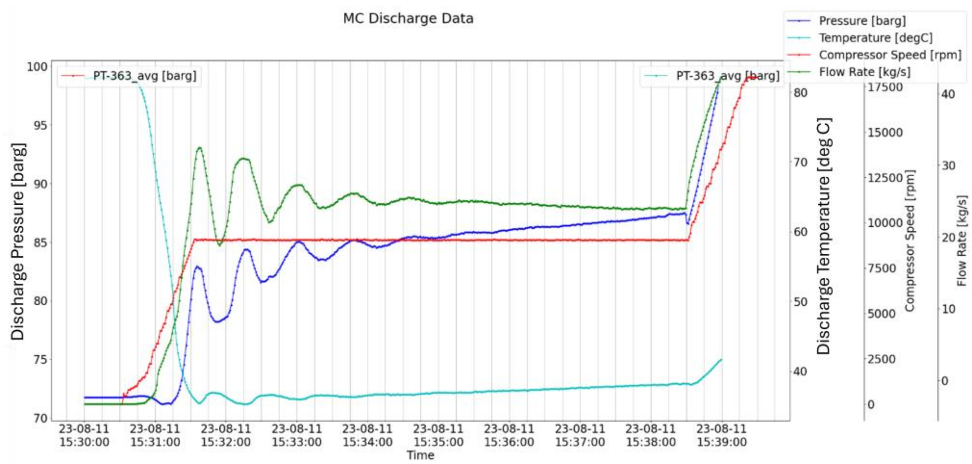

Figure 2. Main Compressor Discharge Conditions [10]

Figure 3 shows the compressor suction conditions at the same time. It is important to note that as the compressor begins to spin, the pressure drops below the critical point, causing the fluid to become high dense CO_2. Figure 4 shows that the CO_2 approaches the vapor dome, possibly causing liquid in the inlet of the compressor. Nonetheless, as previously stated, even if liquid does not form and CO_2 is still in a binary supercritical form, the density nearly doubles at the inlet up to the point where the oscillations disappear.

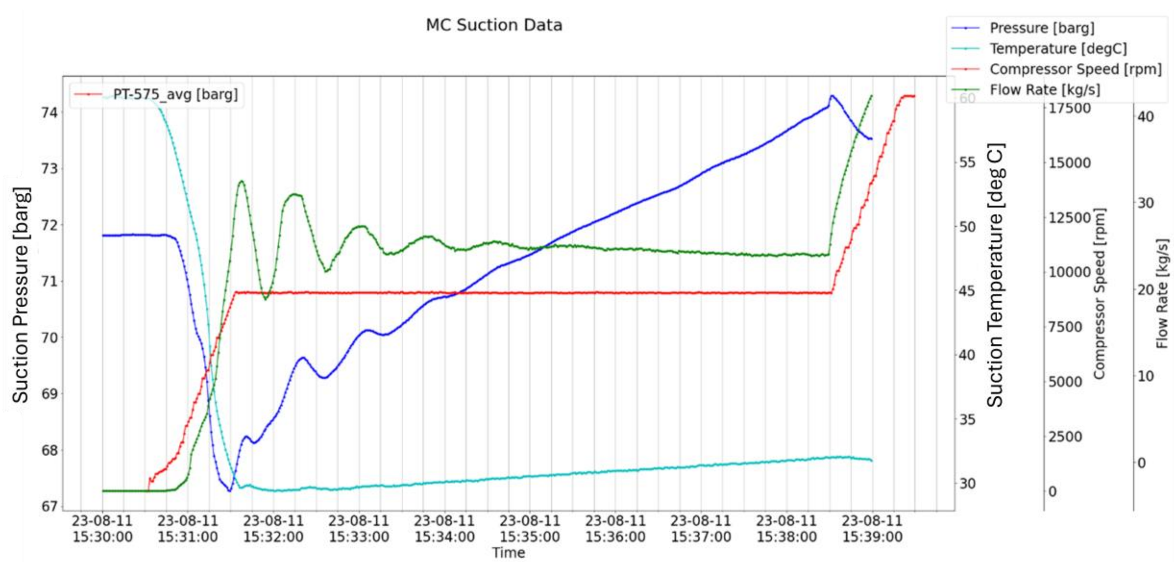

Figure 3. Main Compressor Suction Conditions [10]

With the piping geometry and state points being known, the round-trip travel of the oscillations can be calculated. By knowing the area of the piping, the mass flow rate, and the density, one can calculate the fluid velocity, and by knowing the total piping length, one can calculate the transit time of the fluid. One of the critical items to note is that during this event, it is potentially important for the compressor to operate at slow speeds to minimize potential damage by pockets of fluid circulating at nearly double the density and minimize the risk of instabilities or damage.

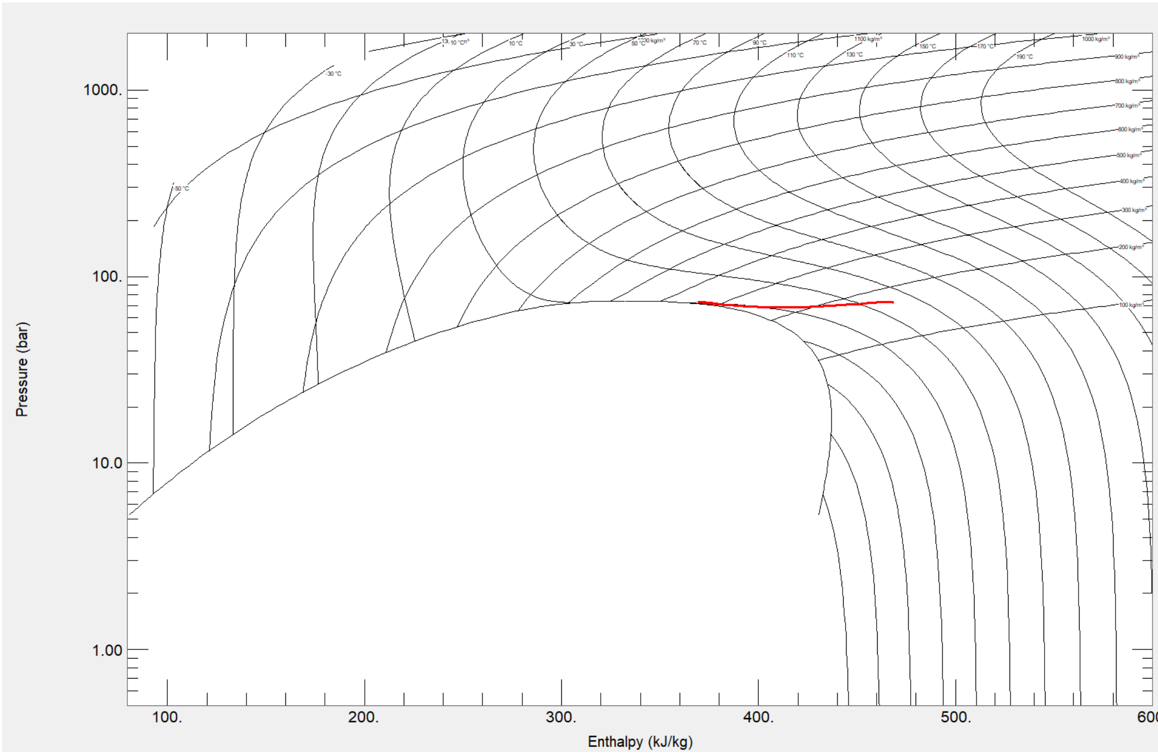

Figure 4. Compressor Suction Conditions Plotted on p-H Diagram [10]

CO_2, being a refrigerant known as R744, has a high Joule-Thompson coefficient at low temperatures and high pressures. For this reason, warm-up by trickling a flow of CO_2 through equipment can be challenging. For example, cracking open the turbine stop/turbine control valve causes CO_2 to undergo isenthalpic expansion, causing cooling; this can cause highly dense or even potentially two-phase CO_2 to form in cavities where it should not be present, and rotation of turbomachinery at these points can be catastrophic. Such an expansion effect can be observed in Figure 5. It is not until the main process heater is started and minimum mass flow is met that the equipment can begin warming up and nominal starting can occur.

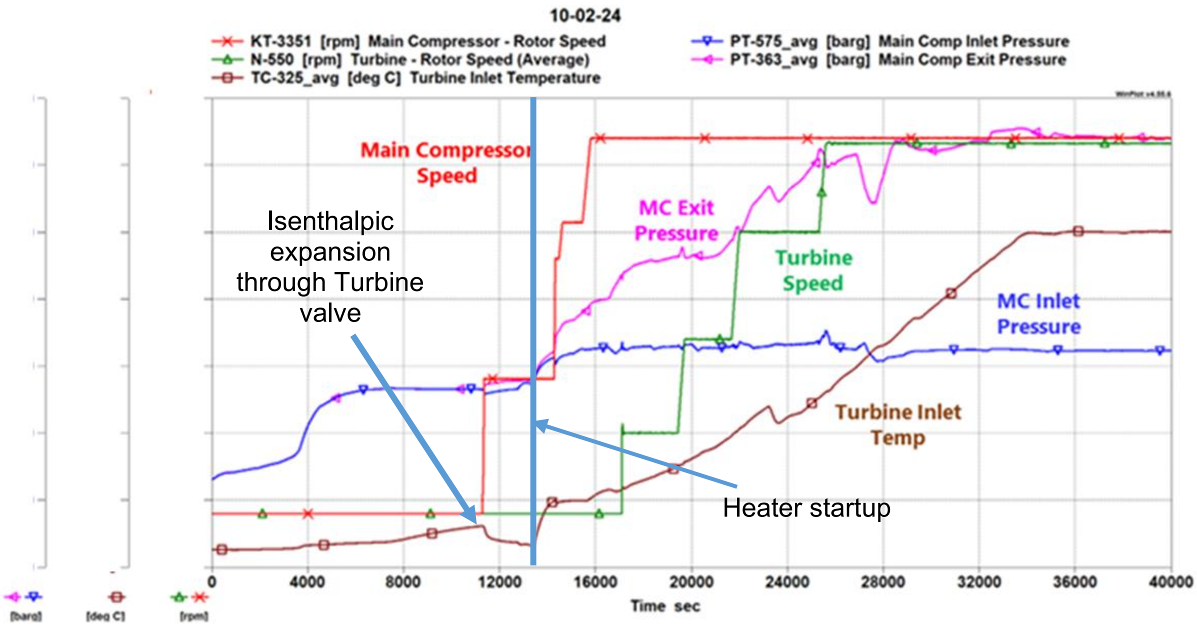

Figure 5. Nominal start

SHUTDOWN DYNAMICS

After nominal operations, an sCO_2 cycle may undergo either a normal or emergency shutdown, each requiring different operational strategies to protect critical equipment and maintain system integrity. In a normal shutdown, the process of bringing machinery and equipment down to a halt is gradual and coordinated. The turbine is brought down from nominal power by reducing heater temperature, which decreases the thermodynamic work of the cycle. Using a combination of mass flow control, inlet guide vanes (IGVs), and anti-surge valves, the plant operators reduce turbomachinery speed in a stable manner, avoiding operation outside safe compressor maps. Shutdown procedures occur within a matter of hours, and temperature and depressurization rates are controlled to be within 0.1–0.2 bar per second and temperature within 1–2°C per minute. This shutdown can be seen in Figure 6. After shutdown of the turbine, dry gas seals are still fed to ensure proper cooldown of machinery if it is observed that thermal gradients are still not settled.

In contrast, an emergency shutdown (e.g., due to high vibrations, overspeed, or grid disconnection) requires immediate action. Heating is terminated abruptly, and both active venting (via fast-acting control valves) and passive relief systems may be triggered. These events can induce rapid pressure and temperature changes, which may result in severe thermal shock conditions, and under such events the inventory management system feeds dry gas seals, acting as active cooling for turbomachinery to ensure thermal migration barriers are held for machinery to withstand longitudinal conduction.

According to Welding Research Council (WRC) Bulletin 490 [11], such transients are known contributors to thermal fatigue, brittle fracture, and cracking—especially in thick-walled components like valve bodies, heat exchangers, and turbine casings. Damage mechanisms arise not only from the temperature differential but also from the rate of change and the mechanical constraint of components. For example, it was observed that due to high thermal transients during shutdowns and startups, the turbine control and stop valve seat exhibited damage. Such damage has been mitigated, and design changes for the valve help mitigate these.

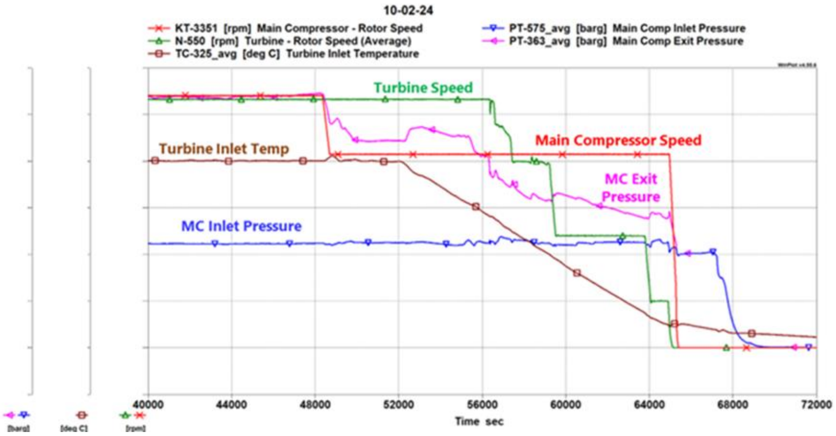

Figure 6. Nominal shutdown

IMPLICATIONS FOR COMPONENT DESIGN

The HTR, LTR, and MPC are all printed circuit heat exchangers (PCHE). They are fabricated by etching flow channels onto metal plates, then diffusion bonding the plates together to form a single block. The PCHEs are extremely compact compared to shell and tube heat exchangers, containing high surface area in small volume. PCHEs are designed to handle the process conditions encountered in an sCO_2 power cycle.

The HTR is subjected to severe thermal stresses, with the high-temperature side subjected to temperatures approaching 600°C and 200°C on the low-temperature side. The HTR, which is one of the largest PCHE in the world, is built from several blocks joined together, which results in a relatively long vessel. This results in the HTR deforming into a curved “banana” shape when at operating temperature. Due to this deformation under the thermal load, special considerations had to be considered when designing the support structure for HTR. The ends of the HTR could not be held fixed, as this would cause high mechanical stresses due to thermal transients, so instead one end was pinned to the structure while the other end was suspended on spring hangers. This resulted in a support structure that allowed the HTR to grow and deform freely.

During startup and shutdown, the HTR hot side temperatures closely follow the heater discharge temperatures, while the cold side temperatures closely follow the main compressor discharge temperatures. At steady state, the HTR high-pressure bypass valve maintains a cold side approach to temperature of 6°C to ensure a uniform temperature gradient and reduce thermal stresses due to longitudinal conduction. However, during a cold startup it was noted that the low-pressure discharge temperature was cooler than the high-pressure inlet temperature, which seems to suggest that the sCO_2 fluid is warming up the thermal mass of the HTR.

Startup conditions, particularly ramp rates, should be considered at an early stage of the plant design and are heavily influenced based on the expected operations of the plant. Thus, it is important to simulate expected transient operations in the design process and add transient conditions into component procurement specifications. For example, it is commonly known that a peaker plant or load following operational style will require faster ramp rates, whereas a base load operating profile can accept much slower ramp rates. Figure 7 displays several parameters of the 10 MW plant in operation throughout a day of testing. The plant had operated the prior day, and with insulation maintaining temperatures overnight, the heater inlet temp remained around 100°C. As the plant starts up and CO_2 flows through the heater, a sharp dip in the inlet temperature occurs and begins to climb back up above 100°C as the heater is fired. This type of operation was considered during design and was not considered a high risk. The ensuing ramp rate of temperature increase is controlled by the firing rate of the heater and was highly engineered during the design phase to ensure minimal effect on the shell and tube weld joints in the heater heat exchanger section. The generator coupled to the turbine has a high break-away torque, causing the turbine to not start spinning until double digits of kg/sec start to flow. This gives us control on the ramp rate the heater without spinning the turbine and incurring damage. A normal ramp rate increasing and decreasing temperatures of the heater section and subsequently entering the turbine, can be seen occurring between 50,000s and 85,000s in the figure below. However, the cooling ramp rate caused by an emergency stop of the turbomachinery (~41000s in Figure 7), and thus stopping the firing of the heater, was not highly modeled. This rapid cooling is a side effect of NFPA code compliance requiring a purge of the heater after stopping the flow of natural gas and is difficult to mitigate. This condition necessitates additional monitoring of the system, including counting the number of rapid thermal cycles experienced by the equipment, and may lead to a shorter life expectancy of the equipment overall. Additional methods to mitigate this condition are still under investigation with the manufacturer and the project team.

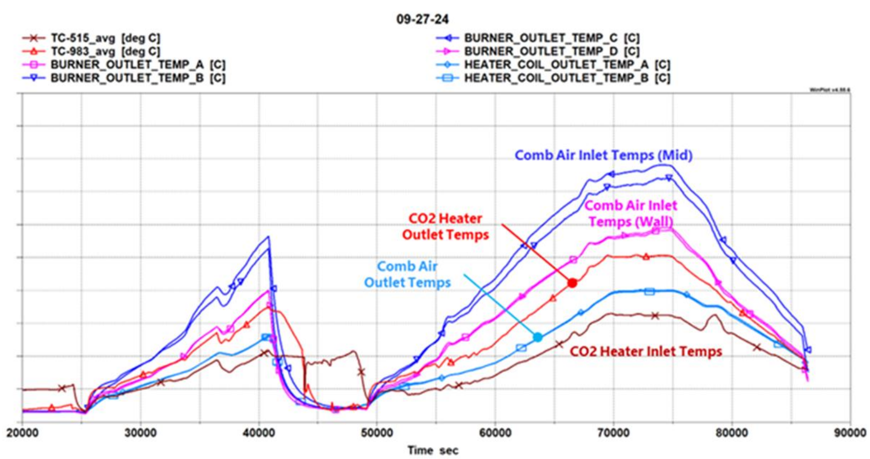

Figure 7. Testing of sCO_2 plant, showing cold startup, trip, and hot restart

The first ramp rate discussed was the bulk system ramp rate, which is determined by the most restrictive piece of equipment. The next ramp rate discussed is localized to a smaller section of the system and can be observed near a process control valve. The data shown in Figure 8 is from the turbine inlet temperature immediately downstream of the turbine control valve (TCV), which is used for speed control until the turbine is synchronized with the grid and load control thereafter.

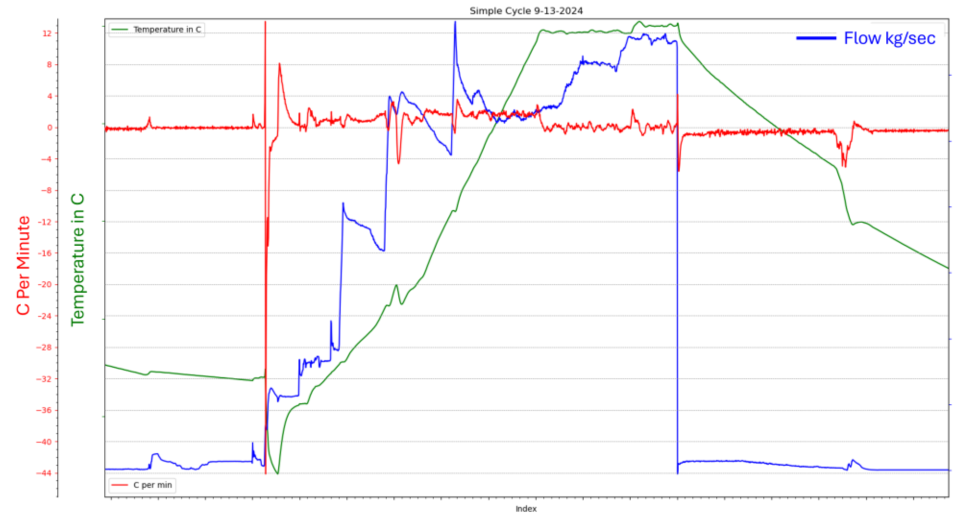

Figure 8. Operational data for turbine inlet temperature and ramp rate

After the system operated, it remained stationary for some time. This allowed some uninsulated sections to cool more than the insulated sections, especially in the heater, which experiences forced cooling after shutdown due to NFPA operating requirements. When the TCV is opened to initiate flow through the turbine for the next startup, the cooler CO_2 begins to flow through the warmer piping, creating a convective cooling effect. This compounds with the natural Joule-Thompson cooling of the CO_2 as it passes through the TCV and loses pressure. In the data shown above, this leads to a rapid cooling of 40°C per minute for a short time, which exceeded the design of 2°C per minute that would allow for greater than 10,000 hours of service life. This rapid localized cooling was not predicted and led to ill-advised decisions in the design of the valve. The result of repeated exposure to this rapid cooling can be seen in Figure 9. The valve was originally designed with welded seats, which led to high thermal stress during these rapid cooling events that resulted in a crack in the valve seat. After the crack was detected, the valve was removed, and the seat design was modified to a clamped style to allow for increased tolerance of rapid cooling events.

Figure 9. Small crack formed on valve seat

CONCLUSION

sCO2 power cycle operation has yielded learning experiences to implement in future generations of such cycles. Some of these could potentially be mitigated through the use of a different heat source, or controls and hazard assessment analysis. For example, the NFPA requirement for heater quenching would not be an issue with CSP or Nuclear heat sources, and improved controls such as a Turbine Control Valve with finer Cv/Kv control could help mitigate cold shocking equipment downstream. Similarly, improved controls would mitigate transients in the hot restarts. It is important to note that during startup dynamics, the ability to provide a heat load promptly could minimize the difference in density and the damage due to inlet density slugs going through turbomachinery; by heating up the CO2, one could potentially reduce such risk. Finally, the Turbine Control and Stop Valves which have been observed to have small cracks on the valve seat, have undergone rebuild and new seat inserts have been manufactured which have not been welded, this is done to allow expansion of dissimilar metals during transients such as startups, shutdowns and trips. It is important to take the lessons learned during plant operations and try to determine ways to make the next generation cycles more resilient to operational events and transients.

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