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# Applied Controls for $sCO_2$ Brayton Cycles

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## ABSTRACT

Sandia has a history of testing supercritical $CO_2$ Brayton Cycles to explore operation fundamentals and provide validation data for computer models. These systems have always had data acquisition and controls features. The Development Platform (DP) has been a flagship system for loop testing and operation but has been limited to manual control of many systems. Manual operation has increased operating complexity and reduced stability and repeatability. This work documents automated control development by linearizing otherwise non-linear valves, addition of closed-loop proportional-integral control software that has been tuned to the Sandia DP. It also describes testing of control methods that have improved test quality and reliability as shown in actual test data.

## ACKNOWLEDGEMENTS

The author is grateful for the software development efforts of Matt Carlson, the test operation of Logan Rapp, and the hands-on work of technologists Rob Sharpe and Ron Hrzich.

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# ACRONYMS AND DEFINITIONS

| Abbreviation | Definition |
|---|---|
| CIT | Compressor inlet temperature |
| DP | Development Platform |
| PID | Proportional-integral-derivative |
| RCBC | Recompression Closed Brayton Cycle |
| $sCO_2$ | Supercritical $CO_2$ |
| TCV | Turbine control valve |

# 1. INTRODUCTION

## 1.1. Sandia Development Platform

The Sandia sCO2 Development Platform (DP) has been a flagship test facility for testing of turbomachinery as well as startup and shutdown operations. It was the first facility in the world to demonstrate the feasibility of electricity generation with the Recompression Closed Brayton Cycle (RCBC) (Conboy, 2011). Efforts began in 2017 to reconfigure the DP into simple cycle operation for testing of the turbo-compressor of industry partner Peregrine Turbine Technologies. This included an updated data acquisition and control system in addition to piping changes around the turbomachinery. A photo of the DP configuration in summer 2019 is shown in Figure 1.


**Figure 1. Development Platform with insulated heaters on right, turbomachinery in foreground, and heat exchanger train in background**

The $sCO_2$ in the DP is cooled in the Gas Cooler immediately upstream of the compressor as shown in Figure 2. The cooling level is controlled by adjusting the water flow rate through the Gas Cooler by changing the positions of the Main and Bypass Valves. The Cooling Tower Fan is another method for adjusting the amount of cooling.

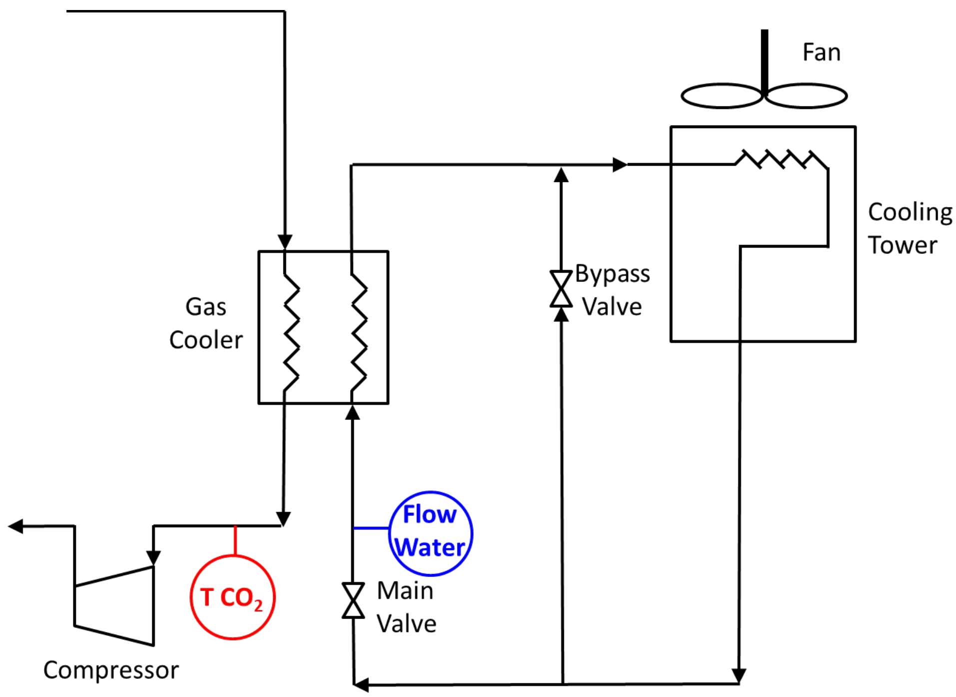


**Figure 2. Development Platform cooling system flow diagram**

The cooling requirements for a test start from zero and increase over time until a steady state is reached. To allow constant water flow for pump health, the Bypass Valve starts fully open. The Main Valve starts closed so the water flow rate through the Gas Cooler is zero. When sCO2 system cooling is required, the Main Valve is opened gradually until fully open, then the Bypass Valve is closed, further increasing water flow through the Gas Cooler. The total valve angle under this operation can be represented by

$$\theta_{Total} = \theta_{Main} + \left(100 - \theta_{Bypass}\right)$$

where $\theta_{Total}$ is the total angle or position, $\theta_{Main}$ is the Main Valve angle, and $\theta_{Bypass}$ is the Bypass Valve angle. This total angle has a range of 0–200%, with larger angles allowing greater cooling. When even greater cooling is required, the fan can be used to increase heat transfer to the air.

The measured Gas Cooler water flow as a function of total valve position is shown in Figure 3 and depicts very nonlinear behavior. The flow rate quickly increases with the opening of the Main Valve at low total valve positions, then is nearly constant through the rest of the Main Valve opening and about 25% into the closing of the Bypass Valve. With further decreases to the Bypass Valve position, the flow rate increases at an increasing rate. These valves are globe type with linear $C_v$ curves that were expected to provide linear flow response in this system. Industrial experience has shown more linear system behavior from equal percentage valves that may have been a better option. Nevertheless, performing software corrections was selected as the solution.

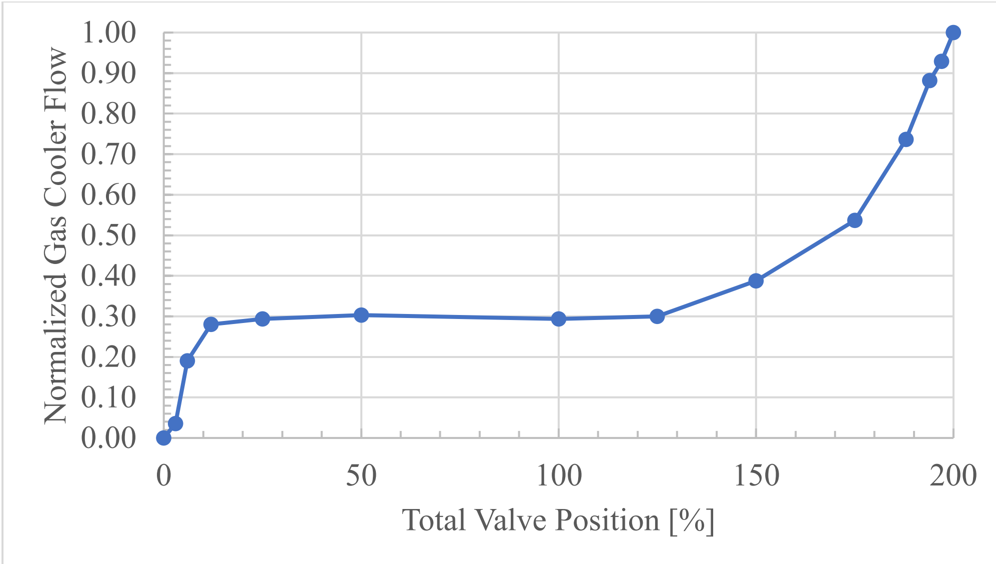


**Figure 3. Normalized Gas Cooler flow as a function of total valve position**

## 1.2. Manual Control Challenges

In recent years, in both configurations, the DP has been limited to manual operation of all systems, most notably the heating and cooling of the closed loop. This required the operator to monitor and continuously adjust these settings in addition to those affecting the turbomachinery speed and secondary systems. This operation load was very demanding and lead to several errors that perturbed planned operation and fortunately did not lead to damaged equipment. The perturbations included step changes to conditions that have the potential to cause turbomachinery instabilities and imposed faster thermal transients than required for equipment, potentially decreasing lifetimes. On several occasions, the operator accidently turned the heaters to 100% instead of an incremental change.

An example of the challenge of manual cooling control is shown in Figure 4 from a test that was run on 2016-02-08 in RCBC configuration. This test shows that the operator had to correct for over-cooling four times by reducing the valve position. The abrupt temperature decreases presented risk for the turbomachinery as the compressor inlet conditions had drastic property variations near the critical point that can affect bearing loads. Ideally, the compressor inlet temperature would be tightly controlled for stable and repeatable operation near the design point.

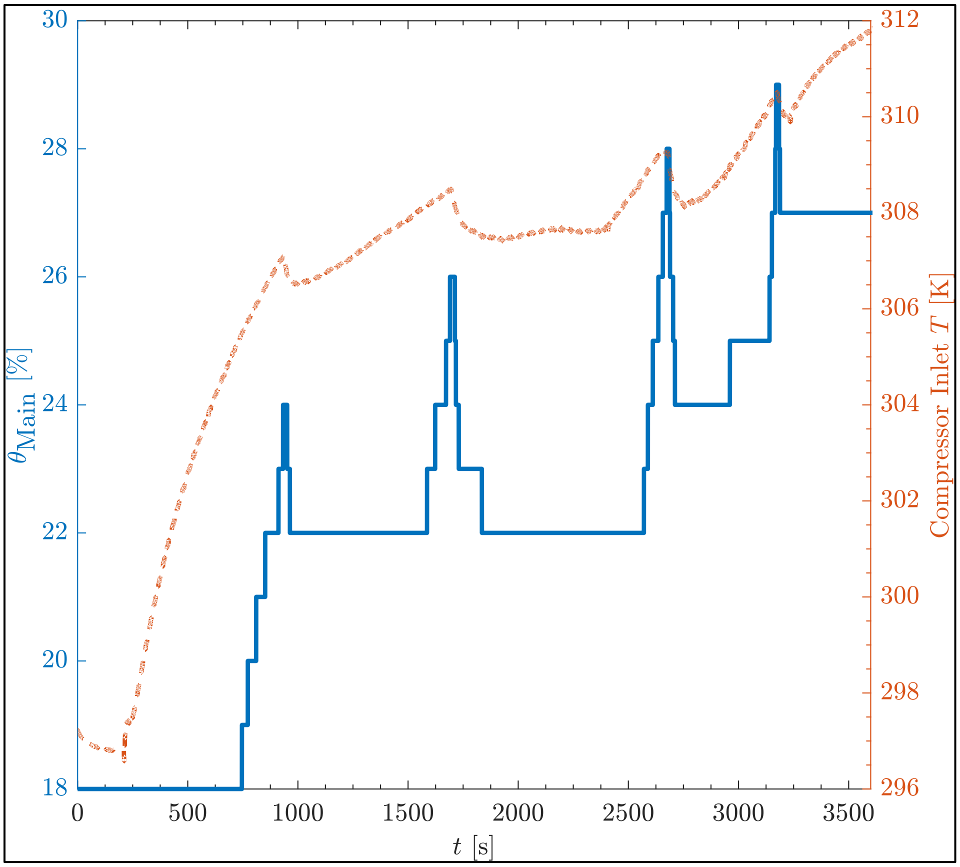


**Figure 4. Main Valve position and compressor inlet temperature for test on 2016-02-08**

## 2. METHODS

### 2.1. Closed-loop Control

Closed-loop control using proportional-integral-derivative (PID) methods has been widely used in industry and shown to work well when tuned to the system. In contrast to open-loop controllers, closed-loop controllers monitor the control variable to adjust the output, correcting for steady-state errors and adapting to systems that have slight variations (Nise, 2008). Both types of control loops are represented in diagram form in Figure 5. The cooling system is subject to dynamic changes due to environmental conditions for the cooling tower from slow but dramatic seasonal changes and rapid, though less dramatic, weather changes such as cloud cover and/or rain. Also, the Gas Cooler water side tends to have fouling that reduces the water flow rate characteristics over the period of months. It has been shown to reduce maximum water flow rate from a nominally clean 190 gpm to 120 gpm.

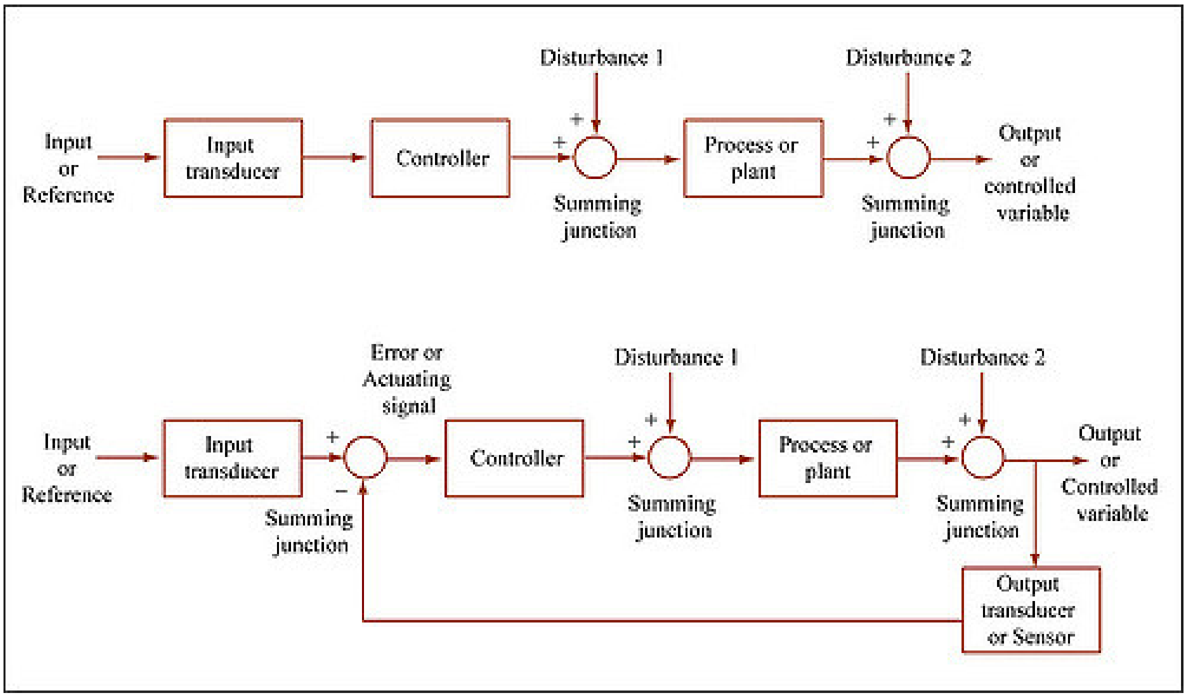


**Figure 5: Diagrams for open-loop (top) and closed-loop (bottom) control systems**

Proportional-integral-derivative controllers change the setpoint based on proportional, time integrated, and time derivative aspects of the error. The controller may be a mix of P, PI, or PID with increasing complexity required with the addition of each aspect. The DP heating and cooling systems are slow and stable enough to require only a PI controller as the derivative aspect is often only needed for fast systems.

### 2.2. Heater Controller Tuning

The tuning of both the heating and cooling controllers was performed following the commonly accepted methods of Ziegler-Nichols (National Instruments, 2019) then followed by manual updates. The method involves live system operation and observations to step changes in the process variable (heater output or cooling required). Steady state conditions are required, then a step change, and system monitoring until it converges to another steady state. The measured parameters are the time constant $T_p$ (time required to reach 63.2% of the system change) and dead time $\tau$ (time between change and observed response). The proportional gain and integral time for a slow system could then be defined by

$$K_c = 0.24T_p/\tau$$

and

$$T_i = 5.33\tau.$$

The tuning was performed by filling the DP with $CO_2$ and pumping it around the loop with a HydroPac reciprocating pump at typical mass flow rates and system temperatures and pressures. The rotating turbo-compressor was removed and an analogue was installed with open flow paths. This configuration allowed for long-term and reliable operation for controller tuning without placing the turbo-compressor in jeopardy. The online heater tuning step change in power and thermal response is shown in Figure 6. It shows steady conditions around 7% heater power and 400°F, a rapid change in heater power to 12%, and the thermal response of the heater outlet temperature that is immediately upstream of the turbine inlet. The thermal response shows general first order behavior but, since this is a closed loop, the later stage tends toward linear behavior. The time constant was still estimated from this result assuming that the first-order section of the change ended at 300 seconds at 435°F. The time constant for 63.2% of the temperature change was estimated as $T_P = 103$ seconds.

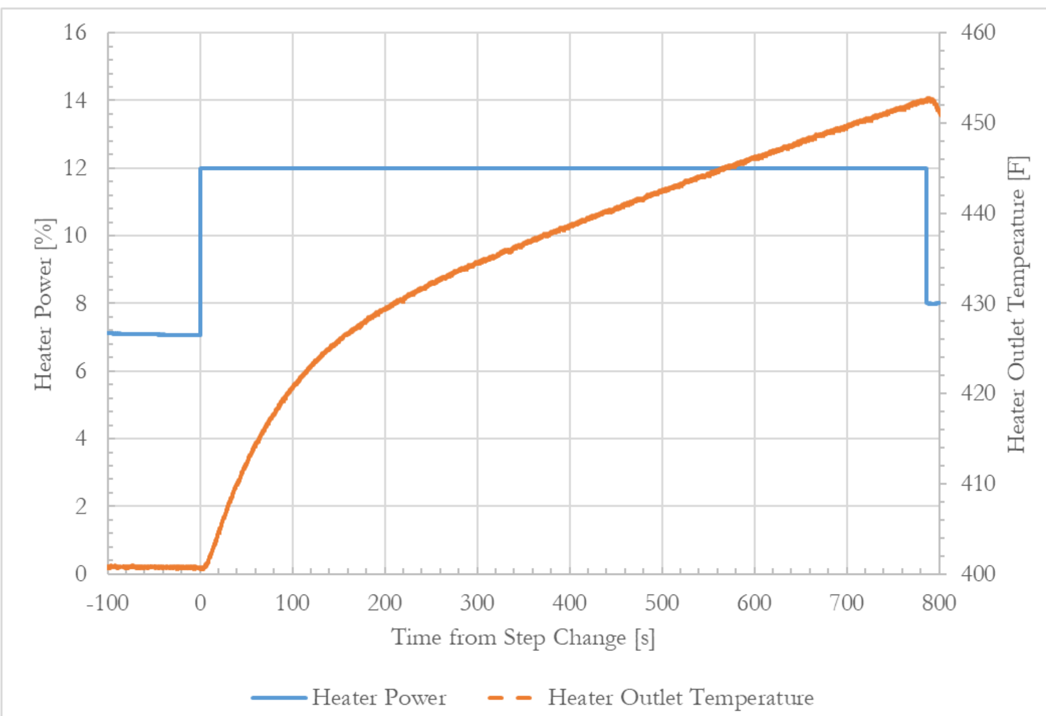


**Figure 6. Heater tuning step change and resulting thermal response**

The time delay $\tau$ can be estimated by zooming into the heater power step change area as shown in Figure 7. Here the time delay was estimated at 4 seconds and is a reasonable estimation of the time delay between commanding a change in heating and realizing a thermal response in the $sCO_2$ downstream.

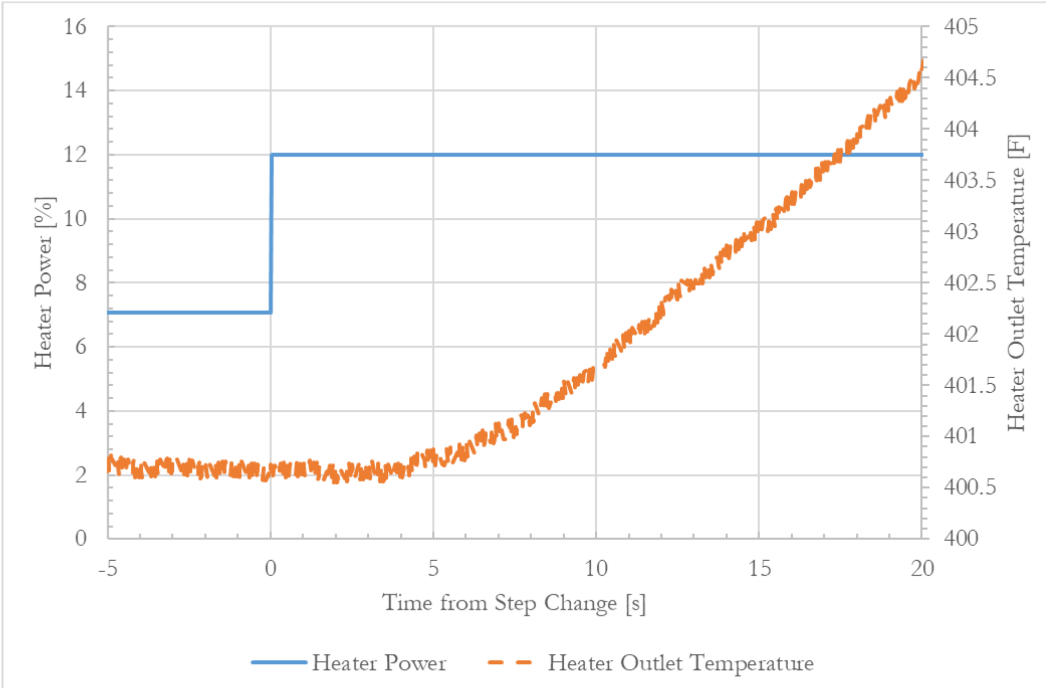


**Figure 7. Heater tuning test zoomed into the step change for time delay estimation**

From these measurements and the equations above, the PI constants were initially estimated to be $K_c = \frac{0.24(103\ s)}{4\ s} = 6.18$ and $T_i = 5.33(4\ s) = 21.2\ s$. These were programmed into the DP control software that used the built-in LabVIEW PID functions and tested. They were found to cause large-scale oscillations in heater power but were useful for initial estimates. These constants were iteratively adjusted by reducing the proportional gain $K_c$ and increasing the integral time $T_i$ until oscillations were deemed small, overshoot was limited to a maximum of approximately 10°F, and convergence was realized to within 1°F within approximately one minute. The final parameters were

$$K_c = 0.5 \text{ and } T_i = 0.5 \text{ minutes}$$

and had excellent stability with a multitude of tested upsets to conditions. The tests included step changes in setpoint temperature up to 50°F up and down and flow rate changes by a factor of 3x. A faster set of constants was also tested but resulted in larger overshoot. If faster control is desired, constants of 1.0 for both $K_c$ and $T_i$ may be used while monitoring for potential instabilities.

One note is that the heater control is not effective when there is no flow because the heaters and the downstream temperature sensor at T600 are physically separated by a small amount of pipe. In the blowdown testing of the turbo-compressor, the flow is initially stagnant and has a rapid flow increase over the period of several seconds. The best working solution is to use manual heater control to change from 0 to a starting power around 20-40% for the first approximately 10 seconds before switching to closed-loop control. The control software implementation has bump-less switching between manual and automatic control so that this live switching provides continuous heater power for smooth operation. It also has capabilities for setting the maximum and minimum controlled heater power, maximum and minimum temperature setpoint, a setpoint ramp to assist with automated cold-starts while avoiding thermal shock, and heater power ramp limitations to also avoid thermal shock.

### 2.3. Cooling Valve Linearization

The operation of the cooling valves was particularly challenging because of their non-linear behavior. Standard closed-loop controllers are relatively easy to tune and control for systems that are mostly linear with little backlash. The valves therefore required linearization for stable integration into an automated control system. The cooling valve non-linearity was corrected by use of a transfer function in software, basically making their behavior appear more linear by adjusting two valve positions with a single input. Figure 8 shows several Gas Cooler water flow curves. The ideal is completely linear. The Original curve shows the large non-linearities shown similarly to that in Figure 3. The Linearized-v1 curve shows significant improvements but not ideal as the water flow rate was only about half what it should be for low values of the input Cooling Required. A second version of the Linearized transfer function was developed by adjusting valve positions in real-time and adapting to the ideal case as shown in Linearized-v2. This second version has excellent agreement to the Ideal curve with the only exception being slightly higher flow rates for Cooling Required around 2%. This linearization was deemed adequate for standard closed-loop control. The table for Linearized-v2 that was implemented in DP control software is shown in Table 1. It is implemented in the LabVIEW control software as an input text file for easy editing and is linearly interpolated.

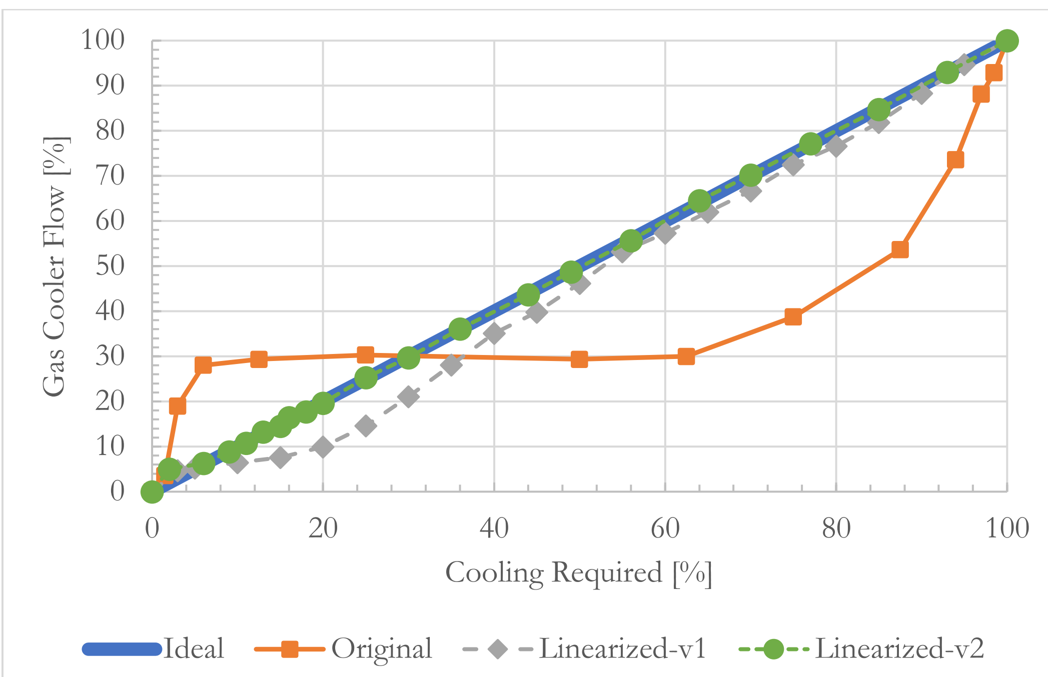


**Figure 8. Linearized valve characteristic curves showing ideal and versions 1 and 2**

**Table 1. Final valve linearization table to map Cooling Required to both valve positions**

| Cooling Required [%] | Water Main Valve Position [%] | Water Bypass Valve Position [%] |
|---|---|---|
| 0 | 0 | 100 |
| 3.5 | 3 | 100 |
| 19 | 6 | 100 |
| 25 | 12 | 100 |
| 26 | 25 | 100 |
| 28 | 50 | 100 |
| 30 | 100 | 100 |
| 32 | 100 | 75 |
| 39 | 100 | 50 |
| 54 | 100 | 25 |
| 74 | 100 | 12 |
| 88 | 100 | 6 |
| 93 | 100 | 3 |
| 100 | 100 | 0 |

## 2.4. Cooling Controller Tuning

The cooling control was based on the compressor inlet temperature (CIT) and actuated the water valves. From the linearization efforts, the controller sees a very linear response when only controlling the Cooling Required parameter. The tuning procedure was conducted similarly to the heater control with online step changes and following the Ziegler-Nichols methods (National Instruments, 2019). When CIT was stable at design conditions, the cooling was reduced from just under 70% to 50% and the CIT was monitored. The results are shown in Figure 9. The CIT shows an influence from the reciprocating pump that had a non-steady flow rate that did not impact the heater control tuning but had a noticeable impact on compressor inlet conditions where it was drawing from and injecting into the loop. Because of this unsteady behavior, a moving average with 200 points (4 seconds) was used to aid in estimating the time constant and time delay. Like the heater control, the closed-loop system had an initial first order response that transitioned to a linear response. From this, the time constant was estimated at 20 seconds. A closer look at the step change as in Figure 10 suggests that the time delay is approximately 5 seconds. These results suggested initial constants of $K_c = \frac{0.24(20\ s)}{5\ s} = 0.96$ and $T_i = 5.33(5\ s) = 26.25\ s$.

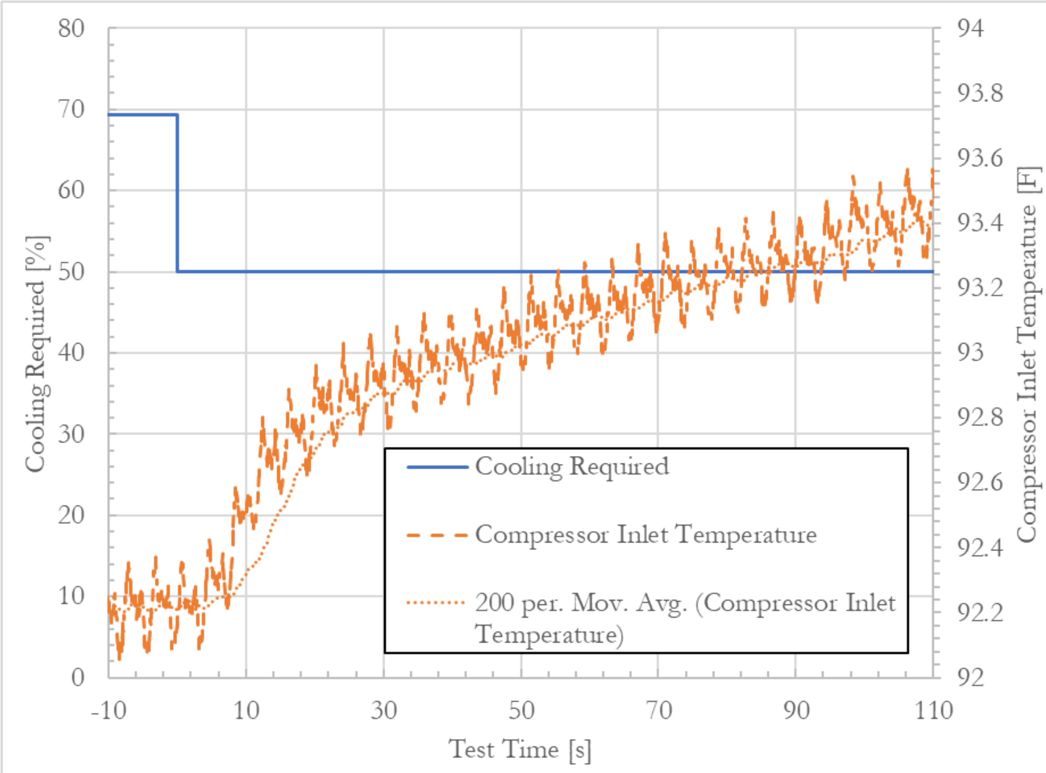


**Figure 9. Compressor tuning results**

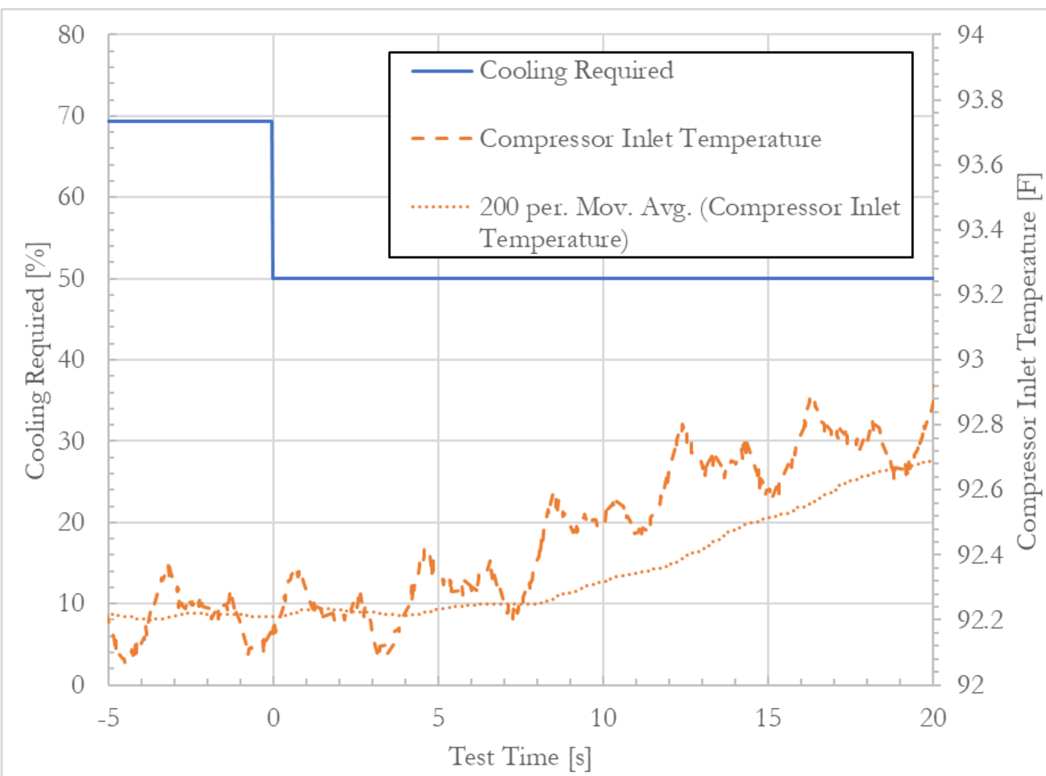


**Figure 10. Compressor tuning results with a focus on step change in cooling**

The cooling control constants were further refined by online testing. It was shown that the proportional gain $K_c$ was excellent but the integral time $T_i$ caused the system to be very slow. Many combinations were tested online until the final parameters

$$K_c = -1.0 \text{ and } T_i = 0.05 \text{ minutes}$$

were obtained. Note that the proportional gain is negative in this case because an increase in the control action (Cooling Required) decreases the CIT temperature, opposite to the control action of the heater. The proportional gain has twice the magnitude of the heater while the integral time is an order of magnitude smaller.

# 3. RESULTS

## 3.1. Heater Controller Testing

The heater control has been used on tests on a total of five days since it was tuned in November 2018 with no issues. The longest duration test was on 2019-05-07 that lasted eight hours. A plot of several important parameters to the heater controller from the first approximately two hours is shown in Figure 11. The mass flow rate transients had the potential to cause instabilities to the heater controller but temperature errors were typically less than 3°F. The temperature setpoint was changed several times with overshoot typically less than 5°F with rapid recovery. The heater power quickly rose when the setpoint was increased but also quickly decreased to maintain temperature. There is a slow decrease in heater power when the setpoint was constant suggesting that the piping and heat exchangers were absorbing less heat over time. Overall the temperature controller was able to maintain stable control for a variety of perturbations at both small and long time scales with acceptable errors.

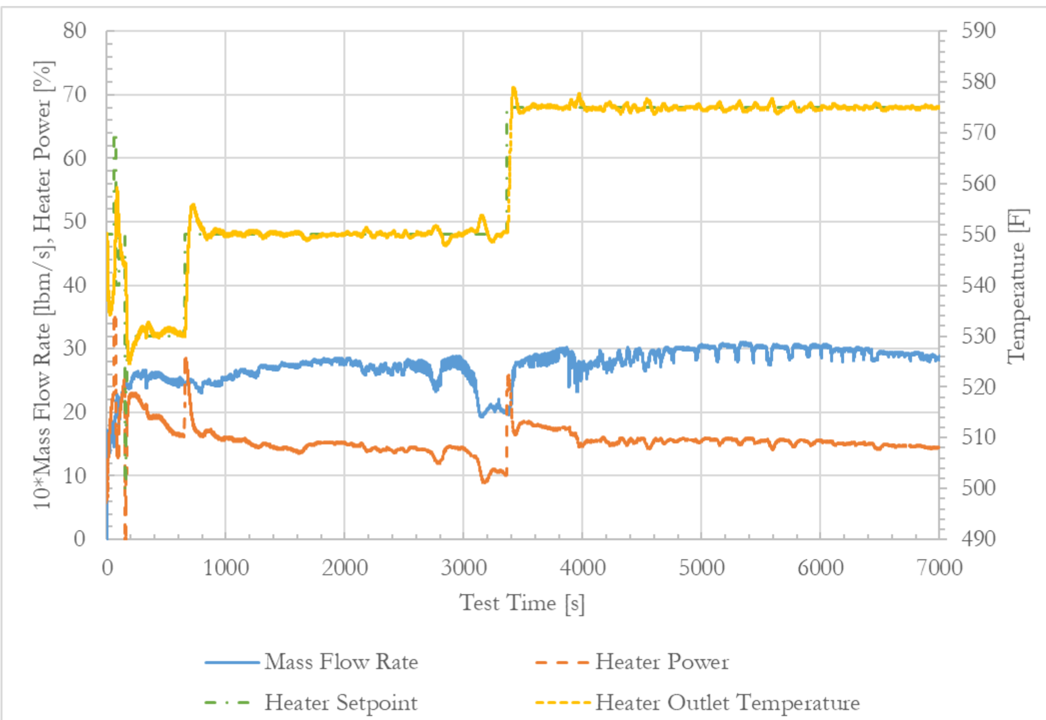


**Figure 11. Heater control data from turbo-compressor test on 2019-05-07**

A closer look at the startup/blowdown part of the test (during the first 1000 seconds) is shown in Figure 12 for a detailed look at the most challenging control conditions with many rapid changes. Contrary to many of the recent test practices, the heater power just prior to and during blowdown was automatically controlled. The heater output was able to adapt to rapid changes in mass flow and heater setpoint during the first 200 seconds of testing with typical errors of 10°F. The setpoint change around 650 seconds provides a good measure of the response when starting at stable conditions. The temperature setpoint was changed from 530°F to 550°F that caused a maximum overshoot of about 6°F and full convergence within 180 seconds. This behavior is thought adequate for the system requirements.

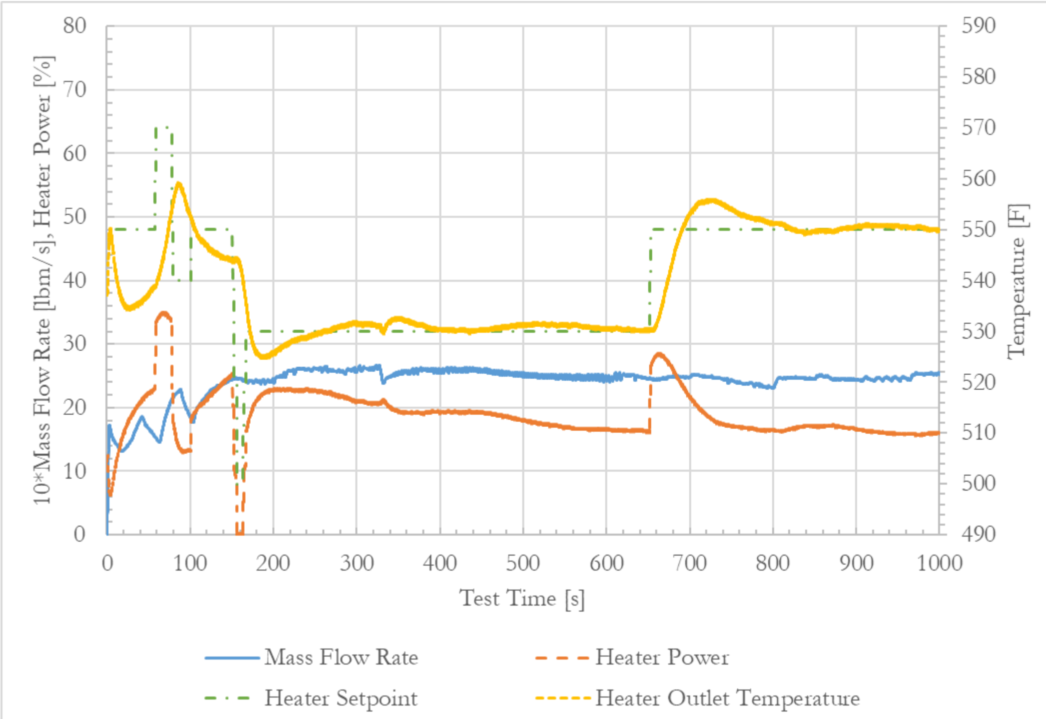


**Figure 12. Heater control data from turbo-compressor test on 2019-05-07 with focus on startup/blowdown**

## 3.2. Cooling Controller Testing

The automated cooling control was tested extensively immediately after the tuning was performed but has not been used on a turbo-compressor test due to the need for predictable behavior. The CIT has a large influence of turbomachinery performance and the properties of $sCO_2$ vary greatly near the critical point where the compressor performs best. Based on the extensive testing, it is believed that it is ready for turbo-compressor test support. The tests included a startup/blowdown simulation where the mass flow rate was quickly stepped from zero to a large value, a heater ramp at both high and low flow rates, and full transients of the cooling fan. The full cooling control shakedown test results are shown in Figure 13. The mass flow rate was ramped up early in the test and the cooling control was not turned on until 17.6 seconds into the test when the CIT was 93.2°F. After this, the CIT only rose another 1.3°F until the controller was able to achieve stability. This is a worst-case scenario and only expected during initial startup. After this, the maximum undershoot was 1°F and maximum overshoot was 1.3°F. The undershoots were typically due to step changes in mass flow rate, something that is expected to only happen during startup/blowdown in turbo-compressor tests. The cooling fan was also ramped up and down rapidly and the cooling control reduced the water flow rate accordingly as the water temperature (not shown) decreased. The thermal mass of the system and the long distance between the cooling tower and the Gas Cooler likely reduce the sharpness of this fast transient. The errors from full on and off step changes were less than 0.2°F, showing that the controller has excellent control for any fan transient. The heater power was also changed rapidly up and down on several occasions with little disturbance to CIT. An example of this is at 500 seconds where the heater power was nearly doubled while maintaining mass flow and the

CIT error was less than 0.3°F. Considering the three upsetting parameters tested, CIT is sensitive to mass flow transients and largely insensitive to heater power and cooling fan transients.

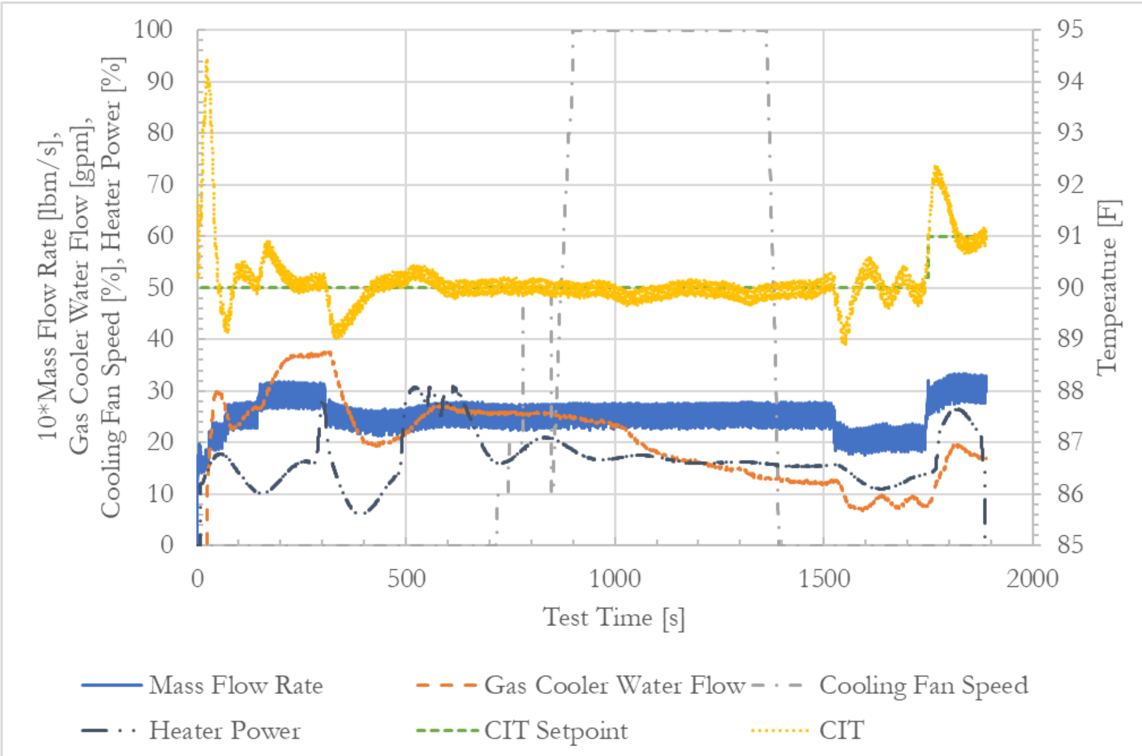


**Figure 13. Cooling control data from shakedown testing on 2019-02-06**

# 4. CONCLUSIONS AND FUTURE WORK

## 4.1. Conclusions

Closed-loop control was implemented on Sandia's $sCO_2$ Development Platform heating and cooling systems successfully. This included software capabilities development, cooling valve linearization, controller tuning, and finally system testing. The PI parameters presented earlier are repeated in Table 2 for quick reference. This applied controls work will increase system stability, reliability, and repeatability for greater quality results for high impact testing. Also, the methods presented herein can be applied to other control aspects of this and other systems for similar impact.

**Table 2. Controller proportional gain and integral time results**

| Control Type | Proportional Gain $K_c$ | Integral Time $T_i$ [minutes] |
|---|---|---|
| Heating | 0.5 | 0.5 |
| Cooling | -1.0 | 0.05 |

## 4.2. Future Work

The cooling tower fan controller was not able to be developed with the limited amount of personnel time available, but this is another opportunity for closed-loop control. The piping network is quite large, so the fan impact on compressor inlet control will have a large delay. The details of this are in a previous milestone report "Control Methods for the Sandia Closed Brayton Cycle" from 2016. The findings showed that a typical system flow rate of 200 gpm had a time delay of approximately 65 seconds. But the fan control can be based on cooling tower outlet water temperature that will have very little delay that can be controlled with a tuned PI controller. Controlling the water temperature will stabilize the system with beneficial results for the water flow rate controller as well as reducing operator load.

The DP has several other systems that would benefit from closed-loop control to increase stability and repeatability while decreasing operator load. These include the Turbine Control Valve (TCV) in addition to potentially Valves E and F. The TCV is currently a ball valve with very nonlinear behavior and sizeable backlash that make closed-loop control challenging. It is recommended that this be replaced with a globe/plug valve with an equal percentage characteristic curve that should provide more linear control with little backlash. Valves E and F currently have these characteristics and are expected to perform well.

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