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SUPERCRITICAL CO_2 BRAYTON RECOMPRESSION CYCLE DESIGN AND CONTROL FEATURES TO SUPPORT STARTUP AND OPERATION

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After graduating from Clarkson University with a BS in chemical engineering Michael joined Knolls Atomic Power Laboratory in 1980. His experience spans component and system design and plant analysis for training and operating power plants. Beginning in 2004 he developed the transient model for the Jupiter Icy Moons Orbiter (JIMO) Brayton power plant. Since 2007 he has been working on supercritical CO_2 Brayton power systems. For the last two years he has been an advanced plant projects advisor as well as a supercritical CO_2 contributor.

ABSTRACT

The efficiency of the Supercritical CO_2 (S-CO_2) Brayton cycle is increased by adding a second compressor (or recompressor) to the simple cycle. The additional compressor works on fluid before heat is removed by the precooler. Although the two parallel compressors have quite different inlet conditions, they must operate at nearly the same pressure ratio to avoid impacting the performance of the other compressor. A relatively small difference in pressure ratio can surge one of the compressors and shutdown the system. While maintaining adequate surge margin is required for any Brayton loop design, the recompression cycle adds the new challenge of balancing main compressor and recompressor performance. New control methods must address starting both compressors and maneuvering the Brayton loop through system heatup, power output changes and plant transients.

This paper presents the transient modeling analysis of the design and control features needed to safely operate the "recompression cycle". For this study a full plant TRACE model provided an ideal platform to develop and verify recompression loop control features. Results indicate that isolation valves and recirculation flow paths are needed for independent main compressor and recompressor startup. Once both compressors are operating, isolation valves are opened and the recompressor speed is controlled to maintain an optimum pressure ratio balance with the measured performance of the main compressor.

INTRODUCTION

The use of S-CO_2 as the working fluid for a closed loop Brayton power cycle has been the topic of three international symposiums [1-3]. For some applications this power cycle has the potential for higher efficiency, compact design, automated operation and reduced life cycle cost. In order to determine the design and control features required for recompression cycle operation a plant transient model must be used.

The Brayton recompression loop (Fig. 1) utilizes two compressors arranged in parallel to increase cycle efficiency [4]. The additional compressor is used to compress a fraction of the working fluid before energy is removed by the precooler. Recuperation is split resulting in low and high temperature units. The CO_2 that is compressed before entering the precooler rejoins the main flow path downstream of the low temperature recuperator. By splitting the recuperation duty each heat exchanger will be designed for a lower heat duty and lower temperature drop than the single recuperator in the simple cycle. Since the recompressor draws off CO_2 before entering the precooler this heat exchanger should also be smaller.

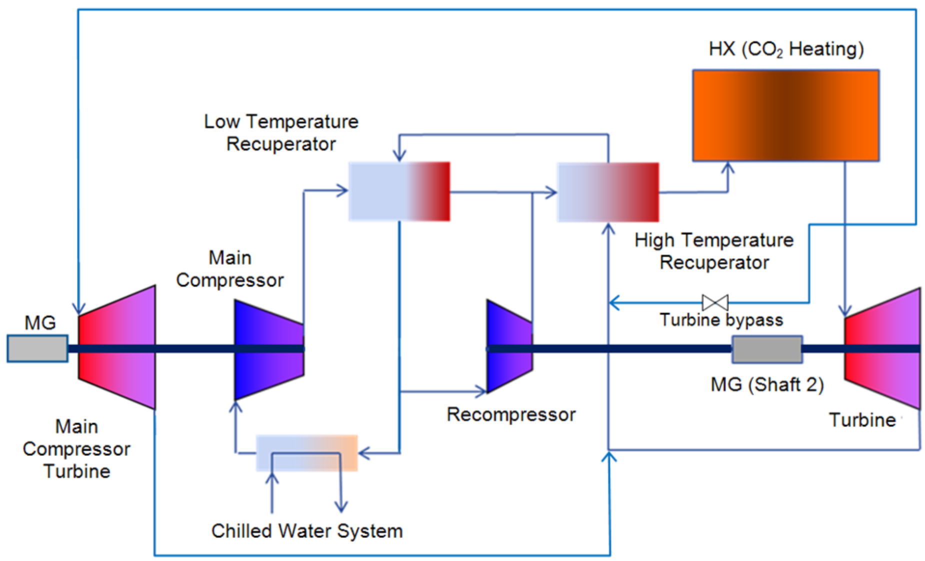

Figure 1: Typical Recompression Brayton Cycle System

The thermal efficiency benefits of this cycle increase with turbine inlet temperature. Figure 2 illustrates the potential for higher efficiency when adding the recompressor to the recuperated Brayton cycle. The heat balance calculations used to define cycle efficiency do not address design differences required to actually operate each Brayton loop variant. However, a basic understanding of compressor control challenges will lead to the conclusion that the recompression cycle control will be more complex. A transient analysis is needed to determine the control complexity and which cycle is best for a given application.

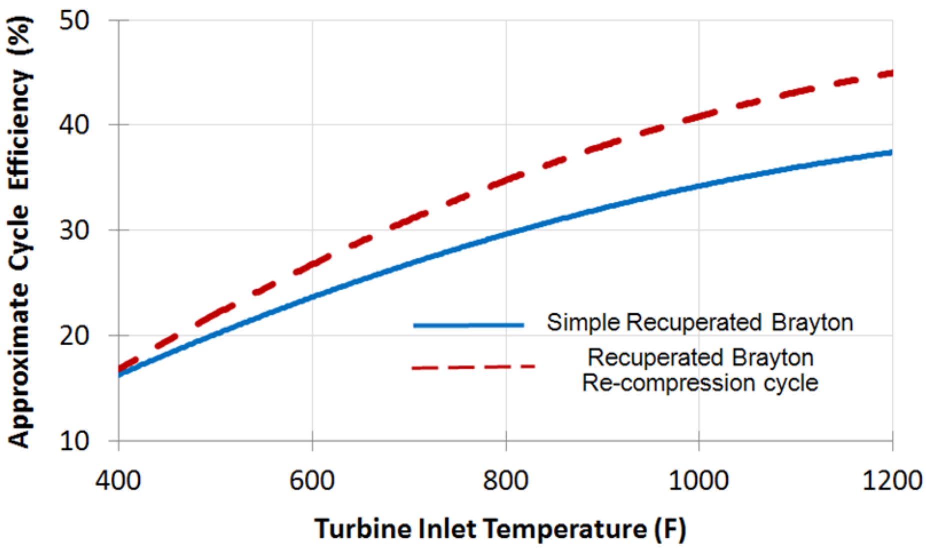

Figure 2: Relative Efficiency between S-CO_2 Brayton Cycles

To determine how to control and operate the Brayton recompression loop a strategy was adopted that builds upon an existing plant design and transient model. The plant design selected is the Integrated Systems Test (IST) shown in Figure 3. The IST [5] demonstrates a closed loop Brayton power cycle with S-CO_2 as the working fluid. The loop, located at the Bettis Atomic Power Laboratory, is a two shaft design that allows the power-turbine generator to operate at constant speed while the compressor speed is varied. Brayton loop power output may be varied between 0 and 100 kWe with compressor speed changes. Both motor-generator and thermal-hydraulic (valve) control features shown in Figure 3 will be used to change compressor speed. The

IST will also demonstrate control features needed to start and heat-up a Brayton loop while providing performance data to qualify component and system attributes of the transient code and model. A qualified transient model will allow Brayton system designs for future applications to be optimized and evaluated relative to other power generating technology.

Transient analysis of the IST is performed using a modified version of TRACE, a plant analysis code that is developed by the United States Nuclear Regulatory Commission (NRC). TRACE is the current successor code to previous NRC plant and accident analysis computer codes. The code utilized for this report is synchronized with TRACE Version 5.0RC3, which is the version that the NRC submitted for Developmental Assessment (a code qualification process for Light Water Reactors).

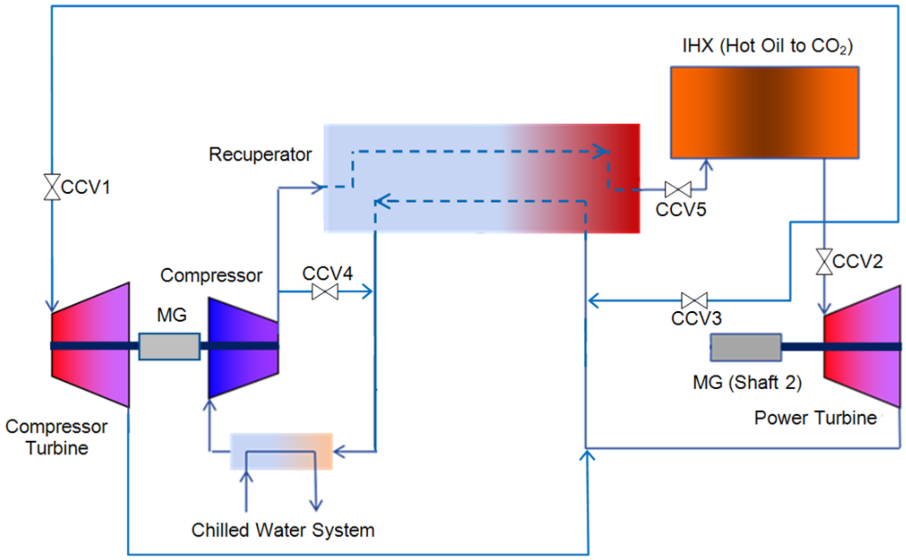

Figure 3: Basic IST Configuration and Control Devices

For this study the IST plant TRACE model [6] was modified by adding a recompressor, recompressor isolation valves, recompressor recirculation system and second recuperator. The Brayton turbomachinery in the new model was not redesigned and resulted in relatively poor cycle performance, but was an effective platform to develop and verify control features required for two compressor systems. The fundamentals of recompression loop operation draw from IST operating principles [6, 7] with the main compressor control tied to desired power output. Control features are added to the TRACE model to address new challenges of starting and operating the recompression cycle.

RECOMPRESSION LOOP DESIGN

Steady state heat balance calculations have typically been used to estimate cycle efficiency, size components and design Brayton power loops. At full power if all Brayton loop components operate at their predicted design point, few control features are needed. However, it is unlikely that first of a kind component and loop designs will function exactly as predicted. More importantly, a power plant must transition from cold shutdown to full power. To account for plant startup, heatup and power changes the Brayton loop must have the capability to modify and control loop thermal/hydraulic conditions as well as turbomachinery performance. Experience developing and testing IST control features [6, 7] has led to the selection of recompression loop design features needed for starting the two compressors and operating the loop at off-design conditions.

The following design features have been added to the basic Fig. 1 layout to support plant operations while avoiding surge in both compressors:

- Recompressor isolation valves
- Recompressor recirculation piping and control valve
- Dynamic recompressor speed setpoint controller

The IST (Fig. 3) model has been modified to create two Brayton recompression loop models. Figure 4 illustrates the first of two hydraulic variations of recompression loop control features. Figure 4 shows the addition of three control valves (CCV6, CCV7 and CCV8) which allow the recompressor to be independently started and hydraulically balanced with the main compressor. To support heat removal when the recompressor is operated in isolation a relatively compact heat exchanger may be needed. In a properly designed system, large recirculation flow fractions would only be needed from startup to low power operation. During these operations the recompressor speed will be low and fluid heating relatively small.

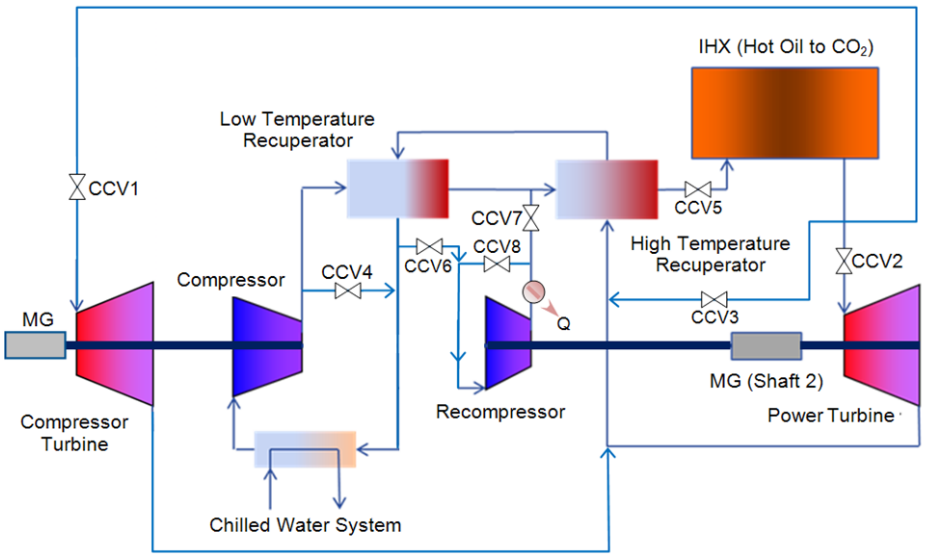

Figure 4: IST2 with Baseline Recompressor Recirculation

While the recompressor recirculation heat exchanger required in Fig. 4 may be relatively small, it still represents an additional component and cost to the system design. Therefore, a variant of the Fig. 4 design (IST2r) has also been developed and evaluated. In the IST2r configuration shown in Fig. 5 the recompressor recirculation path utilizes the low temperature recuperator for heat removal. The potential drawback to the simplified IST2r design is that recompressor startup is not fully independent of main loop operation. For this concern the plant transient model was exercised to determine if a suitable startup procedure can be developed.

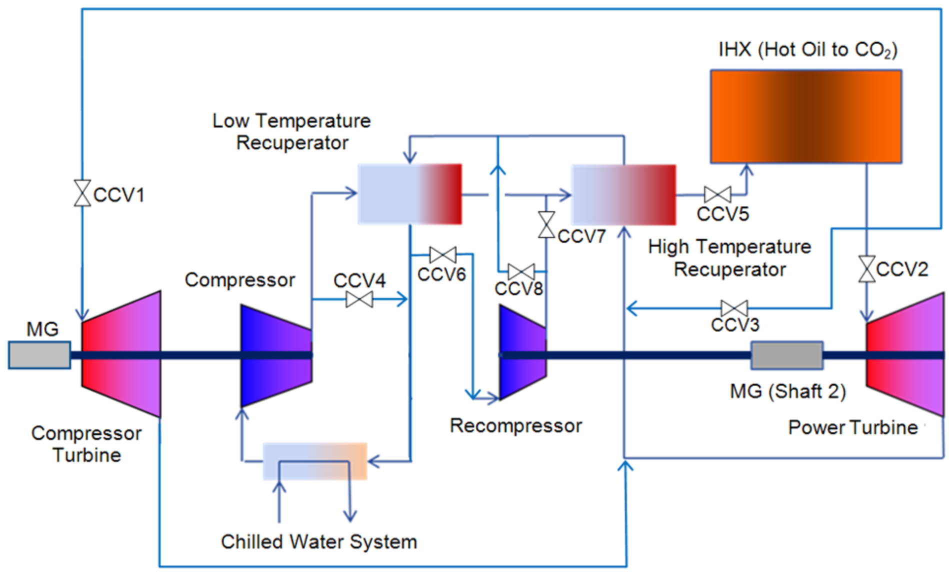

Figure 5: IST2r with Alternate Recompressor Recirculation

Both Figures 4 and 5 illustrate how the recompressor has been added to the IST model. In the original IST design (Fig. 3) the second shaft consists of a turbine generator. In the recompression models (IST2, IST2r) the recompressor wheel has been added to the second shaft, creating two variable speed shafts. Adding the recompressor to the existing IST power turbine shaft will reduce the loop power output and cause both compressors to operate at a lower flow (higher resistance) than originally designed. Table 1 defines the compressor and turbine design points used in the new recompression models. Note that at the design point, total compressor flow would exceed turbine flow by more than 5 lbm/s. This mismatch will tend to push compressor operation toward the surge line and make high power system control more challenging. The process of developing a control system and strategy that can be successfully implemented on the IST2 and IST2r recompression designs will be needed to support off-nominal operation for any (optimized) Brayton loop.

Table 1: IST2 Recompression Loop Turbomachinery Design

Component (Shaft number)	Design inlet temp (°F)	Design inlet density (lbm/ft^3)	Design flow rate (lbm/s)
Compressor (Shaft 1)	96 (36°C)	42 (673 kg/m^3)	12.2 (5.5 kg/s)
Re-compressor (Shaft 2)	140 (60°C)	12 (192 kg/m^3)	5.0 (2.3 kg/s)
Compressor Turbine (Shaft 1)	570 (299°C)	10 (160 kg/m^3)	5.3 (2.4 kg/s)
Re-compressor or Power Turbine (Shaft 2)	570	10	6.1 (2.8 kg/s)

Table 1 also shows that the compressor and recompressor are designed for very different inlet conditions. The much lower recompressor inlet density leads to a relatively large diameter wheel design (compared with main compressor). To illustrate why system control features are needed to operate a recompression loop, consider turbomachinery startup. If both compressors are started and operated at low turbine inlet temperature or low power the inlet conditions (density) at the compressor and recompressor inlet will be nearly the same. For any scenario where the main compressor and recompressor inlet densities are similar their performance will diverge. Since the recompressor has been designed to operate at a much lower density than the main compressor it will produce a larger pressure ratio and may surge the main compressor.

CONTROL SYSTEM DEVELOPMENT

IST recompression control system development began by adopting the SPEED control system used in the simple IST system design. In SPEED control the plant operator selects (changes) the desired Brayton power output. The control system determines the compressor speed required to meet the requested power level and adjusts the power or load applied to the compressor shaft motor generator to achieve the setpoint speed. Through a table lookup function the compressor recirculation valve (CCV4) is automatically repositioned such that when the speed change is complete the compressor shaft motor generator power (load) will return to zero. This simple Brayton loop control is calibrated such that the compressor shaft motor generator is used to drive speed changes but be effectively off during steady state operation. Other Brayton loop control strategies discussed in [6] could also be applied to the recompression cycle. A cooling water system control system uses anticipatory and feedback control to maintain the compressor inlet at a constant temperature. A primary heating loop controller changes the heat source power to maintain constant turbine inlet temperature. Extensive transient model simulations and early testing have demonstrated these control strategies are effective for Brayton loop operation.

In the recompression loop design main compressor control remains essentially unchanged. What is needed is a means to start both compressors and balance their performance so that system efficiency is maximized without driving either into surge. Control system development was an iterative process that evolved during the plant transient analysis. This iterative process used models that incorporated the recompression loop hydraulic design features (piping and valves) to perform startup transient simulations. Once the compressors are started the most effective method to balance their performance had to be found. The first control method tested utilized surge margin, a common control parameter for automatic compressor control. Surge margin is calculated for each compressor based on its performance map and current operating conditions. Surge margin is the ratio of compressor flow to the flow at the surge line at a given speed. Another control parameter tested for balancing main compressor and recompressor performance was measured pressure ratio. Transient simulations have shown that pressure ratio is a better indicator to balance performance than surge margin. More detail on the use of pressure ratio measurement will be described in following sections.

A control system needs an effective control device (action) as well as indication. Two types of control actions were investigated: loop hydraulic and recompressor speed (dynamic setpoint) control. Like the selection of the best performance indication, a transient analysis that included a range of plant operations is required. A comparison of the two control action options is summarized below.

Hydraulic Control

In a single compressor system the recirculation valve can be effectively used to modify compressor speed, pressure ratio and prevent surge. However, recompression loop simulations using recirculation flow control was only effective for setting the overall surge margin. Because the two compressors and recirculation valves are hydraulically coupled, recirculation valve action is ineffective in changing the relative pressure ratio of the two compressors. In addition, when operating near the surge line, feedback control using the recirculation valves was potentially unstable. Figure 6 illustrates an unstable controller that uses the recompressor recirculation valve (CCV8) to set recompressor performance. This figure also shows how sensitive surge margin is to recirculation valve position.

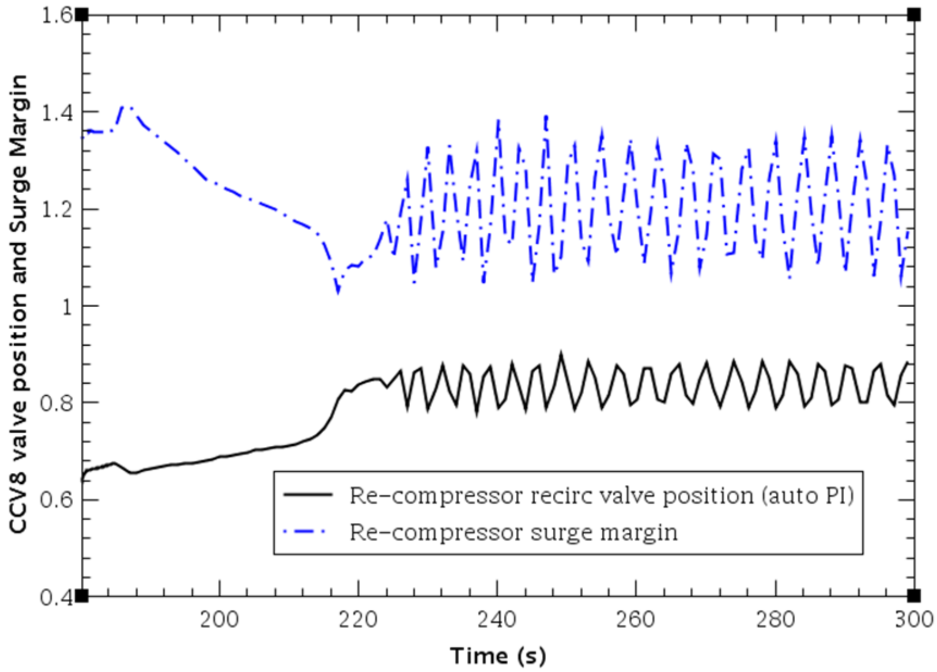

Figure 6: Unstable Control Action Using Recompressor Recirculation Valve

The reason for the controller instability shown in Fig.6 can be understood by reviewing a compressor performance map. Figure 7 shows an IST compressor map with an estimated surge line that defines the minimum safe flow rate. This surge line is created by combining test data (known safe operating conditions) with an estimate for where surge could occur. For example, it may not be practical (or worth risking equipment damage) to determine the exact compressor surge conditions at 70,000 rpm. Instead the map defines surge as the point where the constant speed line has zero slope (flat). A flat constant speed line means that a change in system hydraulic resistance that changes flow rate will have little or no effect on the y-axis parameter (corrected specific ideal enthalpy rise). The y-axis can also be defined in terms of ideal pressure ratio. Therefore hydraulic changes at the surge line will not produce a useful change in pressure ratio and the controller will not be stable. Instead, speed changes should provide the best method to affect recompressor pressure ratio.

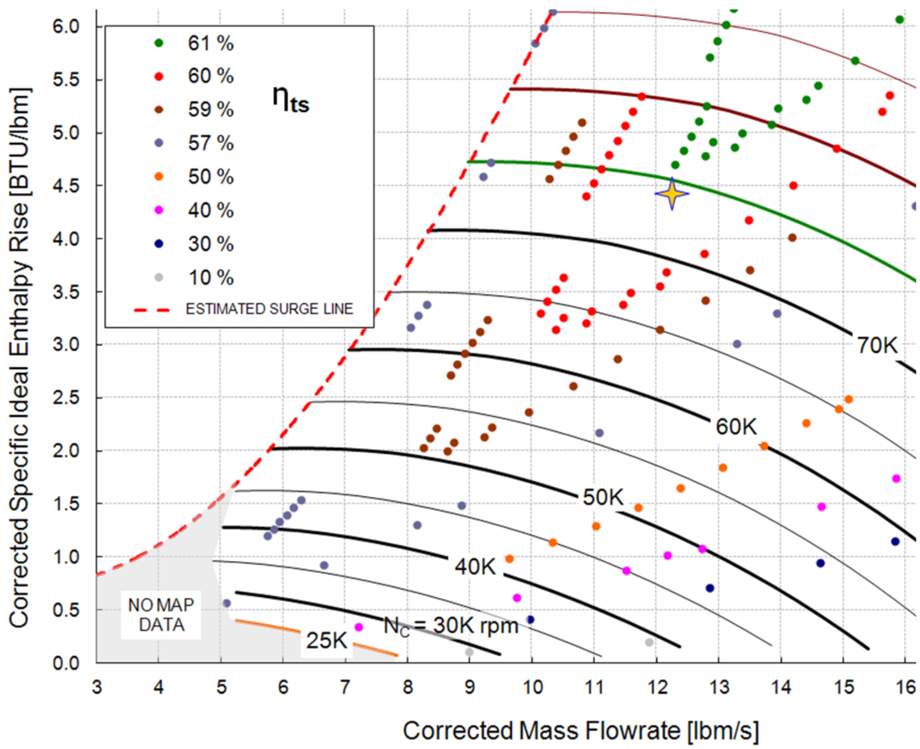

Figure 7: IST Compressor Map (Typical)

Recompressor Speed Setpoint Control

Having established compressor Pressure Ratio (PR) as the best indication of performance and compressor speed change as the best method to adjust PR, a recompressor control strategy has been developed. Brayton system power output is controlled through main compressor speed setpoint. Main compressor and recompressor recirculation flow control valves (CCV4 and CCV8) are set to maintain adequate surge margin as a function of main compressor speed. An additional controller is required to set recompressor speed such that it maintains the correct PR balance with main compressor. This balance is determined through transient analysis and confirmed by subsequent system calibration testing and expressed in terms of Pressure Ratio-Ratio (PRR). The term PRR is defined as:

$$PRR = PR\ recompressor\ /\ PR\ compressor$$

The PRR controller is implemented into the TRACE transient model as shown in Fig. 8. The Fig. 8 control structure allows manual (initial startup to idle speed) and automatic (applied once recompressor reaches idle speed) operation. In automatic operation the controller defines the PRR (a function of main compressor speed) and calculates the desired recompressor PR by multiplying PRR by the measured main compressor PR (see Fig. 8 cb7228). The desired recompressor PR is then used in a PI feedback control block to obtain the desired recompressor PR by modulating recompressor speed. PI control block (cb7166) varies the recompressor speed setpoint until the measured recompressor PR matches the desired recompressor PR. Details of the TRACE code PI controller are defined below:

- Inputs are recompressor PR (measured) and recompressor PR setpoint
- Output is Shaft 2 speed setpoint with units of radians/s
- Gain of 50,000 (based on PI output units of radians/s)
- Integration period (ΔT) of 2 seconds
- Time constant (lag) of 0.100 seconds

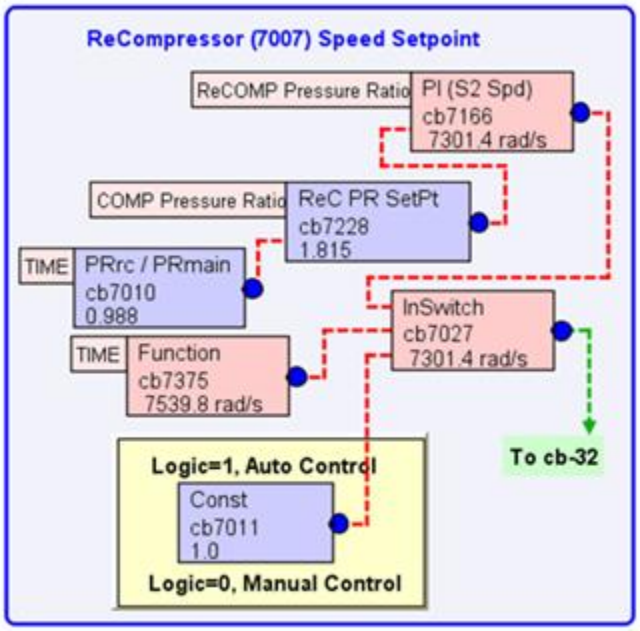

Figure 8: IST2 (IST2r) Recompressor Dynamic Speed Setpoint Control

TRANSIENT RESULTS

The recompression transient analysis has three parts. These parts are:

1. Main compressor and recompressor startups
2. Optimized high and low power heat balances
3. Power maneuvers

The first part simulates starting the main compressor with the recompressor isolated and establishes CO_2 flow for Brayton loop heatup. As the loop is heated the main compressor speed is increased and recompressor started. Once started the recompressor isolation valves are opened and the normal system flow paths established. The second phase of transient analysis is to find optimized high and low power heat balance conditions. The optimization determines the steady state control settings (valve positions and PRR) that produce the best trade-off between cycle efficiency (power) and surge margin. The final phase of transient analysis is to simulate power maneuvers between high and low power control points. This analysis optimizes controller time constants and determines the transient capability of the system design.

Main Compressor and Recompressor Startup

Both recompression models (IST2 and IST2r) are started with the same preheat conditions as the single compressor model. Establishing a turbine inlet temperature of 150°F (65.6°C), compressor inlet of 100°F (37.8°C) and loop pressure of about 1250 psia (8.62 MPa) allows the main compressor to be motored to half speed with adequate surge margin and produce a pressure ratio large enough to assure forward flow through the main compressor turbine. The recompressor isolation valves are shut during this process. With CO_2 flow provided by the main compressor there is no immediate need to start the recompressor. Because the IST turbomachinery bearings are gas foil designs they provide insufficient capacity below half speed. When starting the recompressor to half speed it will produce greater pressure ratio than the main compressor operating at the same speed. Therefore, before starting the recompressor and opening its isolation valves the main compressor speed must be increased to balance their pressure ratios.

Recompressor startup has been successfully modeled for the IST2 and IST2r models. Since the IST2 design allows the recompressor to be started fully isolated from the main loop, the more challenging IST2r startup is shown here. Recompressor startups were evaluated at turbine inlet temperatures of 300°F, 400°F and normal operating temperature of 570°F (149°C, 204°C and 299°C respectively). Startups at high temperature must consider thermal stress issues and may require preheat actions to reduce thermal transients. Figure 9 shows the compressor speeds during the IST2r recompressor startup at 400°F (204°C). To establish the correct initial PR balance the main compressor speed is increased from 37,500 rpm (idle speed) to about 42,000 rpm before starting the recompressor to idle speed. The need for this action is clear by noting the PR values shown in Fig. 10.

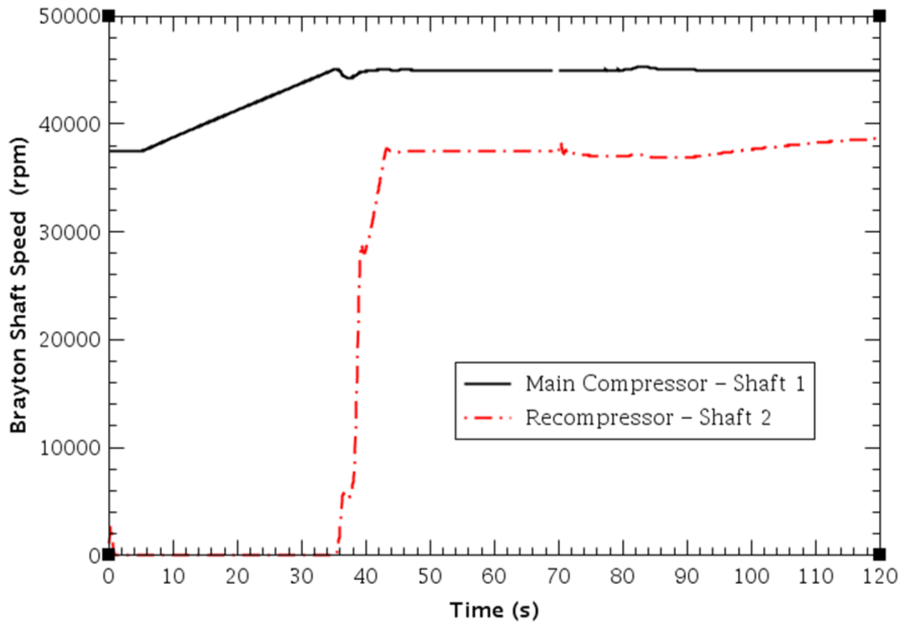

Figure 9: IST2r Shaft Speed during Recompressor Startup at 400°F Turbine Inlet Temperature

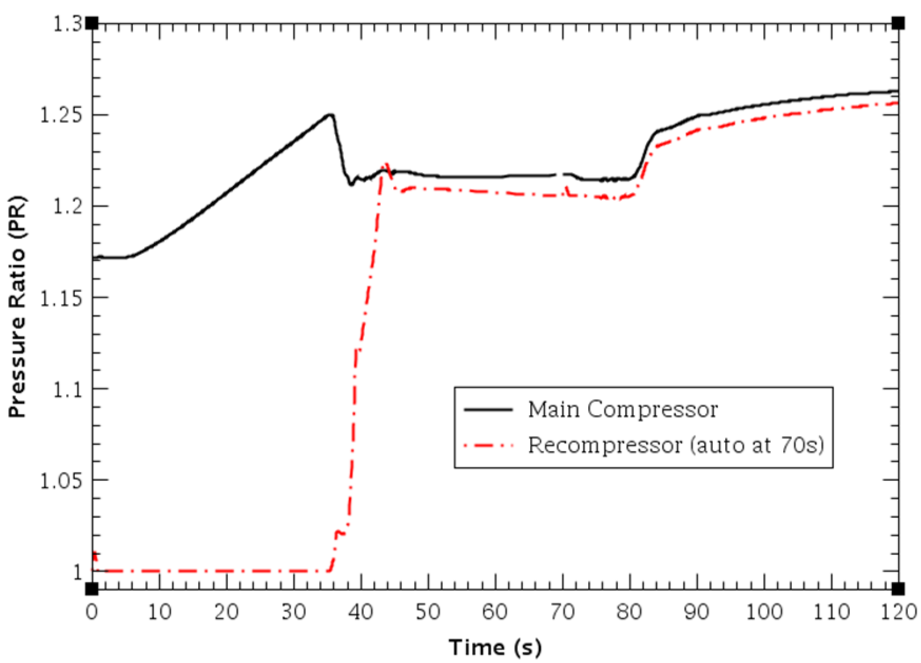

Figure 10: Pressure Ratios during IST2r Startup

Control system actions during IST2r recompressor startup include repositioning a number of valves. Prior to starting the recompressor the main compressor recirculation valve (CCV4) is repositioned to 70% open. As shown in Fig.11, when the recompressor is started (35 to 45 seconds) the power turbine and recompressor inlet isolation valves are opened (CCV2 and CCV6). The recompressor outlet isolation valve (normal flow path) CCV7 remains shut until stable loop conditions are established and automatic recompressor speed setpoint control is engaged at 70 seconds. At 70 seconds the normal recompressor flow path is established by opening CCV7. Although CCV7 is open, recompressor flow will continue to be fully recirculated until the recompressor recirculation valve CCV8 is throttled to 60% open. The CCV8 position change between 80 and 90 seconds produces a significant increase in main compressor pressure ratio (Fig. 10). Without the automatic recompressor speed setpoint control engaged the recompressor PR would not have followed the main compressor PR resulting in potential recompressor surge. Once all valve actions are complete and automatic recompressor speed setpoint control engaged total motor power for both shafts is reduced from 23kW to just 1 kW.

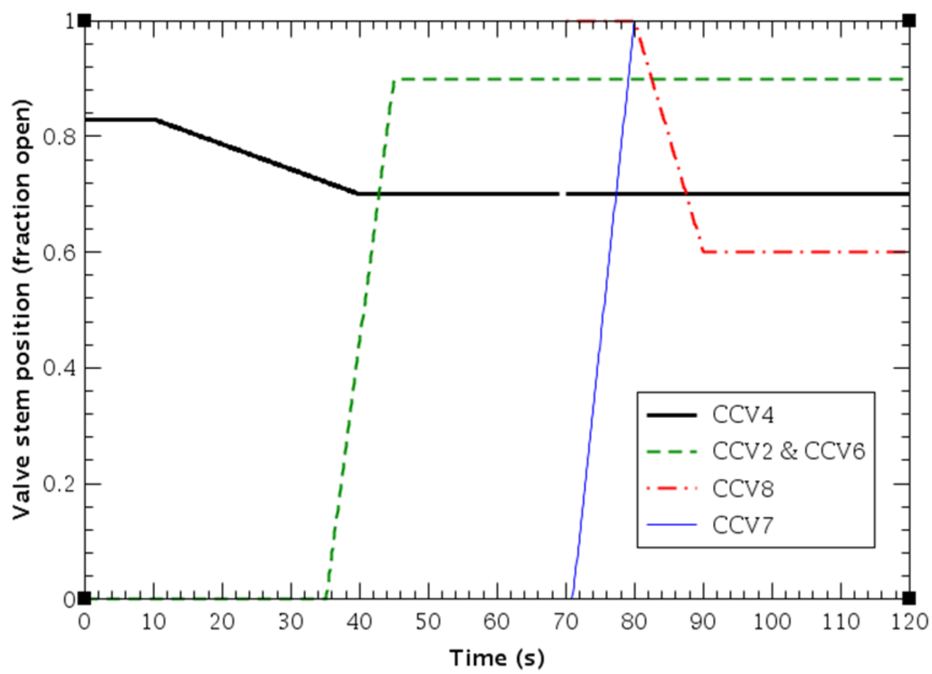

Figure 11: IST2r Recompressor Startup Valve Control

Optimized Heat Balance

Once the IST2 and IST2r models have been started and the Brayton loop maneuvered to normal operating temperature a sensitivity study is conducted to reach the highest power output with acceptable surge margin for both compressors. Because the IST recompression models have undersized turbines the system is surge margin limited and control settings were established to provide a main compressor and recompressor surge margin of 1.0. For a typical clean sheet design the target surge margin would be at least 1.1. For maximum power the main compressor speed is increased to its design value of 75,000 rpm. To set a surge margin of 1.0 the recirculation valve positions were 24% open for CCV4 and 45% open for CCV8. For the IST2 model the full power PRR setpoint is 0.988, resulting in a recompressor speed of 69,723 rpm.

Once the full power heat balance was found the model was maneuvered to an optimum low power heat balance. For example, dropping main compressor speed to 47,630 rpm produces about 10% power. To optimize cycle efficiency and power output at this condition the recirculation flow control valves are repositioned to 70% open for CCV4 and 60% open for CCV8. The PRR setpoint is also adjusted to 0.993. With high and low power heat balance control settings defined the next step is to perform transient power maneuvers. During transient conditions the PRR, CCV4 and CCV8 setpoints will continuously change. Setpoint values for PRR and recirculation control valve positions are linearly interpolated using main compressor speed setpoint between the low and high power values.

Power Maneuvers

Up and down power maneuvers have been performed at rates greater than 1% per second for both recompression loop designs. PRR dynamic recompressor setpoint control was found to be very effective in maintaining the PR balance during these maneuvers. Normally the surge margin would be maintained above 1.0 for both compressors throughout the transient. However, since the steady state full power surge margin was only 1.0, well controlled small decreases below 1.0 were considered acceptable for this study. Selected power maneuver results are described below.

The first example is an up-power maneuver using the IST2r model from minimum to maximum power. The transient is driven by the plant operator ramping requested power from roughly 13% to 100% in one minute. The Brayton control system interprets this request and ramps the main compressor speed from 45,000 to 75,000 rpm. Since recompressor dynamic speed setpoint control is engaged, the recompressor shaft speed is automatically increased to maintain the desired PRR. The resulting speed changes are shown in Fig. 12. The Brayton control system also repositions the two compressor recirculation flow control valves from their low power to high power settings. The resulting changes in Brayton loop flow are shown in Fig. 13. Finally, the effectiveness of PRR control is shown in Fig. 14. Figure 14 shows that the desired relationship between recompressor and compressor PR is controlled and surge margin is maximized for this design. An IST2 up-power maneuver demonstrated similar effective control.

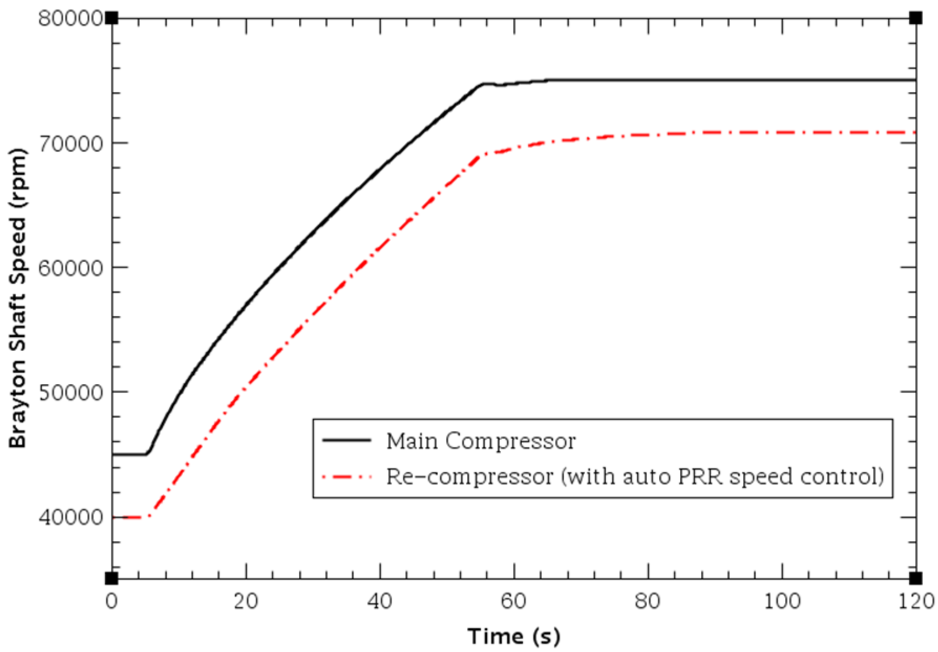

Figure 12: IST2r Up-Power Shaft Speeds

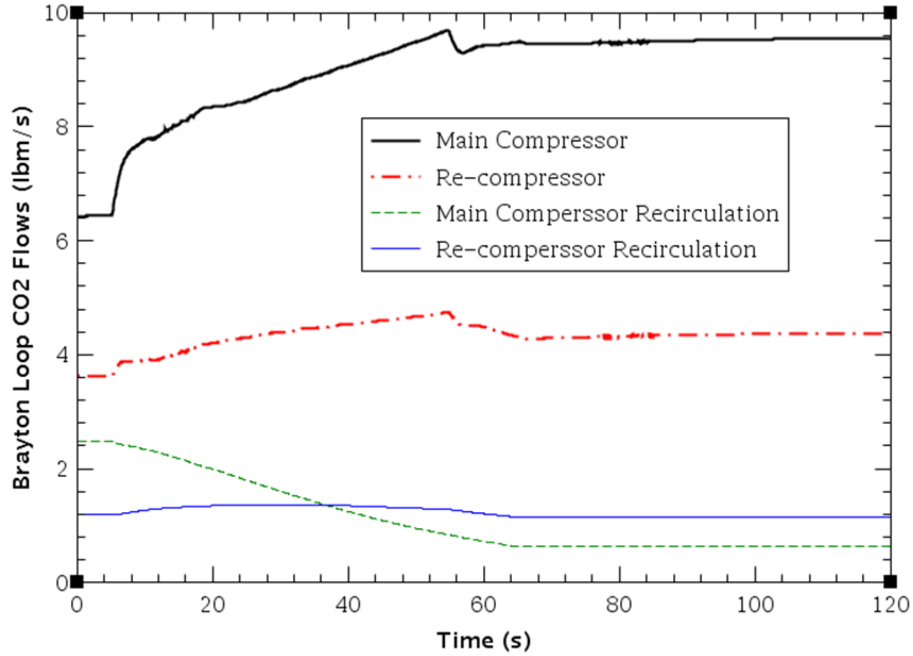

Figure 13: IST2r Up-Power Brayton Loop CO_2 Flows

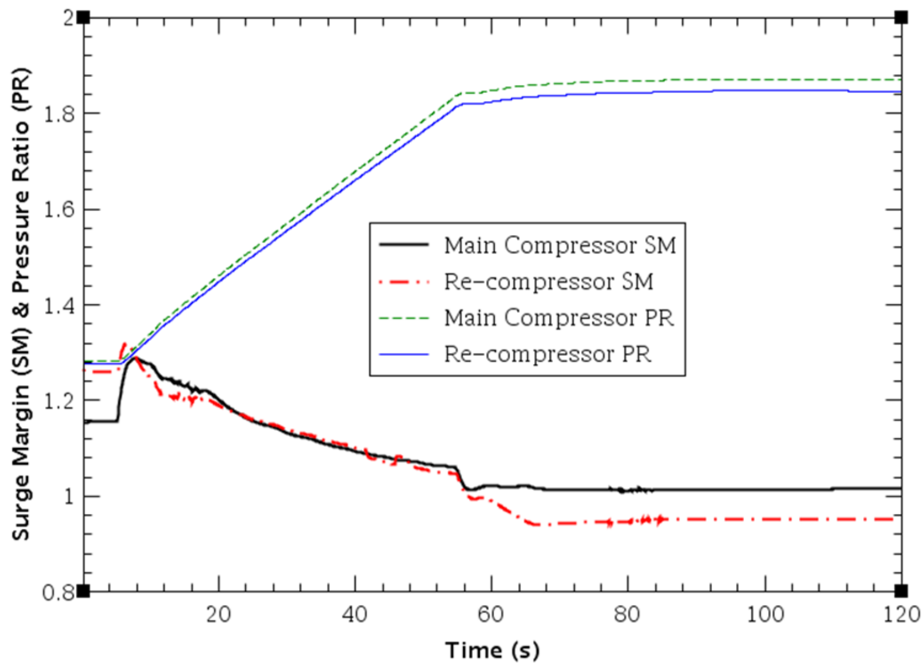

Figure 14: IST2r Up-Power Surge Margin and PR

The second example is a down-power maneuver using the IST2 model from maximum to low power. The transient is driven by the plant operator ramping requested power from 100% to about 15% in 60 seconds. The Brayton control system interprets this request and ramps the main compressor speed from 75,000 to 47,630 rpm. Since recompressor dynamic speed setpoint control is engaged, the recompressor shaft speed is automatically decreased to maintain the desired PRR. The resulting speed changes are shown in Fig. 15. The Brayton control system also repositions the two compressor recirculation flow control valves from their high power to low power settings. The resulting changes in Brayton loop flow are shown in Fig. 16. Finally, the effectiveness of PRR control is shown in Fig. 17. Figure 17 shows that the desired relationship between recompressor and compressor PR is controlled and surge margin is maximized for this design. An IST2r down-power maneuver demonstrated similar effective control.

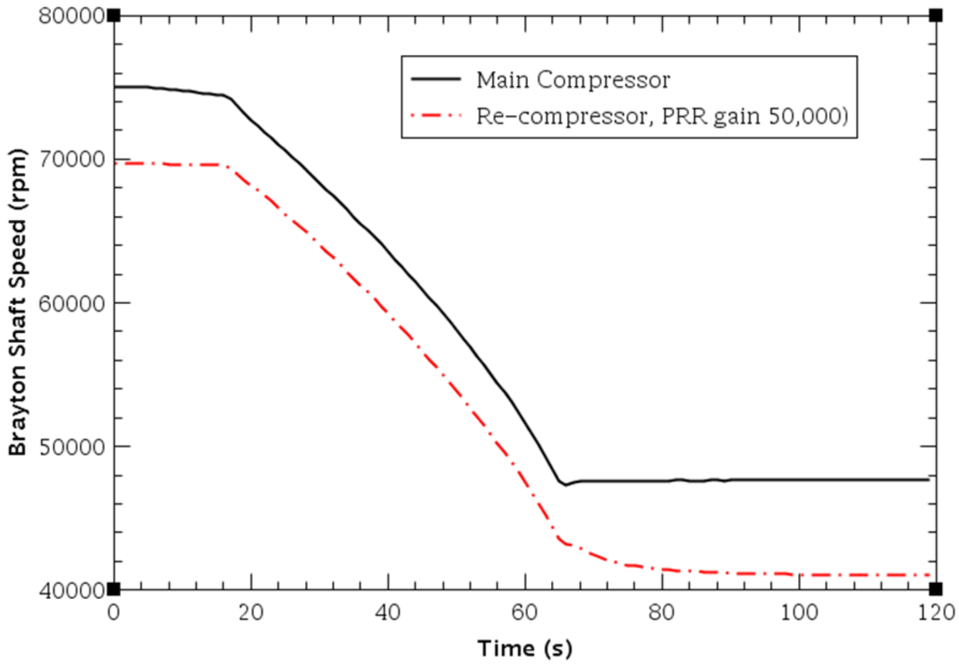

Figure 15: IST2 Down-Power Shaft Speeds

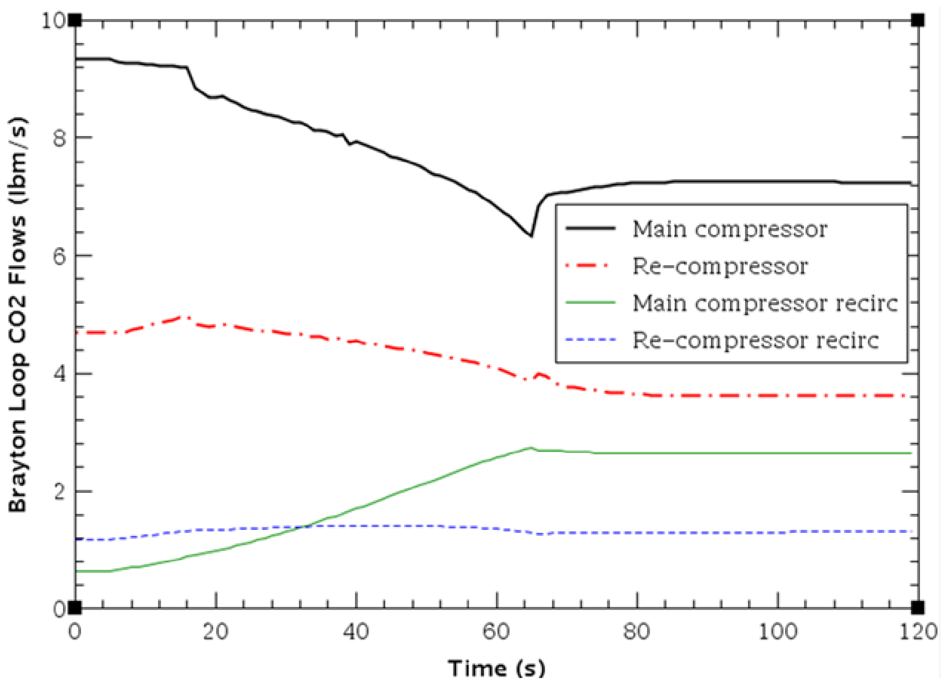

Figure 16: IST2 Down-Power Brayton Loop CO_2 Flows

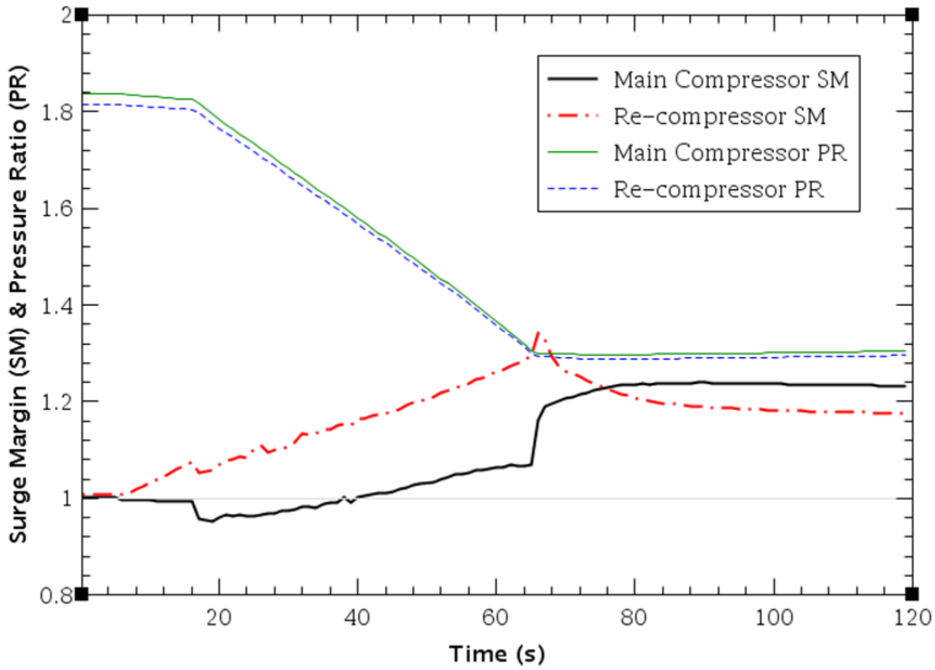

Figure 17: IST2 Down-Power Surge Margin and PR

CONCLUSION

The S-CO_2 recompression Brayton power cycle uses two compressors to increase cycle efficiency over the simple single compressor design. However, operating two compressors in parallel in a closed Brayton loop increases the potential for compressor surge and system shutdown. This study has identified design and control features that allow effective startup, heatup and power maneuvers using the recompression cycle.

A recompression variant of the 100 kWe S-CO_2 single compressor IST design has been developed to determine how to control a two compressor design. The analytic strategy was to minimize IST model changes, adding a second recuperator and compressor to rapidly begin the analysis. The two compressors and two turbines used in this analysis were not designed to operate together, making system control more difficult than an optimized design. To support recompressor operation, isolation valves and a recirculation path (with control valve) were added. Two hydraulic variants of recompressor recirculation were evaluated for loop startup and control. Both variants were successfully used for startup and normal operation. The hydraulic features and procedures needed for recompression loop operation are provided. Transient simulations identified the need to add a control feature that dynamically determines the optimum recompressor speed setpoint based on measured compressor and recompressor performance. This control system is required to successfully perform plant operations without compressor and/or recompressor surge. Without this control system very small differences between compressor and recompressor performance can lead to surge. This paper provides guidance on system design and control that should be used for recompression Brayton loops. The control features described enable startup, heatup, low power and off-nominal operation that places the loop in conditions that are different than the steady state design condition (full power heat balance).

The Brayton power loop designs described in this paper have power conditioning hardware to support variable speed generators. A large power plant application may instead use a constant speed power turbine generator. For this case the dynamic recompressor speed setpoint control would be replaced with another method to set recompressor PR. Recompressor performance might be altered with adjustable inlet guide vanes instead of speed changes. The general control principles provided herein should be adopted to fit the final application.

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