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INITIAL TRANSIENT POWER OPERATION OF A SUPERCRITICAL CARBON DIOXIDE BRAYTON CYCLE WITH THERMAL-HYDRAULIC CONTROL

E. M. Clementoni
Bechtel Marine Propulsion Corporation
West Mifflin, PA, USA
eric.clementoni@unnpp.gov

T. L. Cox
Bechtel Marine Propulsion Corporation
West Mifflin, PA, USA
timothy.cox@unnpp.gov

M. A. King
Bechtel Marine Propulsion Corporation
West Mifflin, PA, USA
martha.king@unnpp.gov

ABSTRACT

The Bechtel Marine Propulsion Corporation (BMPC) Integrated System Test (IST) is a two shaft recuperated closed Brayton cycle using supercritical carbon dioxide (sCO_2) as the working fluid. The IST is a simple recuperated Brayton cycle with a variable speed turbine driven compressor and a constant speed turbine driven generator designed to output 100 kWe. The main focus of the IST is to demonstrate operational, control, and performance characteristics of an sCO_2 Brayton power cycle over a wide range of conditions.

IST operation has reached the point where the system can be run with the turbine-compressor thermal-hydraulically balanced so that the net power of the cycle is equal to the turbine-generator output. In this operating mode, power level is changed by using the compressor recirculation valve to adjust the fraction of compressor flow that goes to the turbines. Operational data from transient power operations are presented for various system power level transients using a thermal-hydraulic based control method. The operational data showed very good agreement between commanded and actual turbine-generator output power and turbine-compressor speed for changes in the system load setpoint. The system also showed very stable response to poorly tuned controllers which resulted in large oscillations in the compressor recirculation valve position that directly controls turbine-compressor speed.

INTRODUCTION

sCO_2 power cycles offer the potential for higher system efficiency than other energy conversion technologies, especially when operating at temperatures above 450°C (842°F) [1]. Additionally, sCO_2 power cycles enable more compact arrangements than other technologies due to the small turbomachinery size enabled by higher fluid density at the turbine. These benefits have led to sCO_2 power cycles being actively developed for a wide range of applications including waste heat recovery, concentrated solar power, and fossil power sources.

sCO_2 power cycle development efforts at BMPC focus on Brayton cycle thermodynamics, system control modeling, power cycle demonstration, and heat exchanger development. BMPC has designed and operated the IST to demonstrate operational, control, and performance characteristics of an sCO_2 Brayton power cycle over a wide range of conditions [2-10]. The IST is a simple recuperated closed loop sCO_2 Brayton system (Fig. 1) with a variable-speed turbine-compressor and a constant-speed turbine-generator. The IST is designed to generate nominally 100 kWe at a relatively modest turbine inlet temperature of

570°F (299°C), as shown in the design full power heat balance (Fig. 2). The modest turbine inlet temperature and low turbine and compressor efficiencies resulting from the small scale of the IST equipment cause the overall loop efficiency to be much lower than is predicted for larger sCO_2 Brayton cycles.

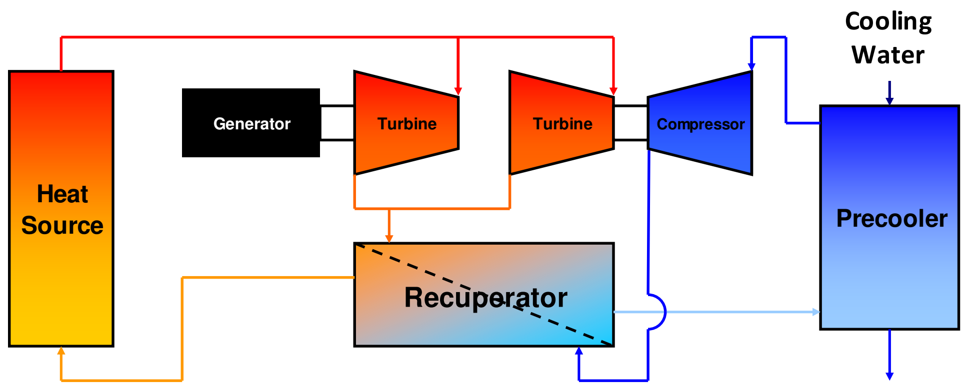

Figure 1. Simple Recuperated Brayton Cycle

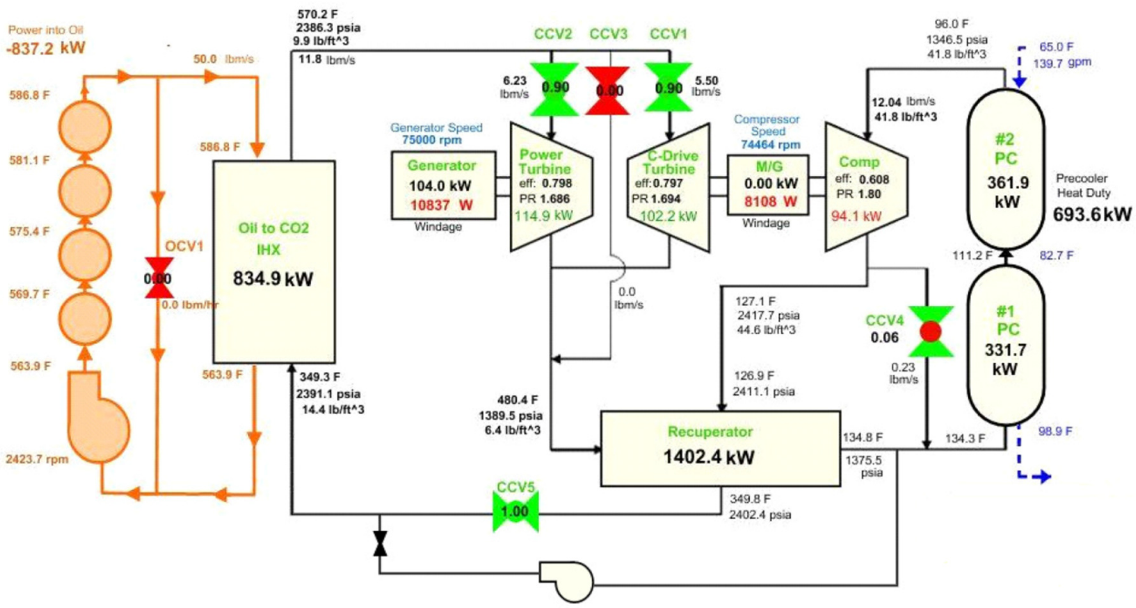

Figure 2. IST Design Full Power Heat Balance

The IST component layout and the physical arrangement of the test loop are shown in Fig. 3 and Fig. 4. The heat source for the IST is a 1 MW electrically heated organic heat transfer fluid system which provides heat to the CO_2 through a standard shell-and-tube heat exchanger. The heat sink is a chilled water system which rejects heat from the precooler and other heat loads to a refrigerated chiller. This chilled water system is broken into two loops so that cooling flow can always be provided to auxiliary heat loads throughout the system while the precooler conditions are regulated to maintain the desired compressor inlet conditions.

The turbomachinery for the IST was designed and fabricated by Barber-Nichols, Inc. (BNI), Arvada, CO. The design speed is 75,000 rpm for both the turbine-compressor and turbine-generator. Both machines incorporate a permanent magnet motor-generator on the shaft which provides the ability to maintain the desired shaft speed independent of loop thermal-hydraulic conditions. While the IST turbine-generator is designed to operate with a generator output of 100 kWe at a speed of 75,000 rpm, limitations associated with the motor-generator controller and higher than anticipated windage heating [9] have limited generator output to 40 kWe while operating at a speed of 60,000 rpm.

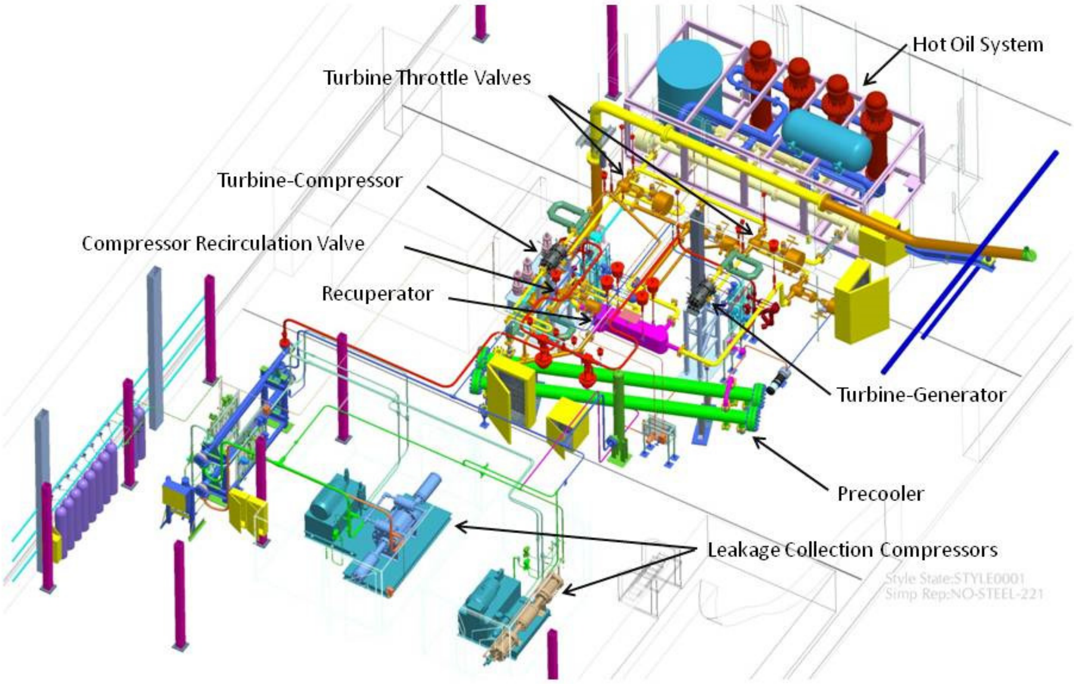

Figure 3. IST Component Arrangement

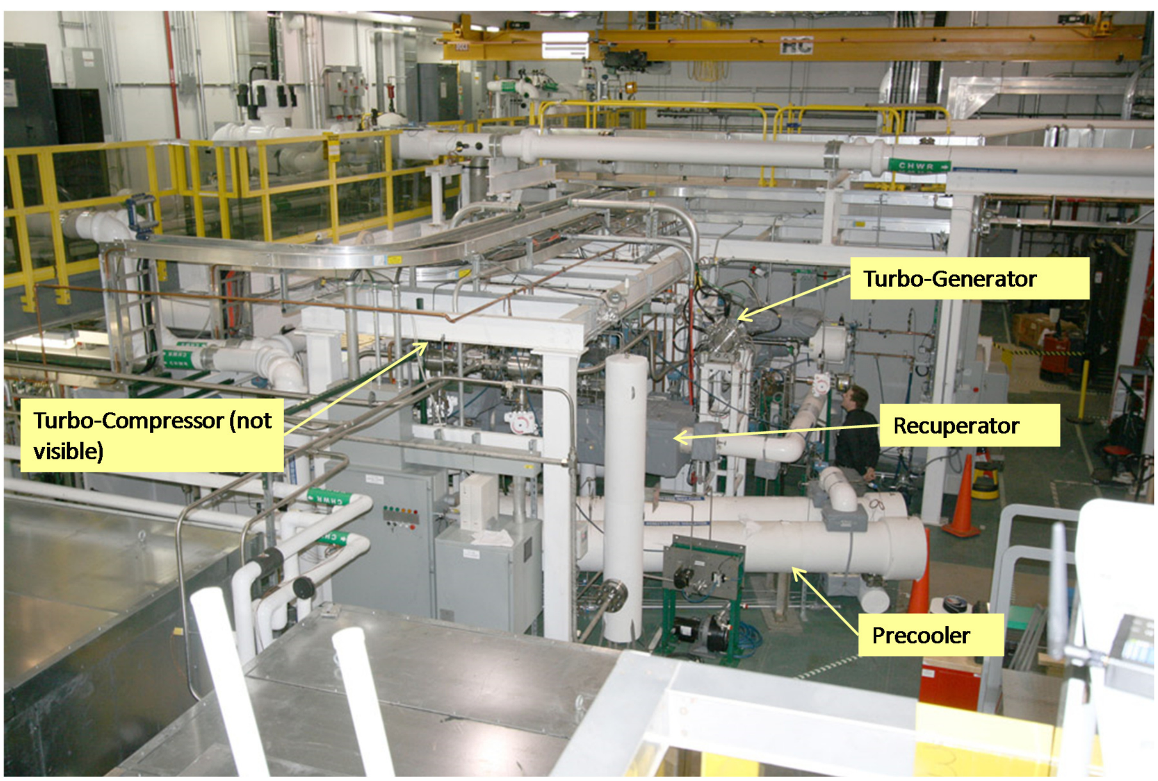

Figure 4. IST Physical Layout

SYSTEM CONTROL

A system transient model [10] was used to develop control strategies which would provide stable steady-state and transient control of the sCO_2 Brayton power cycle. Three control strategies for the IST were developed that utilize different methods for maintaining turbine-compressor and turbine-generator shaft speeds. The three control strategies build upon each other starting from direct shaft speed control and working toward a control strategy capable of following changes in the generator load where the shaft speed for both machines is controlled by the loop thermal hydraulics using turbine inlet and compressor recirculation control valves.

For normal IST operations, the turbine-generator is maintained at a constant shaft speed and the turbine-compressor speed is varied to change system power. Additionally, the compressor recirculation valve (CCV4) position changes as a function of system power level in order to provide adequate compressor surge margin over the range of power levels. At low power levels and turbine-compressor speeds, adequate compressor surge margin is maintained by having the compressor recirculation valve open wider to allow more compressor flow to bypass the turbines and circulate through the precooler and back to the compressor. At high power levels and turbine-compressor speeds, the compressor recirculation valve is closed further, sending more of the compressor flow to the turbines for power production. Closing the compressor recirculation valve and increasing the compressor speed also result in an increase in compressor pressure ratio. Ideally, the compressor recirculation valve would be fully closed at maximum system power such that all compressor flow would be sent to the turbines to produce power. In practice, the minimum compressor recirculation valve position for stable system control is 5-10% depending on the valve dead band, hysteresis, and slope of the flow coefficient (C_v) curve at low valve positions.

The cycle control strategy utilized during the testing presented in this paper is a thermal-hydraulic based cycle control method. In this control mode, the turbine-generator speed is maintained at a commanded setpoint by the motor-generator controller which applies the necessary load to regulate speed as turbine hydraulic power changes. The turbine-compressor operates in a thermal-hydraulically balanced mode where the power of the compressor turbine is equal to the compressor work plus internal losses. The turbine-compressor motor controller does not motor or generate; instead, turbine-compressor speed is maintained by modulating the compressor recirculation valve using proportional-integral (PI) control and a turbine-compressor speed setpoint based on the desired system power level. The turbine-compressor speed setpoint as a function of system power level is provided in a lookup table. This lookup table was initially developed using model predictions and ultimately refined using loop operating data.

In addition to the controls necessary to regulate system power level and turbomachinery speed, the main cycle process control system has several additional PI controllers to maintain important loop operating conditions. The turbine inlet temperature is established by maintaining the intermediate heat exchanger (IHX) CO_2 outlet temperature at a desired setpoint by modulating the heater power. The compressor inlet temperature is maintained at a setpoint by controlling the flow of chilled water through the precooler. The chilled water temperature into the precooler is automatically controlled to a fixed value by using the warm return flow from the precooler to heat a portion of the chilled water flow heading to the precooler inlet through a plate and frame heat exchanger. CO_2 loop pressure is not directly controlled but is a function of the mass of the entire CO_2 system which is held constant during normal operation.

The pressure setpoint for the motor-generator cavities for both turbomachines is maintained using feedback control to adjust the stroke speed of the leakage collection compressors. The motor-generator cavity pressure setpoint is minimized to reduce shaft windage losses, but the minimum achievable cavity pressure is limited by the flow capacity of the leakage collection compressors. CO_2 flow is also injected into the cavities to provide cooling for the bearings and stator to augment the cooling provided by shaft seal leakage. This augmented cooling flow is regulated based on table lookup control which sets the position of leakage collection system control valves to maintain cavity temperatures within operating limits.

SYSTEM OPERATING CONDITIONS

The system operating conditions have been altered from the design conditions due to limitations with the turbine-generator output power which is limited to approximately 40% of design power. As explained above, the thermal-hydraulically based cycle control method utilizes compressor recirculation valve position to control system power level. In order to utilize the maximum available range of the compressor recirculation valve without exceeding the maximum turbine-generator output power, the IHX CO_2 outlet temperature setpoint is reduced from the design point of 570°F (299°C) to 440°F (227°C). The compressor inlet temperature is maintained at 96°F (36°C) and the turbomachinery cavity pressure is set to 200 psia (13.8 bar). The total system mass is adjusted once these conditions are achieved to obtain a target compressor inlet pressure of 1310 psia (90.3 bar) with both shafts spinning at 37,500 rpm. Compressor inlet pressure increases as normal turbine-generator speed and thermal-hydraulically balanced turbine-compressor operation are achieved but then holds relatively steady throughout the power operating range due to the fixed system mass and the stable control of precooler inlet temperature. Minor changes in compressor inlet pressure with power level are seen due to changes in the average system density as small amounts of mass shift between the main loop and leakage collection system as a result of changing turbomachinery motor-generator cavity temperatures and densities caused by windage heating.

DISCUSSION OF RESULTS

Initial transient operations were performed with the IST operating in the thermal-hydraulic based control mode discussed above. The turbine-generator load setpoint was raised from 5% load (5 kWe) to increasing setpoints of 10% (10 kWe) to 40% (40 kWe), held to establish steady-state conditions, and returned back to 5% load. The rate of change in the turbine-generator load setpoint was automatically regulated in the control system at 0.5% per second to allow for gradual changes between the initial and final steady-state conditions. As shown in Figure 5, the turbine-generator output was in very good agreement with the load setpoint throughout the transients. Output power tracked very well with the changes in load setpoint and stabilized quickly at the setpoint power levels.

During the power level transients, the generator output lagged the commanded load setpoint and took up to 50 seconds to reach the commanded power level after the load setpoint reached the target value. This lag can be attributed to the performance of the motor-generator controller for the turbine-generator. As the turbine-compressor speed changes with the load setpoint, the flow to the turbines and the system pressure ratio change which affect the power produced by the generator turbine.

For an up-power event (increasing compressor speed and flow), the shaft power produced by the generator turbine increases which, in turn, causes the shaft speed to increase. The motor-generator controller responds by increasing the load to maintain the desired shaft speed. Similarly, the motor-generator controller reduces load in response to speed decreases during down-power transients. However, the response of the motor-generator controller to these speed excursions caused by the transients was not sufficient to adequately maintain the 60,000 rpm speed setpoint. Turbine-generator speed excursions as large as 2,500 rpm from the 60,000 rpm setpoint were observed during the transients due to the inability of the motor-generator controller to quickly regulate the load on the machine. The turbine-generator speed eventually returns to the 60,000 rpm setpoint around the same time that the generator output reaches the load setpoint. These factors indicate that the lag in generator output experienced during the power level transients could be reduced or eliminated by tuning the motor-generator controller to more quickly respond to changes in shaft speed caused by changes in turbine power.

Generator output at the 5% load setpoint gradually increased through the test with each return to this setpoint. This change was the result of changing turbomachinery cavity conditions. The abradable bushings that make up part of the turbomachinery shaft labyrinth seals degraded through this test such that the leakage collection compressors could not maintain the cavity pressure setpoint at high power levels and the cavity pressure increased above the setpoint. At the high power levels the compressor discharge pressure increases which creates a larger differential pressure across the seals, further increasing shaft seal leakage. The inability to maintain cavity pressure at high power levels caused the PI controller, which controls reinjection compressor stroke speed to regulate cavity pressure, to undershoot the target cavity

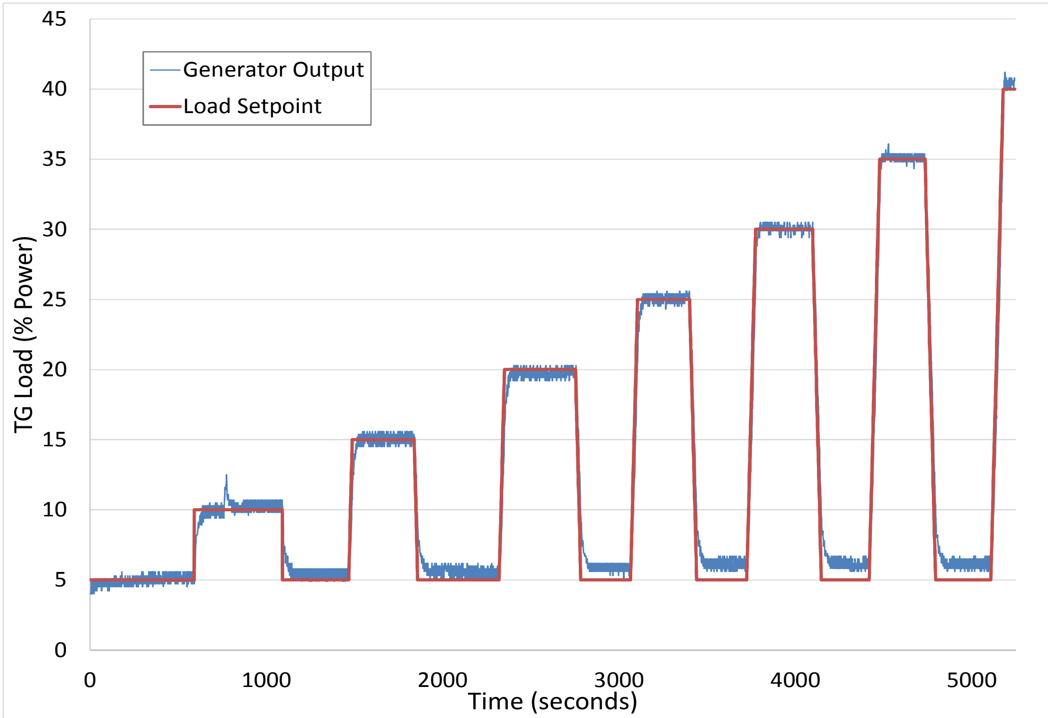

Figure 5. Turbine-Generator Load Control

pressure when the system power level was reduced back to the 5% level, thereby reducing the windage losses on both shafts resulting in additional system power output for the same turbine-compressor speed. An unexpected spike in generator output power can be observed in Figure 5 at around t = 760s. This spike corresponds with a brief stall of one of the reinjection compressors which caused the average pressure within the turbomachinery cavities to drop with an accompanying decrease in windage losses and increased turbine-generator output.

The power output of the turbine-generator is controlled by changing the position of the compressor recirculation valve, which changes the speed of the turbine-compressor. The system controller uses a lookup table to determine the turbine-compressor speed setpoint that corresponds to the generator load setpoint. Figure 6 shows the control of the turbine-compressor speed relative to the speed setpoint. The turbine-compressor speed follows very well with the speed setpoint and stabilizes quickly at each new steady-state value. Brief overshoots in speed of up to 250 rpm above the normal speed band are seen for many of the transient maneuvers but these overshoots are corrected within a matter of seconds.

It can be seen in Figure 6 that the turbine-compressor speed was more stable at high power levels than low power levels. For example, at the 5% load setpoint the turbine-compressor speed varied by as much as 550 rpm from the desired speed while at the 35% load setpoint this deviation is less than 325 rpm. The turbine-compressor speed is controlled by modulating the compressor recirculation valve position. Figure 7 shows the commanded compressor recirculation valve position determined by the PI controller based on turbine-compressor speed. The valve position had very drastic oscillations for steady-state power levels with the magnitude of the oscillations decreasing with increasing system load setpoint and decreasing valve opening. At the 5% load setpoint, the compressor recirculation valve commanded position had a range of 10% valve travel and the setpoint was changing by as much as 3% open per second.

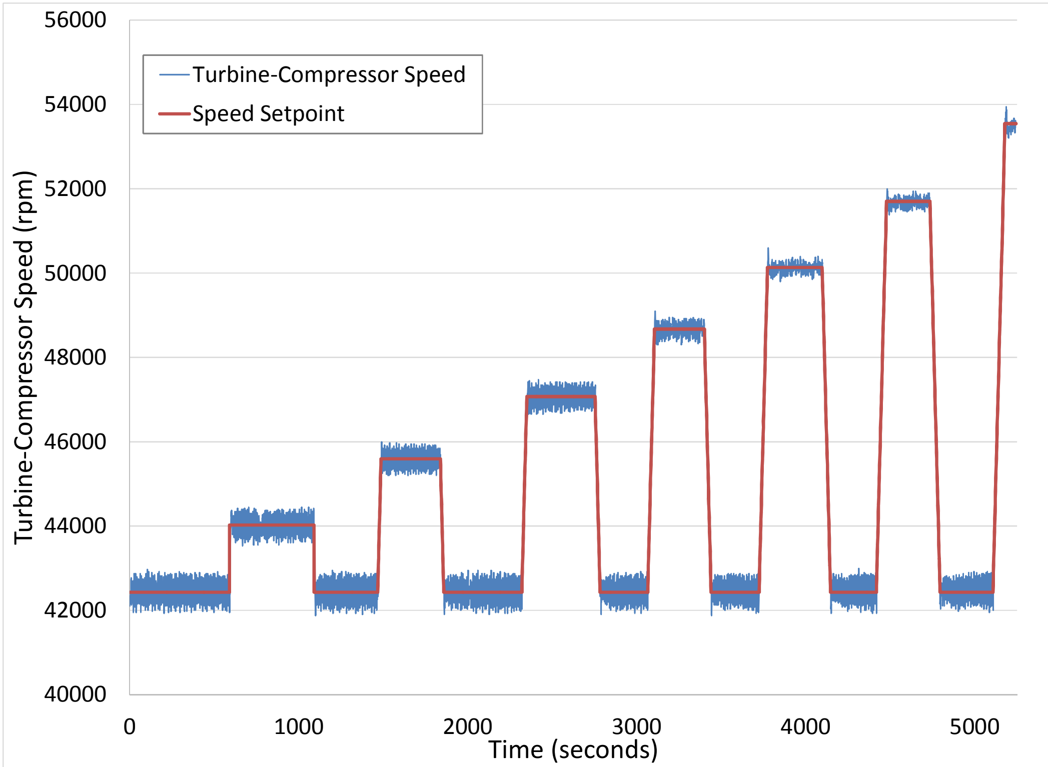

Figure 6. Turbine-Compressor Speed Control

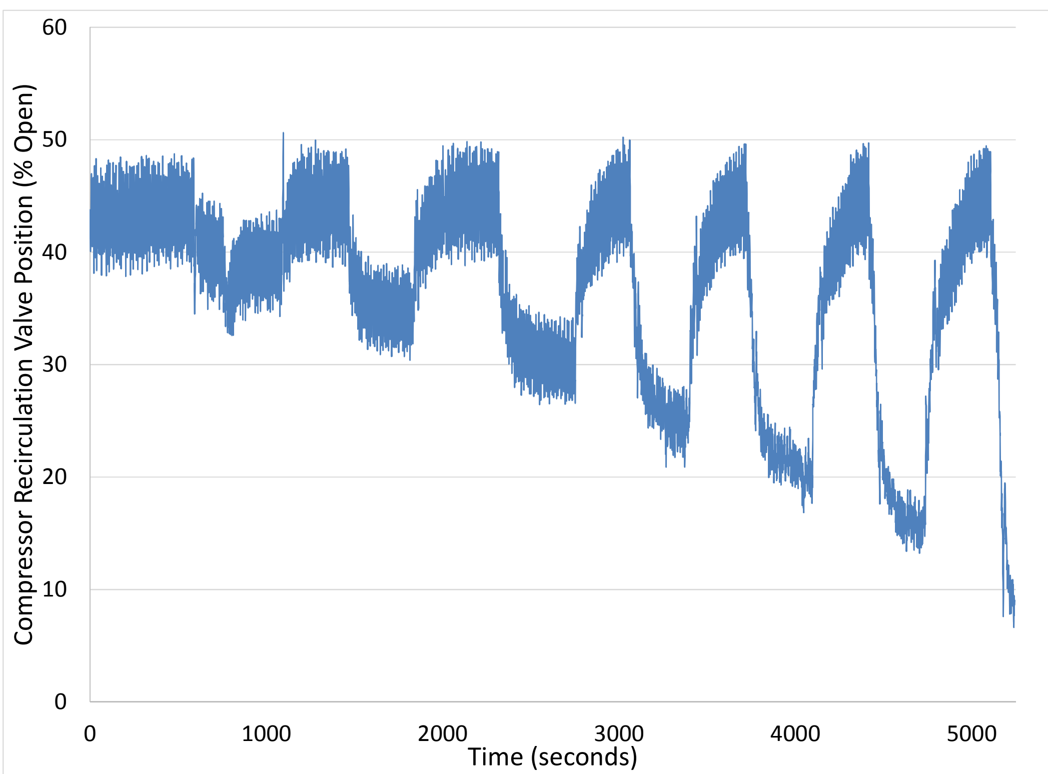

Figure 7. Compressor Recirculation Commanded Valve Position

The reason for the oscillations in commanded compressor recirculation valve position is a combination of the turbine-compressor speed signal delay, the process control PI constants in use for valve position, and the inconsistent response of the valve's internal controller. The turbine-compressor speed is calculated by the motor-generator controller and provided to the test loop process control system every 250 milliseconds. With this low update rate and unsteady windage losses resulting from use of reciprocating compressors to regulate cavity pressure, the reported speed can change by up to 250 rpm away from the setpoint even with the motor-generator controller regulating shaft speed. The process control PI constants for valve position in use during the testing were developed using the system transient model which assumed that the shaft speed was stable unless intentionally perturbed by a change in setpoint or turbine and compressor conditions. The constants were then selected based on the ability to control system power level for design power transients. Additionally, erratic behavior has been observed in the compressor recirculating valve's internal controller, which would oscillate unpredictably even for a steady-state setpoint. Further tuning of the PI controller constants and valve control is expected improve turbine-compressor speed control.

The turbine-compressor speed oscillations at the low load setpoints were further exacerbated by the fact that the flow coefficient curve for the compressor recirculation valve (equal percent flow characteristic) is much steeper in the valve position range for low power operation than at high power. The flow coefficient (C_v) corresponding to the valve position through the test is shown in Figure 8. At the 5% load setpoint, the flow coefficient was fluctuating by as much as 25% from the average value during steady-state operation. However, due to the smaller valve oscillations and the flatter response of the C_v curve at low valve positions, the flow coefficient was only fluctuating by less than 10% of the average value while operating at the 35% load setpoint.

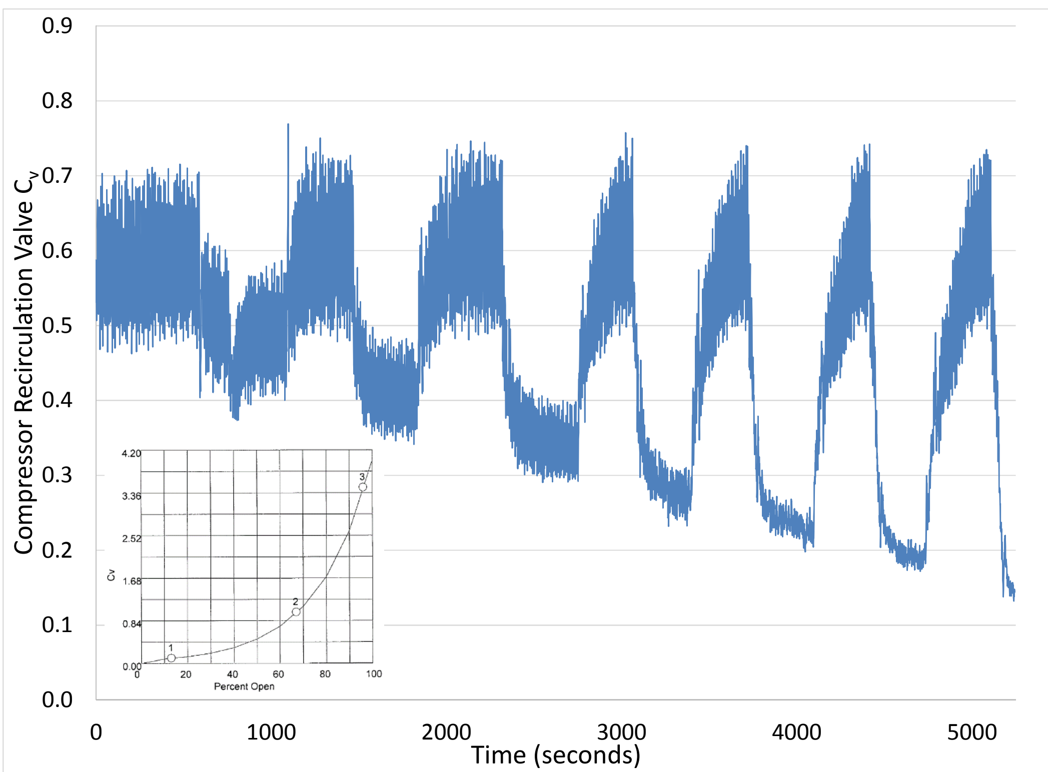

Figure 8. Compressor Recirculation Valve Flow Coefficient

Despite the large oscillations in compressor recirculation valve position, the overall system response is very stable. The generator output and compressor speed both were in very good agreement with the setpoints. This indicates very good stability of the sCO_2 simple Brayton power cycle even when operating with non-optimum control values. This behavior may not be translatable to other more complex Brayton Cycle configurations, such as a recompression cycle, where the performance of the two compressors must be very closely controlled to ensure stable parallel operation.

CONCLUSIONS

The IST has been operated for a range of power transients between 5% and 40% of design power rating using a thermal-hydraulic based control system. System power level is controlled by regulating the compressor recirculation valve position to attain the desired turbine-compressor speed and turbine-generator output. The overall response of the turbine-generator output and the turbine-compressor speed were in very good agreement with the setpoints for these parameters despite large oscillations in compressor recirculation valve position caused by controlling with PI constants which were developed using stable modeling assumptions and had not yet optimized for actual system operation. Tuning of the PI constants for the compressor recirculation valve control of turbine-compressor speed is expected to reduce the oscillations on valve position and provide even more stable system response. However, tuning of the PI constants may make the system less responsive to changes in load setpoint.

IST operation continues to demonstrate the overall feasibility and controllability of the sCO_2 Brayton cycle. Operation of the system has not identified any inherent issues with the cycle. Issues observed in the IST to date have been associated with the small scale and high speed of the turbomachinery and the associated power electronics that control the equipment. As discussed in this paper, stable system transient control can be accomplished with overly responsive control features.

NOMENCLATURE

BMPC	=	Bechtel Marine Propulsion Corporation
IHX	=	Intermediate Heat Exchanger
IST	=	Integrated System Test
PI	=	Proportional-Integral
sCO_2	=	Supercritical Carbon Dioxide

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