The 6th International Symposium - Supercritical CO_2 Power Cycles
March 27-29, 2018, Pittsburgh, Pennsylvania

SIMULATION OF IST TURBOMACHINERY POWER-NEUTRAL TESTS WITH THE ANL PLANT DYNAMICS CODE

Anton Moisseytsev
Principal Nuclear Engineer
Argonne National Laboratory
Argonne, IL USA
amoissey@anl.gov

James J. Sienicki
Manager, Innovative Systems and Engineering Assessments
Argonne National Laboratory
Argonne, IL USA
sienicki@anl.gov

Anton Moisseytsev is a Principal Computational Nuclear Engineer in the Nuclear Engineering Division of Argonne National Laboratory (ANL). He has fourteen years of experience in modeling and simulation of various systems, including design and analysis of the advanced reactors and energy conversion systems, safety analysis of nuclear reactors, and code development for steady-state and transient simulations of nuclear power plants. Anton has been involved in the development of the supercritical carbon dioxide Brayton cycle at Argonne since 2002.

James J. Sienicki is the Manager of the Innovative Systems and Engineering Assessments Section and a Senior Nuclear Engineer in the Nuclear Engineering Division at ANL. He has been leading the development of the supercritical carbon dioxide Brayton cycle at ANL with funding from the U.S. Department of Energy since 2002. He is involved in the design of sCO_2 Brayton cycle power converters for Sodium-Cooled Fast Reactors as well as the design and analysis of experiments on fundamental phenomena involved in heat exchangers for supercritical CO_2 cycles.

ABSTRACT

The validation of the Plant Dynamics Code (PDC) developed at Argonne National Laboratory (ANL) for the steady-state and transient analysis of supercritical carbon dioxide (sCO_2) systems has been continued with new test data from the Naval Nuclear Laboratory's (operated by Bechtel Marine Propulsion Corporation) Integrated System Test (IST). The common feature of these tests is power-neutral operation of the turbine-compressor shaft, where no external power through the alternator was provided during the tests. Two tests were analyzed: compressor-inlet temperature variation and turbine-compressor speed variation. The new test data turned out to be important for code validation by allowing validation of the shaft dynamics equations in asynchronous mode, as well as validation of both the compressor surge control and the turbine bypass control actions. Although the steady-state and transient simulations of both tests showed good agreement with the test data, several areas of future improvements for the PDC simulation of the IST are identified.

INTRODUCTION

The Plant Dynamics Code (PDC) has been developed at Argonne National Laboratory (ANL) for system level transient analysis of supercritical carbon dioxide (sCO_2) Brayton cycle power converters (Moisseytsev and Sienicki, 2006). The code has been used extensively for cycle design analysis, as well as for development and refinement of cycle control strategies. Recent work on the PDC development has been focused on code validation with the experiment data obtained from the Recompression Closed Brayton Cycle (RCBC) sCO_2 Loop (Moisseytsev and Sienicki, 2015) at the Sandia National Laboratory (SNL), and from the Integrated System Test (IST) (Moisseytsev and Sienicki, 2016) at the Naval Nuclear Laboratory (NNL) operated by Bechtel Marine Propulsion Corporation.

The IST loop configuration is shown in Figure 1 and described in detail in Clementoni and Cox, 2014a, 2014b; Kimball, 2014; and Rahner, 2014. The most recent IST test results are presented in Clementoni et al., 2017.

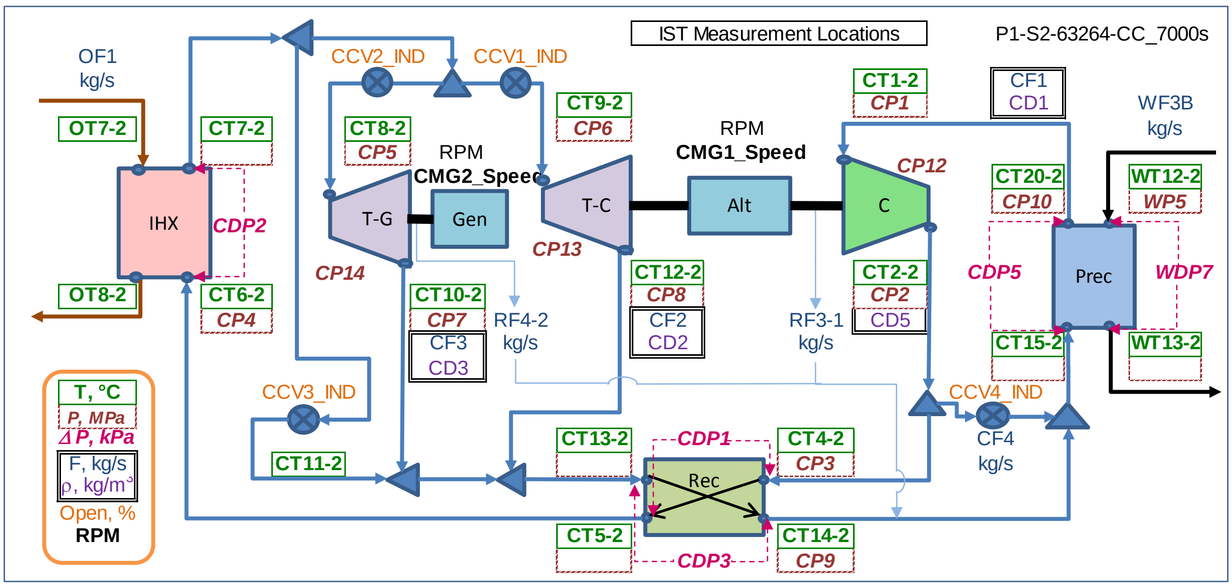

Figure 1. IST Configuration and Measurements.

New IST Test Data

The work described in this paper is based on the simulation of new IST test data provided by NNL to ANL. The common characteristic of these new tests is that no direct speed control on the turbine-compressor shaft was implemented, i.e., the power applied to or taken from the alternator on that shaft was exactly zero. For these reasons, the tests are referred to as “power-neutral” because no external power was applied to the turbine-compressor shaft; instead the shaft speed was allowed to change as dictated by the power balance between the turbine and the compressor. The shaft speed, though, was indirectly controlled by means of the compressor recirculation valve (CCV4 in Figure 1) which acts similarly to the turbine bypass control action investigated previously in the PDC sCO_2 cycle control analysis. The generator turbine shaft speed was actively controlled all of the time by the shaft speed control.

The first test, numbered 64661, is the test where the shaft speeds were held at constant values and the *compressor-inlet temperature* was varied in a ramp-and-hold fashion to investigate the system response to the changing CO_2 conditions at the compressor inlet. The second test, 65261-P, held the compressor-inlet temperature fixed. At the same time, the *turbine-compressor shaft speed* was commanded to change in a step-wise fashion to investigate the effect of the compressor recirculation flow fraction on the system flow rate and the generator power. This test is basically the equivalent to the turbine-bypass control analyzed previously with the PDC.

When the test data were provided to ANL, it was stated that the loop configuration for these new tests was exactly the same as for the test data analyzed previously. Because of that, no PDC input changes for the components and the loop data were implemented for this work. However, it was later discovered that some changes had taken place in the test facility. For example, the turbines were rebuilt several times between those tests. During one of these rebuilds, the generator turbine and shroud were replaced with Inconel components. Although the dimensions remained the same, this change may have altered clearances in the turbine. However, these changes are hard to quantify; therefore, for the PDC simulations, no changes in the turbine design input were implemented. Also, it was noticed by the IST operators that the generator turbine outlet density measurements began to "drift" sometime between these two tests, making the readings less reliable than they were in the previous tests.

SIMULATION OF THE VARYING COMPRESSOR-INLET CONDITIONS TEST

Test Data

The Test 64661 is the compressor-inlet temperature sensitivity test. In this test, the compressor inlet temperature was commanded to change to several levels by means of the water flow control. The precooler-outlet temperature during the test is shown in Figure 2 along with the target value for the PDC simulation. This target value is slightly different from the measured value since, similar to the previous analysis (Moisseytsev and Sienicki, 2016), the target temperature is calculated based on the measured pressure and density which are believed to provide a more accurate indication of the CO_2 properties than the temperature near the critical point.

The rest of the external inputs, including the shaft speeds, oil flow rate, and IHX-outlet temperature, are basically held constant (within the control bands) during the entire test. For the PDC simulation, the inlet water temperature is a given boundary condition for which the measured data is used. The water flow rate will be calculated by the PDC to maintain the target precooler-outlet temperature. The data for the test cover 5,500 seconds; it is the time for which the PDC simulation was run.

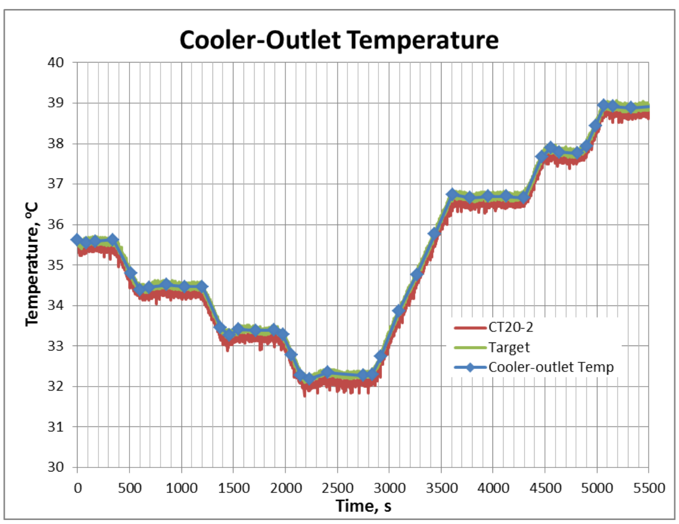

Figure 2. Precooler-Outlet Temperature in Test 64661.

Steady-State Results

The PDC analysis of the test started with simulation of the steady-state conditions prior to the test. The PDC results were generally close to the test data, with three exceptions.

First, the flow split between the two turbines was predicted to be 49%/51% for the generator/compressor turbines in the code, while in the test, it is 52%/48%. This is believed to be a result of the recent generator turbine modifications, which are difficult to characterize quantitatively and which are not currently implemented in the PDC.

Second, the code underpredicted the heat transfer rate in the cooler. Further investigation revealed that this difference is most likely due to assuming a pure counter-flow arrangement for the shell-and-tube heat exchanger. In the IST precooler, however, there is a mixture of a counter- and cross-flow pattern due to staggering baffle plates on the shell (CO_2) side. A modification of a heat exchanger subroutine to properly account for the cross-flow effect of the baffle plates is planned for future work. In this analysis, a multiplication factor for the heat transfer coefficient on the CO_2 side was introduced. Based on the steady-state results for this test, a factor of 1.4 is adopted, although the simulation of the other test in this paper suggests that this value is overestimated and a smaller correction factor could be more appropriate.

Lastly, the measured generator turbine outlet density was not predicted very well by the code. This, however, is believed to be a result of an instrumentation issue, rather than a code problem. Both the temperatures and pressures at the two turbine outlets were close to each other, which should result in close values for the densities, as the PDC calculated, but was not exhibited in the test data from the Coriolis flow meters. As discussed previously, this difference is believed to be a result of a "drift" in the density readings at this location.

Transient Results

The transient simulation of Test 64661 was carried out in two steps. In the first step, the turbine-compressor shaft speed control was not implemented. Instead, the shaft speed and the recirculation valve position versus time were provided as an input to the PDC. This is the same way that the PDC simulation was carried out in previous analysis (Moisseytsev and Sienicki, 2016). This step allows one to investigate the PDC prediction of the loop behavior without complications of the shaft speed control. The shaft speed control was added in the second step.

In the first simulation, the turbine-compressor shaft speed and the compressor recirculation valve position (Figure 3) were provided as inputs. Since this is the way the simulation was run previously, no code or input modifications (aside from actual numbers) were needed for these calculations. Note that as described in Moisseytsev and Sienicki, 2016, the PDC uses a simplified treatment of the valve in dynamic calculations. So, the actual valve position was corrected for the PDC input to account for these differences, as shown in Figure 3.

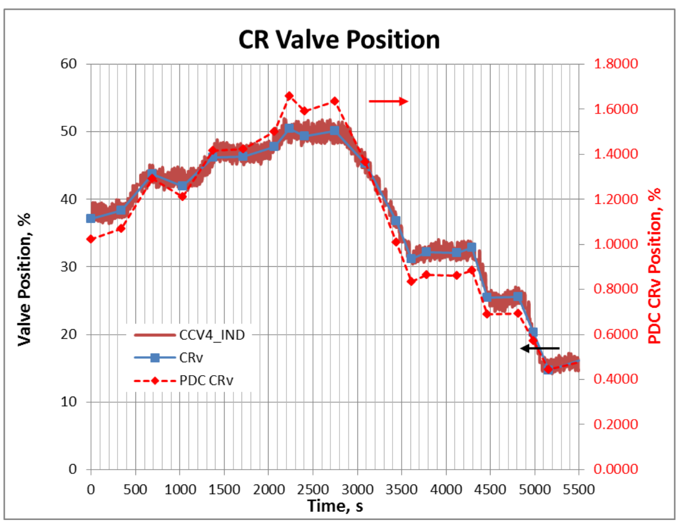

Figure 3. PDC Input for Compressor Recirculation Valve.

The first results of the transient simulation of Test 64661 showed a much slower response of the precooler-outlet temperature than was measured in the test and thus required much larger variations in the water flow rate. An attempt to improve the precooler performance by means of optimizing the water flow rate PID control coefficients was not successful. Eventually, the problem was traced to the wrong calculation in the PDC of the tube mass for a shell-and-tube heat exchanger for the transient equations of the tube wall temperature. After correcting the error, the dynamic response of the precooler was much closer to the test as the results below will show. This correction proved the value of this test for PDC validation. Since both the SNL RCBC loop and all the previous sCO_2 cycle control calculations for full-size plants used a Printed Circuit Heat Exchanger (PCHE) for the cooler, this error did not affect those results and could not be identified prior to this test simulation.

The transient simulation of Test 64661 with corrected heat exchanger tube mass showed much better agreement with the test data. With the exception of the generator turbine outlet temperature, density, and flow rate, all other results are within or very close to the uncertainty margins of the test data. The issues with the generator turbine performance prediction have already been discussed previously in the steady-state section.

Next, the full simulation of Test 64661 with active shaft speed control was initiated. To simulate the power-neutral operation of the turbine-compressor shaft with compressor recirculation valve control, the turbine bypass control logic, already implemented in the PDC, has been reactivated. The turbine bypass control has been used for the plant output control or, in case of asynchronous grid connection or disconnection from the grid, for the shaft speed control. Therefore, no changes to the PDC control logic were required to simulate Test 64661, except for assigning the compressor recirculation valve to effectively be a "turbine bypass" valve for this control. The turbine-compressor shaft is simulated as being disconnected from the grid to simulate zero generator control in the test.

Still, some changes were necessary to the way the shaft power is calculated. The original version of the PDC does not include windage loss calculations in the shaft. It is simply assumed that those losses are small for a commercial-sized plant and contribute a given fraction (like 1%) of the turbine or compressor power. For small-scale loops like the IST, though, the windage losses were found to be large, in relative terms, and could not be assumed to be a fixed and small fraction of a total power. To more accurately account for these windage losses, the following approach was implemented. During the steady-state initialization, the windage losses are calculated as the difference between the turbine power and the compressor power, such that the net shaft power is always zero at steady state. Then, this loss is assumed to scale with the third power of the shaft speed according to the windage loss equation from Vranick, 1968. This scaling, however, is not important for Test 64661 where the shaft speed is maintained close to the steady-state conditions, but will be important in the next test where the shaft speed is changing.

The compressor recirculation valve control in the following calculations will use the same PI coefficients as in the actual IST control. Since the PDC valve position (open fraction) is different from the actual IST valve position, the same scaling implemented previously (see Figure 3) was also applied to the PI coefficients for the PDC input. The manual control of the CR valve is disabled for the automatic control. The code will calculate the required valve position, which will again be converted to the equivalent IST valve position for comparison with the test.

The results of the full Test 64661 simulation are shown in Figure 4 for the shaft speed and the compressor recirculation valve position and in Figure 5 for some other measurements. The calculated shaft speed in Figure 4 confirms that the modeling of the IST shaft speed dynamics and control is a close representation of the test setup. Smaller shaft speed variation in the PDC can be attributed to some effects which are not included in the model, such as controller dead band and delays in electronics. Figure 4 also shows that the PDC predicts the compressor recirculation valve position very closely relative to the test data.

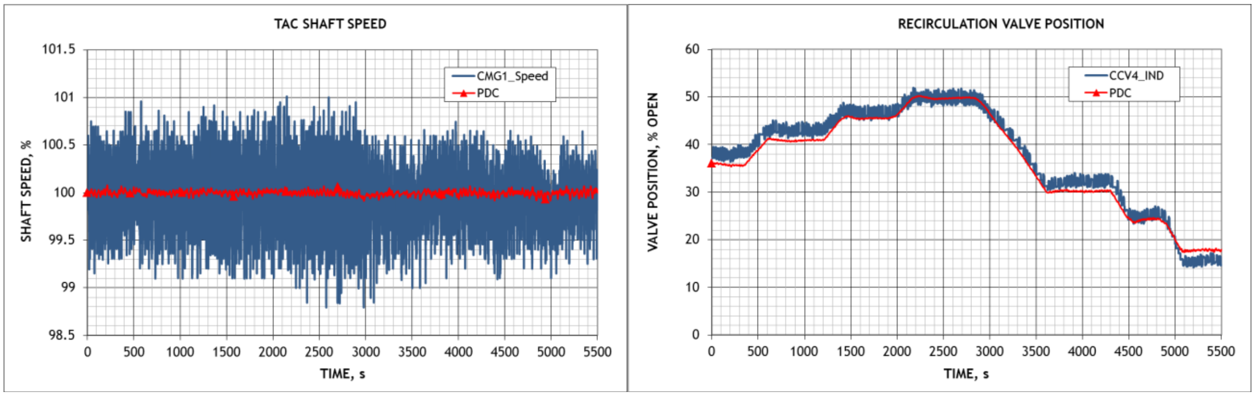

Figure 4. Shaft Speed and CR Valve Position Results.

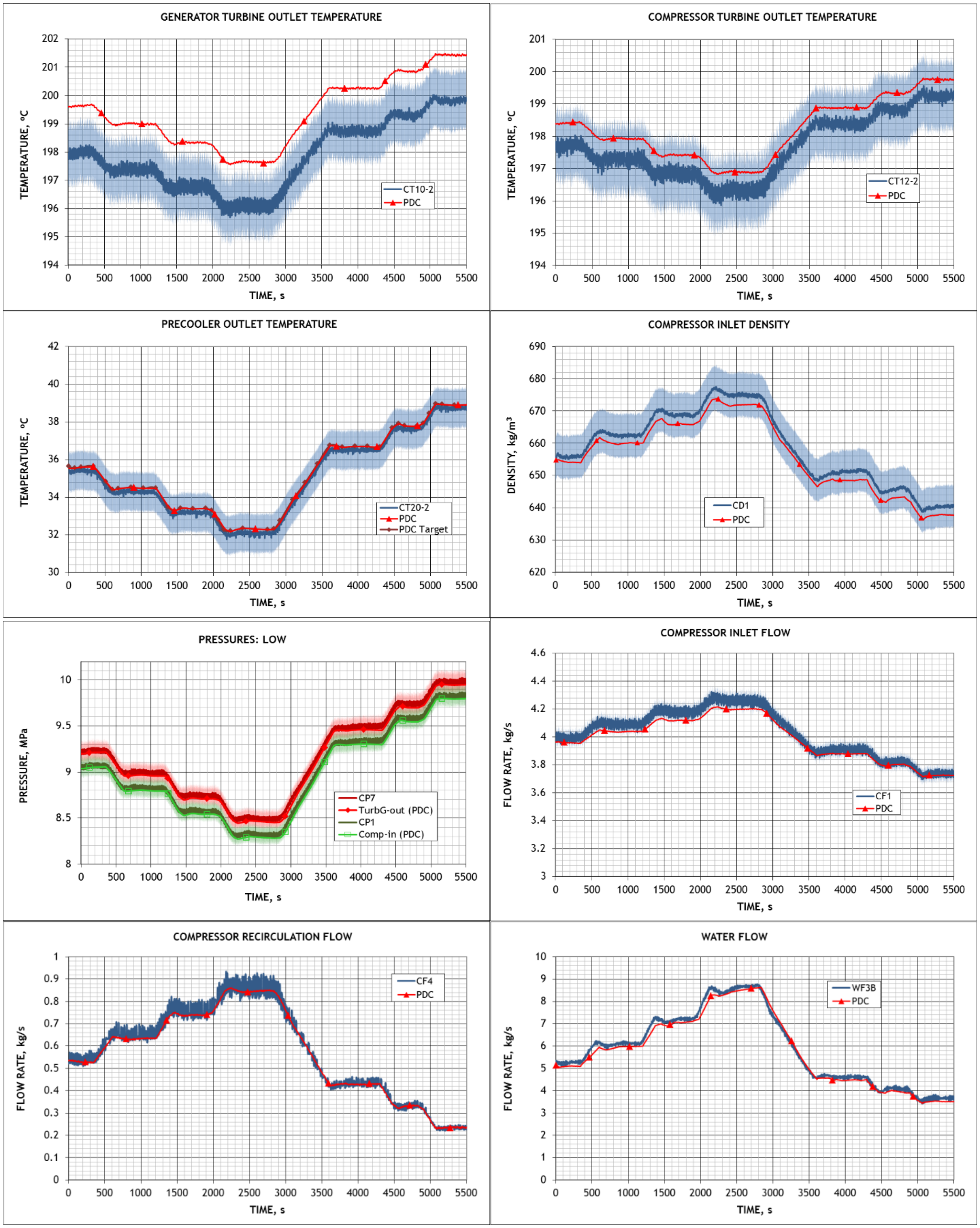

Figure 5. Results of the Test 64661 Full Simulation.

The PDC results, shown in Figure 5, are close to the experiment data and to the previous results with given shaft speed: all of the PDC results (including those not shown in Figure 5) show good agreement with the test, with the exception of the generator turbine performance. Precooler outlet temperature matches the target value very accurately thus verifying the precooler water control setup. Water flow rate, which is close to the measured value, confirms the water flow rate control and also the precooler thermal inertia and performance (although, with a multiplication factor on the CO_2 side heat transfer coefficient). Agreement in the compressor recirculation flow rate confirms the correct mapping between the actual valve position in the IST loop and the valve open fraction in the PDC calculations.

SIMULATION OF THE VARYING POWER LEVEL TEST

Test Data

For Test 65261-P, the generator turbine output was varied in a step fashion. The generator output was changed by a commanded change in the turbine-compressor shaft speed. Figure 6 shows the recorded shaft speed ("Nr_TC" line) as well as the commanded speed ("RPM" line), both normalized to the conditions at the start of the transient. As in the previous test, the turbine-compressor shaft speed is automatically controlled by the compressor recirculation (CR) valve position. When the turbine-compressor shaft speed needs to be increased, the CR valve closes. Higher compressor speed results in higher flow rate in the loop. In addition, less compressor recirculation flow means that more flow is sent to the turbines. Both these factors mean that more flow is going through the generator turbine increasing its power output. Overall, this test is equivalent to the turbine bypass control action considered previously in the PDC analyses of full-scale systems with the exception of asynchronous (free-changing speed) operation of the compressor. The test was run for 6,000 seconds, of which the first 1,400 seconds were used to obtain steady-state conditions. So, the PDC transient simulation will be run for 4,600 seconds of the test.

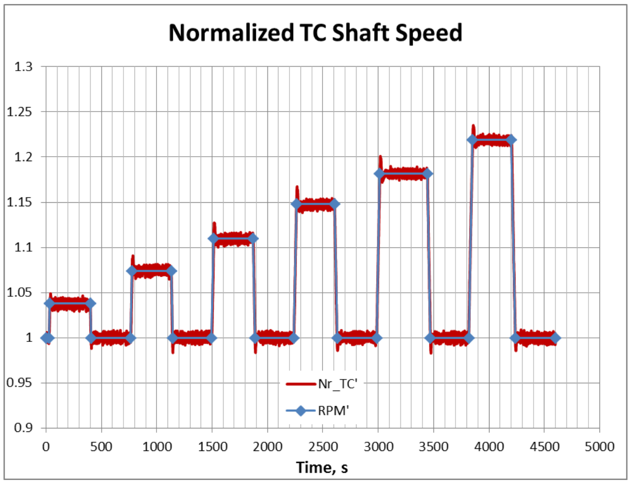

Figure 6. Turbine-Compressor Shaft Speed in Test 65261-P.

The rest of the external input for this test was similar to the test described previously. The water precooler-inlet temperature, oil flow rate, and CO_2 IHX-outlet temperature are all held constant during the test. In this test, though, target CO_2 precooler-outlet temperature was also held constant during the test. As Figure 7 shows, the PDC input for the target precooler-outlet temperature (blue line) is different from the measured values to be consistent with the precooler-outlet density, as in the previous calculations. The generator shaft speed was commanded to a constant speed as in the previous test. However, as Figure 7 demonstrates, it experienced some fluctuations at the points of the turbine-compressor shaft speed changes. Unlike the turbine-compressor shaft speed, the generator turbine shaft speed was actively controlled by the shaft speed controller which is not modeled in the PDC. Therefore, as in all previous calculations, the generator turbine shaft will be simulated in the PDC in a synchronous mode with a given shaft speed versus time. That PDC input was selected to include the spikes in the shaft speed as shown in Figure 7.

Aside the data in Figure 7 and the conditions for the steady-state calculations, the PDC simulation of this test was done in exactly the same way as the previous test. The only differences from the previous simulation are the changing turbine-compressor shaft speed (rather than 100% in Test 64661) and a fixed precooler-outlet target temperature. The rest of the PDC input, including the input for each individual component and the loop setup, was preserved from the previous simulation. Also, the PDC control logic and the control PID inputs are exactly the same as in the previous test.

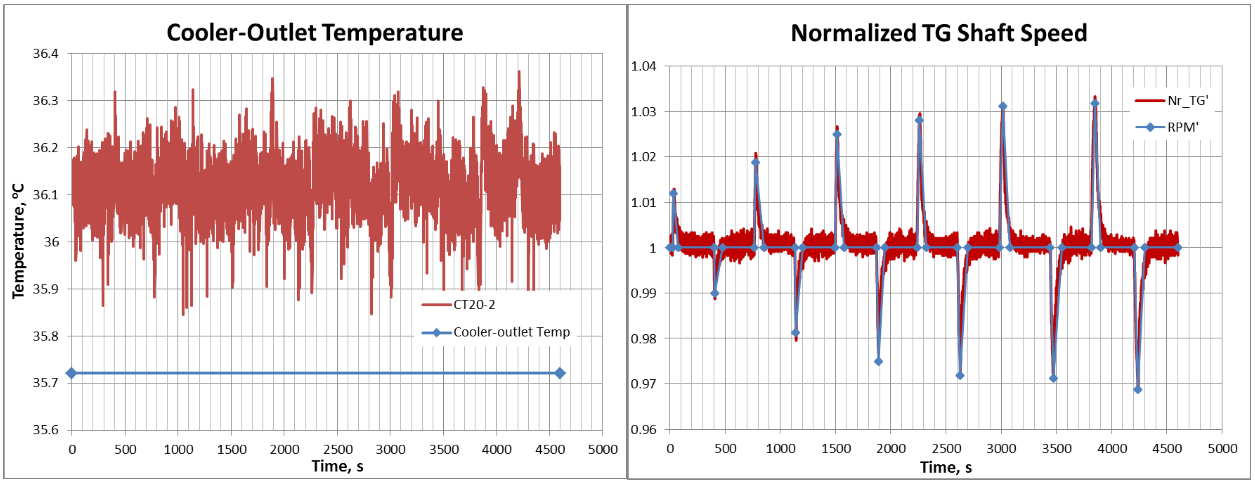

Figure 7. Other External Input for Test 65261-P.

Steady-State Results

The results of the steady-state PDC calculations of the conditions at the beginning of the test run showed good agreement with the test data everywhere, except for the generator turbine performance and the flow split between the turbines. This split is 45%/55% for generator/compressor turbines in the code, compared to the 50%/50% split in the test. Therefore, the same issue with the generator turbine performance remains in this simulation. Also, the correction factor for the heat transfer coefficient on the CO_2 side of the precooler appears to be too large for this test since the PDC results require a lower water flow rate than the measured value. Both of these issues, though, are minor and are not expected to affect transient simulation significantly.

Transient Results

Although the simulation of the previous test showed that the code predicted the generator shaft speed change very accurately with an active control, the transient simulations of the Test 65261-P were still done in two steps, similar to the previous simulation. In the first step, the turbine-compressor shaft was set to operate in a synchronous mode at the speed defined in Figure 6 with a given CR valve position. The results of this simulation show that, similar to the previous test and the steady-state results, the majority of the test data is predicted within the uncertainty margins, with the noticeable exception of the generator turbine performance, which affects the outlet temperature and the flow split between the turbines.

In the second step, the shaft speed control by means of the CR valve was reinstated. The turbine-compressor shaft mode was switched to “disconnected from the grid” to invoke the shaft speed dynamic equations. The control setup is the same as in the previous test simulation, with the only difference being the target shaft speeds from Figure 6. The results of the PDC prediction of the shaft speed and the corresponding CR valve position are shown in Figure 8 with the rest of the transient results shown in Figure 9. Even though the target shaft speed was maintained, the code predicted progressively lower CR valve positions as the shaft speed increases. By about 4,000 seconds, the code predicts that the CR valve would be almost closed (while the test data shows it about 25% open).

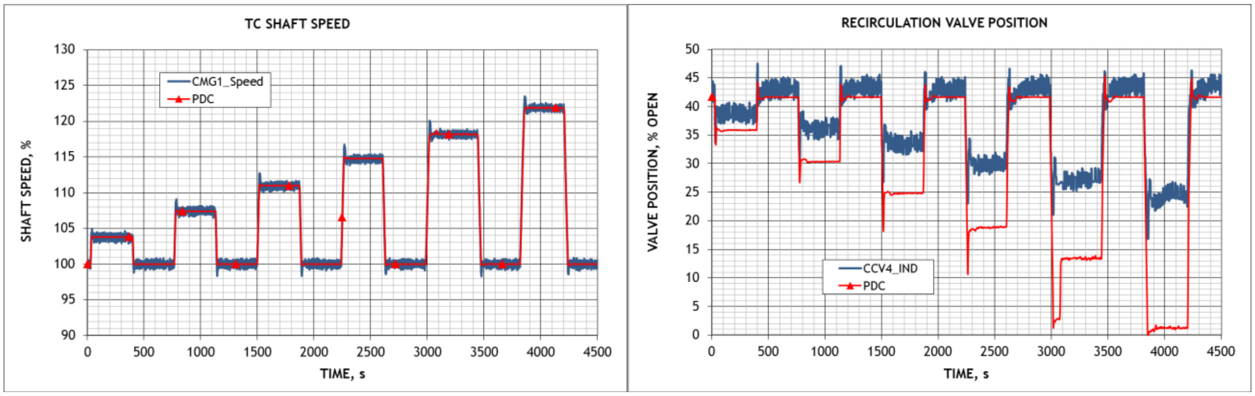

Figure 8. Shaft Speed and CR Valve Position Results for Test 65261-P.

The analysis of the test simulation results in Figure 9 in comparison with the results from step one with given shaft speed suggests that the issue of underpredicting the CR valve position is not originating in the valve modeling itself. It is more likely that the issue is with the windage loss modeling. As described above, the current modeling is based on scaling of the windage losses with the third power of shaft speed. That scaling, though, assumes that the fluid density and clearances in the shaft cavity remain constant. The Test 65261-P description stated that the shaft cavity pressure was maintained at 200 psia during the test. It is possible, however, that the shaft cavity temperature changes during the test resulting in a changing density. This possibility is confirmed in another note in the test description, suggesting that the CO_2 mass may be shifting between the cavity and the main loop, which is possible only if the cavity density is changing. If this hypothesis is correct, then the changing cavity density will result in an inaccuracy in the windage loss prediction and, therefore, an incorrect shaft power balance for the same speed. To compensate for this, the code demands more flow through the compressor and, consequently, less flow through the CR valve. This assumption is also confirmed by much better agreement with the test data for the CR flow rate in step one, where the shaft power balance was not calculated. To resolve this issue, a better prediction of the shaft windage losses, not just scaling with shaft speed, would be required, which, in turn, might require more detailed modeling of the shaft cavity leakage collection system.

The rest of the PDC results in Figure 9 are consistent with the previous results in step one. The majority of the test data is predicted within the uncertainty margins with the noticeable exception of the generator turbine performance, which affects the outlet temperature and the flow split between the turbines. The water flow rate is underpredicted during the entire transient confirming that the heat transfer coefficient correction factor is too large. Most of the PDC results in Figure 9 show significantly larger variations at the points of the shaft speed changes than the test data. These variations are a result of the generator turbine shaft speed spikes in Figure 7. The difference with the test data is believed to be due to a combination of an assumption of instantaneous turbomachinery response on the modeling side and sensor delays on the experiment side. In most cases, though, those spikes occur within the measurement uncertainty range.

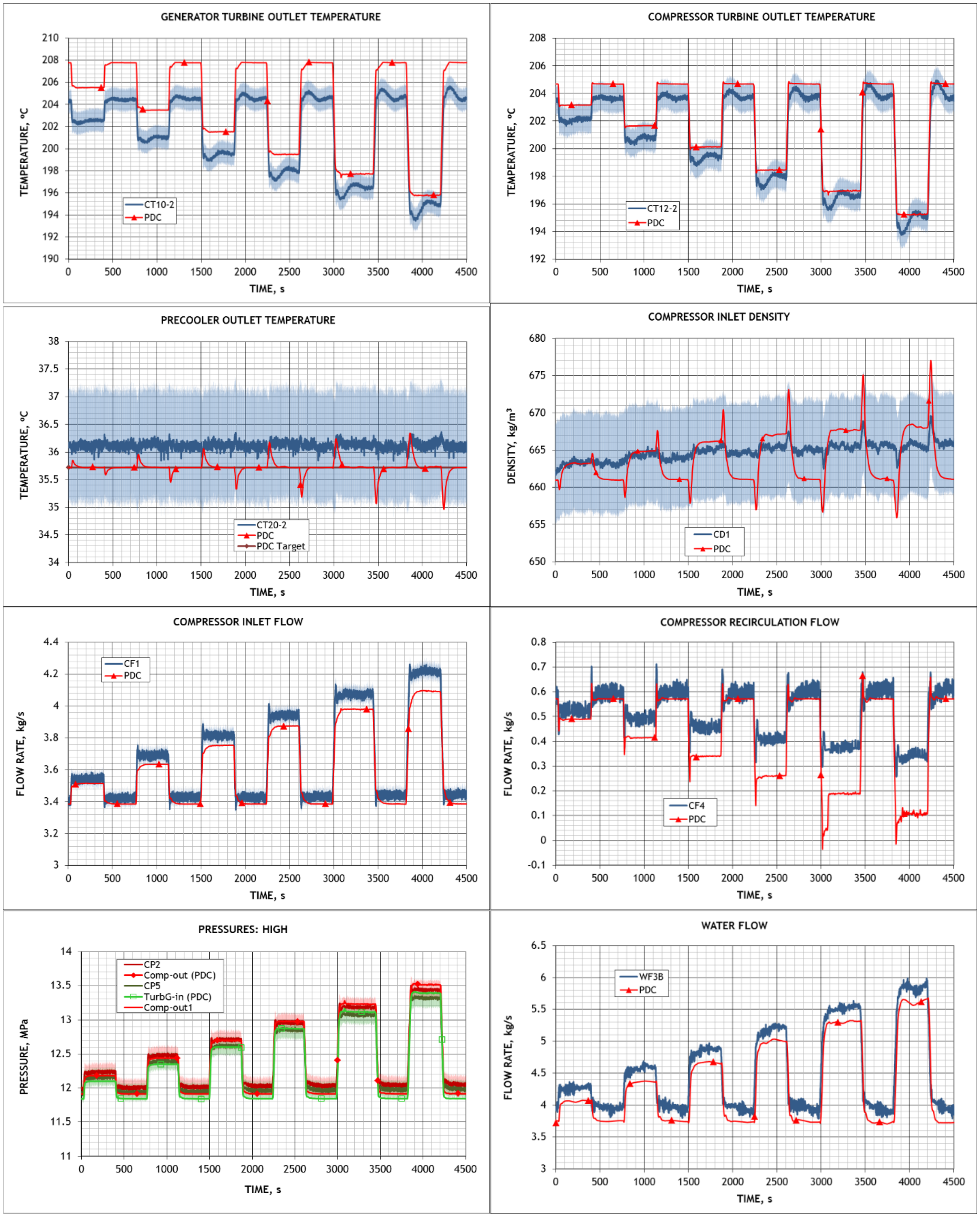

Figure 9. Results of the Test 65261-P Full Simulation.

SUMMARY

Validation of the PDC) developed at ANL for steady-state and transient analysis of sCO_2 systems has been continued with the new test data from the NNL's IST. Data from two more runs were provided to ANL. The common feature of these tests is the power-neutral operation of turbine-compressor shaft, where no external power through the alternator was provided during the tests. Instead, the shaft speed was allowed to change dictated by the power balance between the turbine, the compressor, and power losses in the shaft. The shaft speed was indirectly controlled by means of the compressor recirculation (CR) valve to divert some flow from the compressor outlet back to compressor inlet, an action similar to both compressor surge control and a turbine bypass action. Of the two tests, in Test 64661 the CO_2 temperature at the compressor inlet was commanded to change in a ramp-and-hold fashion, while the speed of both the turbine-compressor and turbine-generator shafts was maintained constant. In the second test, 65261-P, the compressor-inlet temperature was maintained at the same level, but the turbine-compressor target shaft speed was commanded to change in a step fashion.

The new test data turned out to be important for code validation for several reasons. First, the power-neutral operation of the shaft speed allows validation of the shaft dynamics equations in asynchronous mode, when the shaft is disconnected from the grid. Second, the shaft speed control with the CR valve not only allows for testing the code control logic itself, but it also serves as a good test for validation of both the compressor surge control and the turbine bypass control actions, since the effect of the CR action on the loop conditions is similar to both these controls. Third, the varying compressor-inlet temperature change test allows validation of the transient response of the precooler, a shell-and-tube heat exchanger.

It was originally stated that the new test data was obtained in the same loop configuration and the same equipment as the previous data analyzed with the PDC, such that no PDC input modification was required, with the exception of the external control input defining the tests. However, later discussion with NNL personnel revealed that the turbomachinery equipment was rebuilt several times, including a material change at some point. Although those modifications preserved the design dimensions, they could still influence the turbine and compressor performance, for example, through the changes in clearances. Because those effects were not yet characterized quantitatively for the IST loop, the PDC calculations presented in this paper were still carried out with the same input as before.

The steady-state simulation of both test initial conditions showed good agreement with the test data, with two noticeable exceptions. The turbine performance is not predicted as well as it was in the previous IST simulation, resulting in the prediction of a somewhat different flow split between the two turbines. This is believed to be the result of not including the recent turbomachinery modifications in the model. Second, the precooler performance was underpredicted by the code, resulting in a lower water flow rate needed to achieve the same precooler-outlet conditions. This issue is temporarily dealt with by the introduction of a multiplication factor for the heat transfer coefficient on the CO_2 (shell) side of the heat exchanger.

The first transient simulation of the compressor-inlet temperature variation Test 64661 showed a much slower response of the precooler than the recorded test data. Further investigation revealed an error in calculating the heat exchanger tube mass for the PDC dynamic equations that resulted in a slower change in the tube wall temperature. When the error in the tube mass was corrected, the response of the precooler was much closer to the test data.

The transient calculations for both tests were carried out in two steps. The first step was done in the same fashion as in previous analysis, where the CR valve position and the turbine-compressor shaft speed were specified through the PDC input based on the test values. This step proved to be useful in validating other system characteristics.

In the second step of a transient simulation, the turbine-compressor shaft dynamic equations were invoked by specifying that the shaft is disconnected from the grid. Also, the CR valve control was used to control the shaft speed. Both of these features already existed in the PDC, so no code changes were needed for the shaft speed equations and the control logic. The IST CR valve control PI coefficients were directly implemented in the PDC input. The only change introduced into the PDC to simulate the IST components was the revised calculation of shaft friction (windage) loss. Rather than providing the friction loss power fraction in the input, the loss is calculated based on the shaft balance at steady-state conditions and is assumed to be scaled with the third power of the shaft speed in the transient.

The full simulation of the compressor-inlet temperature variation Test 64661 showed close agreement with the test data, including the CR valve position and the shaft speed. The only noticeable differences from the test data were the same issues observed in the steady-state calculations, such as turbine performance and the flow split between the turbines.

The full simulation of the turbine-compressor speed variation Test 65261-P with shaft speed control showed a greater difference with the test data later in the transient than the other test. Further analysis of the results revealed that the difference is most likely due to scaling the shaft windage losses only with the shaft speed and ignoring the dependency on the fluid density in the shaft cavity.

Based on the results of the steady state and transient calculations of Tests 64661 and 65216-P, the following possible improvements for the PDC simulation of the IST are identified:

- More accurate turbine performance prediction to reflect recent changes to the turbomachinery components;
- More accurate modeling of the precooler with a mixed cross- and counter-flow pattern on the shell side with baffle plates; and
- Better prediction of turbomachinery windage losses at changing conditions, which might require more detailed simulation of the shaft leakage collection system.

ACKNOWLEDGEMENTS

Argonne National Laboratory's work was supported by the U. S. Department of Energy Advanced Reactor Technologies (ART) Program under Prime Contract No. DE-AC02-06CH11357 between the U.S. Department of Energy and UChicago Argonne, LLC. The work presented here was carried out under the Energy Conversion Technology area of the ART Program. The authors are grateful to Gary Rochau (SNL), the Technical Area Lead, Bob Hill (ANL/NE), the National Technical Director, as well as Brian Robinson (U.S. DOE), Headquarters Program Manager for the project. The authors are especially thankful to Patrick Fourspring and Eric Clementoni of Naval Nuclear Laboratory for providing new IST test data and their continuous support during the analysis presented in this paper.

REFERENCES

Clementoni, E.M. and Cox, T.L., 2014a, "Steady-State Power Operation of a Supercritical Carbon Dioxide Brayton Cycle," *4th International Symposium - Supercritical CO_2 Power Cycles*, Pittsburgh, Pennsylvania, September 9-10.

Clementoni, E.M. and Cox, T.L., 2014b, "Practical Aspects of Supercritical Carbon Dioxide Brayton System Testing," *4th International Symposium - Supercritical CO_2 Power Cycles*, Pittsburgh, Pennsylvania, September 9-10.

Clementoni, E.M., Cox, T.L., King, M. A., and Rahner, K.D., 2017, "Transient Power Operation of a Supercritical Carbon Dioxide Brayton Cycle," *GT2017-63056, Proceedings of ASME Turbo Expo 2017, GT2017, Charlotte, NC, USA, June 26-30*.

Kimball, K.J., 2014, "Supercritical Carbon Dioxide Power Cycle Development Overview," Presentation at *4th International Symposium - Supercritical CO_2 Power Cycles*, Pittsburgh, Pennsylvania, September 9-10.

Moisseytsev, A. and Sienicki, J. J., 2006, "Development of a Plant Dynamics Computer Code for Analysis of a Supercritical Carbon Dioxide Brayton Cycle Energy Converter Coupled to a Natural Circulation Lead-Cooled Fast Reactor," ANL-06/27, Argonne National Laboratory.

Moisseytsev, A. and Sienicki, J. J., 2015, "Lessons Learned and Improvements in ANL Plant Dynamics Code Simulation of Experimental S-CO_2 Loops," *PowerEnergy2015-49182, Proceedings of ASME Power & Energy 2015,* San Diego, CA, June 28-July 2.

Moisseytsev, A. and Sienicki, J. J., 2016, "Simulation of S-CO_2 Integrated System Test with ANL Plant Dynamics Code," *5th International Symposium - Supercritical CO_2 Power Cycles,* San Antonio, TX, March 28-31.

Rahner, K.D., 2014, "S-CO_2 Brayton Loop Transient Modeling," Presentation at *4th International Symposium - Supercritical CO_2 Power Cycles*, Pittsburgh, Pennsylvania, September 9-10.

Vranick, J. E., 1968, "Prediction of Windage Power Loss in Alternators," National Aeronautics and Space Administration, Washington, D.C., NASA Technical Note D-4849.