Simulation of IST Turbomachinery Power-Neutral Tests with the ANL Plant Dynamics Code

Nuclear Engineering Division

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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 (sCO₂) systems has been continued with new test data from the Naval Nuclear Laboratory (operated by Bechtel Marine Propulsion Corporation) Integrated System Test (IST). Although data from three runs were provided to ANL, only two of the data sets were analyzed and described in this report. The common feature of these tests is the power-neutral operation of the 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 the power losses in the shaft.

The new test data turned out to be important for code validation for several reasons. First, the power-neutral operation of the shaft 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 compressor recirculation (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 for both of 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.

The first transient simulation of the compressor-inlet temperature variation Test 64661 showed a much slower calculated response of the precooler in the calculations than the 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 than measured.

The transient calculations for both tests were done in two steps. The first step was done in the same fashion as the FY15 IST analysis, where the CR valve position and the turbine-compressor shaft speed were specified through the PDC input based on the test values. On the second step, the turbine-compressor shaft dynamics equations were invoked by specifying that the shaft is disconnected from the grid. In addition, the CR valve control was used to control the shaft speed, based on the turbine bypass control logic already implemented in the PDC. For the shaft power balance, the friction (windage) loss is calculated based on the shaft balance at the steady-state conditions and is assumed to be scaled to the third power of shaft speed in the transient.

Both the steady-state and transient simulations of both tests showed good agreement with the test data. The only significant difference was the turbine performance, which was 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 flow split difference is believed to be the result of not addressing the recent turbine modifications in the model.

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

Based on the results of steady state and transient calculations of the Tests 64661 and 65216-P, several areas of future improvements for the PDC simulation of the IST are identified.

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The difference between those two tests is the rate of change in the turbine-compressor shaft speed, which was changed more aggressively in the 65261-U test. Because these two tests are similar and due to time constraints, only one test, 65261-P was analyzed with the PDC for this report.

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 in FY15. Because of that, no PDC input changes for the components and the loop data were implemented for this work. However, later discussion with the Naval Nuclear Laboratory personnel revealed that some changes took 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.

2 Simulation of the IST Varying Compressor-Inlet Conditions Test

2.1 Test Data

The Test 64661 (full name is P1-S3-64661-F) 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 summary of the test, as provided by Naval Nuclear Laboratory, is presented on the next page. 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 [1], the target temperature is calculated based on the measured pressure and density which are believed to provide a more accurate indication of the CO₂ properties than the temperature near the critical point. Because the density is measured in the IST closer to the precooler outlet than to the compressor inlet, the precooler outlet temperature is used in the PDC control.

Summary for Integrated System Test (IST) Data Set**Data File Name:** P1-S3-64661-F

Summary of Test: This test was performed to determine the effect of compressor inlet temperature on system performance with a fixed turbine-compressor speed. Compressor inlet temperature was varied between 90°F and 102°F in 2°F increments by adjusting water flow through the precoolers. Steady-state operating conditions were achieved at each operating condition.

Initial Conditions:

- Turbine-Compressor speed setpoint of 48,675 rpm; speed controlled via PI feedback control of CCV4
- Turbine-Compressor motor-generator has no applied motor or generator current
- Turbine-Generator speed setpoint of 60,000 rpm; speed controlled via load regulation by motor-generator controller
- CCV1, CCV2, and CCV5 set to 100% open
- CCV3 set to 0% open
- IHX outlet temperature (CT7) set to 440°F via automatic control of heater duty cycle
- Compressor inlet temperature (CT1) set to 96°F via automatic control of WCV3 to regulate precoolers water flow
- Precoolers water inlet temperature set to 75°F via automatic control of WCV1
- Oil system flow rate set to 50 lbm/s via automatic control of OPUMP1 speed
- Turbomachinery cavity pressure set to 200 psia via automatic control of RCOMP2 stroke speed
- Initial conditions were held for greater than 10 minutes prior to start of data file to obtain steady-state operating condition

Control Actions:

Timestamp	Action
370 sec	Compressor inlet temperature setpoint set to 94°F at a rate of 0.01°F/sec
1190 sec	Compressor inlet temperature setpoint set to 92°F at a rate of 0.01°F/sec
1955 sec	Compressor inlet temperature setpoint set to 90°F at a rate of 0.01°F/sec
2825 sec	Compressor inlet temperature setpoint set to 98°F at a rate of 0.01°F/sec
4300 sec	Compressor inlet temperature setpoint set to 100°F at a rate of 0.01°F/sec
4875 sec	Compressor inlet temperature setpoint set to 102°F at a rate of 0.01°F/sec

General Notes:

- Main loop temperatures (CT#), pressures (CP#), and differential pressures (CDP#) have been adjusted for data acquisition system bias and zero flow corrections. As a result of these adjustments, the measured compressor inlet density and the density calculated using temperature and pressure via REFPROP differ by less than 1% for all data points.
- Total system CO₂ mass was held constant during this test. Additionally, the speeds of both turbomachines were held constant resulting in only minimal changes in turbomachinery cavity conditions which should result in a nearly constant Brayton loop mass.
- Turbine-generator AC output power has been found via independent instrumentation to be ~25% greater than DC power calculated using DC current and voltage.

Figure 2. Precooler-Outlet Temperature in Test 64661.

Figure 3 shows more of the Test 64661 data used to define the transient in the PDC, along with the PDC input fits. The shaft speeds, oil flow rate, and IHX-outlet temperature are basically held constant (within the control bands) during the entire test. The inlet water temperature is a result of how the IST water cooling loop works at changing conditions; for the PDC simulation, the water temperature is a given boundary condition. 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 3. Test 64661 Data.

2.2 Steady-State Results

The results of the steady-state PDC calculations of the conditions prior to the test run are shown in Figure 4 in comparison to the test data. The test data were averaged over the first 100 seconds. The PDC results are generally close to the test data, with three exceptions:

- First, the flow split between the two turbines is 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.
- The code under predicts the heat transfer rate in the cooler. Last year, that difference was dealt with by reducing the water inlet temperature by 1°C within the measurement uncertainty. 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 (CO₂) side. A modification of a heat exchanger subroutine to properly account for the cross-flow effect of the baffle plates is planned for the future work. In this analysis, a multiplication factor for the heat transfer coefficient on the CO₂ side was introduced. Based on the results for the Test 64661, a factor of 1.4 is adopted, although the simulation of the other test in this report suggests that this value is overestimated and a smaller correction factor could be more appropriate.
- The measured generator turbine outlet density is not predicted very well by the code. This, however, is believed to be a result of an instrumentation issue, rather than a code problem. As the data in Figure 4 show both the temperatures and pressures at the two turbine outlets are close to each other, which should result in close values for the densities around 112 kg/m³, as the PDC calculates. However, the reading from the generator turbine outlet density sensor (Coriolis flow meter) is 117 kg/m³. As discussed previously, this difference is believed to be a result of a “drift” in the density readings at this location.

Figure 4. Steady-State Conditions at the Start of Test 64661 (top) Compared to the PDC Results (bottom).

2.3 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 FY15 [1]. 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.

2.3.1 Results with Given Shaft Speed

In this simulation, the turbine-compressor shaft speed (Figure 3) and the compressor recirculation valve position (Figure 5) were provided as inputs. Since this is the way the simulation was run in FY15, no code or input modifications (aside from actual numbers) were needed for these calculations. Note that as described in the FY15 report [1], 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 5.

Figure 5. 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, as demonstrated in Figure 6. 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 error¹, the dynamic response of the precooler was much closer to the test as the results below will show.

¹ Both the SNL loop and all the previous sCO₂ cycle control calculations for full-size plants used Printed Circuit Heat Exchanger (PCHE) for the cooler, so this error did not affect those results.

Figure 6. First Transient Results of the Test 64661 Simulation.

Due to a significant change in the precoolers transient behavior after the tube mass calculations were corrected, the water flow rate control coefficients have to be re-optimized to achieve a transient response close to the actual performance. Note that the actual water flow rate control implemented in the IST could not be directly simulated in the PDC since in the test the control is applied by a throttling valve, while in the PDC the controlling variable is the pump head. Besides, the PDC uses a much more simplified treatment of the water loop that does not simulate all of the IST water loop equipment (with the exception of the water pump) outside the precoolers.

The results of the transient calculations in comparison with the test data are shown in Figure 7. The test data are shown in Figure 7 with the measurement uncertainty margins defined in the FY15 report [1]. 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.

For this simulation, the results of particular interest are:

- *Precooler outlet temperature*, which matches the target value very accurately thus verifying the precooler water control setup,
- *Water flow rate*, which is close to the measured value, which again confirms the water flow rate control and also the precooler performance (although, with a multiplication factor on the CO₂ side heat transfer coefficient)
- *Compressor recirculation flow rate*, which confirms the correct mapping between the actual valve position in the IST loop and the valve open fraction in the PDC calculations (see Figure 5).

In addition, the good prediction of all other parameters, including pressures and flow rates, confirms that the PDC is able to simulate the test conditions very accurately.

Figure 7. Results for Test 64661 with Given Shaft Speed.

Figure 7. Results for Test 64661 with Given Shaft Speed. (Continued)

Figure 7. Results for Test 64661 with Given Shaft Speed. (Continued)

Figure 7. Results for Test 64661 with Given Shaft Speed. (Continued)

Figure 7. Results for Test 64661 with Given Shaft Speed. (Continued)

Figure 7. Results for Test 64661 with Given Shaft Speed. (Continued)

2.3.2 Transient Simulation with Shaft Speed Control

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 the Test 64661, except for assigning the compressor recirculation valve to effectively be a “turbine bypass” valve for this control. The turbine-compressor shaft will be simulated as being disconnected from the grid to simulate zero generator control in the test.

However, 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 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 Reference [2]. This scaling, however, is not important for the 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 shaft speed dynamic equation already implemented in the PDC is used without any modification. The total shaft moment of inertia is a PDC input and was provided by the Naval Nuclear Laboratory to be equal to $5.3 \text{ lb}\cdot\text{in}^2$ ($1.55\cdot 10^{-3} \text{ kg}\cdot\text{m}^2$). In the grid disconnection mode, the shaft speed input becomes the target speed value, while the actual speed is calculated by the code.

The compressor recirculation valve control in the following calculations will use the same PI coefficients as in the actual IST control. But, since the PDC valve position (open fraction) is different from the actual IST valve position, the same scaling implemented previously (see Figure 5) 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.

With implementation of the shaft speed control, the time step for the dynamic calculations decreased from 10^{-3} - 10^{-2} seconds to 10^{-6} - 10^{-4} seconds due to the very low moment of inertia of the small IST turbine-compressor unit and how fast the shaft speed can change (the test data in Figure 3 also demonstrates that the shaft speed changes very fast in the test). Accordingly, the PDC calculation times increased from a few hours to several days for the 5,500 seconds duration of the test.

The results of the full Test 64661 simulation are shown in Figure 8 for the shaft speed and the compressor recirculation valve position and in Figure 9 for all other measurements. The shaft

speed results in Figure 8 demonstrate why the time step had to be smaller in this simulation, - the shaft speed changes very fast, which is a result of the low moment of inertia of the IST turbine-compressor shaft. The calculated variation in shaft speed is similar, although smaller, than the actual measured value. This result also 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 8 also shows that the PDC predicts the compressor recirculation valve position very closely to the test data. Again, the valve position in Figure 8 for the PDC results is the estimated IST valve position, which was obtained using a reverse polynomial fit used previously (see Section 2.3.1).

Figure 8. Shaft Speed and CR Valve Position Results.

The rest of the PDC results, shown in Figure 9, are close to the experimental data and to the previous results in Figure 7. Due to oscillations in the turbine-compressor shaft speed, most of the PDC results in Figure 9 show some oscillations. However, again, those oscillations are smaller

than in the test data. Similar to Figure 7, all of the PDC results in Figure 9 show good agreement with the test, with the exception of the generator turbine performance.

Figure 9. The Rest of the Test 64661 Full Simulation.

Figure 9. The Rest of the Test 64661 Full Simulation. (Continued)

Figure 9. The Rest of the Test 64661 Full Simulation. (Continued)

Figure 9. The Rest of the Test 64661 Full Simulation. (Continued)

Figure 9. The Rest of the Test 64661 Full Simulation. (Continued)

Figure 9. The Rest of the Test 64661 Full Simulation. (Continued)

3 Simulation of the IST Varying Power Level Test

3.1 Test Data

For the Test 65261-P (full name is P1-S4-65261-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. 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 summary of the test, as provided by Naval Nuclear Laboratory, is presented on the next page. The transient is defined by the changing (commanded) turbine-compressor shaft speed shown in Figure 10. 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 10. Turbine-Compressor Shaft Speed in Test 65261-P.

Summary for Integrated System Test (IST) Data Set

Data File Name: P1-S4-65261-P

Summary of Test: This test was performed to determine system response to power level transients between different power levels with the turbine-compressor speed changing at 250 rpm/sec (equivalent to ~0.8%/sec or ~0.8 kW/sec). Power level was changed between 5% and a range of higher powers from 10% to 35% in 5% increments. Steady-state operating conditions were achieved at each power level.

Initial Conditions:

- Turbine-Compressor speed setpoint of 42,426 rpm; speed controlled via PI feedback control of CCV4
- Turbine-Compressor motor-generator has no applied motor or generator current
- Turbine-Generator speed setpoint of 60,000 rpm; speed controlled via load regulation by motor-generator controller
- CCV1, CCV2, and CCV5 set to 100% open
- CCV3 set to 0% open
- IHX outlet temperature (CT7) set to 440°F via automatic control of heater duty cycle
- Compressor inlet temperature (CT1) set to 96°F via automatic control of WCV3 to regulate precooler water flow
- Precooler water inlet temperature set to 75°F via automatic control of WCV1
- Oil system flow rate set to 50 lbm/s via automatic control of OPUMP1 speed
- Turbomachinery cavity pressure set to 200 psia via automatic control of RCOMP2 stroke speed
- Initial conditions were quasi-steady-state with no control actions taken during the first 14 minutes of the data file to obtain steady-state operating condition

Method of Control:

The system is operating in T/H LEAD control mode where the compressor speed setpoint is determined via lookup table from the operator input generator load setpoint. Generator load setpoint is in % of full power and directly corresponds to CMG2 DC Current. The speed setpoint for each power level was determined via testing and as a result differences in operating conditions between tests due to different system masses, increased seal leakage, and degrading cavity pump down compressors performance can result in system output varying from commanded value.

Control Actions:

Generator load setpoint was changed between 5% and 10% to 35% with a corresponding turbine-compressor speed rate of change of 250 rpm/sec. The commanded load setpoint and resulting turbine-compressor speed setpoint are appended to the standard data set (Columns 321 and 322 for Excel).

General Notes:

- Main loop temperatures (CT#), pressures (CP#), and differential pressures (CDP#) have been adjusted for data acquisition system bias and zero flow corrections. As a result of these adjustments, the measured compressor inlet density and the density calculated using temperature and pressure via REFPROP differ by less than 1% for all data points.
- Total system CO₂ mass was held constant during this test. Turbine-compressor speed changes with load setpoint with windage heating possibly resulting in mass shifting into and out of the Brayton loop.
- Turbine-generator AC output power has been found via independent instrumentation to be ~25% greater than DC power calculated using DC current and voltage.

Figure 11 shows more of the Test 65261-P data used to define the transient in the PDC, along with the PDC input fits. The water precooler-inlet temperature, target CO₂ precooler-outlet temperature, oil flow rate and CO₂ IHX-outlet temperature are all held constant during the test. As Figure 11 shows, the PDC input for the target precooler-outlet temperature is different from the measured values to be consistent with the precooler-outlet density, as in the previous calculations. The water flow rate basically follows the turbine-compressor shaft speed and the CO₂ flow rate, but the PDC input is not needed since the water flow rate will be calculated by the code.

The generator shaft speed was commanded to a constant speed as in the previous test. However, as Figure 11 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 11.

Aside for the data in Figure 11 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, will be 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 11. Test 65261-P Data.

3.2 Steady-State Results

The results of the steady-state PDC calculations of the conditions at the beginning of the test run are shown in Figure 12 in comparison to the test data. The test data were averaged over 100 seconds prior to the first introduction of the shaft speed change at 1,400 seconds. Similar to the previous tests, the agreement is good 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 CO₂ side of the precoolers 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 so the transient simulation can proceed from the steady-state conditions in Figure 12.

3.3 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 10 with a given CR valve position. The results of this simulation are shown in Figure 13. 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. It also has some effect on the recuperator-inlet and IHX-inlet temperatures, but to a lesser degree. Also, the water flow rate is under predicted during the entire transient confirming that the heat transfer coefficient correction factor is too large. Most of the PDC results in Figure 13 show significantly larger variations at the points of the shaft speed changes than the test data. This difference is a result of the generator turbine shaft speed spikes in Figure 11. 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 experimental side. In most cases, though, those spikes occur within the measurement uncertainty range.

Figure 12. Steady-State Conditions at the Start of Test 65261-P (top) Compared to the PDC Results (bottom).

Figure 13. Results for Test 65261-P with Given Shaft Speed.

Figure 13. Results for Test 65261-P with Given Shaft Speed. (Continued)

Figure 13. Results for Test 65261-P with Given Shaft Speed. (Continued)

Figure 13. Results for Test 65261-P with Given Shaft Speed. (Continued)

Figure 13. Results for Test 65261-P with Given Shaft Speed. (Continued)

Figure 13. Results for Test 65261-P with Given Shaft Speed. (Continued)

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 10. The results of the PDC prediction of the shaft speed and the corresponding CR valve position are shown in Figure 14 with the rest of the transient results shown in Figure 15. 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 completely closed (while the test data shows it about 25% open), leading to an inability to maintain the target shaft speed and to unstable code results.

Figure 14. Shaft Speed and CR Valve Position Results for Test 65261-P.

The analysis of the test simulation results in Figure 15 in comparison with the results in Figure 13 (in particular, the compressor recirculation flow rate) suggests that the issue of under predicting 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 in Section 2.3.2, 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 in the shaft cavity is constant. The Tests 65261-P Summary above states 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 Summary, suggesting that the CO₂ 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 pressure 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 in Figure 13, 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 15 are consistent with the previous results in Figure 13 and are close to test data prior to 4,000 seconds. These results again confirm that there is room for improvement in the turbine modeling, similar to all of the previous results in this report.

Figure 15. The Rest of the Test 65261-P Full Simulation.

Figure 15. The Rest of the Test 65261-P Full Simulation. (Continued)

Figure 15. The Rest of the Test 65261-P Full Simulation. (Continued)

Figure 15. The Rest of the Test 65261-P Full Simulation. (Continued)

Figure 15. The Rest of the Test 65261-P Full Simulation. (Continued)

Figure 15. The Rest of the Test 65261-P Full Simulation. (Continued)

4 Summary

Validation of the Plant Dynamics Code (PDC) developed at Argonne National Laboratory (ANL) for steady-state and transient analysis of supercritical carbon dioxide (sCO₂) systems has been continued with the new test data from the Naval Nuclear Laboratory (operated by Bechtel Marine Propulsion Corporation) Integrated System Test (IST). Data from three 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 three tests, with test, 64661, the CO₂ 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 last two tests, 65261-P and 65261-U, 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. Since the only difference between these two tests was the rate of the shaft speed changes and due to time limitations, only Test 65261-P was analyzed for this report.

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 the Naval Nuclear Laboratory 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 report 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 under predicted 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 CO₂ (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. Note that this error was specific to the shell-and-tube heat exchanger only and thus was not discovered before this test in the SNL loop simulation nor did it affect the results for the full-scale system transients with the PDC, since both of those systems use compact diffusion-bonded heat exchangers.

The transient calculations for both tests were carried out in two steps. The first step was done in the same fashion as the FY15 IST 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 particular, for Test 64661, the water flow rate control was optimized to achieve a system response similar to that in the test (the IST water flow control logic could not be directly used in the PDC; since in the test, the control is applied to a throttle valve in a rather complicated water loop, while the PDC assumes direct control of the water pump head in a simple water loop).

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 PDC results even showed fast variations in the shaft speed, comparable to those observed in the test. 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 the Tests 64661 and 65216-P, the following improvements for the PDC simulation of the IST are:

- 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, that might require more detailed simulation of the shaft leakage collection system.

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