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# Steady State and Transient Modeling for the 10 MWe $SCO_2$ Test Facility Program

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## Abstract

The Gas Technology Institute (GTI), General Electric Global Research (GE-GR), and Southwest Research Institute (SwRI) have been working together to perform steady state and transient modeling analysis for the Supercritical Carbon Dioxide ($SCO_2$) 10 MWe Pilot Plant Test Facility, to be located in San Antonio, Texas. Both GTI and GE-GR are using AspenTech software, with GTI using Aspen Plus and GE-GR using Aspen Hysys. SwRI is using Numerical Propulsion System Simulation (NPSS). The benefit of having each organization performing steady state analysis in different tools is an assurance of the repeatability of the results as well as an increased confidence in the analysis. For the transient analysis, GTI is using a software called Flownex and GE-GR, collaborating with SwRI on one model, is using NPSS. As was done with the steady state models, the results of the two transient models will be reviewed and compared. The purpose of this paper is to provide insight into the results of the steady state models and a status of the transient models for the facility.


## Introduction

Interest in closed Brayton cycles for power generation has been rekindled in the last 20 years as practical limits have been reached in improving the efficiency of steam Rankine cycles. Heat integration strategies, turbine efficiency improvements, and advanced materials to increase turbine inlet temperatures have increased cycle efficiencies; however, further improvement requires very significant investment for incremental returns. As combustion and process technologies are pursued for reducing carbon emissions in fossil fueled plants, one of the simplest strategies for reducing these emissions is to increase the power cycle efficiency, and thus reduce the amount of fuel required. Several independent studies[1,2,3] have predicted higher cycle efficiencies for closed Brayton cycles, and in particular that the Recompression Closed Brayton Cycles (RCBC) is 3 to 5 points higher in cycle efficiency than steam Rankine cycles at temperatures above 500°C. $SCO_2$ is an attractive working fluid for these cycles as its critical point, at 31°C and 74 bar, allows for effective heat rejection and achievable cycle pressures, and its high density over the range of cycle conditions reduces the size of the turbomachinery. This offers the potential for lower capital costs and more rapid transient response.

The $SCO_2$ Facility is a 10 MWe net $SCO_2$ Brayton test facility to be built in San Antonio, Texas. The effort to design, construct, and build this facility is funded by the DOE National Energy Test Laboratory (NETL). The objectives of this project and for this facility are to demonstrate the operability of the $SCO_2$ cycle, verify the performance of components (turbines, recuperators, and compressors, etc.), show the potential for producing a lower cost of electricity in relevant applications, and demonstrate the potential and pathway for a thermodynamic cycle efficiency greater than 50% in commercial applications. Ultimately, this project will demonstrate a 700°C or higher turbine inlet temperature, and produce a 10 MWe net RCBC configuration that can be used to demonstrate and evaluate system and component design and performance capabilities (including turbomachinery and recuperators in steady state, transient, load following, and limited endurance operation). The facility will also be capable of being reconfigured to accommodate potential future testing of system/cycle upgrades, new cycle configurations, and new or upgraded components (turbomachinery, recuperators and heat exchangers).

Steady state modeling was performed for each configuration of the facility. The purpose of the steady state model is to provide the input for the technical specifications for each equipment unit in the facility to ensure successful target operation. Sizing and optimization of the components were based on the design point case of a 10 MWe RCBC cycle configuration with 715°C turbine inlet temperature. The sized components were then used in the off-design cases for both the RCBC and simple cycle configuration to determine the range of boundary conditions for each equipment unit. Three tools are used for the steady state modeling: Aspen Plus[7] by GTI, Aspen Hysys[8] by GE-GR, and NPSS[9] by SwRI. Aspen Plus and Hysys software are commercially validated codes that provide reliable thermodynamic analysis and a wide range of property method calculations, such as REFPROP, which is the best property method available for calculating $SCO_2$ properties at temperatures and pressures required for high efficiency Brayton cycles. NPSS is another powerful tool with sophisticated solving capability that can import numerous property methods, including REFPROP. With each organization modeling in different tools, the different modeling results will be reviewed and compared to ensure accuracy and repeatability of results.

Transient modeling will also be performed for the facility. Two different codes will be used: Flownex by GTI and NPSS by GE-GR and SWRI who are collaborating on one model. As with the steady state model results, the results from both transient models will be reviewed and compared. The results from the transient model will provide insight into the operational analysis for the facility.

## Steady State Modeling

### *Aspen Models*

Two $SCO_2$ cycle configurations will be tested in the pilot facility: a simple recuperated Brayton cycle and a recompression Brayton cycle. Both configurations were modeled. The $CO_2$ thermodynamic properties for both the Plus (GTI) and Hysys (GE-GR) models were based on REFPROP from the ASPEN properties bank coming with the ASPEN and HYSYS software packages. The $CO_2$ thermodynamic properties for NPSS was also based on REFPROP.

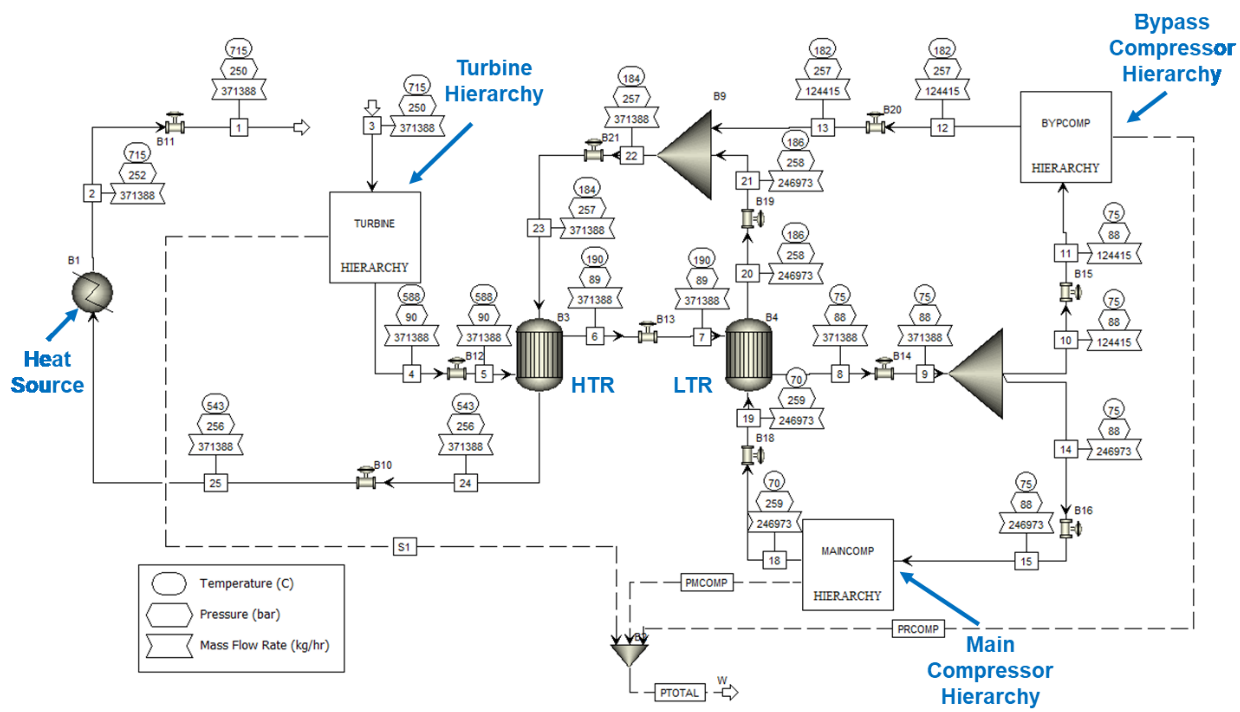


Figure 1. 10 MWe RCBC Pilot Plant Aspen Plus Model

For both configurations, hierarchy blocks were used to model the turbomachinery for ease of viewing the main process flowsheet, as the submodels contained secondary flow losses. The turbine model includes turbine balance piston leakage and the compressor models include compressor recycle lines for recycle mode operation.

The Aspen Plus model for the RCBC cycle is shown in Figure 1 above. $SCO_2$ is heated by the heat source (Stream 3) and is expanded through the turbine (Stream 4). It is then cooled initially in the high temperature recuperator (HTR) (Stream 6) and then further in the low temperature recuperator (LTR) (Stream 8). The $SCO_2$ flow is then split between the bypass compressor (Stream 10) and the main compressor (Stream 14). Before entering the main compressor, the $SCO_2$ is cooled by the main process cooler. It is then compressed up to system pressure (Stream 18) and then heated in the LTR (Stream 20) where it mixes with the exit of the bypass compressor (Stream 12). The mixed $SCO_2$ stream then recuperates additional heat in the HTR before it is recycled back to the heat source to be heated to system turbine inlet temperature (Stream 25).

The Aspen Plus model for the Simple cycle is shown in Figure 2 below. As with the RCBC cycle, $SCO_2$ is heated by the heat source and then expanded through the turbine (Stream 4). It then is cooled initially in the high temperature recuperator (HTR) and then further by the main process cooler. It is then compressed in the main compressor back up to system pressure. It recuperates heat in the HTR and is recycled back to the heat source to be heated to system turbine inlet temperature (Stream 25).

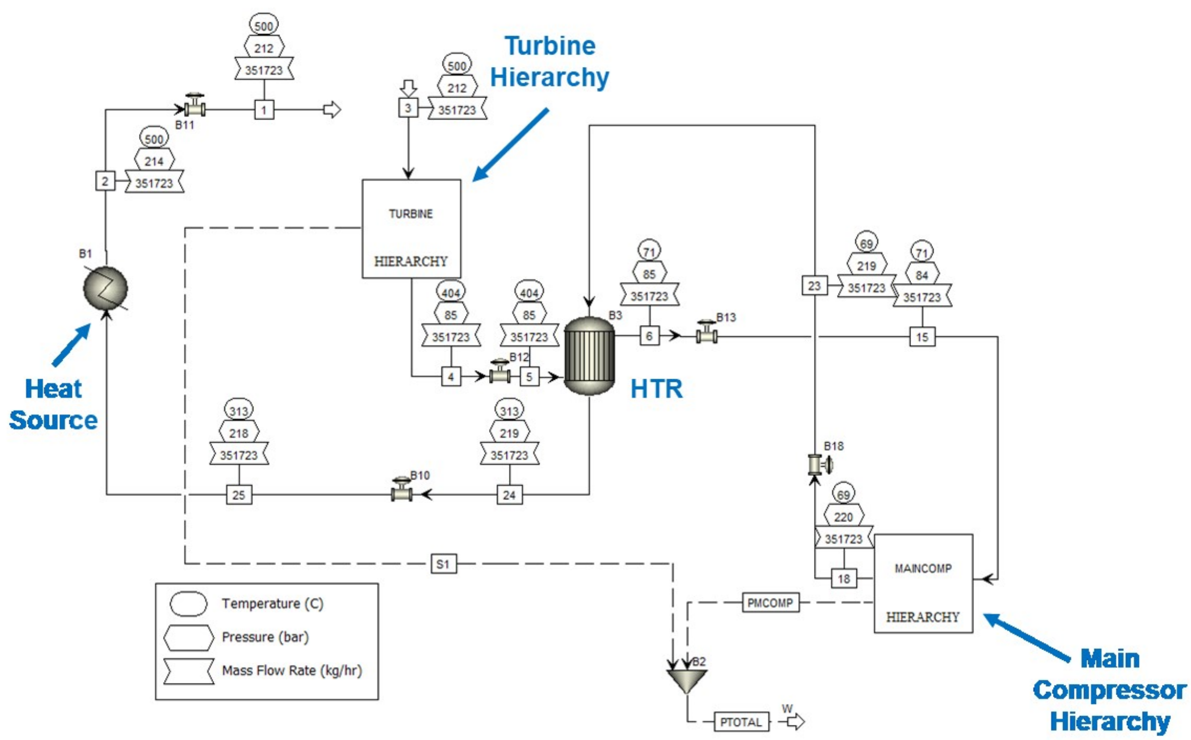


Figure 2. Pilot Plant Simple Cycle Aspen Plus Model

### *Model Assumptions and Fidelity*

Turbomachinery performance maps were provided by GE using their proprietary Oil and Gas compressor design tool. The inputs for the tool were the boundary conditions for the design point case, shown in Table

2. Recuperator parameters were chosen based on the best that vendors could design to. An approach temperature of 5°C and a pressure drop of 0.7 bar was chosen. Initial piping pressure drops were also included and were based on a past GTI layout of a commercial scale RCBC cycle. Pressure drop through the heat source was set to 4 bar.

### *Optimization Parameters*

The three elements to achieving a high cycle efficiency are maximizing turbine power production, minimizing compressor power, and minimizing the heat input into the system while maintaining net electrical output. There are several parameters that affect those elements and can be adjusted to optimize system performance, shown in Table 1 below.

| Parameter | Simple Cycle | Recompression Cycle |
| --- | --- | --- |
| Turbine Inlet Temperature | X | X |
| Turbine Inlet Pressure | X | X |
| Turbine Pressure Ratio (Exit Pressure) | X | X |
| Compressor Inlet Guide Vane (IGV) Setting | X | X |
| Flow Split between Compressors | | X |

Table 1. Optimization Parameters for Simple and Recompression Cycles

Turbine power is dependent upon the inlet and exit conditions, mass flow rate, and efficiency of the turbine. Choosing a turbine pressure ratio defines the turbine exit pressure, which in combination with the inlet condition sets the mass flow rate for the system based on the turbine flow function. A lower exit pressure is desirable to maximize turbine pressure ratio, which aids in increasing cycle efficiency. The lower limitation on the exit pressure is the combination of the critical pressure of $CO_2$, which is 74 bar, and the pressure drops in the cycle from the turbine exit to the main compressor inlet. Margin must be allocated to ensure two phase flow does not occur at the inlet to the main compressor. A 15% margin on critical pressure was allocated for this facility.

As has been well proven thermodynamically by many[2], system efficiency varies directly with turbine inlet temperature and pressure, with temperature being a much larger driver on system efficiency than pressure. For reference, the general breakaway point in which Brayton cycle technology surpasses Rankine cycle technology in system efficiency is about 500°C. Currently, the highest temperature and pressure condition for the facility for the Simple Cycle will be 500°C and 212 bar and for the RCBC will be 715°C and 250 bar. The turbine inlet temperature and pressure conditions for the RCBC configuration were chosen to show the highest performance of the system while meeting conditions achievable by the facility. The temperature for the simple cycle was chosen based on industry applicability and the pressure was based on compressor performance at the off-design condition of the simple cycle.

As with the turbine, compressor power is dependent upon the inlet and exit conditions, mass flow rate, and efficiency of the compressors. Inlet guide vane (IGV) settings can be adjusted to vary compressor pressure ratios and efficiencies. The flow split between the compressors determines the mass flow rate through each compressor. For the simple cycle, as there is only one compressor, the flow rate is determined by the turbine flow function. The optimal IGV setting is then chosen based on the point on the compressor map that provides the highest efficiency while meeting the pressure requirement at the turbine inlet. For the recompression cycle, there are two compressors with two different maps and IGV settings. The optimization of each compressor is the point on each map that provides the highest efficiency for that compressor while meeting the pressure requirement at the turbine inlet for that given flow split. As the flow split affects the mass flow rate through each compressor, the performance of each compressor is dependent upon that ratio. Generally, a range of about 30-36% flow split to the bypass compressor is optimal. Given that the flow split is an adjustable parameter, parametric studies need to be done to determine the optimal split and IGV setting combination that minimizes the total compressor power and therefore increases cycle efficiency.

The main method to minimize the heat input into the system is to minimize the flow rate through the heater and achieve the highest cold stream outlet temperature possible out of the HTR. As system flow rate is dependent upon the turbine flow function, which is a function of the turbine conditions already chosen, the mass flow rate is not an adjustable parameter. Thus, the only adjustable parameter is the inlet temperature to the heater. This was maximized by choosing a 5°C approach temperature for the recuperators, as previously described. In order to maximize system performance, this approach temperature was not a varied parameter.

### *Initial Results*

Several cases were modeled for each configuration to define the technical requirements of the equipment units. Sizing and optimization of the components were based on the design point case of a 10 MWe RCBC cycle configuration with 715°C turbine inlet temperature (Case 051 on Table 2). The sized components were then used in the off-design cases at non-optimal conditions for both the RCBC and simple cycle configuration to determine the range of boundary conditions for each equipment unit.

Seven different cycles were modeled in total: two simple cycles (model cases 033, and 036) and five recompression cycles (model cases 051, 052, 053, 054, and 055). The conditions varied were turbine inlet conditions, process cooler exit conditions, and load levels. For the simple cycle, all turbine inlet temperatures were 500°C and cooler exit temperatures were 35°C. The loads were varied to a max and min percentage value of ~7 and 4 MWe. For the recompression cycle, the operating conditions include 100% nominal and 40% loads and turbine inlet temperatures of 715°C and 500°C, while the ambient conditions include hot (cooler exit temperature of 50°C), nominal (cooler exit temperature of 35°C), and cold (cooler exit temperature of 20°C) days. Shown in Table 2 below is each case modeled with the current associated predicted cycle efficiency. Included in the cycle efficiency numbers are generator and gearbox losses.

As mentioned previously, all equipment units were designed for the baseline case 051. In table below, Case 053 is scaled to a higher net power level than Case 051. An additional case will need to be modeled to scale that case down to below 10.3 MWe.

| Model Names | Cycle Configuration | Load % | Net Power Level (MWe) | Turbine Inlet Temperature | Turbine Inlet Pressure | Cooler Exit Temperature | Cycle Efficiency |
|---|---|---|---|---|---|---|---|
| 033 | Simple | Min | 3.6 | 500°C | 170 bar | 35°C | 26.0% |
| 036 | Simple | Max | 6.5 | 500°C | 212 bar | 35°C | 30.8% |
| 051 | Recompression | 100% | 10.4 | 715°C | 250 bar | 35°C | 46.2% |
| 052 | Recompression | 100% | 7.9 | 715°C | 259 bar | 50°C | 42.1% |
| 053 | Recompression | 100% | 13.6 | 715°C | 250 bar | 20°C | 45.2% |
| 054 | Recompression | 40% | 4.0 | 715°C | 170 bar | 35°C | 39.1% |
| 055 | Recompression | 100% | 7.7 | 500°C | 250 bar | 35°C | 36.6% |

Table 2. Steady State Cases Modeled for STEP Facility

Comparison of the GTI, GE-GR, and NPSS models show the results are in good agreement for the design point Case 051 recompression cycle, as shown in Table 3 below. The differences for each of the parameters is less than 2%, with the exception of the LTR, whose large difference is mainly associated to differences in pressure drop assumptions. For some of the off-design cases, there are differences in the results, which lie mainly in the detailed performances of the equipment, such as the inlet guide vane settings of the compressors or the recuperator heat duties, as shown in Table 4. The differences in the results is as large as 20%. The reason for the discrepancies is due to the off-design modeling method of the recuperators, as well as minor differences in flow rate. The assumption made affects the heat transfer in the recuperators, which affects the performance of both compressors, which then affects the system. Although all three organizations assumed a constant surface area or effectiveness parameter for a shell and tube heat exchanger geometry, the assumptions for the shell and tube heat exchangers in each model were not verified to be the same, such as number of passes and diameter of tubes. Also, although there were discrepancies with component parameters, the discrepancy of the results on the overall performance of the system, such as the cycle efficiency or net power produced, was relatively small. In addition, since the design of the recuperators is currently unchosen, there is no agreed upon good method to model the recuperators for off-design operation. Thus, no further effort was made to look into the detailed assumptions for the off-design recuperators until further discussions are had with the recuperator vendors and a recuperator design selected so that a better decision can be made with respect to modeling recuperators off-design. One benefit of modeling different recuperator assumptions is that the range of results provides a range of operation for the recuperators and compressors that can be used in better defining the equipment specifications.

| Parameter | Unit | Aspen Plus | Aspen Hysys | NPSS | | Largest Delta |
|---|---|---|---|---|---|---|
| Net Electric Power | MWe | 10.4 | 10.3 | 10.3 | | 0.96% |
| Cycle Efficiency | % | 46.2 | 45.7 | 45.8 | | 1.1% |
| Turbine Power | MWe | 15.6 | 15.6 | 15.6 | | 0.06% |
| Main Comp Power | MWe | 2.15 | 2.15 | 2.16 | | 0.47% |
| Bypass Comp Power | MWe | 2.68 | 2.67 | 2.65 | | 1.1% |
| Optimized Flow Split to Main Comp | % | 66.5 | 66.7 | 66.8 | | 0.45% |
| HTR Heat Duty | MWth | 48.0 | 48.3 | 48.1 | | 0.48% |
| LTR Heat Duty | MWth | 15.4 | 14.8 | 14.7 | | 4.5% |
| Heater Heat Duty | MWth | 22.5 | 22.5 | 22.6 | | 0.44% |

Table 3. Comparison of Design Case 051 Results in Aspen Plus, Aspen Hysys, and NPSS

| Parameter | Unit | Aspen Plus | Aspen Hysys | NPSS | | Largest Delta |
|---|---|---|---|---|---|---|
| Net Electric Power | MWe | 4.0 | 4.0 | 3.9 | | 2.5% |
| Cycle Efficiency | % | 39.1 | 39.5 | 39.7 | | 1.4% |
| Turbine Power | MWe | 7.75 | 7.47 | 7.49 | | 3.9% |
| Main Comp Power | MWe | 1.54 | 1.50 | 1.56 | | 4.0% |
| Bypass Comp Power | MWe | 2.06 | 1.77 | 1.85 | | 14% |
| Optimized Flow Split to Main Comp | % | 61.0 | 64.0 | 63.0 | | 4.9% |
| HTR Heat Duty | MWth | 31.2 | 30.6 | 32.95 | | 7.7% |
| LTR Heat Duty | MWth | 8.28 | 9.13 | 7.32 | | 20% |
| Heater Heat Duty | MWth | 10.3 | 10.2 | 9.9 | | 3.9% |

Table 4. Comparison of Off-Design Case 054 Results in Aspen Plus, Aspen Hysys, and NPSS

## *Impact of Results*

### *Commercial System Performance*

An ultimate goal of the project is to develop the pathway to show an efficiency target of greater than 50% at a commercial scale. Due to economies of scale and reduced turbomachinery performance at the pilot plant level, the highest potential achievable efficiency is 46.2%, as shown in Table 2 for Case 051. This cycle efficiency value includes generator and gearbox losses. To determine the efficiency of a commercial scale plant, the RCBC configuration baseline case was rerun and scaled up to 450 MWe net plant capacity. The same pressure drops were assumed in the commercial case as in the pilot plant for piping, valves, instrumentation, and equipment units. Figure 3 shows the Aspen model for the commercial scale plant.

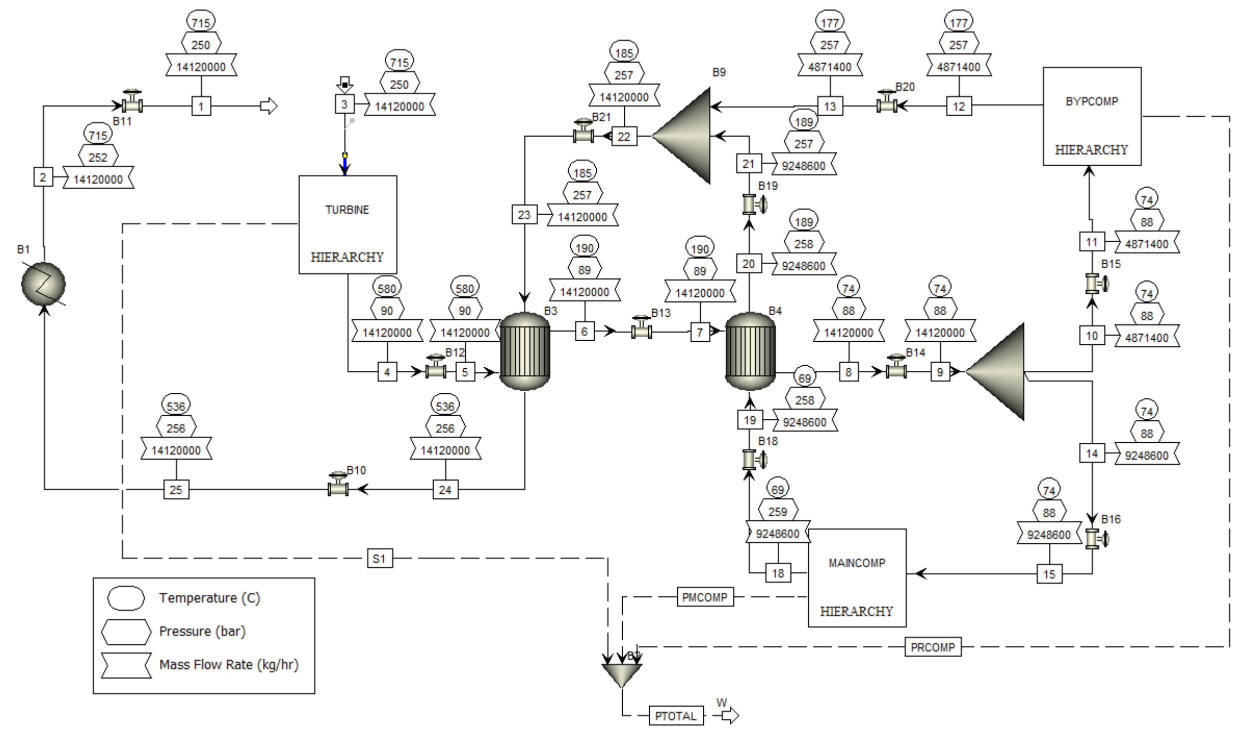


Figure 3. 450 MWe RCBC Commercial Plant Aspen Plus Model

Table 5 below summarizes the changes that were made in scaling the model from pilot scale to commercial scale. Commercial turbine isentropic efficiencies were taken from Part 1 of a study by Bidkar et al.[4] for a 450 MWe reheat cycle. An average efficiency was used based on the high and low pressure turbine efficiencies referenced. Commercial compressor efficiencies were taken from Part 2 of the same study by Bidkar et al.[5] for a 450 MWe reheat cycle. Pressure ratios for each of the turbomachinery components were held constant in the scale up. Generator efficiency increased from 97.6% to 99% for commercial scale, as referenced in GE's hydrogen-cooled TOPGAS generator.[6] Due to the increased turbomachinery and generator performances, the potential achievable cycle efficiency at the 450 MWe commercial scale increases to 50.5%. Other configurations, such as a reheat cycle studied by Bidkar et al., may provide additional efficiency improvements and will be evaluated in the future.

| Parameter | 10 MWe Pilot Plant | Commercial Plant |
|---|---|---|
| Net Plant Capacity | 10 MWe | 450 MWe |
| Turbine PR | ~3 | Same as Pilot |
| Turbine Efficiency | 87.3% | 91% |
| Main Compressor PR | ~3 | Same as Pilot |
| Main Compressor Efficiency | 78.5% | 83% |
| Bypass Compressor PR | ~3 | Same as Pilot |
| Bypass Compressor Efficiency | 78.2% | 80.1% |
| Generator Efficiency | 97.6% | 99% |
| Gearbox Efficiency | 99.3% | N/A |
| System Pressure Drop | ~12 bar | Same as Pilot |
| Cycle Efficiency | 46.2% | 50.5% |

Table 5. Comparison of Pilot vs Commercial Scale Plant Parameters

### *Equipment Technical Specifications*

Results from the steady state models were used to build the technical specifications for each of the equipment units. Requirements for the major equipment specifications, such as the heat source, recuperators, main process cooler, and turbomachinery have been assembled. The technical specifications for each component include the results from each case run to provide vendors with the range of operating conditions for that component. Detailed result inputs include inlet and outlet temperatures, pressures, and mass flow rates for the boundary interfaces of that component. Heat duties for the recuperators, process cooler, and heat source were also included. Target compressor efficiencies were also provided for the compressor specifications.

### *Performance Losses Due to Increased Pressure Drop*

The initial cases run had pressure drop allocations of ~12 bar throughout the main process loop. As initial piping layouts were created and flow meter and valve pressure drops identified, the pressure drop throughout the loop increased to ~17 bar. Initial studies were performed to determine the impact of the increased pressure drop on the baseline case 051. Two cases were run: one which maintained the compressor pressure ratios, thereby decreasing the turbine pressure ratio, and the other which maintained the turbine pressure ratio, thereby increasing the compressor pressure ratios. For both cases, the flow split between the compressors were maintained and the turbine exit pressure was increased to maintain the same margin on the main compressor suction pressure as the baseline case. For the first case which maintained the compressor pressure ratio, the turbine pressure ratio dropped by 2.6%, which led to a 0.9% efficiency drop. For the second scenario in which the turbine pressure ratio was maintained, the compressor pressure ratios increased by 3 or 4%. The net impact on efficiency was only a 0.2% efficiency loss. This was mainly due to higher efficiencies of the compressors for this operating point. While more work needs to be done in evaluating these cases, based on the initial results, the better method to account for increased pressure drop and minimize efficiency loss is to maintain the turbine pressure ratio by increasing the compressor pressure ratios.

### *Performance Losses Due to Recuperator Approach Temperature*

As mentioned previously, increasing the recuperator approach temperature generally reduces system performance as it increases the heat input requirement to the system. One exception in which the approach temperature may be increased with minimal impact on the cycle efficiency is with the LTR approach temperature. The cold stream outlet of the LTR mixes with the bypass compressor exit stream before it enters the HTR. As long as the temperature entering the HTR remains the same, the system efficiency will remain unaffected. In each of the cases run, the LTR cold stream outlet temperature was higher than the bypass compressor exit temperature, leading to some energy losses due to mixing. An increased approach temperature in the LTR would have no impact on cycle efficiency up to the point in which the two mixing stream temperatures match and those mixing losses eliminated for a given flow split. For the design point case 051, that approach temperature was 6°C, a 20% increase in approach temperature or potentially log mean temperature difference which could lead to cost savings for the recuperator. While this change will help with LTR recuperator costs and have minimal impact on cycle efficiency, increasing the approach temperature of the LTR will increase the temperature of the $SCO_2$ entering the cooler, thereby increasing the size and cost of the process cooler. System trades will need to be performed to evaluate the lowest cost option.

## Transient Modeling

The purpose of the transient model is to provide numerical results to aid in the operational analysis of the test facility. Several tools were evaluated and rated on various features, such as the possibility of real-time simulation, size of the user base, availability of $CO_2$ properties, time required to develop the models, etc. GTI will be using Flownex and GE-GR, collaborating with SwRI on one model, will be using NPSS. The physics behind the models of each tool will be the same and results will be compared.

Flownex is an object-oriented simulation environment that was developed in South Africa in the late 1980s and is now managed by Phoenix Analysis & Design Technologies. It is ISO 9001:2008 and NQA1 complaint and has been validated in the nuclear industry.

NPSS is an object-oriented, multi-physics, engineering design and simulation environment which enables development, collaboration and seamless integration of system models. NPSS was originally developed by NASA Glenn Research Center in 1995 and is now supported by a consortium led by SwRI. As SwRI has deep expertise in NPSS, SwRI will provide support to GE-GR. SwRI has developed steady state models for the simple and recompression Brayton cycles that GE-GR will use as a starting point for transient modeling.

The path forward for each tool is to build the steady state model, as is required for the Flownex model, or modify the steady state model, as is needed for NPSS model, and benchmark these steady state models to the Aspen Plus and Hysys models. The first step would be to work on the model of each custom component in the system and ensure each individual component is performing as expected for steady state analysis. GE-GR will be taking the lead on defining the physics required for the turbomachinery equipment and GTI taking the lead on defining the physics for the recuperators. The remaining components will be split up amongst the team. Once each individual component is verified, the entire system will be modeled together. A control system methodology will need to be agreed upon by the team and input into the transient model before running transient cases. A Requirements Document for both the steady state and transient models will also be put together by the team to document all assumptions, methodologies, and requirements of the models to ensure the final product meets all expectations.

The intended use of the transient model is to provide insight into the transient operations of the facility, such as the optimized start up sequence or optimized transition sequence between test points. Another use for the model is to resolve issues that may arise during operation, such as back flow through the heat source in the event of an emergency shut down. As the transient model is built and verified, there will be greater fidelity in the operational analysis.

## Conclusion

Steady state models have been performed for the $SCO_2$ 10 MWe Pilot Plant Test Facility. Three different tools have been used to model the simple and recompression cycles: Aspen Plus, Aspen Hysys, and NPSS. Parameters that were varied include turbine inlet temperature, turbine inlet pressure, turbine pressure ratio, inlet guide vane settings, and compressor flow splits. Initial results show good agreement among models with overall performance, but some minor discrepancies with individual component parameters and performance. The results of the steady state models have been incorporated into the technical specifications for the various equipment units. As the program matures and changes to the system are made, the steady state models will be updated to reflect all new changes.

Transient modeling of the facility is in progress. Two different codes will be used: Flownex by GTI and NPSS, by GE-GR and SwRI, who are collaborating on a model. The results of each model will be compared and reviewed to ensure accuracy of the results. The transient model will aid in the operational analysis for the facility to define optimal operational sequences and prevent operation in undesirable conditions.

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