Task Order 20: Supercritical Carbon Dioxide Brayton Cycle Energy Conversion Study

Final Report

RPT – 3011934-000

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REVISION LOG

Rev.	Date	Affected Pages	Revision Description

DISCLAIMER

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LIST OF ACRONYMS

A&E	Architectural and Engineering
ASME	American Society of Mechanical Engineering
AUSC	Advanced Ultra Supercritical
CO2	Carbon Dioxide
DOE	Department of Energy
FOAK	First-of-a-kind
HP	Horsepower
HRSG	Heat Recovery Steam Generator
HVAC	Heating, Ventilation, and Cooling
ISA	Integrated Safety Analysis
L/Ds	Piping Configuration
m	Meter
mm	Millimeter
MW	Megawatt
MWe	Megawatt Electrical
NE	Nuclear Energy
PRA	Probabilistic Risk Assessment
psi	pounds per square inch
psig	pounds per square inch gage
R&D	Research and Development
ROM	Rough Order of Magnitude
RPM	Rotations per minute
sCO2	Supercritical Carbon Dioxide
SOW	Statement of Work

1.0 Purpose

AREVA Inc. developed this study for the US Department of Energy (DOE) office of Nuclear Energy (NE) in accordance with Task Order 20 Statement of Work (SOW) covering research and development activities for the Supercritical Carbon Dioxide (sCO_2) Brayton Cycle energy conversion. The study addresses the conversion of sCO_2 heat energy to electrical output by use of a Brayton Cycle system and focuses on the potential of a net efficiency increase via cycle recuperation and recompression stages. The study also addresses issues and study needed to advance development and implementation of a 10 MWe sCO2 demonstration project.

The initial basis used to develop this study was the following report:
"A Supercritical Carbon Dioxide Cycle for Next Generation Nuclear Reactors,"
MIT-ANP-TR-100, dated March 10,2004; by: V. Dostal, J.J. Driscoll, and P. Hejzlar

TABLE 1.0-1: SOW ITEM/ACTIVITY MAP TO REPORT SECTIONS

SOW Item	Title	Section
SOW Section 2.4, Subtask 4	Energy Conversion - Cycle Configuration	See 3.0 Configuration: Analysis of sCO2 Brayton Cycle Alternatives
SOW Section 2.4, Subtask 4	Energy Conversion - Cycle Heat Recovery	See 4.0 Heat Recovery: Recuperation and Recompression Improvements
SOW Section 2.4, Subtask 4	Energy Conversion -Thermodynamic Diagrams	See 5.0 Thermodynamic Diagrams: Cycle Configuration Comparisons
SOW Section 2.4, Subtask 4	Energy Conversion – Cost Estimates	See 7.0 Cost Estimates: ASME Qualified 1.25-2.5 Chrome Alloy Materials

2.0 Executive Summary

Selective sCO2 Brayton Cycle Energy Conversion modeling was performed using Aspen Plus, a widely used industry process modeling software (Aspen One V. 8.6). Fluid properties were derived from the Refprop equation of state for CO2. Modeling was performed for cycles with gross cycle power levels of 10 & 30 MWe and two heat source temperature ranges: 300-450°C and 450-625°C. Further modeling trade studies explored the level of cycle recuperation, the flow split between main compressor & recompressor, the use of a Rankine bottoming cycle, and the split of the temperature range with topping and bottoming cycles.

The modeling indicates that sCO2 cycle performance improves with higher temperature, with efficiencies that range from approximately 40% (300-450°C heat source) to more than 48% (450 to 625°C heat source). These efficiency results indicate that for the lower temperature case the steam Rankine cycle (efficiency ~42.5%) is better than the sCO2 cycle, while the advantage is with the sCO2 cycle at the higher temperature case (efficiency ~45.5%). Trade studies of cycle recuperation and flow split did not improve cycle efficiency due to the base cases already reflected optimization from past experience. The addition of a steam bottoming cycle or topping cycles increases the efficiency above 50%, but at considerable additional expense for the added components. Topping and bottoming cycles are not recommended for the narrow (up to 175C) heat source ranges studied, in that the single highly recuperated split flow sCO2 cycle matches the heat input very well. In general, the cycle modeling produced no surprises.

Pressure vessels within the sCO2 cycle will be specified and fabricated per the American Society of Mechanical Engineers (ASME) Section VIII code, while the relevant code for piping will be ASME B31.1. The capability of less expensive and commonly available materials, including 1.25-2.5 chromium alloy material, will be suitable for certain portions of the evaluated sCO2 cycles however, their application will be limited to 575 C and below. A maximum ten year life for the demonstration plant is assumed since total operating hours will be restricted depending on material selections. Cycle temperatures reaching 625 C will require higher chrome content materials such as P91.

The study includes Rough Order of Magnitude (ROM) cost estimates for the major components and piping that comprise the sCO2 primary loop. Overall costs for the heat source, construction, electrical tie-in, auxiliary systems, and facility costs are beyond the scope of this study. A subsequent study is recommended to further review these costs.

Determination of a suitable and cost-effective heat source is a key task leading to the implementation of a 10 MWe sCO2 demonstration project. A fossil fuel fired heat source appears to be the most practical method of reaching the desired sCO2 cycle temperature. A study is recommended that will evaluate fossil heat source options.

In addition, the following technology gaps in the sCO2 cycle and components must be addressed:

1. System optimization, including heat source integration
2. Material compatibility with sCO2
3. Turbomachinery – high power density, dry gas seals, compressor operation with inlet conditions close to triple point, and thermal management in sCO2
4. Compact heat exchangers
5. System operation and control

6. Scaling from pilot to commercial sizes
7. Demonstrated performance over sustained operating periods

3.0 Configuration: Analysis of sCO2 Brayton Cycle

3.1 Modeling Approach

All simulations were carried out using Aspen Plus, a widely used industry process modeling software (Aspen One V. 8.6). The fluid property method was based on the Refprop equation of state since pure CO2 was used as the working fluid. Compressor and turbine efficiencies were assumed to be 85% and 90%, respectively. An approach temperature of 5°C between the hot inlet temperature and the cold outlet temperature was used when defining heat exchangers, as this was a minimum temperature approach reasonably achievable. The exit temperature of the cooler was assumed to be 32°C as this was found to be the optimum cooler temperature in a study performed by Aerojet Rocketdyne to Leonardi Technologies, Inc. under contract DE-FE0004002 [1] . Pressure drop was also included within the system balance based on a piping configuration (L/Ds) for a 550 MWe sCO2 plant. The system pressure is 276 bar. There were two heat source temperature ranges: 300-450°C and 450-625°C. These system temperature and pressure conditions are below the Advanced Ultra Supercritical (AUSC) Steam Rankine conditions and therefore appear very doable. Four main simulations were modeled: one for each of the two heat source temperatures at two different power levels of 10 and 30 MWe.

3.2 Process Flow Diagram

The following figures and tables show the flow of the process, identification of Aspen plus units, temperature-entropy, and pressure enthalpy.

FIGURE 3.2-1: ASPEN PLUS SCHEMATIC OF INDIRECTLY HEATED SCO2 BRAYTON CYCLE WITH HEAT SOURCE TEMPERATURE RANGE OF 450-625°C FOR 10 MWE PLANT

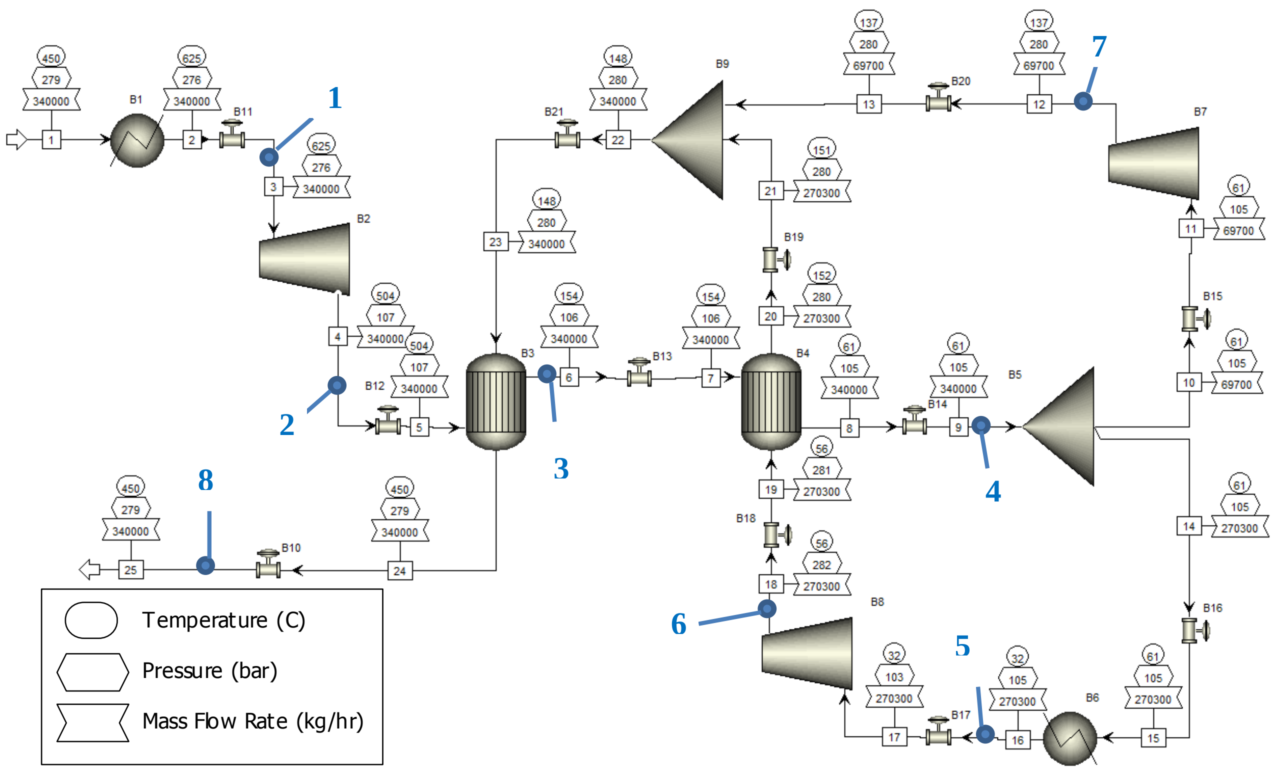


TABLE 3.2-1: IDENTIFICATION OF ASPEN PLUS UNITS

Equipment Unit	Description
B1	Heat Source Heat Exchanger
B2	Turbine
B3	High Temperature Recuperator
B4	Low Temperature Recuperator
B5	Splitter
B6	Cooler
B7	Recycle Compressor
B8	Main Compressor
B9	Mixer
B10 – B21	Equipment unit used to simulate pressure drop

FIGURE 3.2-2: sCO2 TEMPERATURE-ENTROPY DIAGRAM FOR 450-625°C WITH CORRESPONDING NUMBERS TO LOCATIONS IN FIGURE 3.2-1

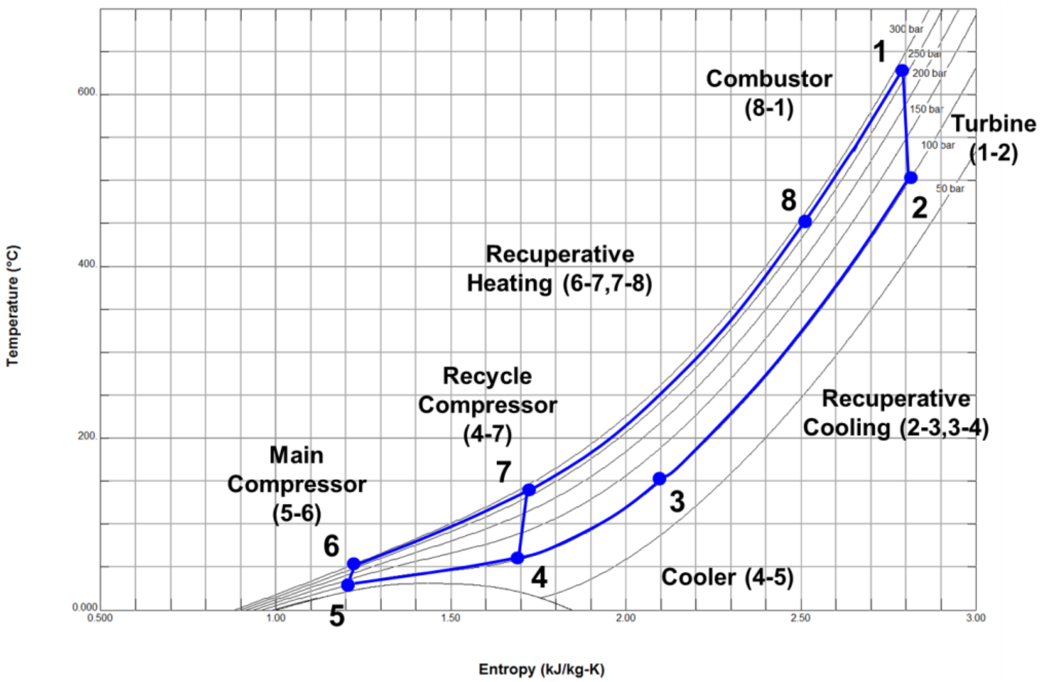
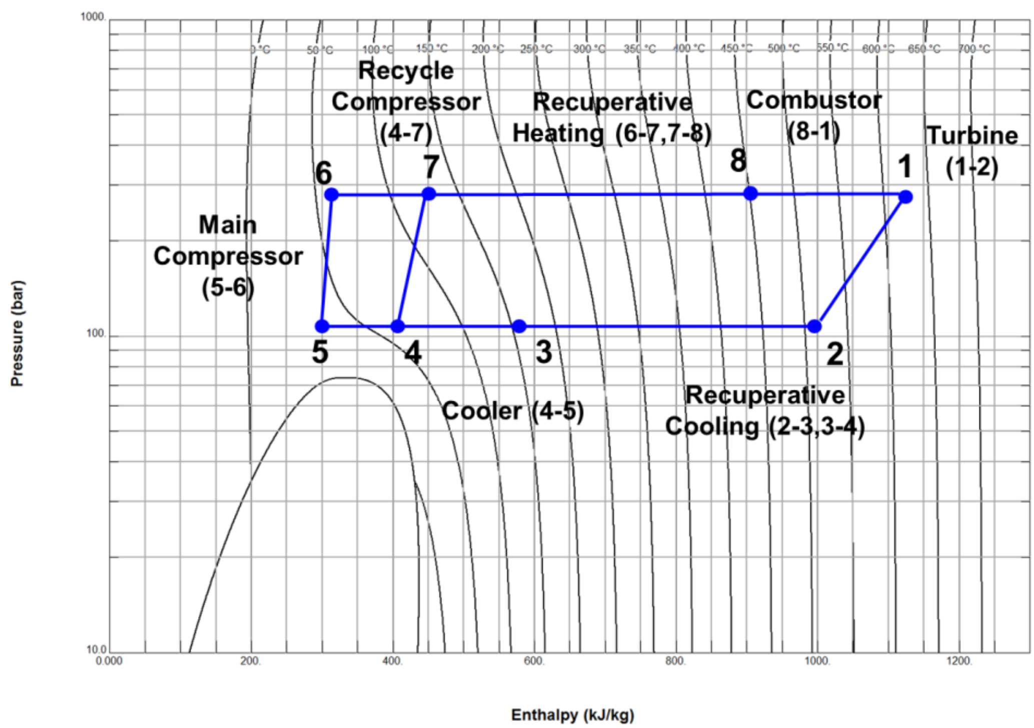


FIGURE 3.2-3: sCO2 PRESSURE-ENTHALPY DIAGRAM FOR 450-625°C



3.3 Cycle Description

The process flow diagram for the 450-625°C, 10 MWe case is shown in **Figure 3.2-1**, with a description of the equipment units in **Table 3.2-1**. All other cases have the same configuration. As indicated in **Figure 3.2-2** and **Figure 3.2-3**, the corresponsing Temperature-Entropy and Pressure-Enthalpy diagrams, sCO2 enters the heat source heat exchanger and is brought up to system temperature (8-1). It is then expanded through the turbine to 107 bar (1-2). This outlet pressure was chosen because it provided the greatest pressure ratio through the turbine, thereby maximizing efficiency, while allowing the hot stream to the high temperature recuperator to provide enough heat to the working fluid to reach the heat source inlet temperature. The sCO2 is then cooled by the high temperature recuperator and the low temperature recuperator (2-3, 3-4). The flow is then split between a main compressor and recycle compressor. With every case, the cycle is optimized for efficiency by adjusting the flow split between the main and recycle compressors. For these cases, the optimum split was at 20%. That is 20% of the fluid enters the recycle compressor (4-7), while the remaining 80% enters a cooler (4-5), and then the main compressor (5-6). The main compressor flow is then heated by the low temperature recuperator (6-7) and mixed with the stream from the recycle compressor. The sCO2 is then reheated by the high temperature recuperator (7-8), returning then to the heat source heat exchanger to be raised to system temperature. The remaining process flow diagrams, Temperature-Entropy diagrams, and Pressure-Enthalpy diagrams for the other cases are shown in the *Appendix A***.**

3.4 Topping and Bottoming Cycles

Two topping cycles and one bottoming cycle configuration was evaluated. The heat source temperature range for these cycles was 450-625°C and the gross plant power was 30 MWe. For the topping cycle configurations, the heat source was split between two separate sCO_2 cycles. The first topping cycle configuration split the heat source temperature at 538°C; one cycle had a heat source temperature range from 450-538°C and the other cycle had a heat source temperature range from 538-625°C. Each cycle contributed 15 MWe to the total plant power. The second topping cycle configuration split the heat source temperature at 582°C, with each cycle contributing 15 MWe as well. The turbine pressure ratio was altered to meet the new heat source temperature range conditions. The split between the main and recycle compressors was also optimized. **Figures 3.4-1** and **Figure 3.4-2** show process flow diagrams for the first topping cycle configuration. The process flow diagram for the second topping cycle can be found in the *Appendix A***.**

FIGURE 3.4-1: TOPPING CYCLE 1 WITH THE HEAT SOURCE TEMPERATURE SPLIT AT 538°C; 450-538°C CYCLE

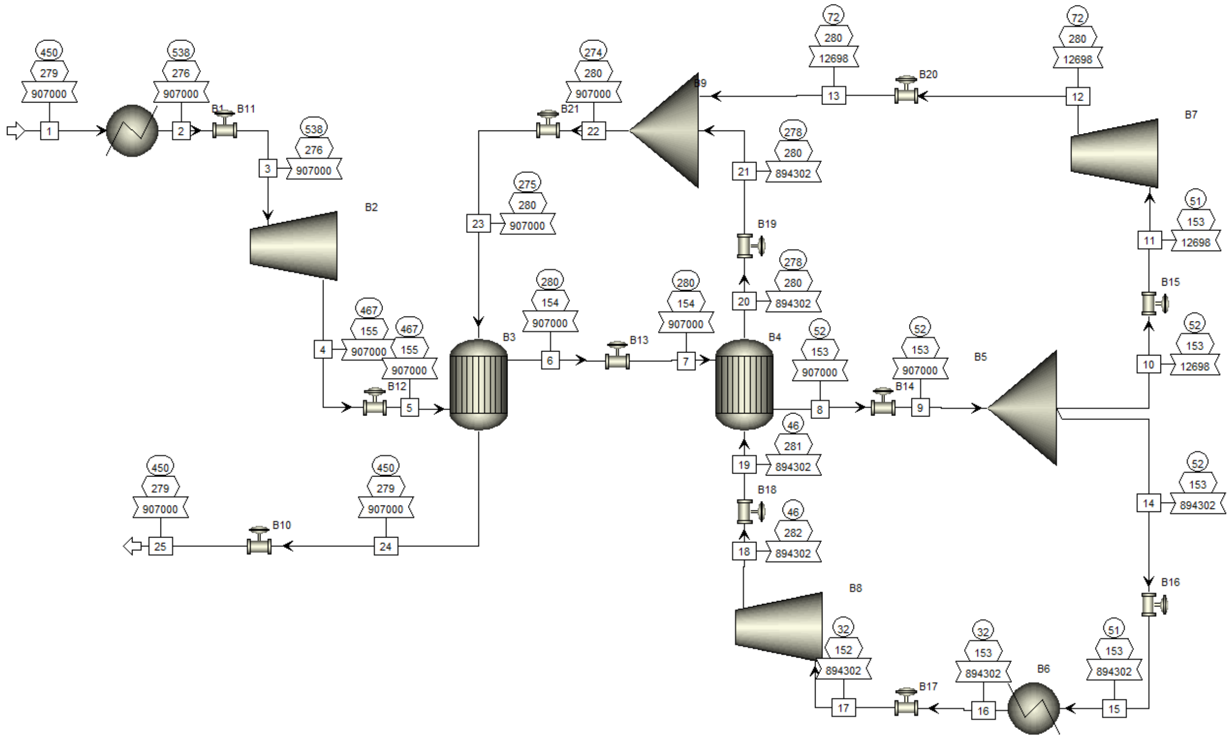
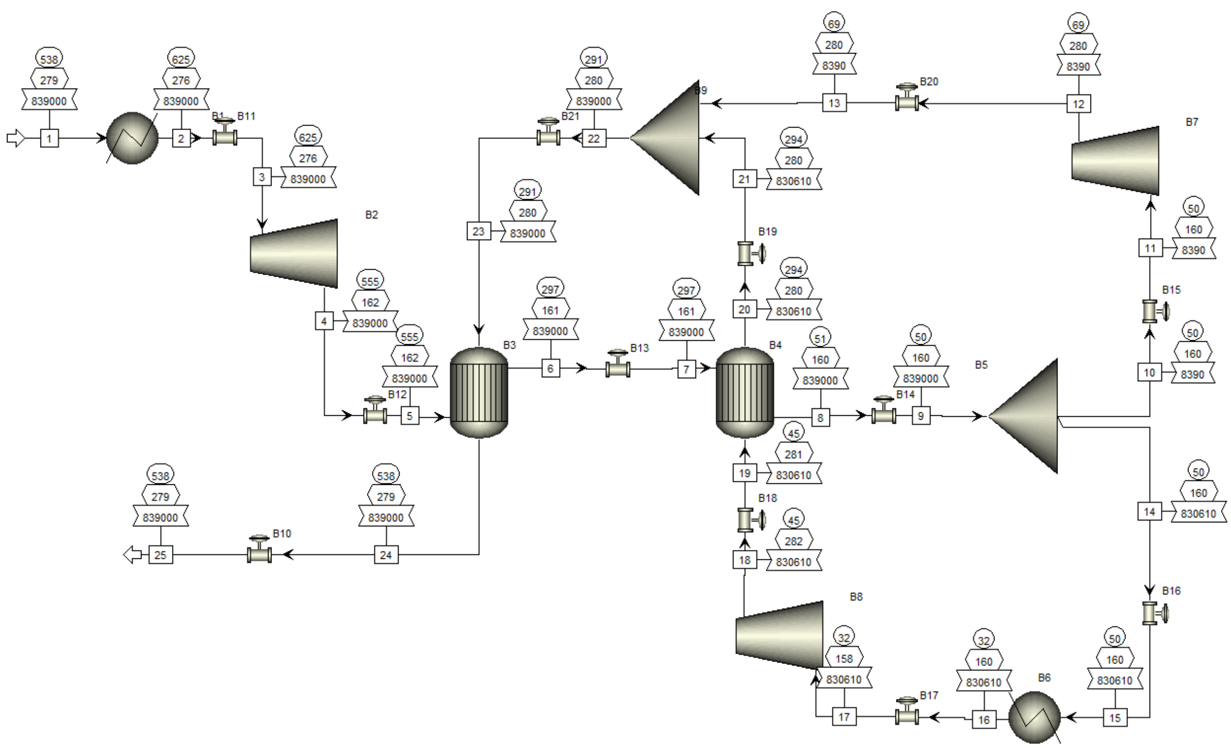
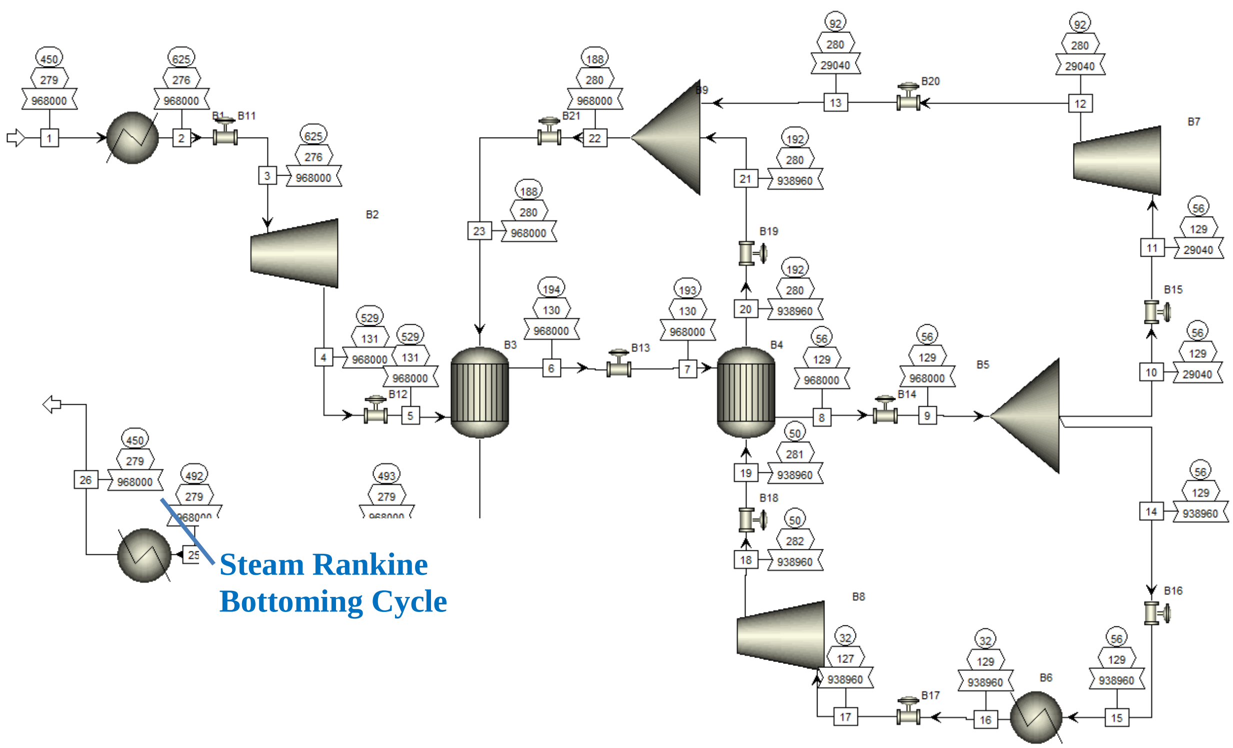


FIGURE 3.4-2: TOPPING CYCLE 1 WITH THE HEAT SOURCE TEMPERATURE SPLIT AT 538°C; 538-625°C CYCLE



The bottoming cycle configuration has both a sCO_2 cycle as the main cycle and a steam Rankine cycle to capture additional heat in the working fluid before it re-enters the heat source heat exchanger **Figure 3.4-3**. The turbine pressure ratio was altered to provide additional heat to the working fluid to be captured by the bottoming cycle. The full steam Rankine cycle was not modeled in Aspen Plus. It was represented by a heat exchanger, in which a curve fit of data from a steam Rankine power plant was used to calculate efficiency based on the inlet temperature to the heat exchanger. The inlet temperature to the heat exchanger was 492°C, which equates to a 44% efficiency for the bottoming cycle.

FIGURE 3.4-3: sCO2 CYCLE WITH A STEAM RANKINE BOTTOMING CYCLE



3.5 Cycle Efficiency

Cycle efficiency was calculated by taking the turbine power and subtracting the cycle compression power and dividing by the total heat input. Other auxiliary loads were not included as part of the cycle efficiency calculation.

3.6 Results

The results for the 4 main cycles and the topping and bottoming cycles are shown in **Figure 3.6-1**. The four main cycles, 1-4, follow the configuration of **Figure 3.2-1**. All have the same turbine pressure ratio, split to recycle compressor, and approach temperature for the recuperators. The total heat recuperated per MW of power produced is also constant for a given temperature range. Thus, for a given heat source temperature range, the only difference between a 10 and 30 MWe plant is the system mass flow rate. **Figure 3.6-1** indicates that a higher turbine inlet temperature leads to a higher cycle efficiency.

The bottoming cycle configuration has the same heat source temperature range, gross plant power, and recuperator approach temperature as Cycle 4. It also has a lower system mass flow rate and slightly higher cycle efficiency. However, plant efficiency and cost are not included in this analysis. The bottoming cycle requires more equipment, which means a larger capital cost, and a larger auxiliary load, which means a lower plant efficiency. Thus, the slight gain in efficiency may be outweighed by these effects.

The topping cycle configuration also shows a higher cycle efficiency as compared to Cycle 4. This is due to a lower thermal input required. However, as with the bottoming cycle, these topping cycle configurations require double the equipment units, leading to a higher capital cost, which may offset the efficiency gains.

For the small temperature ranges provided, a topping and bottoming cycle configuration may not be the best solution. This work does not include auxiliary loads in the efficiency calculation, nor was any cost analysis performed. Adding an additional cycle, whether an sCO2 or steam Rankine cycle, will increase those auxiliary loads and capital costs to potentially outweigh any efficiency benefit.

FIGURE 3.6-1: RESULTS SUMMARY TABLE FOR MAIN CONFIGURATION, TOPPING CONFIGURATION, AND BOTTOMING CONFIGURATION CYCLES

	Cycle 1	Cycle 2	Cycle 3	Cycle 4	Bottoming Cycle Config	Topping Cycle Config 1	Topping Cycle Config 2
Heat Source Temperature Range (°C)	300 - 450	300 - 450	450 - 625	450 - 625	450 - 625	450 - 625	450 - 625
Gross Plant Power (MWe)	10	30	10	30	30	30	30
Cycle Efficiency (%)	40	40	48.1	48.5	50.7	56.1	52.3
System Mass Flow Rate (kg/hr)	470,000	1,410,000	340,000	1,006,000	968,000	907,000 839,000	644,000 1,837,000
Total Heat Recuperated (MWth)	50	147	54	158.5	179	327	518
Turbine Pressure Ratio	2.58	2.58	2.58	2.58	2.11	1.78 1.7	2.14 1.28
Split to Recycle Compressor (%)	20	20	20	20	3	2 1	3 1
Approach Temperature Recuperators (°C)	5	5	5	5	5	5	5
Capital Cost	Nominal	Nominal	Nominal	Nominal	High	High	High

4.0 Recuperation and Recompression Parametric Study

4.1 Modeling Approach

The same modeling approach mentioned in *Section 3.1* was applied to the recuperation and recompression parametric cases with a few exceptions. The temperature approach of the recuperators was not held constant at 5°C, however was varied from 10-27°C for the recuperation cases. The inlet temperature of the heat source heat exchanger was allowed to float instead of being held at 425°C.

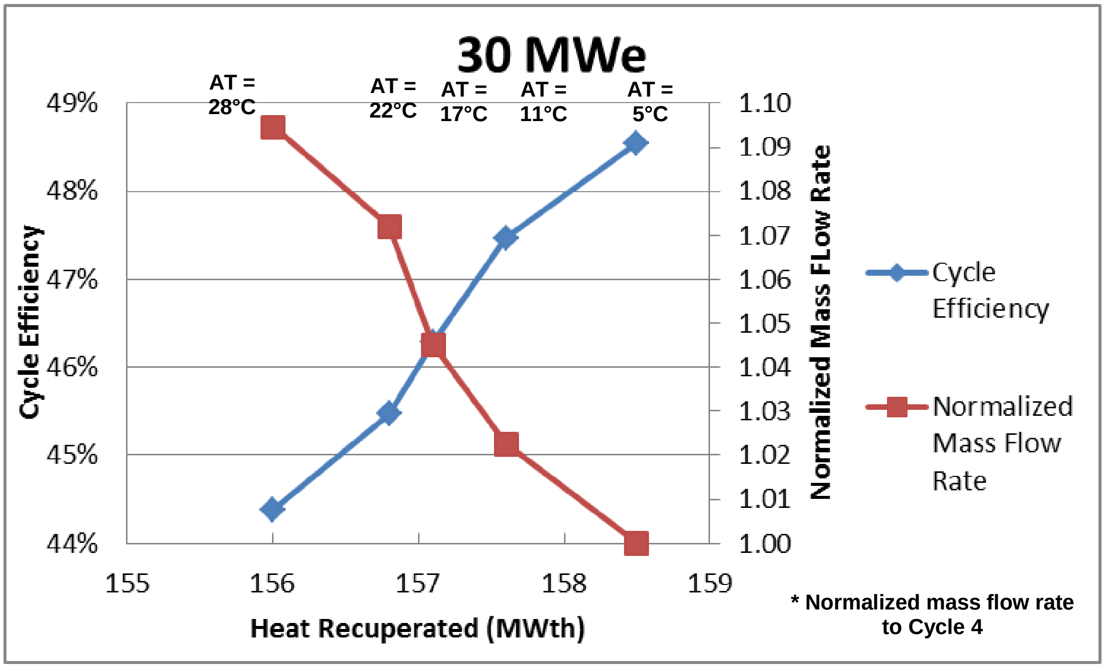
4.2 Recuperation Parametric Study

The double recuperation Brayton cycle has great benefit in splitting the sCO2 fluid between a main and recycle compressor. The purpose of splitting the flow between these two compressors is to reduce the compression power of the system, thereby increasing the overall cycle efficiency. Before the sCO2 enters the main compressor it is cooled to 90F, which is close to the critical temperature of CO2. The closer the CO2 conditions are to the vapor-liquid dome in a temperature-entropy diagram, the smaller the compression power required to bring it back up to system pressure. The limitation to the amount of flow that can enter the main compressor is set by the amount of heat that can be transferred from the "hot" to the "cold" sCO2 flowing through the low temperature recuperator, which is a function of the heat capacity of the sCO2 at those temperatures. If too high a flow is sent to the main compressor, there will be a pinch point issue with the recuperator. Previous cycle designs with lower temperatures than used in this study have flow streams with lower heat capacity and only 50% of the flow could be sent to the main compressor without having temperature crossover. Since the cycles addressed in this study have a higher temperature, the sCO2 streams have a higher heat capacity. Thus, more flow can be sent to the main compressor without having those issues.

The purpose of the recuperation parametrics was to evaluate the effect of heat recovery on cycle efficiency. One method of altering the amount of heat recovered was to vary the approach temperature. Thus, the approach tempeature was varied from 10-27°C. Increasing the approach temperature above 27°C led to pinch point issues within the recuperators and therefore was chosen as the upper limit. For each of the parametric cases run, the flow split between the main and recycle compressor was optimized. The system mass flow rate was adjusted for each case to achieve a 30 MWe plant output.

The results for the recuperation parametric study are shown in **Figure 4.2-1**. As expected, the greater the amount of heat recuperated, the higher the efficiency. Although a lower approach temperature provides only a slight increase in the amount of heat recovered, it leads to an efficiency increase of as much as 5%. The system mass flow rate also decreases by as much as 10%, which means cost savings in terms of piping and equipment units lowering capital cost.

FIGURE 4.2-1: RECUPERATION PARAMETRIC RESULTS FOR VARIOUS APPROACH TEMPERATURES



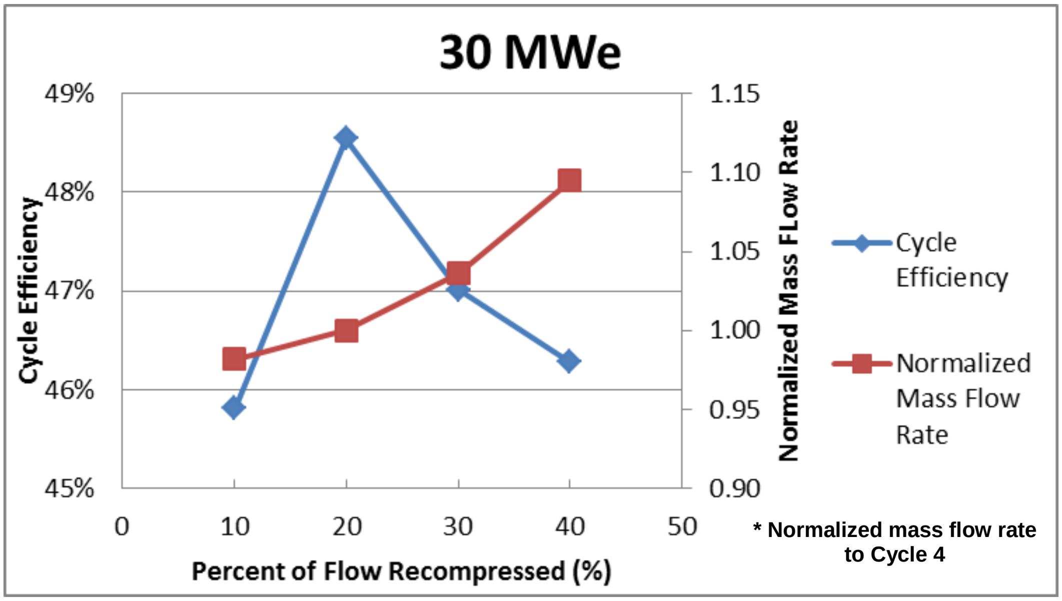
One trade that should be evaluated is the cost of the recuperator on the overall performance of the system. In terms of system performance, a lower approach temperature yields better results. However, a larger approach temperature decreases the cost of the recuperators. Therefore, a deficit in performance could be outweighed by the cost savings of the recuperator. Without taking the cost of the recuperators into account, the lowest reasonable approach temperature is 5°C, which was the approach used in the main configuration cycles.

4.3 Recompression Parametric Study

The recompression parametric study was performed to evaluate the effect of the flow split between the main and recycle compressor on cycle efficiency. The turbine pressure ratio was maintained at 2.6 as in Cycle 4. The split between the two compressors was varied at 10%, 30%, and 40% flow sent to the recycle compressor and the temperature of the sCO2 recycle to the heat source was allowed to float, which ranged from 436 - 458°C. The amount of recuperation was held constant among each case. The system mass flow rate was adjusted for each case to achieve a 30 MWe plant output.

Figure 4.3-1 shows the results for the recompression parametric study. The flow split that shows the highest cycle efficiency is at 20%, which is what was used in Cycle 4. At 20% recycle, the combined amount of compression work for both the main and recycle compressors is at a minimum. Decreasing the flow split from 20% led to a lower cycle efficiency. This lower cycle efficiency is due to a larger flow through the main compressor, which means a lower reheat temperature from the low temperature recuperator due to that higher flow rate. This then lowers the reheat temperature through the high temperature recuperator, decreasing the inlet temperature to the heat source heat exchanger. This lower inlet temperature increases the temperature delta across the heat source, causing the system to require more heat to reach the 625°C turbine inlet temperature, leading to a lower efficiency. Conversely, as you increase the percent of flow sent to the recycle compressor, the amount of flow through the main compressor is decreased. Thus, the reheat temperature through both the low and high temperature recuperators are increased and the temperature delta across the heat source heat exchanger is decreased. However, even though the temperature delta is decreased, there is an increase in system mass flow rate as you increase the recycle split to 40%. This increase in mass flow rate outweighs the lower temperature delta and leads to an increase in heat required to reach the 625°C, causing a decrease in cycle efficiency as indicated the right hand side of **Figure 4.3-1**.

FIGURE 4.3-1: RECOMPRESSION PARAMETRIC RESULTS FOR VARIOUS FLOW SPLITS BETWEEN THE MAIN AND RECYCLE COMPRESSOR



4.4 Recuperation and Recompression Parametric Study Results

Figure 4.4-1 demonstrates the results for all cases evaluated; four main configuration cases, one bottoming cycle configuration, two topping cycle configurations, four heat recovery recuperation parametric cases, and three recompression parametric cases. The recuperation cases illustrate that the more heat recuperated, the higher the efficiency. The recompression cases show that there is an optimum point of recompression in which the compression work for both the main and recycle compressors are at a minimum. This optimum point varies depending on the conditions of the system. Overall, the recuperation and recompression cases show a lower efficiency than Cycle 4, indicating that Cycle 4 was optimized.

FIGURE 4.4-1: RESULTS SUMMARY TABLE FOR ALL CASES EVALUATED

	Cycle 1	Cycle 2	Cycle 3	Cycle 4	Bottoming Cycle Config	Topping Cycle Config 1	Topping Cycle Config 2	Heat Recovery Parametric 1	Heat Recovery Parametric 2	Heat Recovery Parametric 3	Heat Recovery Parametric 4	Recompression Parametric 1	Recompression Parametric 2	Recompression Parametric 3
Heat Source Temperature Range (°C)	300 - 450	300 - 450	450 - 625	450 - 625	450 - 625	450 - 625	450 - 625	450 - 625	450 - 625	450 - 625	450 - 625	436 - 625	451 - 625	458 - 625
Gross Plant Power (MWe)	10	30	10	30	30	30	30	30	30	30	30	30	30	30
Cycle Efficiency (%)	40	40	48.1	48.5	50.7	56.1	52.3	47.5	46.3	45.5	44.4	45.8	47	46.3
System Mass Flow Rate (kg/hr)	470,000	1,410,000	340,000	1,006,000	968,000	907,000 839,000	644,000 1,837,000	1,030,000	1,052,000	1,080,000	1,102,000	990,000	1,043,000	1,102,000
Total Heat Recuperated (MWth)	50	147	54	158.5	179	327	518	157.6	157.1	156.8	156	156	158	160
Turbine Pressure Ratio	2.58	2.58	2.58	2.58	2.11	1.78 1.7	2.14 1.28	2.58	2.58	2.58	2.58	2.58	2.58	2.58
Split to Recycle Compressor (%)	20	20	20	20	3	2 1	3 1	27	31	35	37	10	30	40
Approach Temperature Recuperators (°C)	5	5	5	5	5	5	5	10	15	20	25	5	5	5
Capital Cost	Nominal	Nominal	Nominal	Nominal	High	High	High							

5.0 Thermodynamic Diagrams: Cycle Configuration Comparisons

The remaining process flow diagrams, Temperature-Entropy diagrams, and Pressure-Enthalpy diagrams for the other cases are shown in *Appendix A.*

6.0 sCO2 System Operation and Auxiliary System Considerations

The challenges and opportunities for commercial deployment are discussed in detail in the following summaries of risk by category. Challenges (risks) were considered in design, availability (supply), costs, manufacturing, construction, and operation.

Opportunities highlight any advantages for these same criteria as mentioned before and will also assess the cost/benefit relationship.

Challenges (Risks)

Risks in the design may be related to design of the sCO2 process or related to the technology supporting the sCO2 process. Design-specific and technical-specific risks may include, but are not limited to, the risks shown below and were considered as based on AREVA Fuels sCO2 experience.

Design-Specific Risks

- Inadequate data to support design
- Designer/detailer experience
- Design inadequacies in function
- Detail, precision, and suitability of the design or equipment specifications
- Likelihood of changes to design after work started, especially with safety interpretation
- Design issues caused by or causing issues with execution methods
- Sheer size or complexity of the project

Technical-Specific Risks

- Complexity introduced as a result of new technology increases, installation, and operating costs
- Performance of the technology in a commercial environment
- Quality assurance possible with the technology
- Rate of production possible with the technology
- Reliability and total operating efficiency (up time) of the technology
- Risks specific to project's technology implementation during fabrication and installation and operation of the technology

Supply Risks

Selected supply risks related to the sCO2 technology may include, but are not limited to:

- Cost and schedule impacts due to fabrication with specialty metals and methods
- Suppliers delivery schedule may affect project schedule, especially that of extruded piping
- Difficulty in obtaining high pressure gaskets and seating materials
- Cost risk if special instrumentation is not available and must be fabricated

Cost Risks

Selected cost risks related to the sCO2 technology may include, but are not limited to:

- Management and/or workforce experience
- Lack of sCO2 experience and understanding mechanical process
- Underestimation of scope
- Changes due to First-of-a-kind (FOAK) implementation and unknowns
- Changes in operational strategy or safety expectations
- Unforeseen site conditions
- Sponsor/user scope changes
- Start-up, turnover, and/or launch difficulties
- Inadequate planning or unrealistic scheduling
- Once pressurized, lack of access increases downtime due to safety enclosures
- Lack of dedicated project and operations team

Operating Risks

Selected operating risks related to the sCO2 technology may include, but are not limited to:

- Reduced operating efficiency as the result of leaks and shutdown for repairs
- Equipment failure and downtime for repairs result in reduced operating efficiency
- Operating efficiency reduced due to startup and shutdown issues

Construction Risks

Construction risks that may be experienced during sCO2 project execution are as follows:

- Resource risks - Staff changes and lack of good organizational structure
- Legal risks; including license, patent, contract, permits, regulatory, and proprietary information issues
- Delays occur due to licensing or permitting delays
- Inadequate project management

Operational Risks

CO2 exposure risks, and resulting consequences, must be considered for the fastest releasing failure in the supercritical CO2 loop and/or associated process equipment. Consequences and probabilities must both be analyzed and mitigated as appropriate. Analyses can be done in a number of ways such as Integrated Safety Analysis (ISA), Probabilistic Risk Assessment (PRA) or other appropriate methodology. Engineered and passive controls (barriers), or administrative controls, are all used to reduce risk. High consequence events, generally described for the purposes of this study, are those that could lead to fatality.

Employees can be exposed to high concentrations of carbon dioxide (CO_2). Such exposures can be mitigated by quick annunciation and egress. Analyses must estimate the maximum concentration expected and time for egress. This risk can have a variety of consequences depending on the CO2 concentration and leak location.

Minimizing exposure time will mitigate the consequences. Analyses must have conservatisms built in for safety to be assured. The warning properties of the released CO2, and therefore other components, are excellent. CO2 will have visual and auditory warnings of a line or vessel leak. Risk exists that egress could be difficult depending on the location of the leak. However, this can also be mitigated by providing equipment enclosures.

An ISA of all hazards is needed for the sCO2 process. Any leaks from piping or components can pose significant risk to personnel due to the extreme heat and high pressure. Each general condition will likely have bounding initiating event scenarios. Provided below are examples of mitigated risks and small release risks for cases where enclosures and ventilation are provided.

Example Scenario #1

Vessels suffer a catastrophic failure and leak: This is assumed to blow off some enclosure panels and release all the contents into the room virtually instantaneously. Room levels of CO2 will exceed high consequence exposure levels. Assuming no escape, asphyxiation occurs. The mitigating actions, including alarms and quick egress can help ensure low accident consequences.

Example Scenario #2
A process line in the loop or auxiliary system breaks or backflows through other lines to release CO2 into an occupied area or enclosure: CO2 is released into the enclosure. The CO2 leak rate exceeds the ventilation capacity and leaks into the occupied area. CO2 concentrations may rise to high consequence exposure levels. Assuming no escape, asphyxiation occurs. The mitigating action includes alarm and quick room evacuation, resulting in low consequence.

Example Scenario #3

CO2 gas is released into the exhaust hood from a pipe break or a pump failure in the liquid CO2 feed system: The resulting release could exceed enclosure ventilation capacity. At 3,000 psi the incoming CO2 flow rate would be high and if less than the exhaust rate, no appreciable CO2 escapes the exhaust hood. This would be a low consequence event. But if the release rate exceeds the exhaust rate then **Scenario #2** could play out.

Example Scenario #4

CO2 and filtered solids are released into an enclosure: Dust from this release should be easily controlled by exhaust ventilation and would be a low consequence event. However, if **Scenario #2** plays out where there is CO2 leak rate that exceeds the enclosure exhaust capacity.

Example Scenario #5

Potential high-pressure injection of sCO2 fluid that might result from pinhole leaks in tanks or tubing: The likelihood of this risk should be investigated. However, the use of an appropriate protective barrier, such as an exhaust hood or other enclosure, makes it a highly unlikely risk. In the event that there is such an injection, it may very well result in a high consequence event, though the probability is low due to sCO2 and CO2 gas properties.

Auxiliary Systems Description and Considerations

In addition to the sCO2 Brayton Cycle loop, there are numerous auxiliary systems which are required in order to support it.

A general flow diagram is provided that provides a high level view of such auxiliary systems and how they might be arranged is shown in **Figures 6-1, 6-2, and 6-3. Figure 6-3** is a conceptual facility layout indicating approximate space requirements. **Table 6-1** offers a list of the auxiliary systems that should be considered with the loop.

FIGURE 6-1: AUXILIARY SYSTEMS FLOW DIAGRAM (PAGE 1)

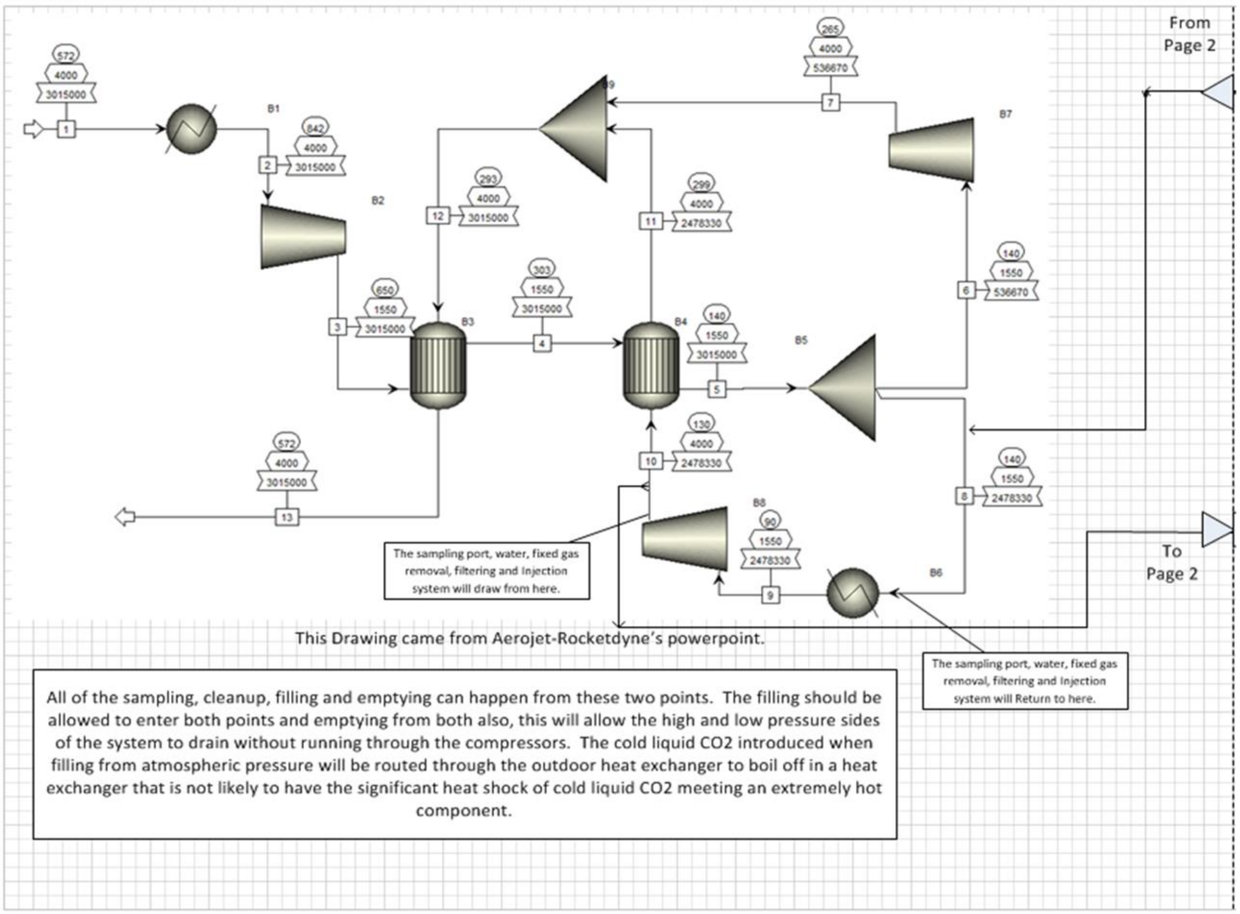


FIGURE 6-2: AUXILIARY SYSTEMS FLOW DIAGRAM (PAGE 2)

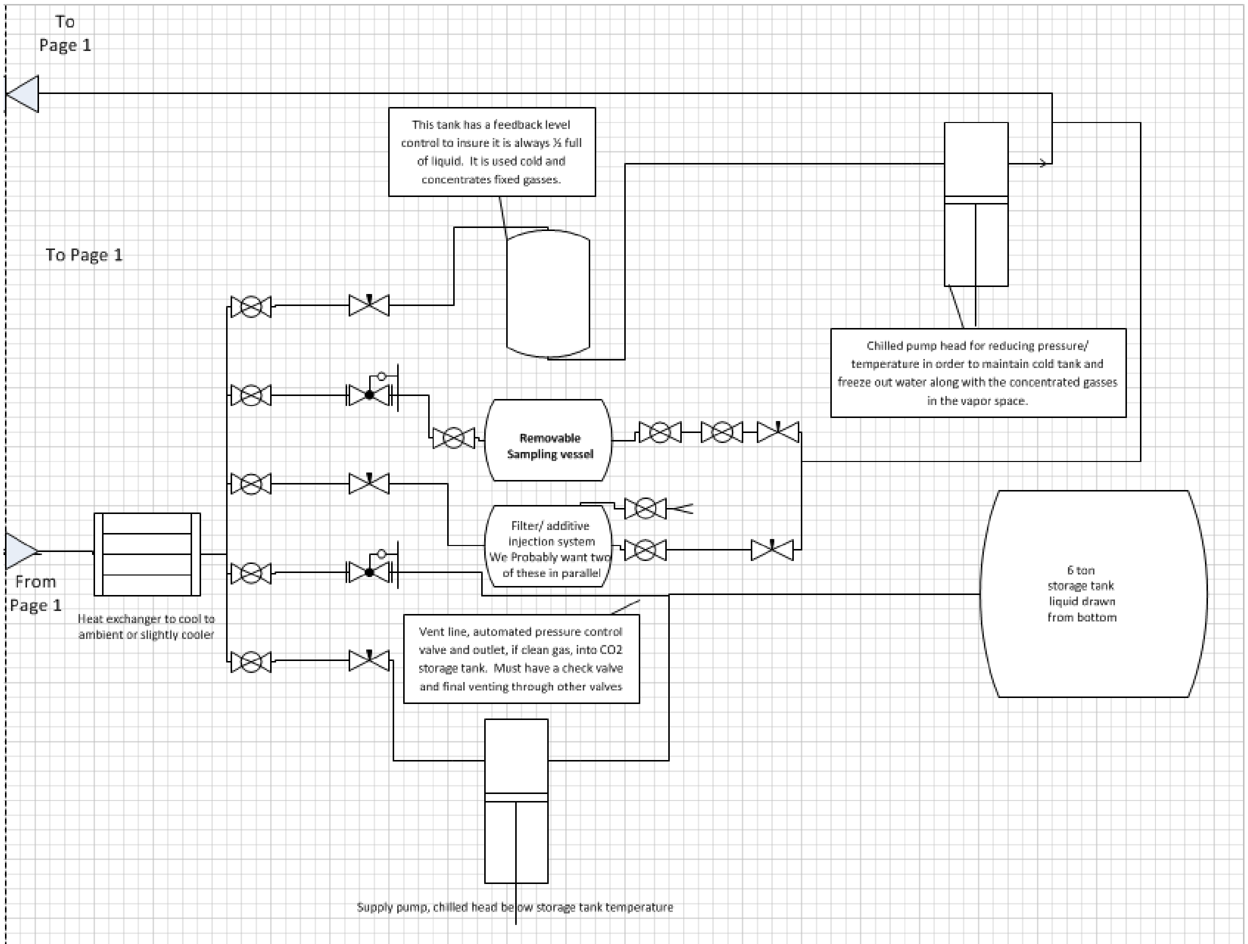


FIGURE 6-3: CONCEPTUAL SCO2 FACILITY LAYOUT

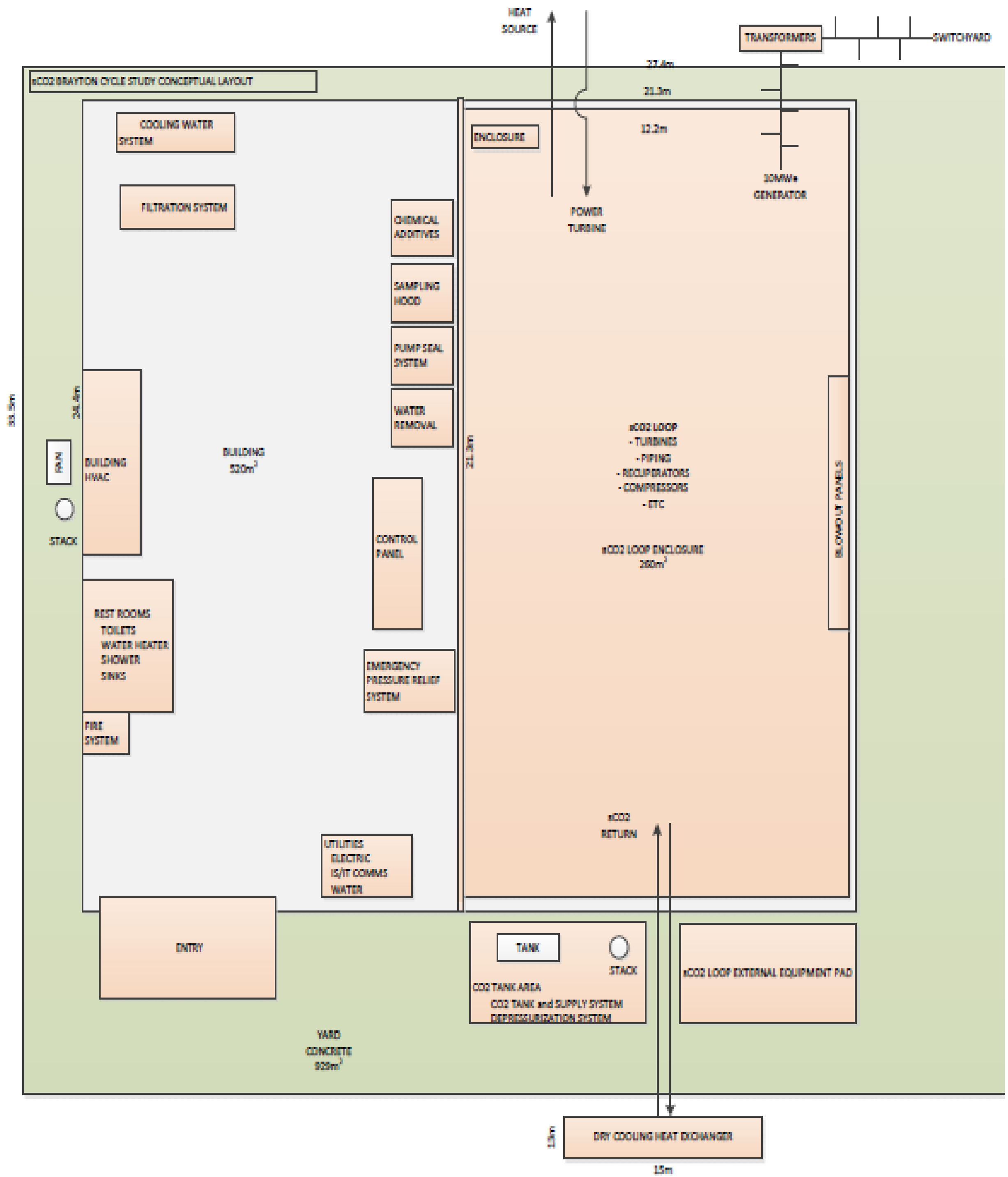


TABLE 6-1: AUXILARY SYSTEM CONSIDERATIONS

Auxiliary System Considerations
Heat Sink
CO2 Feed System
Pump Seal System
Pressure Management System
Pressure Control System
Depressurization System
CO2 Recovery System Option
Standby CO2 Surge Management System Option
Pressure Relief System
CO2 Quality Management System
Particulate Filtration System
Chemical Additive System
Sampling System
Water and Gas Removal System
Process Control Considerations
Flow Measurement
Leak Detection
Pressure Measurement
CO2 Monitors/Interlocks
Cooling Water System Option
Ventilated Enclosures
Building, HVAC, Yard, Utilities

Heat Sink

The assumed heat sink for the sCO2 cooler is supplied by dry cooling. The dry cooling system considered in this study is provided by fan driven air flow across finned tubes containing the sCO2 flow. This allows it to be cooled before entering the main compressor. The required air flow will, of course, vary depending on the ambient air conditions. Cooling system control will involve turning fans off and on to dissipate heat and keep the sCO2 discharge from the cooler at the required temperature. The cooler will require adequate space and separation from other structures to allow sufficient air flow and minimize fan noise impact. The estimated cooler footprint is 15 m in length and 13 m in width as illustrated in **Figure 6-3**.

CO2 Feed System

The system is filled from a bulk storage tank containing about 6 tons of CO2 filled to no more than 5 tons and operated at 250 psig using an oversized chiller. Flow meters will be used to measure gas or liquid flows.

CO2 delivery pumps sized to about 5 HP and rated at 5000 psig will be used to deliver up to 20 kg/min. for filling, after pressure has equilibrated. This system will also replace CO2 lost through leaks or sampling so it must have a smooth startup and shutdown.

Pump Seal System

If auxiliary system liquid pumps that contain seals are used a pressurized system seal lubrication system is recommended. To reduce leakage and to protect the seals' clean cool liquid, CO2 would be injected into auxiliary pump seals.

Pressure Management System

The pressure management system includes pressure control system, a depressurization system, an optional CO2 Recovery System, an optional standby surge management option, and a pressure relief system.

- **Pressure Control Systems** - The pressure is controlled primarily by adjusting heat input and economizer flow ratio. An air-to-close valve should be used to vent excess system pressure. This valve will open upon loss of system power or air pressure, as well as, act as a safety feature. The cool section of the high-pressure system is the best location for this control.
- **Depressurization System** - Venting for shutdown is needed to remove the sCO2 from the loop. This system will vent sCO2 to a header. The header may be equipped with CO2 in the event of a control system failure, the header should also be equipped with a hand valve placed in a convenient outdoor location.
- **CO2 Recovery System Option** - If CO2 recovery is desired, it must be pressurized, chilled to ambient temperature, and de-pressurized to about 350 psi. This will condense about half of it and the other half may be re-injected into the CO2 storage tank where it can be partially liquefied using the tank refrigeration system. Excess CO2 will then be vented from the tank to atmosphere. With this approach, the system will be capable of recovering about 2/3 of the CO2.
- **Standby sCO2 Surge Management Option** - If standby storage for sCO2 surge management is needed it should be in the flow path on the low-pressure section of the economizer loop. It should be cool enough to use less expensive alloys and hot enough to provide a pressure leveling response.

- **Pressure Relief System** - To allow the CO2 to expand and cool as it approaches the valve during venting, each pressure relief valve should be capable of venting more than the pipe feeding it. To permit the relief valves to be rated for lower temperatures, the pressure relief system can be built with standoffs. Vent sizing is primarily a function of heat input during upset conditions. Vents do not capture any of the vented gasses. Each vent valve, or rupture disk, is supplied for each isolatable volume or section of the system. The loop may contain 3 or 4 isolatable volumes, and several minor isolatable volumes, in the auxiliary system. If the vents combine into a common header, the backpressure during a vent will affect flow or relief pressure on adjacent circuits. It is much simpler to vent each relief valve with its own pipe.

CO2 Quality Management System

- **Particulate Filtration System -** The contaminant management system relies on filtration to remove any metal oxides and other solids. The entire volume should be filtered at least once per day. The entire loop need not be directly filtered. Using a circulation pump that removes a slip stream of CO2 from the coldest part of the system, the slip stream of CO2 can be filtered and delivered clean downstream to a section at lower pressure. The pump will draw the CO2 through the filter in order to avoid building up excess pressure. This pump may be a centrifugal pump with a magnetic coupling to make it seal-less.
- **Chemical Additive System Option -** This is an optional system that provides a means of chemical addition for research and development (R&D) purposes. Such chemicals would be circulated in the sCO2 loop. For example, this could support studies of conditioners, corrosion inhibitors, leak detection tracers and/or other additives.
- **Sampling System -** This is an optional system to collect, cool, and sample sCO2 from the loop for R&D purposes. Samples can be collected to test the level of contaminants, such as particulates or material degradation products that are contained in the operating fluid.
- **Water and Gas Removal System-** Water and entrained gases can be removed by allowing a portion of the low temperature CO2 to undergo partial depressurization and re-heating, as well as to re-condense in order to separate the water and gas from the CO2. The purified CO2 is then pumped back into the system. The loop will be vented and can then be heated to remove water and gas from the system.

Process Control Considerations

The selected process control considerations shown below include flow measurement, leak detection, and pressure measurement.

- **Flow Measurement -** Circulating flow in the system can be measured by placing orifices in the lines and measuring the pressure drop across the orifices. This will not work if the flow is two-phase and will be less sensitive once in the supercritical range. Ultrasonic flow meters will work if setup properly but the speed of sound in supercritical CO2 is much slower than steel so the impedance mismatch is not easy to overcome. Although, coriolis meters are effective due to their ability to correct for density, in large sizes they are expensive.
- **Leak Detection -** Pipes very rarely leak over their length, however, the connections are the most common leak points. A sniffer-tube can be placed adjacent to each joint and allowed to pass through the containment panels and insulation. A portable hand-held infrared CO2 monitor can be used to detect leaks in these joints or other potential leak sources. The room, exhaust hoods, and containment areas should also be monitored for CO2.
- **Pressure Measurement -** The pressure measurement system should include taps for pressure sensing lines at the bottom of each vessel. These taps will allow pressure data to be collected by the process control system. These same lines will be used to drain water out after hydro-tests.
- **CO2 Monitors and Interlocks** - CO2 monitors will be placed in key areas of the loop, auxiliary systems, and worker-occupied areas. Alarms will sound if CO2 is detected in excess of alarm set points. This will alert workers to respond per appropriate operating procedures.

Cooling Water System Option

A water chiller is recommended in order to provide cooling to auxiliary equipment where needed. The system includes a refrigeration unit, water tank, and piping system to deliver chilled water to equipment and coils. Heat exchangers will be used to remove heat.

Ventilated Enclosures

All worker occupied and enclosed areas should have ventilation system intakes located in or near the floor in order to remove CO2 quickly. CO2 is denser than air and will settle near the floor. This system may leak a little, especially through seals, and any leak will release CO2. Therefore, ventilation intakes designed to remove CO2 can also be placed near the areas prone to seal leaks.

Building, HVAC, Yard, and Utilities

A steel building with pre-fabricated sandwiched panels equipped with at least 5,600 sq. ft. is envisioned for the loop and auxiliary systems. The building will be sited on a 10,000 sq. ft. site, consisting mostly of a concrete pad for the sCO2 process building and auxiliary systems. The building will be equipped with a heating, ventilation, and air conditioning (HVAC) unit capable of serving the building and the sCO2 process. The building will be equipped with blast doors in the event of a sudden sCO2 release.

7.0 Material Selection and Associated Cost Estimate

7.1 Material Selection Approach

The material selection had the principle aim to propose adequate materials that both provide the necessary technical properties required by the cycle boundary conditions and are codified according to the stipulations of the ASME code. From a technical point of view, the material pre-selection of the sCO_2 piping system was based on the service conditions with two maximum system temperatures being 450 °C and 625 °C, respectively. In that context, both the outer pipe surface being in contact with ambient air and the inner pipe surface being in contact with sCO_2 were taken into consideration with adequate allowances.

It is important to understand that the relatively early state of the study, where certain aspects of the Brayton Conversion Cycle are not yet technically finalized, necessitated certain assumptions in order to achieve the goals of this subtask. The assumptions have been made with focus being placed on reasonable material recommendations and a rough order of magnitude cost estimation for the piping. Additional details about the assumptions are summarized in *Section 7.2*.

7.1.1 General Remarks to Applicable ASME Code

All pressure vessels that are within the sCO2 cycle will be specified and fabricated per the ASME Section VIII code. Since piping sections of the cycle are the primary focus for this sub-task, the relevant construction code was identified to be ASME B31.1 covering power piping. This construction code defines the minimum requirements that need to be met for the construction of pressure retaining piping, therefore stipulating the design conditions and allowable materials.

The material recommendations that were prepared made it necessary to consider the structural integrity of certain alloys for piping. This is given that relatively high pressures at high temperatures might prevail depending on the actual position in the cycle. At this point, a first dimensioning had to be performed where the operational boundary conditions, in combination with so-called allowable stress levels, result in inner diameter and wall thicknesses. This was done by using the minimum required internal diameter for mass flow purposes (input data that was provided by thermo-hydraulic considerations) and the sizing approach as defined in ASME B31.1 paragraph 104.1.2. As the input data for internal diameters was solely based on thermo-hydraulic boundary conditions, the resulting dimensions did not necessarily represent typical piping dimensions that can be considered readily available. Hence, a certain pooling of dimensions was performed to limit the number of diameter-wall thickness combinations and to facilitate procurement and minimize costs.

7.1.2 General Remarks to the Consideration of Ageing

A general screening of the Brayton layouts with the relevant system boundary conditions revealed that the following are major potential degradation mechanisms for ageing:

- Thermal ageing
- Creep
- Corrosion
- Fatigue

However, a detailed analysis was not performed in the frame of this task. Nevertheless, a rough screening of open published literature revealed that there is a lack of research that studies complex ageing mechanisms such as; environmentally assisted cracking, including stress corrosion cracking; and environmentally assisted fatigue (corrosion fatigue). This may be an indication for the nil-existence of these mechanisms in this specific environment, or lacking total operational time, as these mechanisms mostly reveal sluggish reaction kinetics and thus low velocities.

As lifetime and number of operating cycles (i.e. number of start-ups) of a demonstration plant are low as compared to commercial plants, the consideration of ageing is considered less important, except when safety issues are concerned. However, this will be an issue of detailed plant design.

Consequently, at this stage of basic design, the predominant ageing mechanisms of concern, which are necessary as input data for cost estimations, include any type of global loss of wall thickness due to general corrosion attack of the surface. Under the conditions of concern, this consideration must comprise both inner and outer wall surface. The inner surface might be attacked by sCO2 causing the mechanism of metal dusting. At higher temperatures (> 500 °C) oxidation by gaseous oxygen contained in air might also lead to significant loss of metal due to high-temperature oxidation.

As the pooling of dimensions to meet standard sizes will lead to slightly larger dimensions than needed, based on purely thermodynamic requirements, a margin for other ageing mechanisms may be included.

7.1.3 Determination of Corrosion Allowance

A study of the literature revealed that only limited data for corrosion behavior of carbon steel in supercritical CO_2 is available. Only one reference of concern was found, which reported testing of a X65 carbon steel at 50 °C in water saturated CO_2 at pressures from 40 to 80 bar [2]. The corrosion rate at only one parameter set for supercritical conditions (50 °C, 80 bar) was determined to be approximately 0.5 mm/y. For low-alloy steels, no literature references were found for steels with a Chromium content in the desired range of 1.25 to 2.5 wt.-%. Therefore these were investigated only by general low alloy steel reference material. Typical investigated materials for high temperature service in supercritical CO_2 systems are 9 wt.-% Cr steels like P91 and P92, as well as 12 wt.-% Cr steels like HCM12A. Surprisingly enough, HCM12A higher corrosion rates at 650 °C compared to P 91 were found. CO_2 at 650 °C and 207 bar, resulted in a corrosion rate of 0.18 mm/y for P 91 and 0.3 mm/y for HCM12A [3]. For another 12 wt.-% Cr steel a lower corrosion rate of 0.02 mm/y at 600 °C and 200 bar supercritical CO_2 was measured [4]. In another study the corrosion resistance of P92 in sCO2 at 200 bar and 450 °C and 550 °C was examined. The raise of temperature from 450 °C to 550 °C resulted in a ten times higher corrosion rate of 0.2 mm/y at 550 °C, compared to 0.02 mm/y at 450 °C [5].

Another study examined the corrosion behavior of austenitic stainless steel (316SS) and nickel base alloys (718, 738 and 625) in supercritical CO_2 environment at 550 °C and 700 °C. It was found that corrosion rates at 550 °C are negligible for both austenitic and nickel base alloys. At 700 °C a corrosion rate of 0.04 mm/y was measured for 316SS. The nickel base alloys did not show any significant corrosion [6].

The scaling behavior of carbon steels and low alloy steels were extensively examined at Siemens in the 1990's. As per these studies, the scaling tendency depends on the Cr content and the type of operation that can be distinguished between continuously or cyclic. For cyclic operation the parabolic scaling rules cannot be applied and a linear time dependency of the wall thickness losses must be considered. This change in dependency is attributed to scale spalling. For an experimental application with numerous parameter tests and start-ups, the effect of spalling must be considered. The scaling and spalling effects on SA-106 grade B and SA-335 grade P11 and P22 with the consequent loss of wall thickness is summarized in **Table 7.1-1**.

Considering the above and the results for carbon and low alloy steels shown in **Table 7.1-1** the following conclusions can be drawn:

- The use of carbon and low alloy steels (such as 1.25-2.5 chromium alloy), in an experimental supercritical CO_2 cycle plant, is only suitable at < 575 °C
- At temperatures above 500 °C, the corrosion allowance on the external surface (outer diameter) for carbon and low alloy steels is set with 2 mm (valid for discontinuous experimental operation within 10 years)
- The corrosion allowance on the internal surfaces (inner diameter) is set with 2 mm due to metal dusting. (this is valid for discontinuous experimental operation within a period of 10 years)

TABLE 7.1-1: SCALING OF CARBON AND LOW ALLOYED STEEL AT AIR, RELEVANT FOR EXTERNAL SURFACES.

Material	Estimation of wall thickness reduction due to scaling in mm after constant or cyclic operation at the outer surface in contact with air						
	100 h	1000 h	10,000 h	10 x 100h= 1000h	100 x 100 h	10 x 1000 h	Temp. in °C
106B	0.0050	0.018	0.05	0.05	0.25	0.18	550
P11	0.0037	0.010	0.035	0.037	0.185	0.10	550
P22	0.0035	0.010	0.030	0.035	0.175	0.10	550
106B	0.0085	0.026	0.085	0.085	0.425	0.26	575
P11	0.0095	0.030	0.095	0.095	0.475	0.30	575
P22	0.0045	0.017	0.045	0.045	0.225	0.17	575
106B	0.024	0.076	0.24	0.24	1.2	0.76	600
P11	0.033	0.105	0.33	0.33	1.65	1.05	600
P22	0.0075	0.025	0.075	0.075	0.375	0.25	600
106B	0.047	0.15	0.47	0.47	2.35	1.5	625
P11		--					625
P22	0.047	0.15	0.47	0.47	2.35	1.5	625
106B	0.13	0.35	> 1.0	1.3	6.5	3.5	650
P11							650
P22	0.13	0.35	> 1.0	1.3	6.5	3.5	650

In the next step of material selection the operational stresses for four system parameter sets were evaluated:

1. 450 °C, 10 MW
2. 450 °C, 30 MW
3. 625 °C, 10 MW
4. 625 °C, 30 MW

The resulting operating parameters (pressure, temperature and flow rate) are addressed in *Section 3* of this report.

The materials listed in **Table 7.1-2, 7.1-3, 7.1-4, and 7.1-5** were selected based on the allowable stresses of B31.1, the scaling of the external surface at ambient air and the corrosion of the inner pipe surface in supercritical CO_2. The material selection also considers a generally sound weldability. For some parameter combinations a lower material grade might be used. However, by doing so the consequential higher wall thickness would be counter-productive particularly when talking about market availability, weldability, and installation time (e.g. material P92 for 275 bar and 625 °C resulting in a wall thickness of 117.25 mm). The material selection is only valid for experimental set-up with a few thousand hours of operation and is not intended for long-term commercial operation. The minimum wall thickness listed in **Table 7.1-2, 7.1-3, 7.1-4, and 7.1-5** does not include any corrosion allowance.

TABLE 7.1-2: SUPERCRITICAL CO2 CYCLE WITH 10 MW AT 450 °C

No.	T in °C	p in bar	Inner diameter in mm	Allowable Stress in MPa	Material Grade	Minimum wall thickness in mm
1	300	275	173.1	164.0	SA-335 P91	16.1
2	450	275	215.9	141.0	SA-335 P91	23.8
3	343	107	284.2	161.0	SA-335 P91	9.8
4	150	107	215.9	118.0	SA-106 grade B	10.4
5	60	107	173.1	118.0	SA-106 grade B	8.3
6	60	107	66.7	118.0	SA-106 grade B	3.2
7	129	275	54.0	118.0	SA-106 grade B	7.3
8	60	107	131.8	118.0	SA-106 grade B	6.3
9	32	107	87.3	118.0	SA-106 grade B	4.2
10	54	275	87.3	138.0	SA-106 grade B	9.9
11	148	275	109.6	138.0	SA-106 grade B	12.4
12	146	275	131.8	138.0	SA-106 grade B	14.9
13	300	275	173.1	164.0	SA-335 P91	16.1

TABLE 7.1-3: SUPERCRITICAL CO2 CYCLE WITH 30 MW AT 450 °C

No.	T in °C	p in bar	Inner diameter in mm	Allowable Stress in MPa	Material Grade	Minimum wall thickness in mm
1	300	275	284.2	164.0	SA-335 P91	26.5
2	450	275	490.6	144.0	SA-335 P92	52.9
3	343	107	490.6	161.0	SA-335 P91	17.0
4	150	107	366.7	118.0	SA-106 grade B	17.6
5	60	107	257.2	118.0	SA-106 grade B	12.3
6	60	107	109.6	118.0	SA-106 grade B	5.3
7	129	275	87.3	118.0	SA-106 grade B	11.8
8	60	107	257.2	118.0	SA-106 grade B	12.3
9	32	107	173.1	118.0	SA-106 grade B	8.3
10	54	275	173.1	138.0	SA-106 grade B	19.6
11	148	275	215.9	138.0	SA-106 grade B	24.4
12	146	275	215.9	138.0	SA-106 grade B	24.4
13	300	275	284.2	164.0	SA-335 P91	26.5

TABLE 7.1-4: SUPERCRITICAL CO2 CYCLE WITH 10 MW AT 625 °C

No.	T in °C	p in bar	Inner diameter in mm	Allowable Stress in MPa	Material Grade	Minimum wall thickness in mm
1	450	275	173.1	141.0	SA 335 P91	19.1
2	625	275	215.9	173.0	SB 444 N06625	18.6
3	504	107	257.2	122.0	SA 335 P91	11.8
4	152	107	215.9	118.0	SA 106 grade B	10.4
5	60	107	131.8	118.0	SA 106 grade B	6.3
6	60	107	54.0	118.0	SA 106 grade B	2.6
7	129	275	42.9	118.0	SA 106 grade B	5.8
8	60	107	109.6	118.0	SA 106 grade B	5.3
9	32	107	87.3	118.0	SA 106 grade B	4.2
10	54	275	87.3	118.0	SA 106 grade B	11.8
11	150	275	109.6	118.0	SA 106 grade B	14.8
12	146	275	109.6	118.0	SA 106 grade B	14.8
13	450	275	173.1	141.0	SA 335 P91	19.1

TABLE 7.1-5: SUPERCRITICAL CO2 CYCLE WITH 30 MW AT 625 °C

No.	T in °C	p in bar	Inner diameter in mm	Allowable Stress in MPa	Material Grade	Minimum wall thickness in mm
1	450	275	284.2	141.0	SA 335 P91	31.4
2	625	275	325.5	173.0	SB 444 N06625	28.1
3	504	107	450.9	122.0	SA 335 P91	20.7
4	152	107	325.5	118.0	SA 106 grade B	15.6
5	60	107	215.9	118.0	SA 106 grade B	10.4
6	60	107	109.6	118.0	SA 106 grade B	5.3
7	129	275	87.3	138.0	SA106 Grade C	9.9
8	60	107	215.9	118.0	SA 106 grade B	10.4
9	32	107	131.8	118.0	SA 106 grade B	6.3
10	54	275	131.8	138.0	SA 106 Grade C	14.9
11	150	275	173.1	138.0	SA 106 grade C	19.6
12	146	275	215.9	138.0	SA 106 grade C	24.4
13	450	275	284.2	141.0	SA 335 P91	31.4

For the 450 °C cycle, the complete primary system can be made of carbon steel, except for the pipe spool between heat source and turbine. The allowable stresses for carbon and low alloyed steels are not sufficiently high enough. Material grade P91 will be necessary for the spool between heat source and turbine.

For the 625 °C cycle, carbon steels can be used only for the system parts below 150 °C. For the other system parts 2.5 wt.-% Cr steel (P22) and 9 wt.-/ Cr steels (P91) must be used. Between the heat source and the turbine, the allowable strength of P91 is not sufficient to assure a pipe with an appropriate wall thickness that can still be welded. In that instance, a nickel based alloy 625 is the material of choice.

7.2 Cost Estimation from a Materials Point of View

The cost estimation for materials required the use of certain assumptions as the level of information available represented a pre-basic design state. The assumptions shown below were separated into general assumptions, as well as assumptions for delivery and erection. Anything not explicitly mentioned below can be considered as not taken into account.

General Assumptions:

- The version with 10MWe power and 625°C shall be the basis for the cost estimation
- Only seamless pipes, including fittings and valve bodies were considered
- No components, such as compressors or pumps, were taken into account
- Also excluded in the estimates are civil work, supporting structure, measuring systems (including sensors, make-up, auxiliary or supporting systems), draining, I&C, etc.
- The distance between major components for piping cost estimating is assumed as approximately 12 meters (approximately 38 ft)
- Due to the high temperatures, material selections will have accelerated deterioration. Therefore, to assure safe operation, the 10MWe cycle is assumed to operate as an intensive test facility with a maximum of 200 starts over a 10 year span
- Some material replacement will be necessary if the number of starts is exceeded or operation extends beyond 10 years

TABLE 7.1-6: COST ESTIMATION IS BASED ON THE FOLLOWING SELECTED DIMENSIONS FOR 10MW, 625°C SUPERCRITICAL CO2

No.	T in °C	p in bar	Inner diameter in mm	Outer Diameter	DN	Wall Thickness	Schedule	Material Grade
1	450	275	150.0	219.1	200	23.01	160	SA 335 P91
2	625	275	170.0	273.1	250	28.58	160	SB 444 N06625
3	504	107	250.0	323.9	300	17.45	80	SA 335 P91
4	152	107	170.0	219.1	200	12.70	80	SA 106 grade B
5	60	107	120.0	168.3	150	10.97	80	SA 106 grade B
6	60	107	50.0	88.9	80	11.13	160	SA 106 grade B
7	129	275	40.0	88.9	80	11.13	160	SA 106 grade B
8	60	107	110.0	168.3	150	10.97	80	SA 106 grade B
9	32	107	70.0	114.3	100	13.49	160	SA 106 grade B
10	54	275	70.0	114.3	100	13.49	160	SA 106 grade B
11	150	275	90.0	168.3	150	18.26	160	SA 106 grade B
12	146	275	100.0	168.3	150	18.26	160	SA 106 grade B
13	450	275	150.0	219.1	200	23.01	160	SA 335 P91

Assumptions for Delivery

- The length of the pipes are estimated roughly at 12.0 m per system design area and the usual delivery length of pipes (2 x 6 m delivery unit)
- Layout, as well as allocation of components of the test facility are not known
- Four elbows are considered per system design area
- Assumed number of valves: 1 control valve and 1 gate valve per system design area, i.e. a total of 12 control valves, 12 gate valves and 1 quick closure-type valve
- Materials are delivered from stock with common dimensions
- Special documentation or certification is not required
- There is no increased price for minor quantities
- Corrosion allowance for inner surface is in general 2.0 mm
- Corrosion allowance for outer surface is 2.0 mm for temperatures > 160°C; for temperatures < 160°C it is to be considered 0.0 mm
- All connections are welded, there are no flanges considered
- Supporting structures are not to considered

Assumptions for primary sCO2 loop piping welds

- Welding of the piping was reviewed without consideration of supports or any inline equipment
- There will be no erection of equipment
- Quantity of welds per system design area was estimated with 10 (derived from 4 elbows plus connection to equipment or tee at start and end)
- All welds are volumetrically tested
- Final documentation or elaboration of isometrics will not occur
- There is no as-built documentation

Utilizing the above mentioned assumptions and under consideration of the recommended practice no. 17R-97 "cost estimate classification system" class 4, the rough order of magnitude cost estimation is $3,500,000 for fabrication, delivery, and welding of primary loop pipework and valves.

8.0 Major sCO2 Component Sizing and ROM Cost Estimates

The recuperator sizing was based on a conceptual design provided by Brayton Energy, Inc. to Aerojet Rocketdyne in a contract to Leonardo Technologies, Inc [7]. The general architecture of the recuperators can be described as counterflow unit-cell plate-fin heat exchangers. The heat duty of the high and low temperature recuperators were 39 and 15 MWth, respectively. Five units of the Brayton design are needed for both the high and low temperature recuperators. The size of each unit is 24 inches in diameter and 204 inches in length. The cost Nth of a kind cost for each unit is $150,000. The total cost for the high temperature recuperator is $750,000. The total cost for the low temperature recuperator is $750,000 **Table in 8-1**. These costs do not include piping and installation of the recuperators.

The main compressor and recycle compressor are on the same shaft. They are powered by the compressor turbine, also located on the same shaft. All three are situated within the same housing. The RPM is 30,000. The power for the main compressor, recycle compressor, and compressor turbine is 2 MWth, 1 MWth, and 3 MWth, respectively. The sizing of the housing was scaled from a design for a 200 MWe Compressor based on the ratio of tip diameters. The total size of this housing is 22 inches in diameter and 71 inches in length. The cost was scaled from a CO2 compressor of an NGCC cycle based on DOE Report 1397 [8]. The Nth of a kind cost is $6,600,000. This cost is shown in **Table 8-1** and does not include piping and installation of the compressors.

The turbine power was 13 MWe and has an RPM of 20,000. The sizing was scaled from a design for a 650 MWe turbine based on the ratio of pitch diameters. The size of the turbine is 22 inches in diameter and 39 inches in length. The cost was scaled from an NGCC cycle based on DOE Report 1397. The Nth of a kind turbine cost is $1,000,000 as depicted in **Table 8-1**. This cost does not include piping and installation of the turbine.

An estimated ROM cost of $3,500,000 for the major primary loop piping, fittings, and valves is discussed in Section 7.2. Additional costs for auxiliary systems, construction, enclosures, etc., are beyond the scope of this report and are discussed in Section 9.

TABLE 8-1: PRIMARY SCO2 LOOP COMPONENTS AND PIPING ROM ESTIMATES

Item	Required Units	ROM Cost Per Unit	ROM Cost
High Temperature Recuperator	5	$150,000	$750,000
Low Temperature Recuperator	5	$150,000	$750,000
Common Shaft Compressor-Turbine	1	$6,600,000	$6,600,000
Power Turbine	1	$1,000,000	$1,000,000
Primary Loop Piping, Fittings, Valves	1 (package)	$3,500,000	$3,500,000
			ROM Estimate $12,600,000

9.0 Future Cost Considerations: sCO2 Auxiliary Systems and Enclosure

The costs for the sCO2 auxiliary systems and enclosure(s) are beyond the scope of this study, as are construction and implementation costs. This section only provides a general overview of the scope to consider when a future estimate of the overall cost of the sCO2 10MWe energy conversion project is developed. Auxiliary systems have been described in the previous section.

The following considerations should be made for Auxiliary Systems and Building ROM cost estimates:

- Selection of an architectural and engineering firm (A&E) to handle development of the energy conversion systems and facility
- Auxiliary systems such as CO2 supply are proportional to loop size or volume
- Auxiliary systems such as CO2 sampling are not greatly affected by loop size or volume
- Building – Consider a metal building with pre-fab panels, blowout doors, with about 5600 sq. ft. to cover both sCO2 loop and auxiliary equipment
- HVAC –Building ventilation requirements are based on safety analyses and may itself be an engineered barrier, or an item relied upon for safety
- Site - Standard yard and site improvements surrounding building would be expected, at about 10000 sq. ft.
- Standard utilities and facility communications such as phone and internet, should be included
- Safety analyses supporting the process should consider chemical, industrial, environmental, and fire accident sequences
- Federal, state, and local regulatory requirements, permits, and licenses apply and should be considered
- Construction power and electric grid tie-in requirements, including plant power systems, bus duct, transformers, controls, etc.

The cost estimate should consider the following needs:

1) sCO2 demonstration project general systems, building, and other facility needs, **Table 9-1**
2) sCO2 demonstration project process design, install, test, startup needs, **Table 9-2**
3) Project scale-up design and operations cost considerations **Table 9-3**

TABLE 9-1: GENERAL SYSTEMS AND BUILDING ROM COST ESTIMATE CONSIDERATIONS

GENERAL REQUIREMENTS
EXISTING CONDITIONS
CONCRETE
MASONRY
METALS
WOOD, PLASTICS & COMPOSITES
OTHER BASIC MATERIALS
THERMAL & MOISTURE PROTECTION
OPENINGS
FINISHES

SPECIALTIES
FURNISHINGS
BUILDING
CONVEYING SYSTEMS
FIRE ALARM, DETECTION, SUPPRESSION
PLUMBING
LIGHTING
COMMUNICATIONS
EARTHWORK
EXTERIOR IMPROVEMENTS
UTILITIES (Water, Electrical, Other)
AUXILIARY EQUIPMENT
Material Handling – hoists, electric overhead
Extensible Jib Crane
CO2 Handling Feed Tank, Tank and System Fill
Pump Seal System - Header
Pressure Relief Systems and Rupture Disks - Header
Depressurization System - Header
Chemical Additive Systems
Sampling System
Water Removal System
Instrumentation and Control Systems
Other Safety Systems - CO2 Monitors/Interlocks
Particulate Handling and Filtration System
Equipment Enclosures and Exhaust Hoods
Cooling Water System
Building, Utilities, Yard
Building HVAC
HEATING, VENTILATION, AIR CONDITIONING (HVAC)
Condensing unit
Reheaters
Exhaust Stack
Stack Sampling System
Vacuum pump and sampling system
Control System and Dampers
Ducts
Damper
Pumps
Blower
Heat pump
OTHER
Equipment Connections
Basic Materials
Integrated Safety Analysis / Hazards Analyses
Chemical Safety
Industrial Safety
Fire Safety
Environmental Safety

TABLE 9-2: sCO2 PROCESS DESIGN, INSTALL, TEST, START-UP ROM COST ESTIMATE CONSIDERATIONS

Process and Technology Cost Estimate Considerations	
Feasibility Study	**Pilot Study and Tests**
Investigation and Data Collection	Scale-up Flow sheet
Feed/Effluent Characterization	Equipment Procurement
Requirements Definition	Fabricate and Install
Regulatory Needs	Procedures, Training
Technical Framework	Startup
Waste and Recovery Needs	Pilot Study and Analysis
Report - Findings, Recommendations	*Preliminary Design*
Cost-Benefit Analysis	Preliminary Flow sheet
Feasibility Study Report	Preliminary Cost Estimate
Proposal and Approval	Detailed Design Proposal for Full Scale
	Preliminary Integrated Safety Analyses
Rough Order of Magnitude (ROM)	Proposal Justification and Funding
	Technical Basis Document - Pilot
Lab (Bench-Scale) Study and Tests	Risk Analysis
	Pilot Scale Integrated Safety Analyses
Test Plans and Equipment	Parametric Study - Design of Experiments
Chemistry and Behavior Tests	Evaluate Scale-up Parameters/Strategy
Technical Basis Document	Cost-Benefit Analysis
Conceptual Design	Capital Investment Proposal
Conceptual Flow sheet	Lifecycle Cost Minimization
Initial Material Balance	
Parametric Analysis	Preliminary Estimate (+/-50%)
Conceptual Process Flow Diagram	Budgetary Estimate (+/- 30%)
Conceptual Design for Lab Scale	Detailed Final Estimate (+/-10%)
Pilot Test Proposal	
Lab Technical Basis Document	
Cost-Benefit Analysis	
2nd ROM Cost Estimate for Full Scale	
Cost Estimate for Pilot-Scale Study	

TABLE 9-3: PROJECT FULL SCALE CONCEPTUAL DESIGN AND OPERATIONS ROM COST CONSIDERATIONS

Full Scale Project Cost Estimate Considerations	
Technology Transfers	**Project Execution for Installation**
License Agreement	Procurement
Technology License	Fabrication
License Fee	Site Preparation
Confidentiality	Civil/Structural Building
Taxes and Duties	Utilities and auxiliary Systems
Warranties	Heating Ventilation and Air
Liabilities	Installation, Functional Testing
Other Terms and Conditions	System Testing
Technology Product Description	Equipment Functional Testing
Scope of Technology	Acceptance Testing
Licensor Technical Assistance	Procedures, Training
	Licenses and Permits Issued
Full Scale Industrialization	Operations and Maintenance Training
	Process Specifications
Licensing/Permit Submittal	Analytical Plan
Detailed Design and Cost Estimate	Operations/Maintenance Procedures
Final Process Flow sheet	Process Qualifications
Equipment Design	Spare Parts
Piping and Instrumentation Diagram s	Readiness Review
Process Controls	Startup Approval
Design Descriptions	
Equipment List	**Operations**
Equipment Specifications	
Instrument Specifications	Operators/Maintenance/Other Resources
Technical Basis Document – Full Scale	Process Engineering Resources
Safety Analyses	Environmental Compliance Plan
Civil/Structural Drawings	Industrial/Chemical Safety Controls
Mechanical Drawings	Criticality Safety Controls
Safety Equipment and Interlocks	Radiological Protection Controls
Code Compliance Review	Fire Safety Controls
Interlock List	Preventive Maintenance and Procedures
Regulatory Support	Instrument Calibrations and Procedures
Project Management Resources	Set Operating Parameters
Project Financial Justification and ROI	Product and Waste Specifications
Risk Analysis and Management	Quality Assurance Plan
Project Proposal and Authorization	Operating Procedures
	Operations Training
	Decommissioning Funding Plan

10.0 Energy Conversion Study Conclusions and Proposed Actions

As addressed in previous sections of this report, an increase in turbine throttle pressure in the 279 bar range results in a performance improvement. Supercritical fossil plant steam conditions have been above 250 bar since the 1960's and have increased over time to pressures that bound the cycle modeling covered in this report.

Availability of a suitable heat source is a logistic constraint for implementation of a sCO2 demonstration project. Due to the high temperature needed for cycle operation, the likely heat source choice will be fossil fired. Heat source requirements along with a need for an electric grid tie-in tend to make a utility generation site, or substantial process and generating complex, a practical approach to locating the demonstration project.

Several power plant fossil fired options could be adapted to incorporate the sCO2 demonstration project. Coal or oil fired boilers can be used for the heat source, but a natural gas fired approach seems to involve fewer installation complications, as well as a less expensive solution, with the added benefit of reduced emissions. Although gas fired boilers could be considered, there are many more gas turbines in service, either as peaking units or combined cycle units that are base loaded or load following. Utility or Independent Power Producer gas turbines in operation have exhaust temperature ranges that can reach above 600 C.

Due to the relatively small sCO2 heat load, compared to the large steam generating capacity in a combined cycle unit, it appears that installation of a sCO2 heat exchanger in the transition from turbine to heat recovery steam generator (HRSG) should be included in a feasibility review. It seems that with proper controls and supplemental firing/duct burning, steam generation could be maintained and sCO2 heating accomplished. Alternatively, a different gas turbine approach would involve placing the sCO2 heat exchanger with duct firing in the exhaust duct of a simple cycle peaking unit. Since peaking units operate infrequently this would place more restrictions on sCO2 cycle demonstration.

While adaptation of existing power generators to incorporate a sCO2 energy conversion demonstation could be successful, this places the utility/operator at risk. Demonstration project delays may keep a commercial unit off line beyond the planned outage or overall unit performance could be affected. A dedicated stand alone heat source may result in a higher estimated demonstration project cost but allows for fewer interface constraints. Therefore, a direct-fired tubular heat exchanger normally used in process applications may be an expensive, but worthwhile investment, and it may be possible to develop an electric heat source that would fulfill the requirements for the demonstration project.

The schedule time and required cost for securing an air/emissions permit needs to be considered. Keeping emissions within an existing permit could sway heat source selection to an existing power generation heat source as opposed to new installation with a new point source of emissions.

A study is recommended that will focus on determination of an optimum sCO2 heat source that is still cost effective. The study would evaluate the fossil fired heat source options including the heat exchanger design requirements, along with necessary installation, mechanical tie-in, and controls requirements. In addition, the following technology gaps in the sCO2 cycle and components must be addressed:

1. System optimization, including heat source integration

2. Material compatibility with sCO2
3. Turbomachinery – high power density, dry gas seals, compressor operation with inlet conditions close to triple point, thermal management in sCO2
4. Compact heat exchangers
5. System operation and control
6. Scaling from pilot to commercial sizes
7. Demonstrated performance over sustained operating periods

11.0 References

1. Aerojet Rocketdyne Report to Leonardo Technologies, Inc. Turbine Technology Program Phase III – Task 1(a) & (b): Supercritical CO2 Turbomachinery (SCOT) System Studies Incorporating Multiple Fossil Fuel Heat Sources and Recuperator Development. June 2013.
2. Yoon-Seok Choi, Srdjan Nesic – Corrosion behavoir of carbon steel in supercritica CO_2 – water environments, paper no. 09256, NACE Corrosion 2009.
3. L. Tan, M. Anderson, D. Taylor, T.R. Allen – Corrosion of austenitic and ferritic-martensitic steels exposed to supercritical arbon dioxide, Corrosion Science 53 (2011) p. 3273 – 3280.
4. Tomohiro Furukawa, Fabien Roullard – Oxidation and carburization of FBR structural materials in carbon dioxide, Progress in Nuclear Energy (2014) 1-6.
5. Kumar Sridharan – Corrosion in Supercritical Carbon Dioxide: Materials, Environmental Purity, Surface Treatment and Flow Issues, Project No. 10-872, Reactor Concepts RD&D University of Wisconsin-Madison.
6. Curtis J. Parks – Corrsion of Candidate High Temperature Alloys in Supercritica Carbon Dioxide, Carleton University, Ottawa (2013).
7. Aerojet Rocketdyne Report to Leonardo Technologies, Inc. Turbine Technology Program Phase III – Task 2(b): Supercritical CO2 Turbomachinery (SCOT) System Studies – Recuperator Development. September 2014.
8. DOE Report 1397. Cost and Performance Baseline for Fossil Energy Plants Volume 1: Bituminous Coal and Natural Gas to Electricity. Revision 2. November 2010.

Appendix A: Process Flow and Thermodynamic Diagrams

Further detail of the process flow diagrams, temperature-entropy diagrams, and pressure-enthalpy diagrams for all of the cases run (referenced in *Section 3.3*) are shown in this Appendix. For each case, the Aspen Plus process flow diagram is first, followed by the temperature-entropy diagram, and then the pressure-enthalpy diagram.

TABLE A-1: IDENTIFICATION OF ASPEN PLUS UNITS

Equipment Unit	Description
B1	Heat Source Heat Exchanger
B2	Turbine
B3	High Temperature Recuperator
B4	Low Temperature Recuperator
B5	Splitter
B6	Cooler
B7	Recycle Compressor
B8	Main Compressor
B9	Mixer
B10 – B21	Equipment unit used to simulate pressure drop

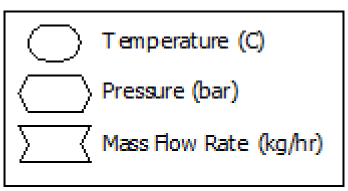


FIGURE A-1: ASPEN PLUS SCHEMATIC OF INDIRECTLY HEATED sCO2 BRAYTON CYCLE WITH HEAT SOURCE TEMPERATURE RANGE OF 300-450°C FOR 10 MWe PLANT

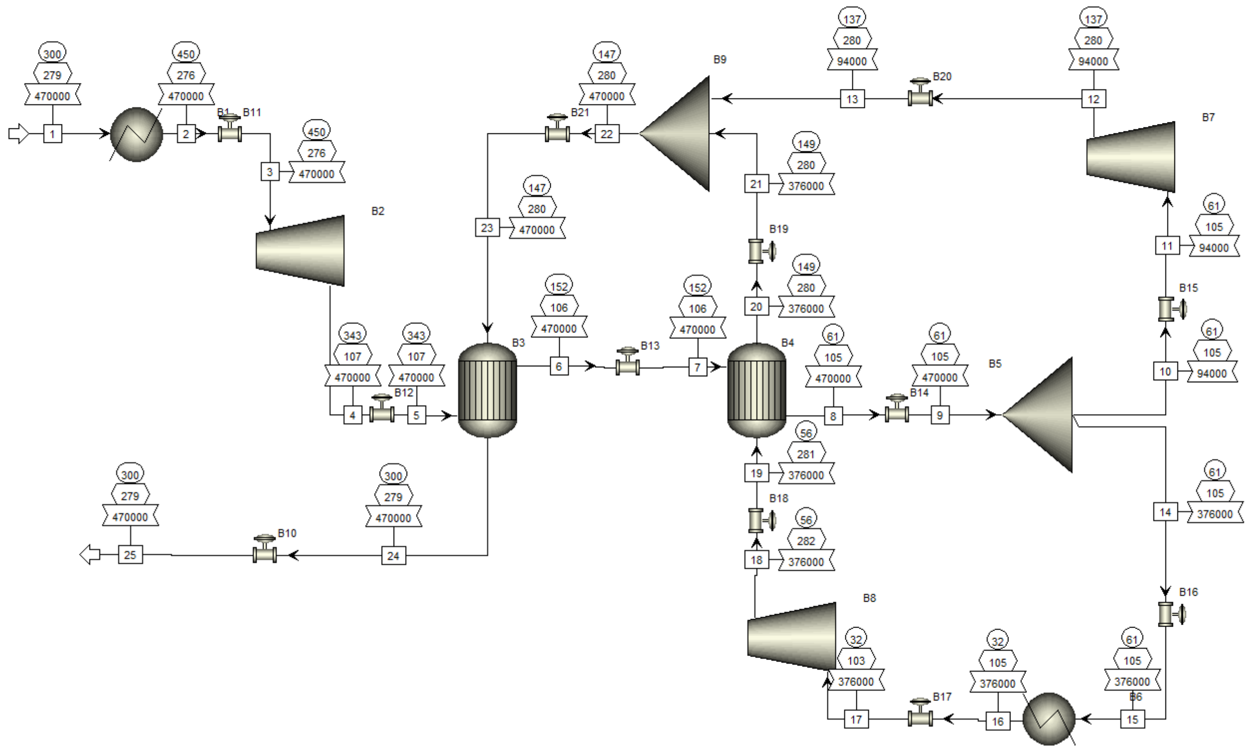


FIGURE A-2: ASPEN PLUS SCHEMATIC OF INDIRECTLY HEATED sCO2 BRAYTON CYCLE WITH HEAT SOURCE TEMPERATURE RANGE OF 300-450°C FOR 30 MWe PLANT

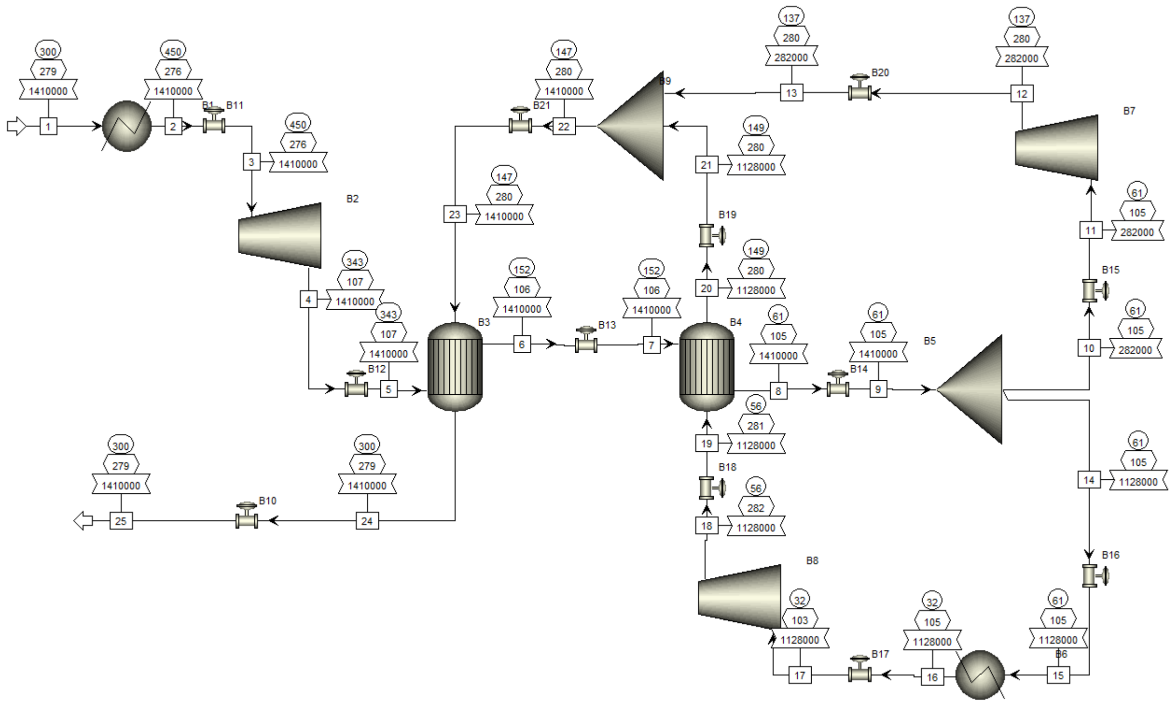


FIGURE A-3: sCO2 TEMPERARTURE ENTROPY DIAGRAM FOR 300-450°C

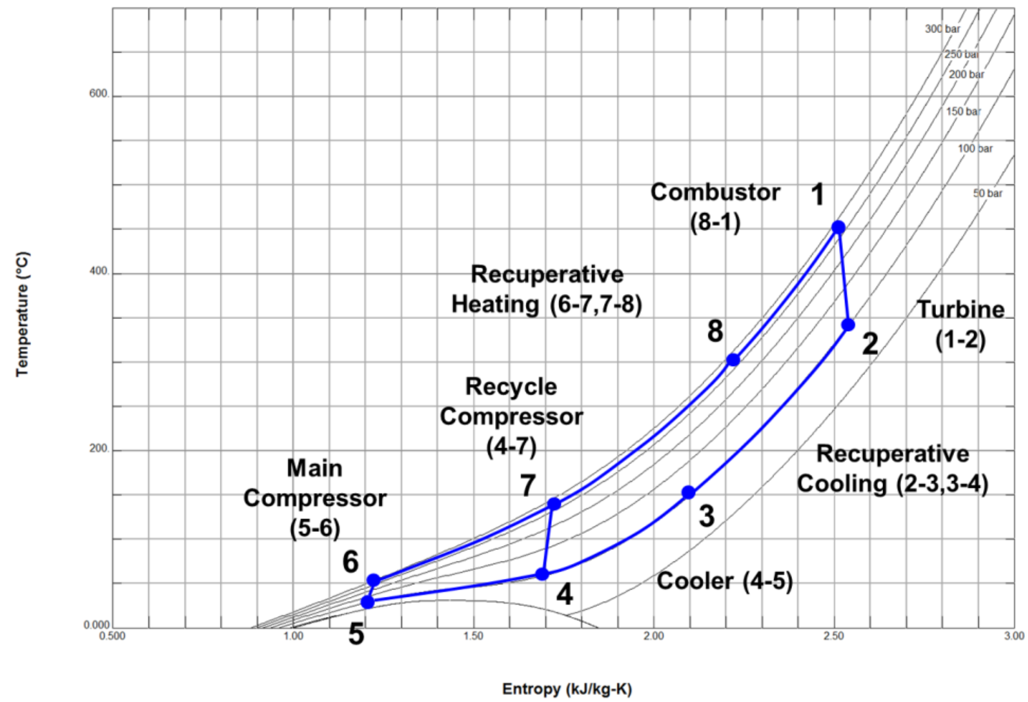


FIGURE A-4: SCO2 PRESSURE-ENTHALPY DIAGRAM FOR 300-450°C

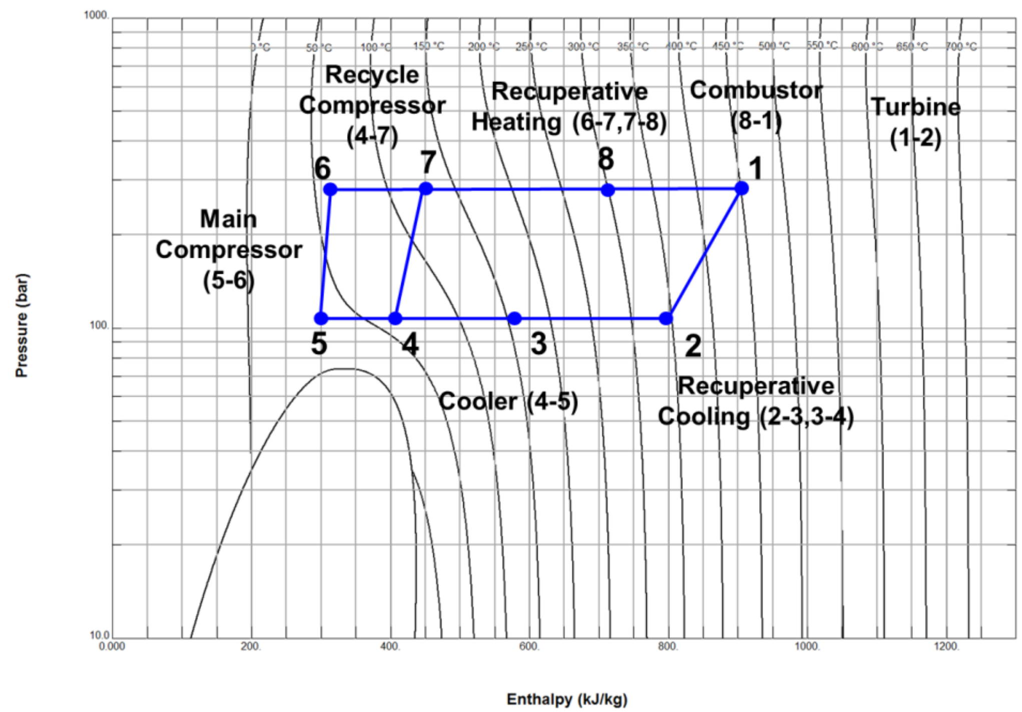


FIGURE A-5: ASPEN PLUS SCHEMATIC OF INDIRECTLY HEATED SCO2 BRAYTON CYCLE WITH HEAT SOURCE TEMPERATURE RANGE OF 450-625°C FOR 10MWe PLANT

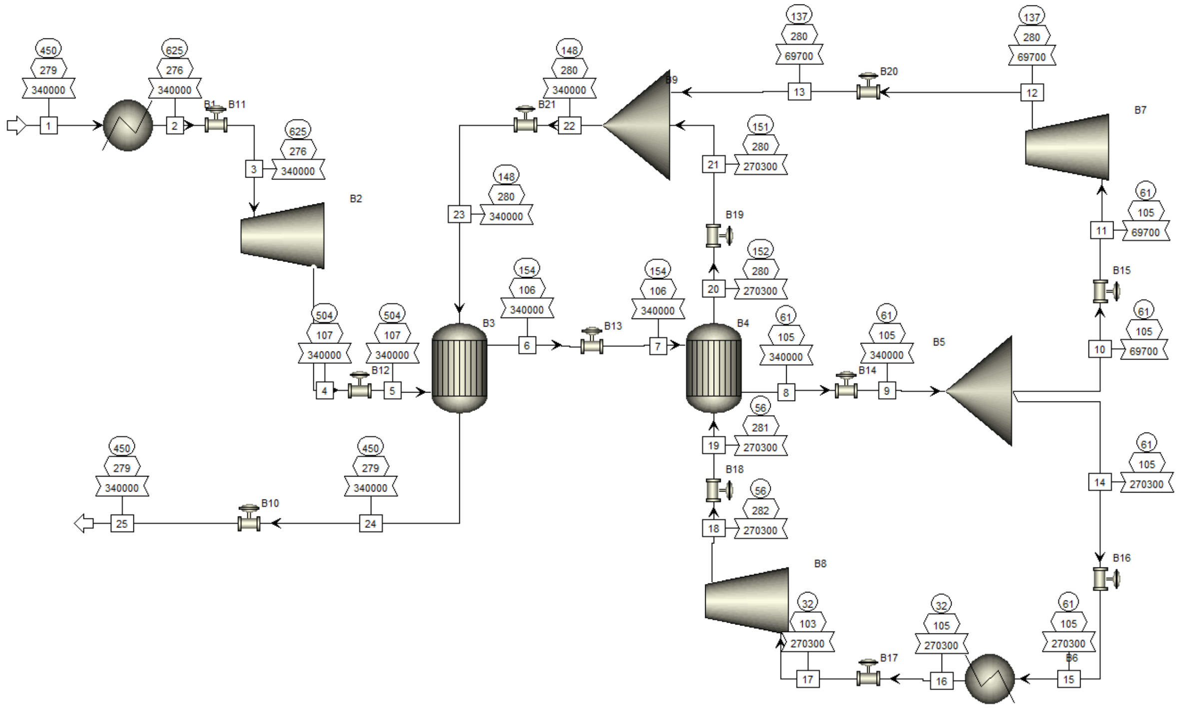


FIGURE A-6: ASPEN PLUS SCHEMATIC OF INDIRECTLY HEATED sCO2 BRAYTON CYCLE WITH HEAT SOURCE TEMPERATURE RANGE OF 450-625°C FOR 30 MWe PLANT

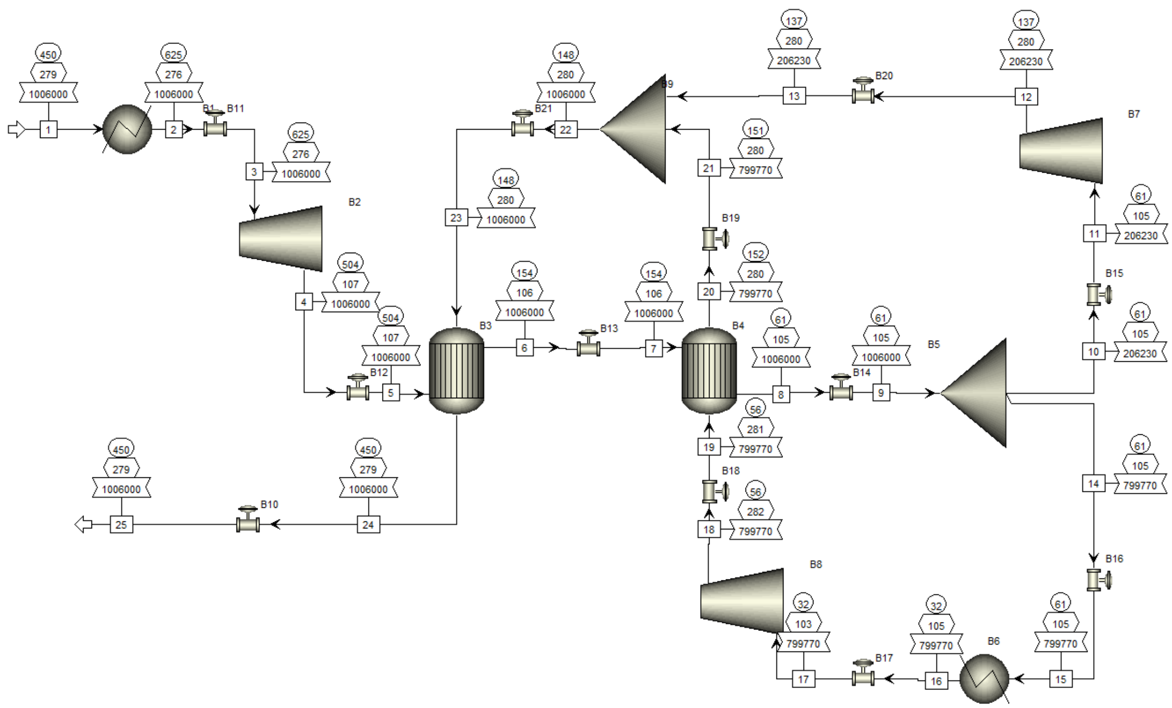


FIGURE A-7: sCO2 TEMPERATURE-ENTROPY DIAGRAM FOR 450-625°C

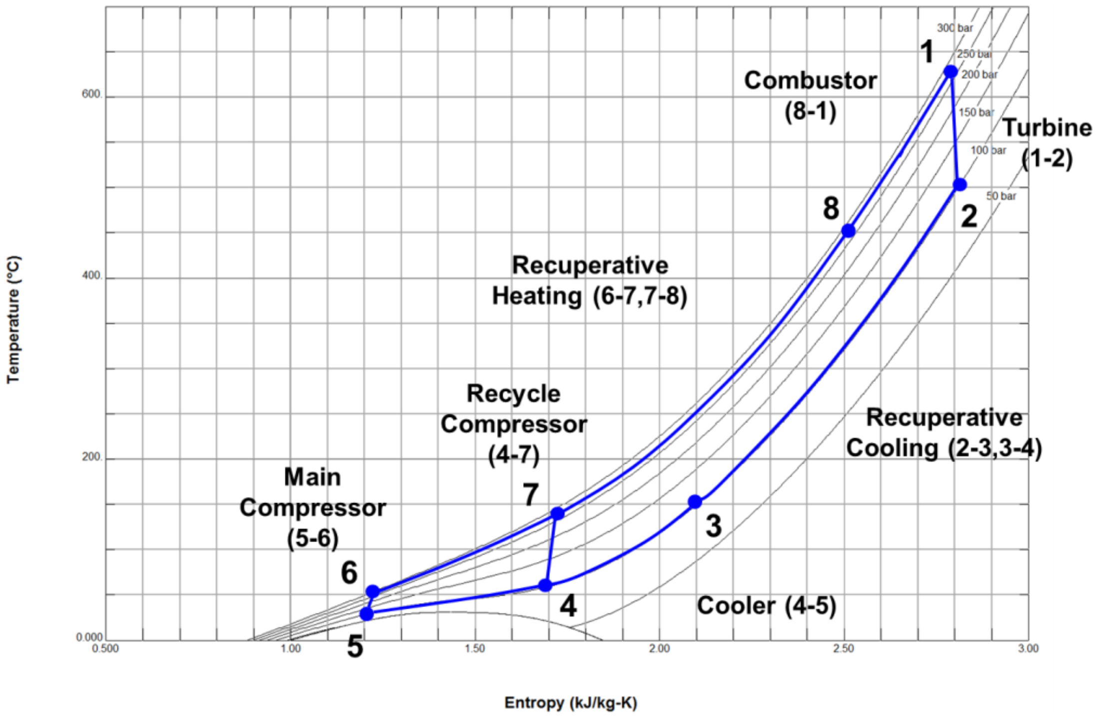


FIGURE A-8: sCO2 PRESSURE-ENTHALPY DIAGRAM FOR 450-625°C

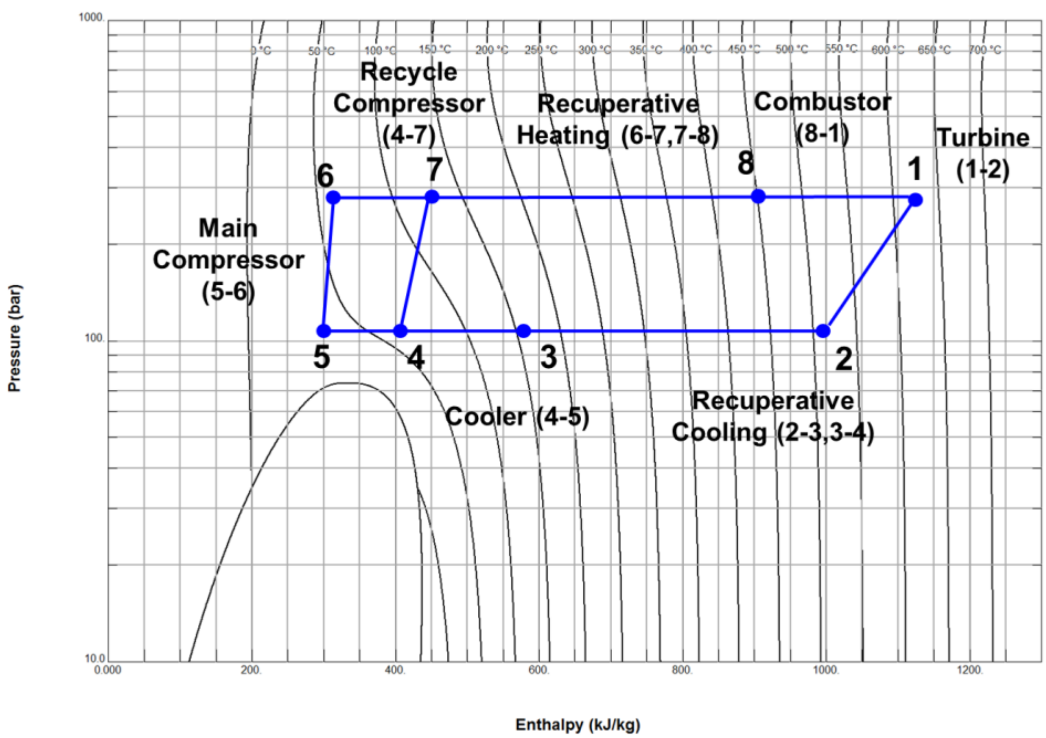


FIGURE A-9: ASPEN PLUS SCHEMATIC OF INDIRECTLY HEATED sCO2 BRAYTON CYCLE WITH STEAM RANKINE BOTTOMING CYCLE

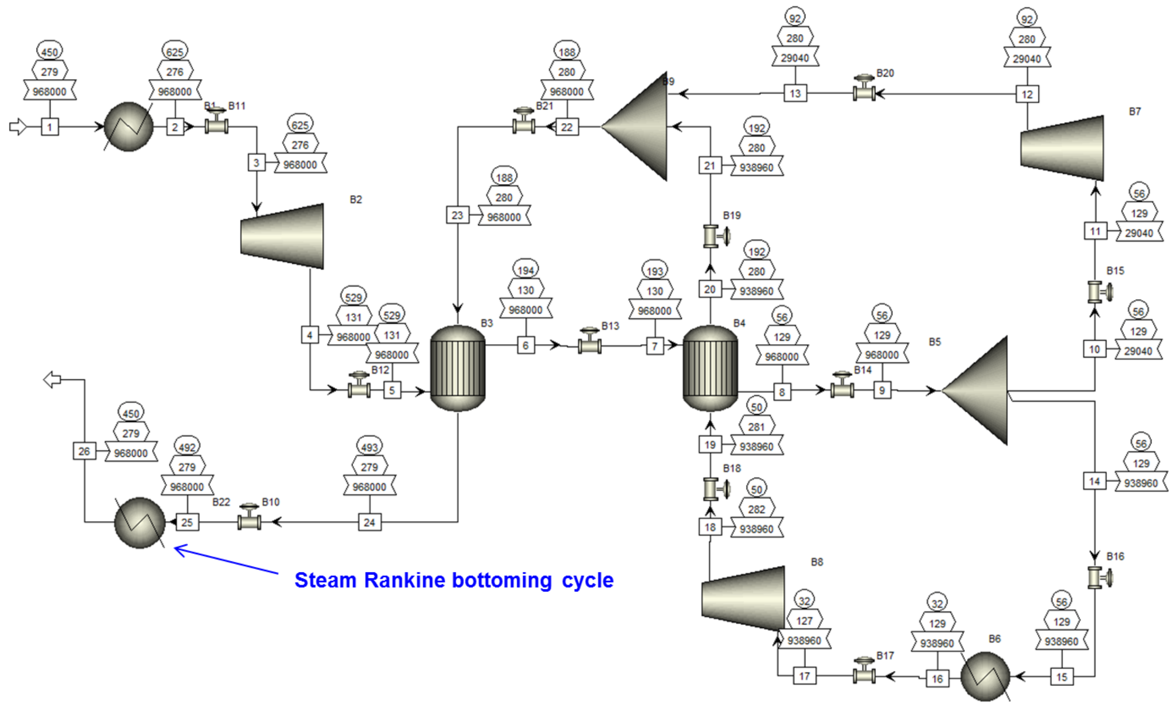


FIGURE A-10: sCO2 TEMPERATURE-ENTROPY DIAGRAM FOR BOTTOMING CYCLE CONFIGURATION

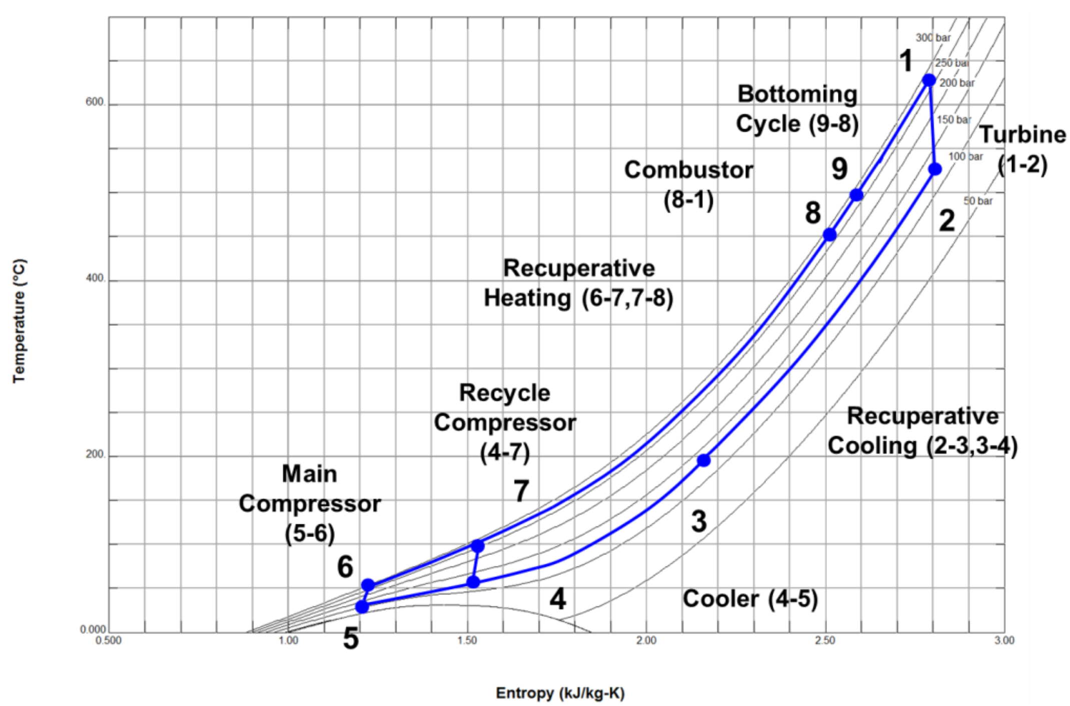


FIGURE A-11: sCO2 PRESSURE-ENTHALPY DIAGRAM FOR BOTTOMING CYCLE CONFIGURATION

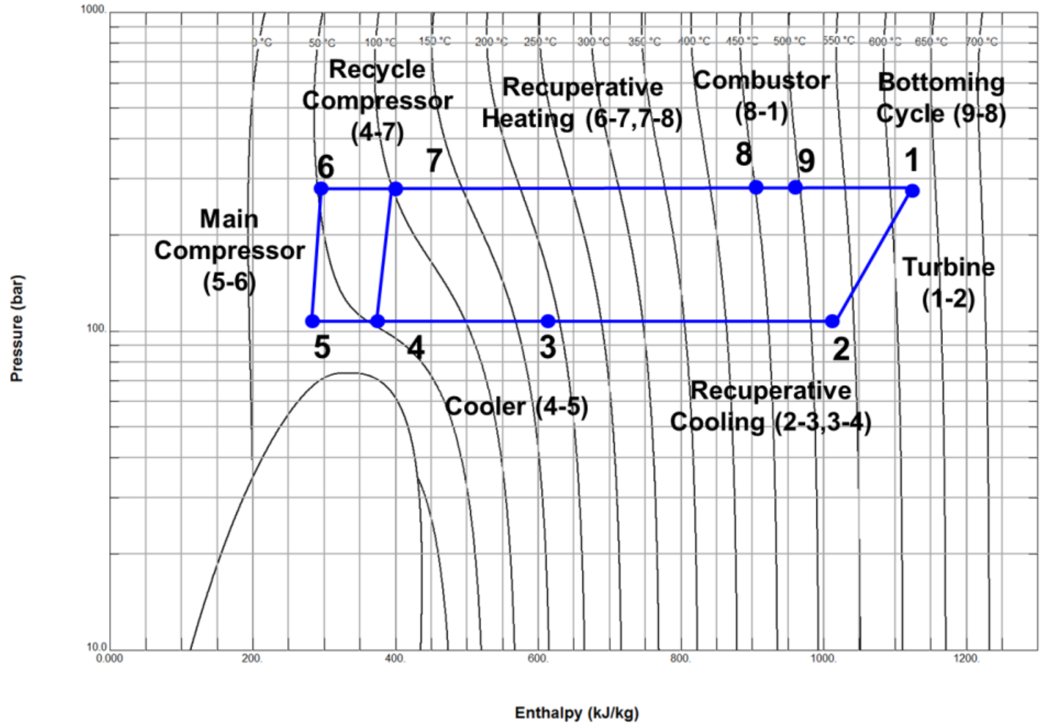


FIGURE A-12: ASPEN PLUS SCHEMATIC OF TOPPING CYCLE 1 WITH THE HEAT SOURCE TEMPERATURE SPLIT AT 538°C; 450-538°C CYCLE

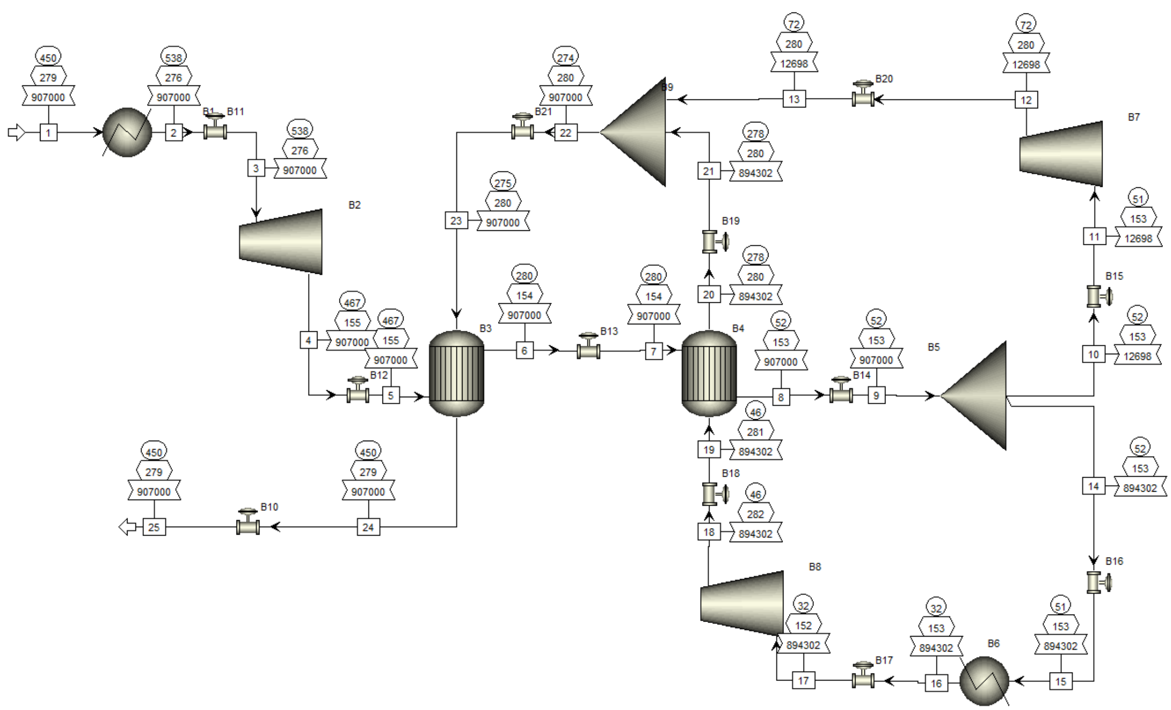


FIGURE A-13: ASPEN PLUS SCHEMATIC OF TOPPING CYCLE 1 WITH THE HEAT SOURCE TEMPERATURE SPLIT AT 538°C; 538-625°C CYCLE

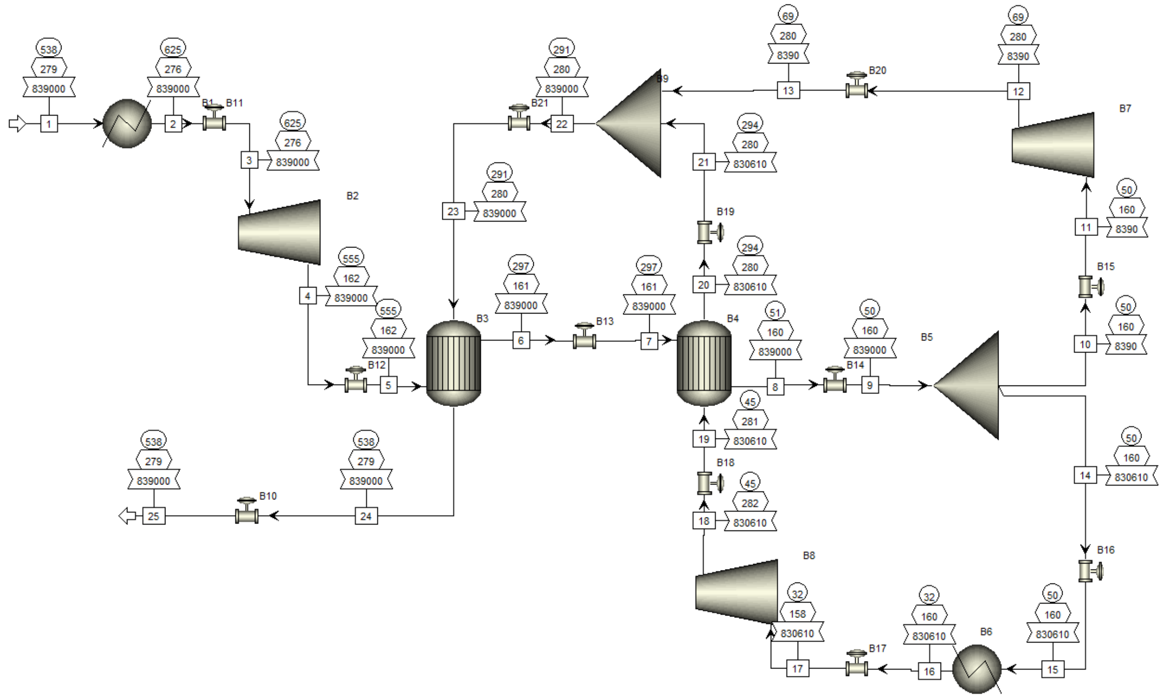


FIGURE A-14: sCO2 TEMPERATURE-ENTROPY DIAGRAM FOR TOPPING CYCLE 1 CONFIGURATION

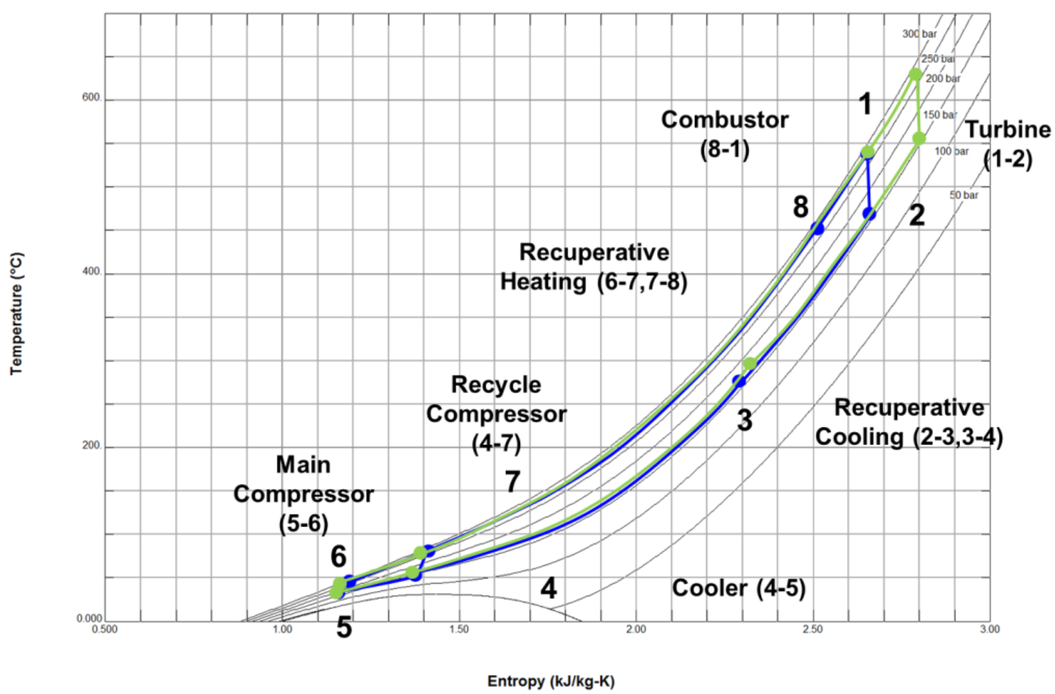


FIGURE A-15: sCO2 PRESSURE-ENTHALPY DIAGRAM FOR TOPPING CYCLE 1 CONFIGURATION

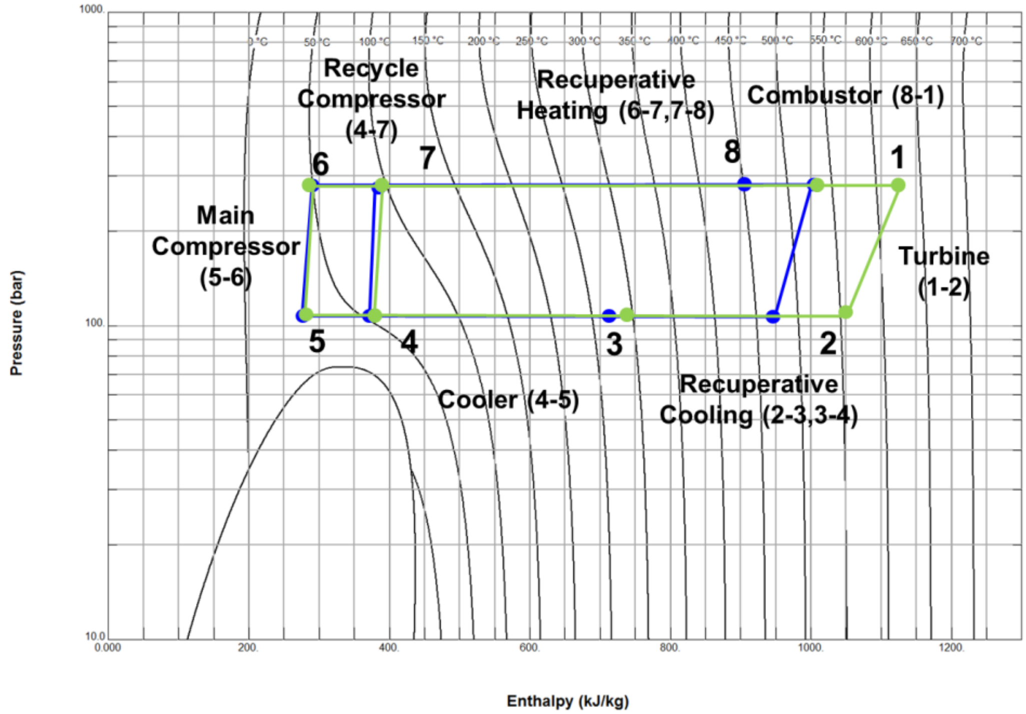


FIGURE A-16: ASPEN PLUS SCHEMATIC OF TOPPING CYCLE 2 WITH THE HEAT SOURCE TEMPERATURE SPLIT AT 582°C; 450-582°C CYCLE

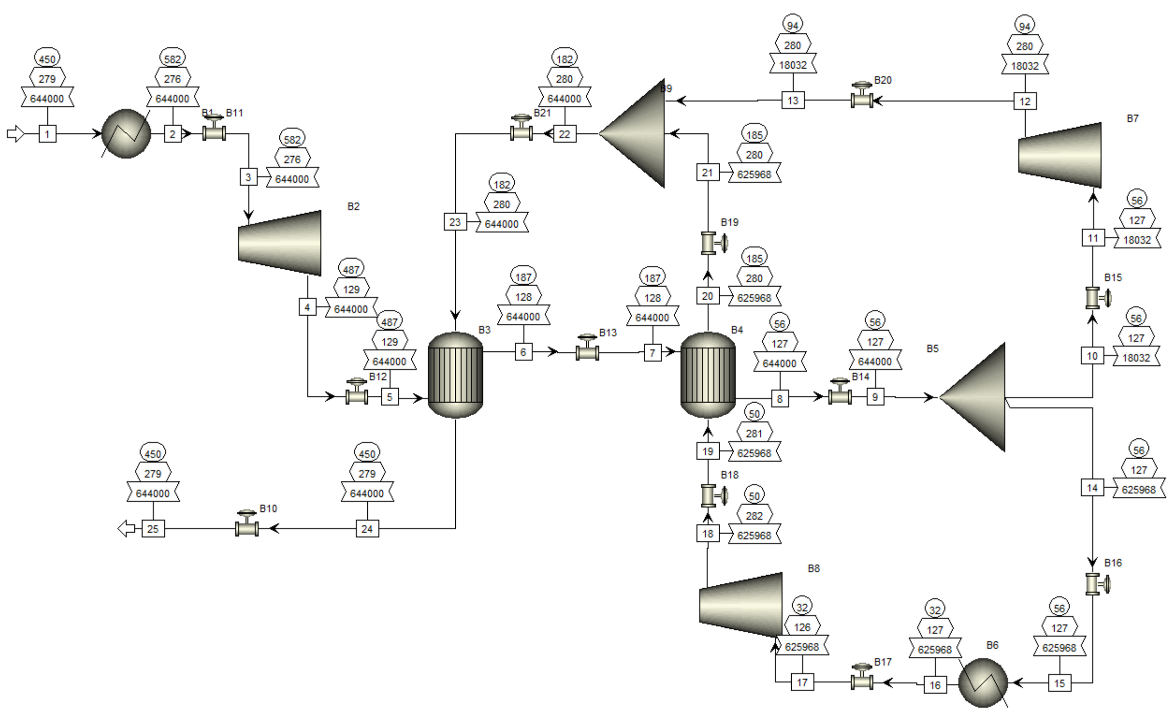


FIGURE A-17: ASPEN PLUS SCHEMATIC OF TOPPING CYCLE 2 WITH THE HEAT SOURCE TEMPERATURE SPLIT AT 582°C; 582-625°C CYCLE

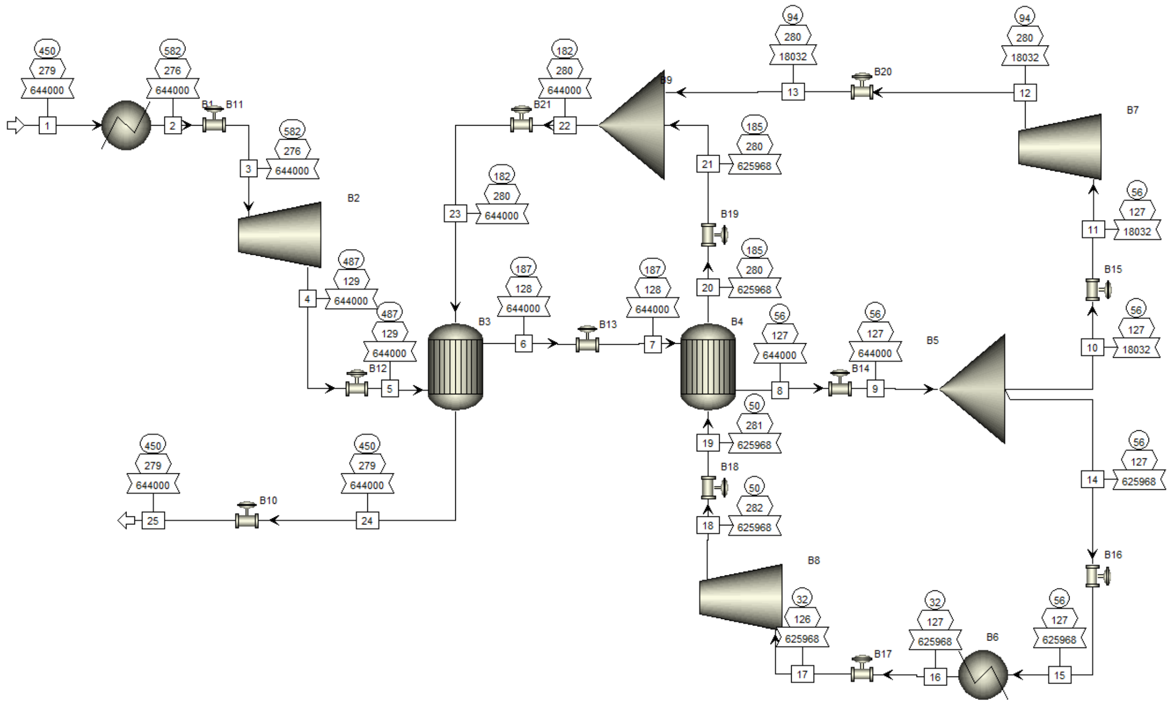


FIGURE A-17: sCO2 TEMPERATURE-ENTROPY DIAGRAM FOR TOPPING CYCLE 2 CONFIGURATION

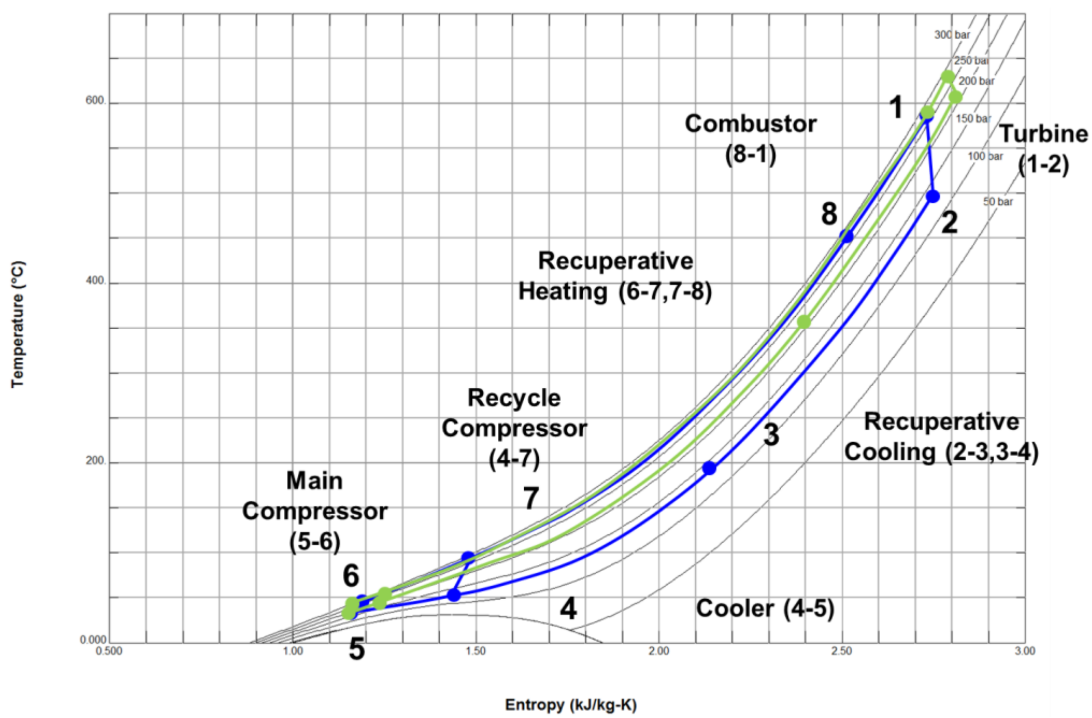


FIGURE A-18: sCO2 PRESSURE-ENTHALPY DIAGRAM FOR TOPPING CYCLE 2 CONFIGURATION

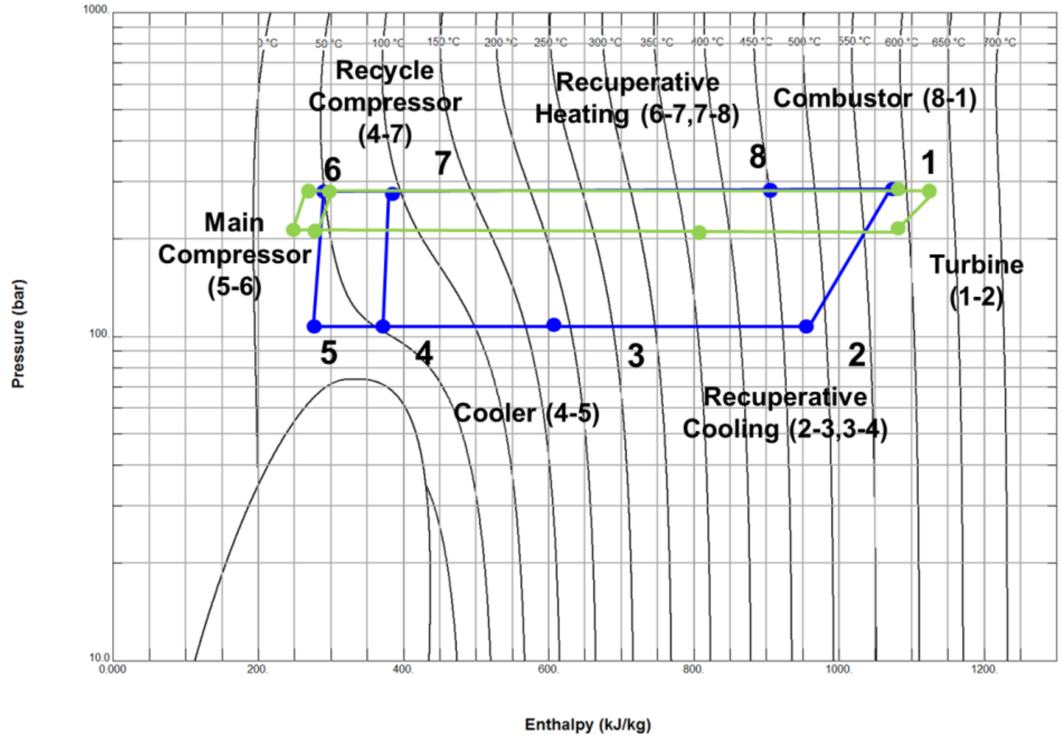


FIGURE A-19: ASPEN PLUS SCHEMATIC OF HEAT RECOVERY PARAMETRIC 1

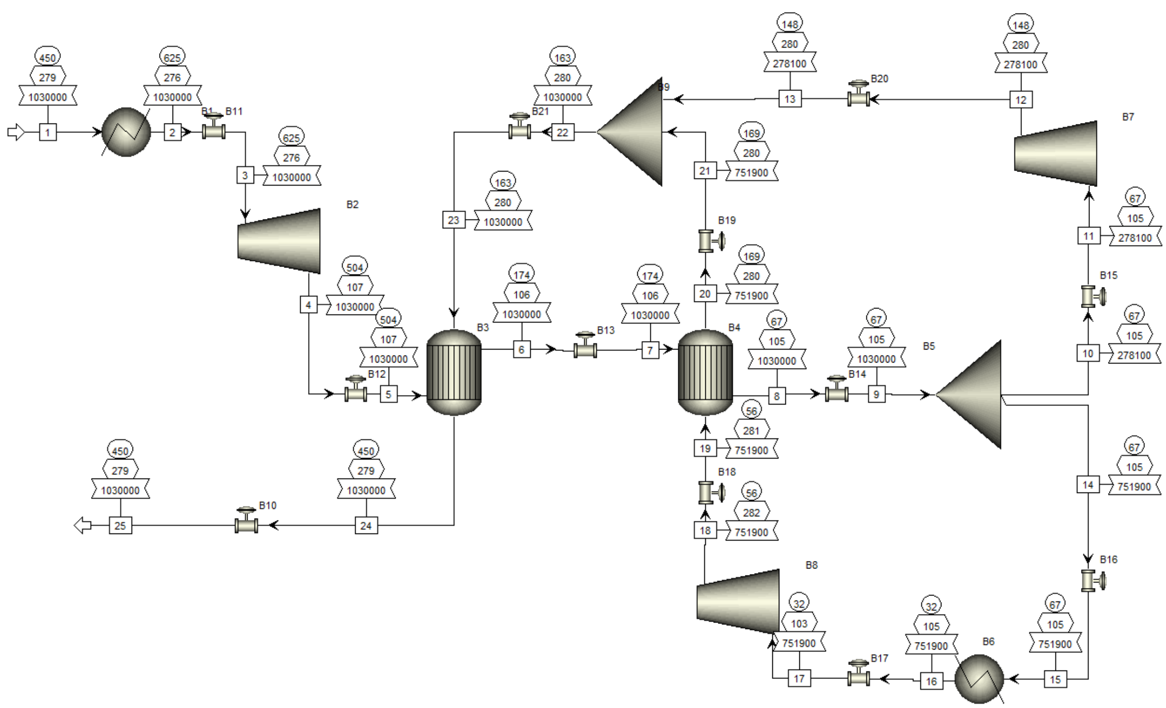


FIGURE A-20: sCO2 TEMPERATURE-ENTROPY DIAGRAM FOR HEAT RECOVERY PARAMETRIC 1

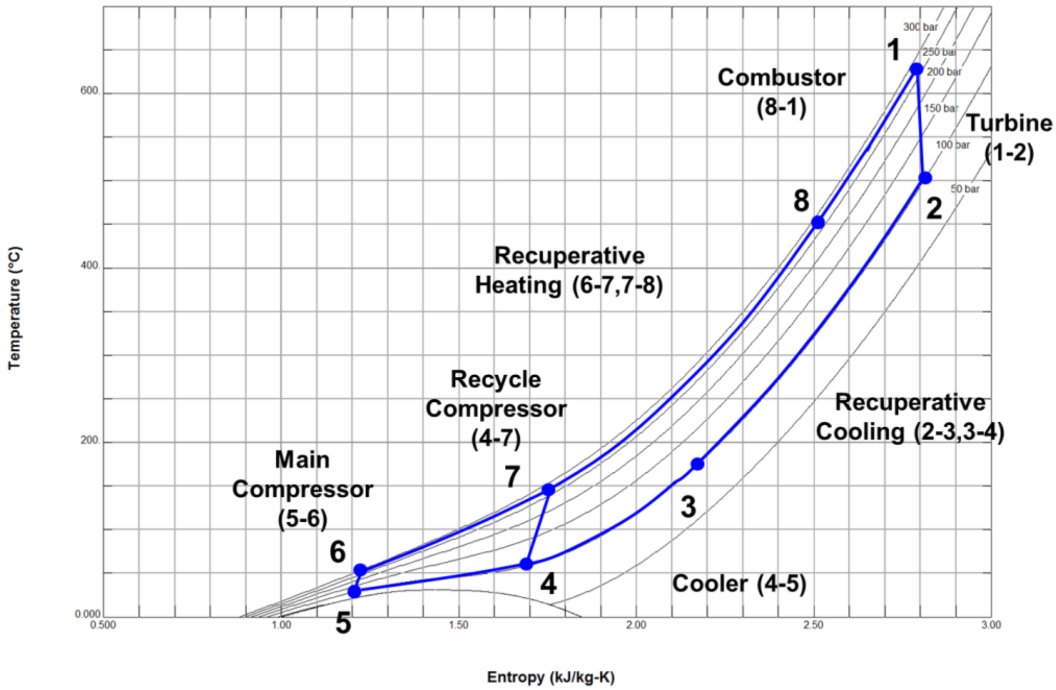


FIGURE A-21: sCO2 PRESSURE-ENTHALPY DIAGRAM FOR HEAT RECOVERY PARAMETRIC 1

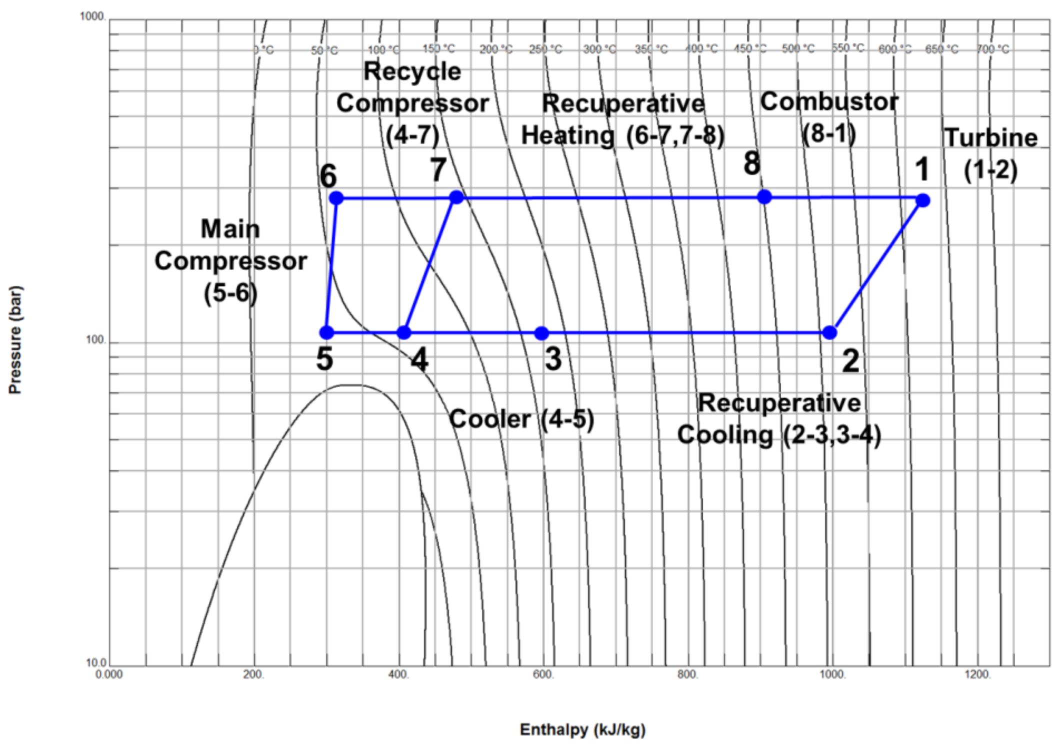


FIGURE A-22: ASPEN PLUS SCHEMATIC OF HEAT RECOVERY PARAMETRIC 2

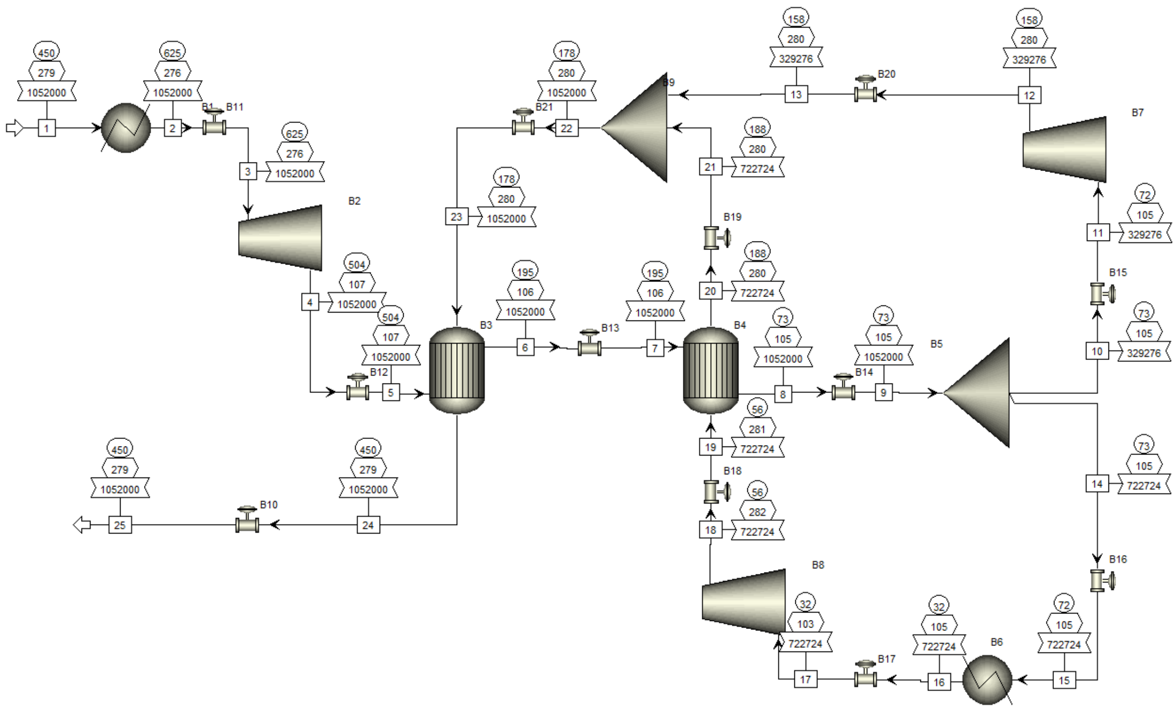


FIGURE A-23: sCO2 TEMPERATURE-ENTROPY DIAGRAM FOR HEAT RECOVERY PARAMETRIC 2

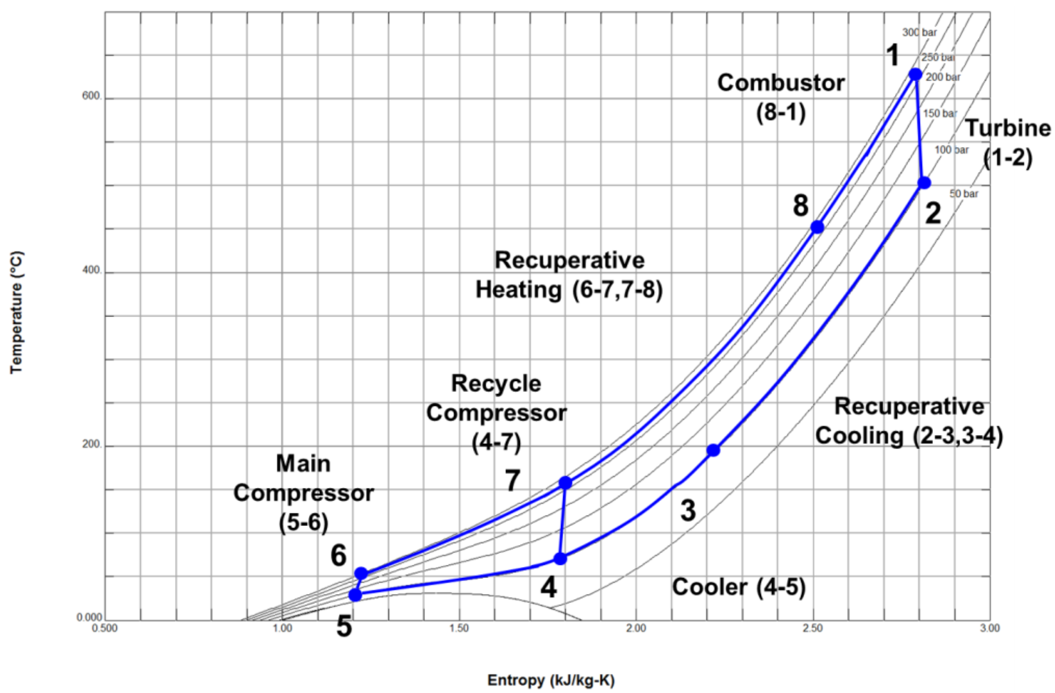


FIGURE A-24: sCO2 PRESSURE-ENTHALPY DIAGRAM FOR HEAT RECOVERY PARAMETRIC 2

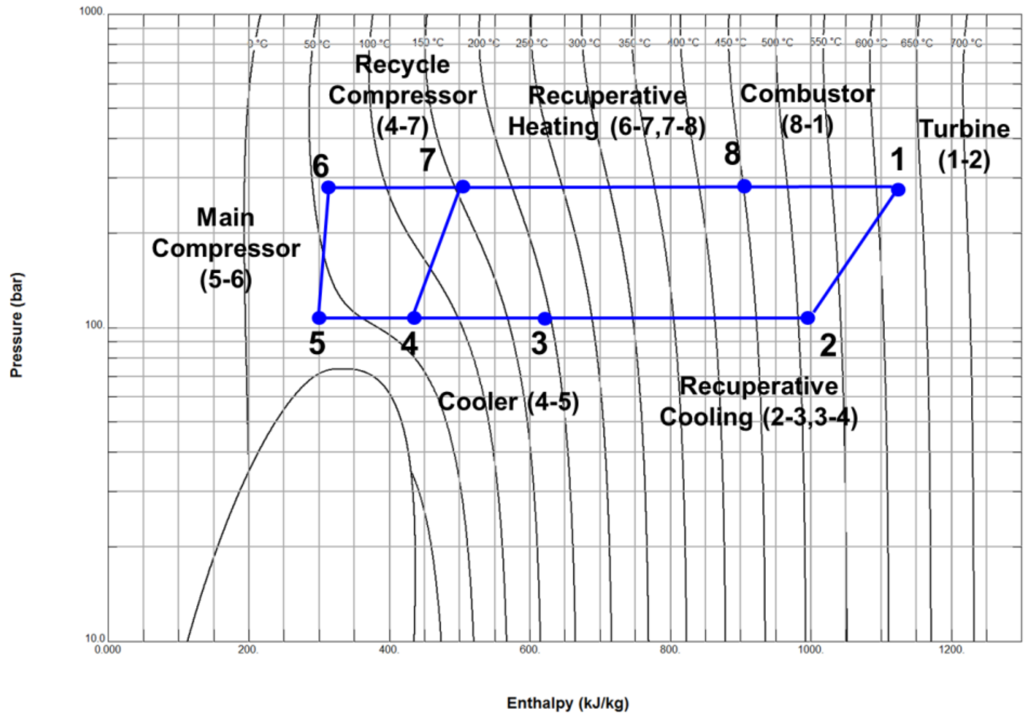


FIGURE A-25: ASPEN PLUS SCHEMATIC OF HEAT RECOVERY PARAMETRIC 3

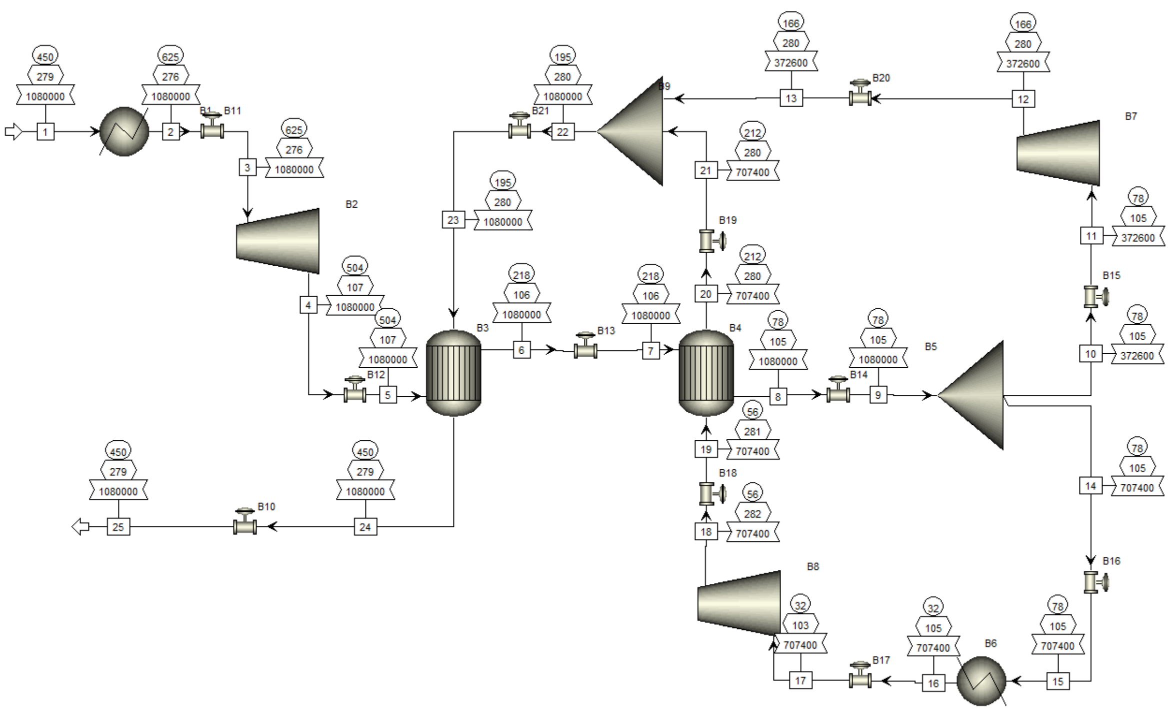


FIGURE A-26: sCO2 TEMPERATURE-ENTROPY DIAGRAM FOR HEAT RECOVERY PARAMETRIC 3

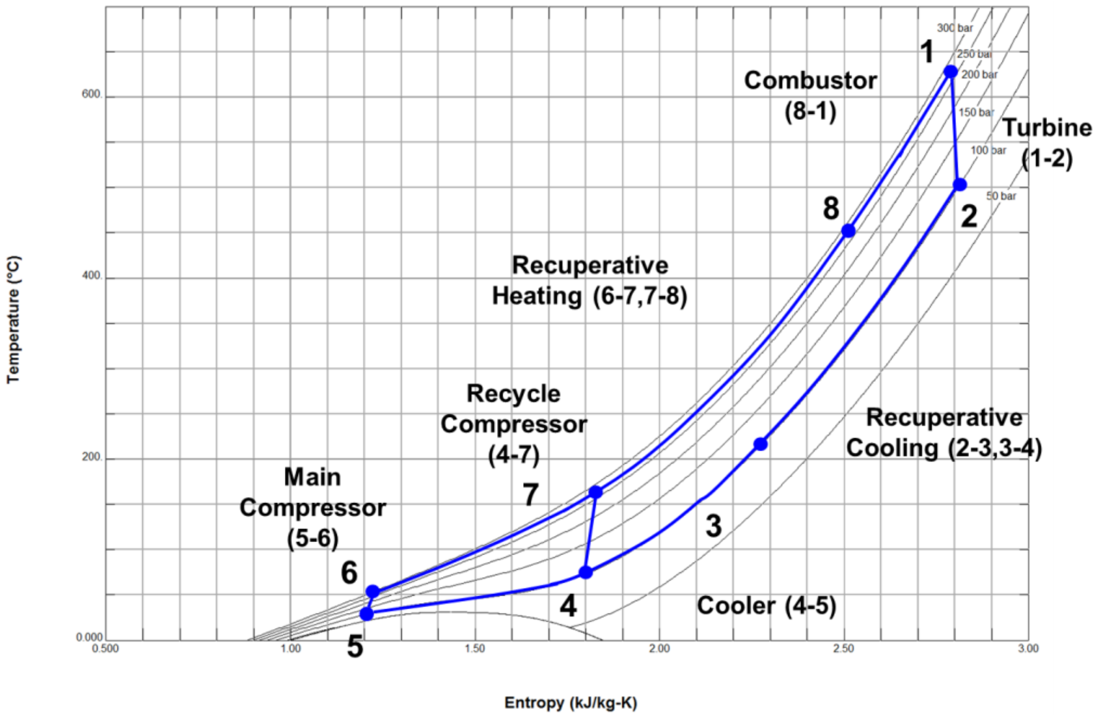


FIGURE A-27: sCO2 PRESSURE-ENTHALPY DIAGRAM FOR HEAT RECOVERY PARAMETRIC 3

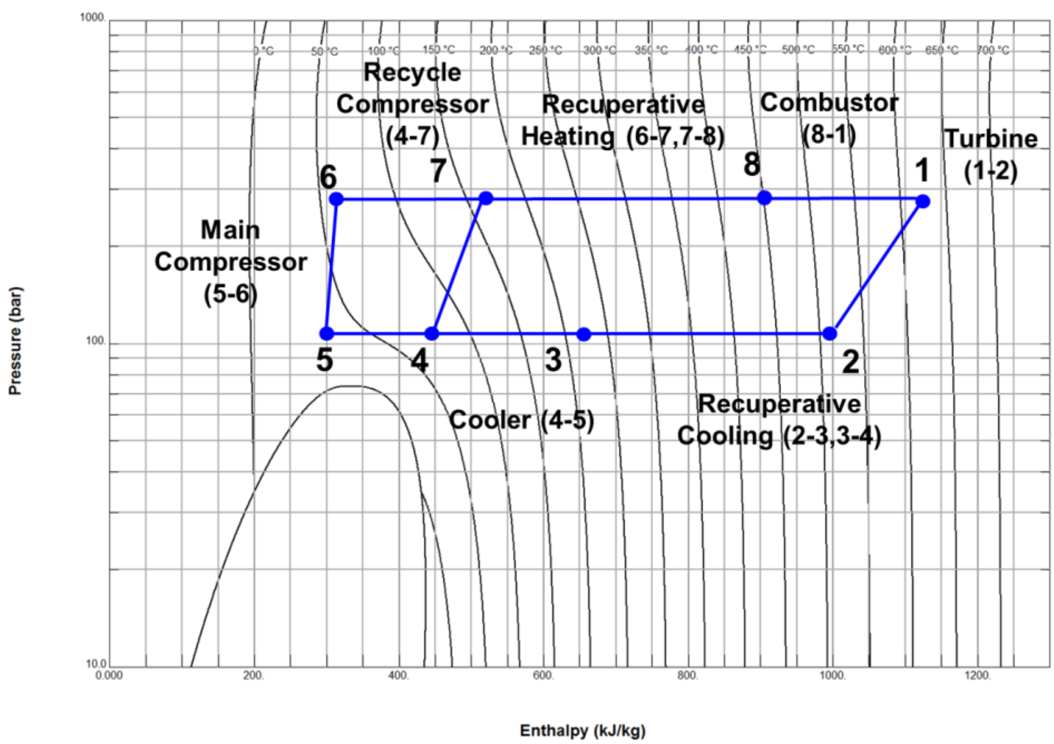


FIGURE A-28: ASPEN PLUS SCHEMATIC OF HEAT RECOVERY PARAMETRIC 4

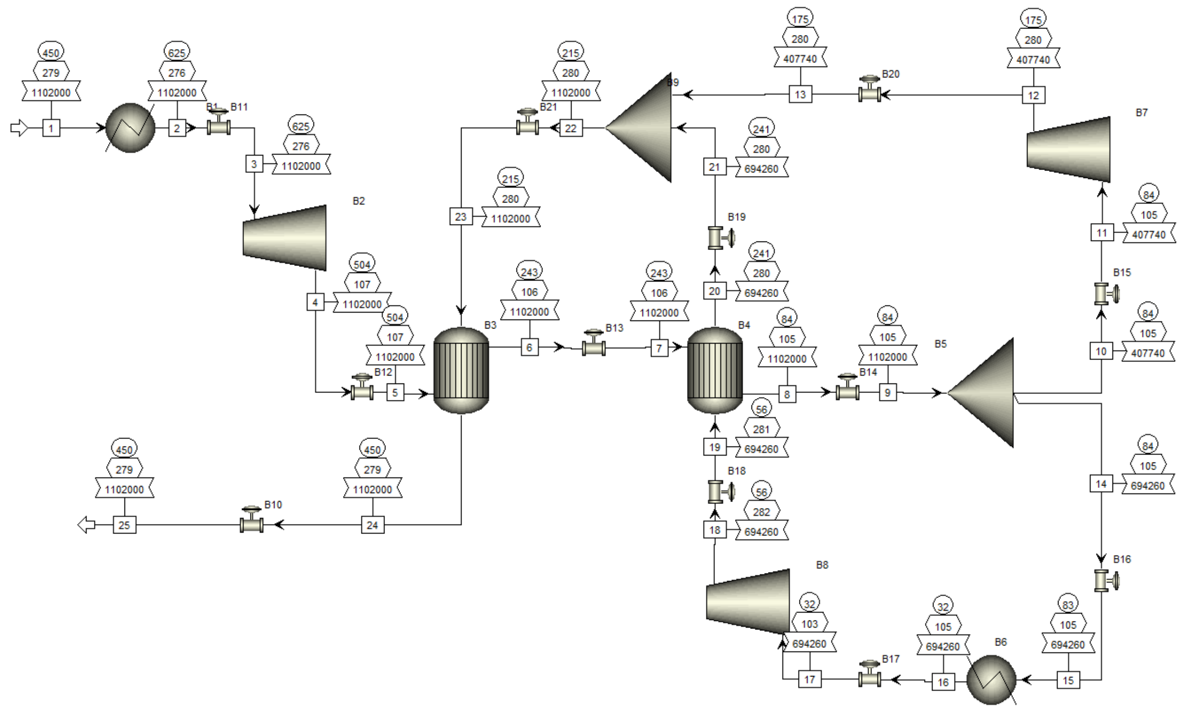


FIGURE A-29: sCO2 TEMPERATURE-ENTROPY DIAGRAM FOR HEAT RECOVERY PARAMETRIC 4

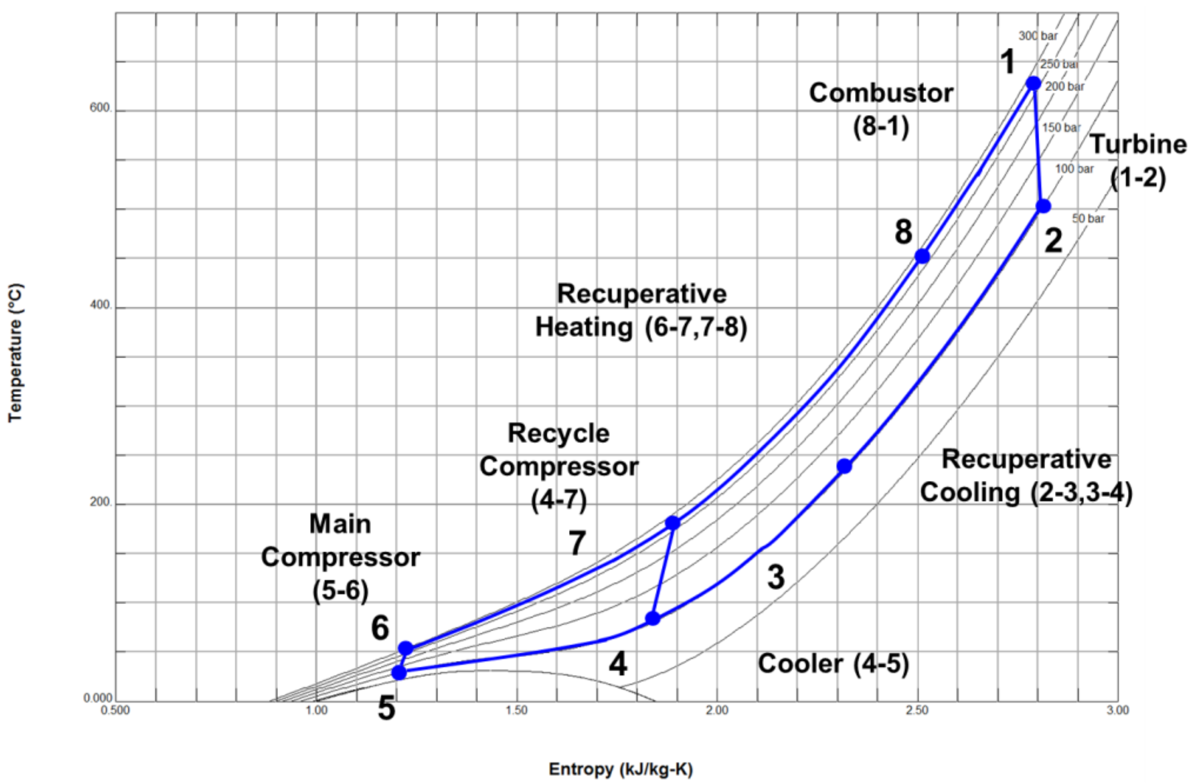


FIGURE A-30: sCO2 PRESSURE-ENTHALPY DIAGRAM FOR HEAT RECOVERY PARAMETRIC 4

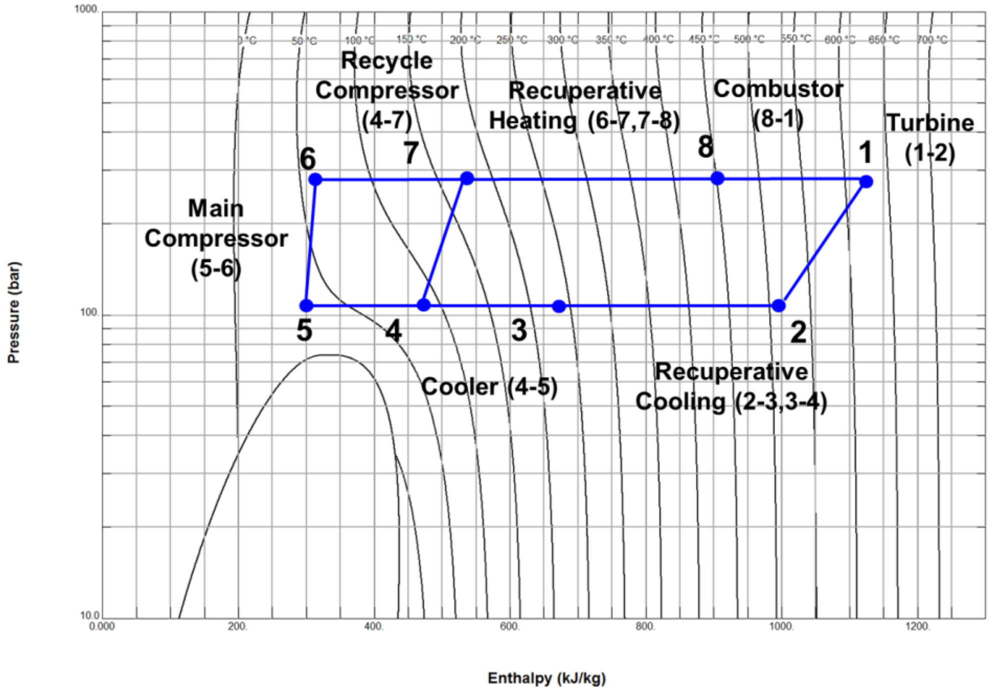


FIGURE A-31: ASPEN PLUS SCHEMATIC OF RECOMPRESSION PARAMETRIC 1

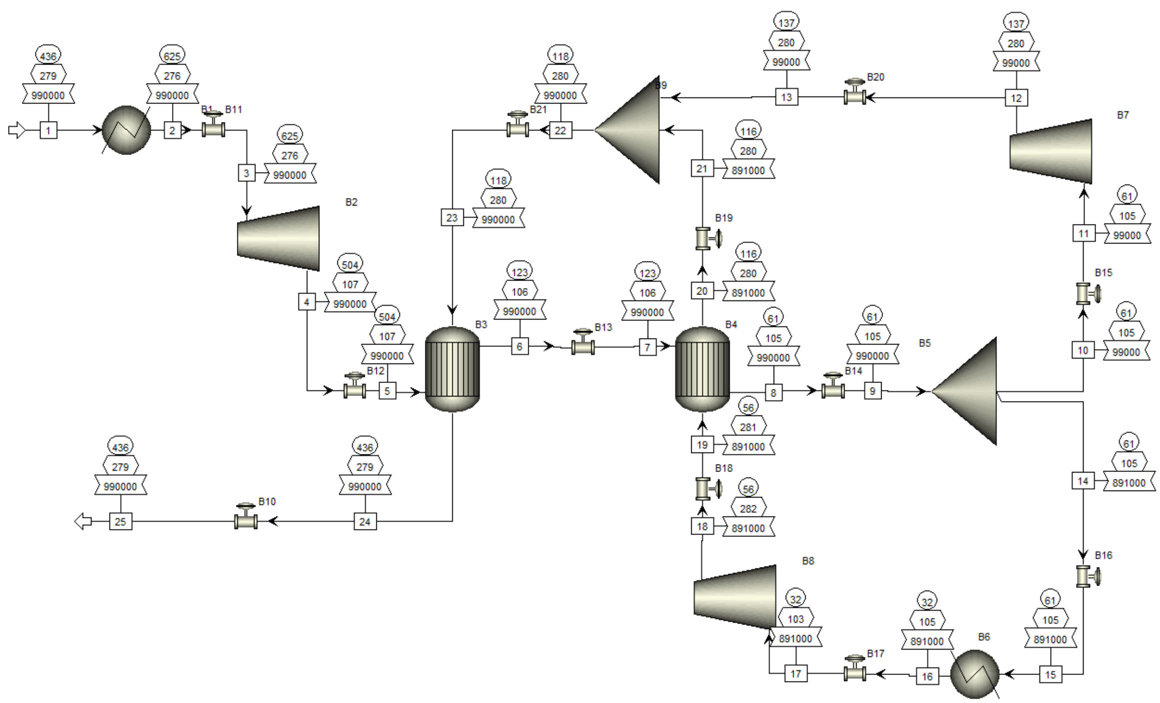


FIGURE A-32: sCO2 TEMPERATURE-ENTROPY DIAGRAM FOR RECOMPRESSION PARAMETRIC 1

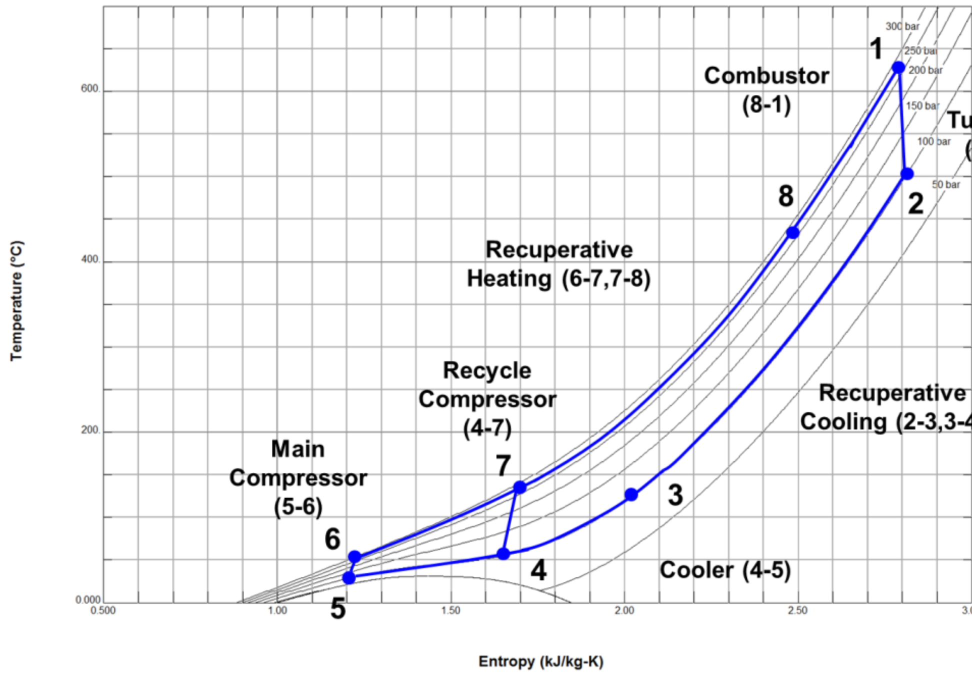


FIGURE A-33: sCO2 PRESSURE-ENTHALPY DIAGRAM FOR RECOMPRESSION PARAMETRIC 1

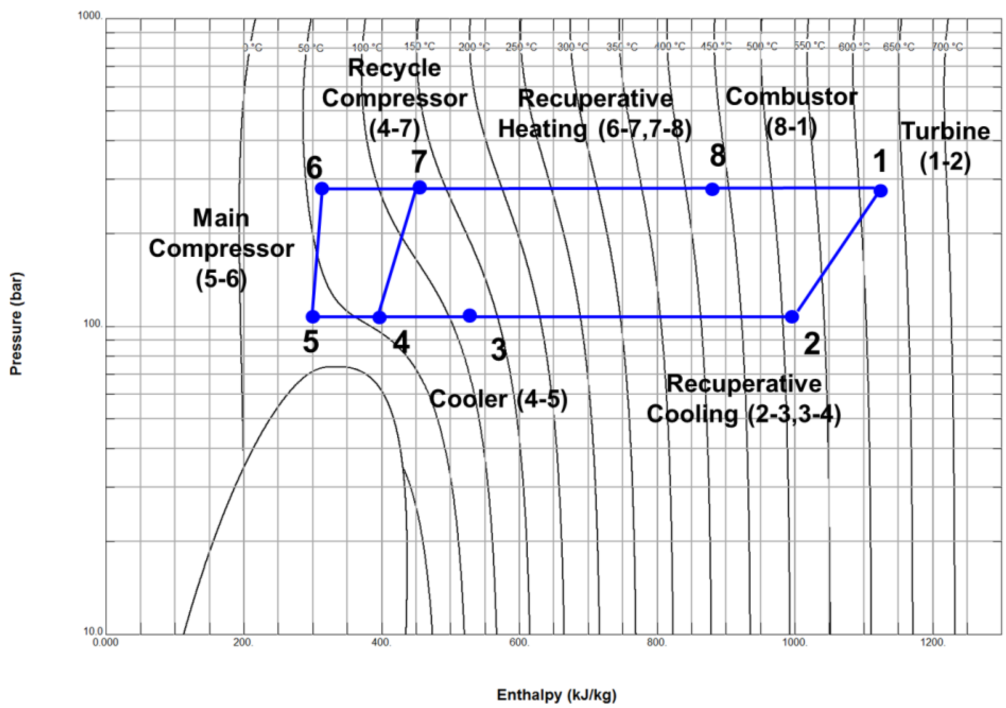


FIGURE A-34: ASPEN PLUS SCHEMATIC OF RECOMPRESSION PARAMETRIC 2

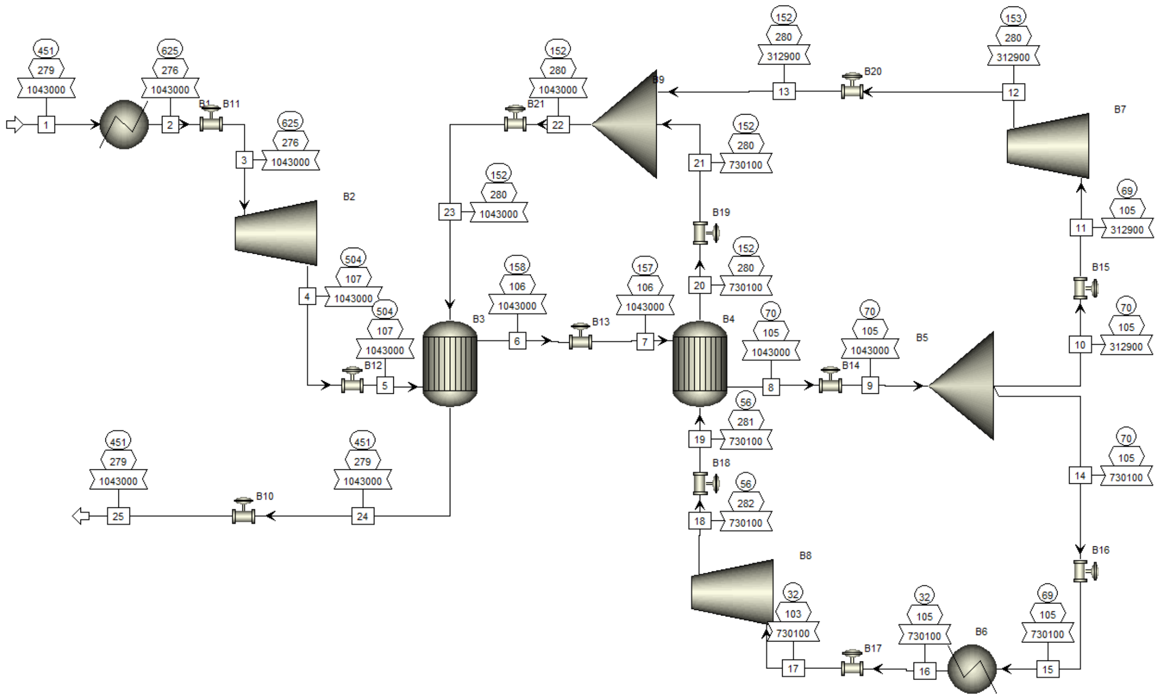


FIGURE A-35: sCO2 TEMPERATURE-ENTROPY DIAGRAM FOR RECOMPRESSION PARAMETRIC 2

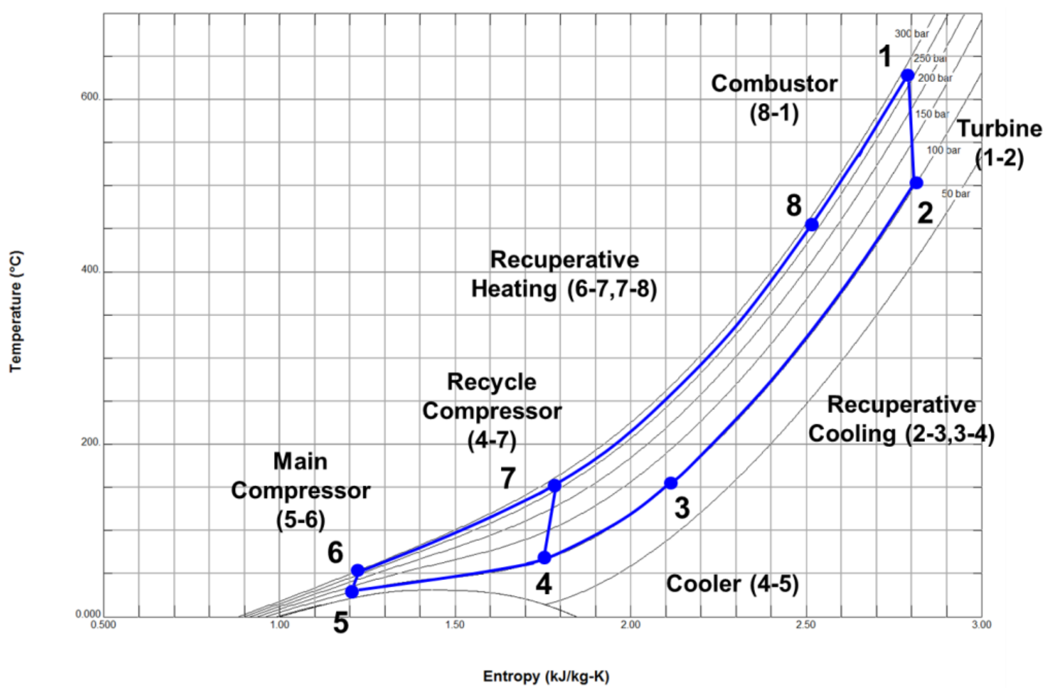


FIGURE A-36: sCO2 PRESSURE-ENTHALPY DIAGRAM FOR RECOMPRESSION PARAMETRIC 2

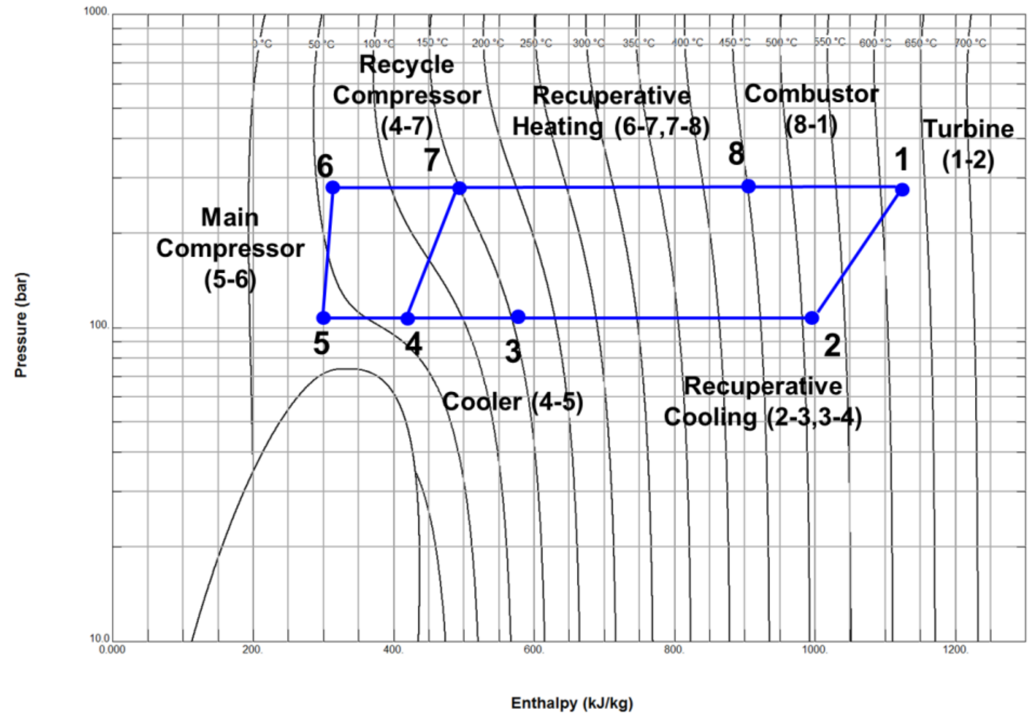


FIGURE A-37: ASPEN PLUS SCHEMATIC OF RECOMPRESSION PARAMETRIC 3

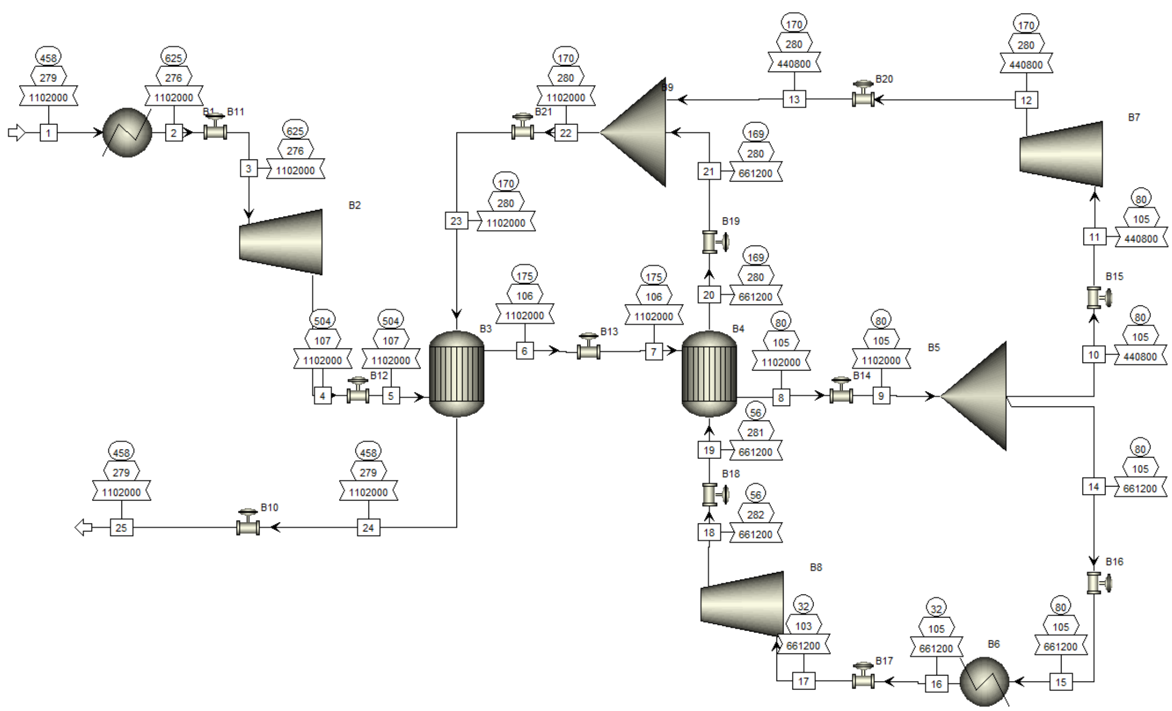


FIGURE A-38: sCO2 TEMPERATURE-ENTROPY DIAGRAM FOR RECOMPRESSION PARAMETRIC 3

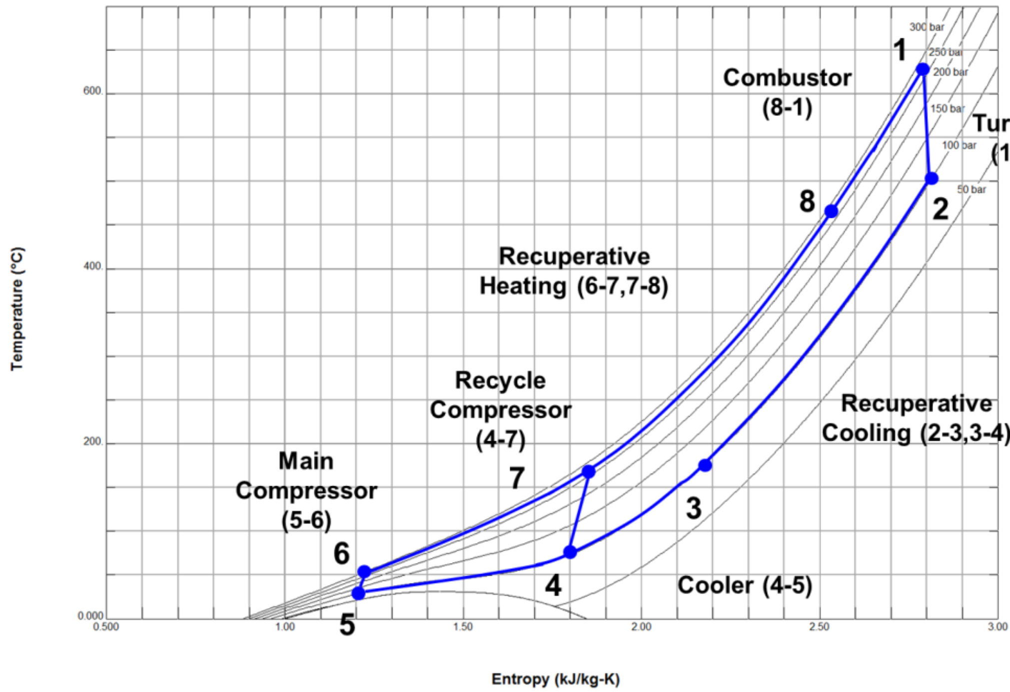


FIGURE A-39: sCO2 PRESSURE-ENTHALPY DIAGRAM FOR RECOMPRESSION PARAMETRIC 3