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A Practical Look at Assumptions and Constraints for Steady State Modeling of sCO_2 Brayton Power Cycles

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ABSTRACT

The National Energy Technology Laboratory (NETL) and the Electric Power Research Institute (EPRI) share a common goal of advancing economically-viable, environmentally-responsible power generation technologies - such as supercritical CO_2 (sCO_2) power cycles - through research, development, and demonstration efforts. The commercial success of Brayton power cycles using sCO_2 as the working fluid will be highly dependent on achieving greater overall power plant efficiencies and/or lower capital costs than plant designs that employ the mature steam-Rankine power cycle. A number of attractive power cycle designs have been explored over the last decade, but it is likely that the optimum power cycle has not yet been identified. Indeed, as EPRI and NETL systems analysis studies have shown, optimizing the power cycle design is likely to be dependent on the specific heating and cooling resources to be exploited.

Modeling sCO_2 Brayton power cycles requires system constraints and component performance metrics such as pressure losses and temperature limitations that are generally unrelated to the underlying thermodynamic characteristics of the cycle. Differing assumptions made by the model designers may lead to significantly different results which are due to the assumptions and not to the underlying thermodynamics which characterize the cycle performance. In addition, some assumptions made by modelers are not likely to be achievable in practice due to limitations on materials or allowable component costs.

As leaders in setting standards for techno-economic analyses, NETL and EPRI have collaborated to propose a set of modeling assumptions that can serve as a common baseline for sCO_2 power plant studies. These will include discussion of turbine inlet temperature specifications (high temperature and pressure limitations), compressor inlet temperature and pressure specifications (cooling limitations), heat exchanger effectiveness/temperature differentials, heat exchanger pressure drop, full- and part-load turbine efficiencies, and full- and part-load compressor efficiencies. Assumptions will be discussed in the context of both indirect and direct sCO_2 cycles.

Introduction

The most recent interest in closed Brayton power cycles using CO_2 as the working fluid started in the early years of the 21st Century. The early interest was largely focused on the Gen IV nuclear application for which (relatively inert) CO_2 might offer safety and cycle efficiency advantages over a comparable steam Rankine cycle in sodium-cooled reactor designs. Other working fluids considered for this application include air, nitrogen, and helium. In comparison to these, CO_2 offers the considerable advantage of a relatively high critical temperature (31°C, 88°F). At temperatures marginally above this and above the critical pressure (7.4 MPa, 1071 psia), CO_2 compressibility is low in comparison with the other gases, resulting in significantly lower compression power requirements which goes directly to increasing overall cycle efficiency. The power cycles studied are usually configured with the compressor inlet pressure marginally higher than the supercritical pressure. These are commonly referred to as sCO_2 Brayton power cycles.

In recent years sCO_2 Brayton power cycles have also been considered for generating power from concentrating solar thermal resources, geothermal resources, “waste” thermal resources, and fuel-combustion products. In addition a “semi-closed” direct oxy/fuel-fired version of the power cycle is under development.

Considerable effort has been undertaken over the last decade to identify configurations of the sCO_2 Brayton power cycle that optimize some figure of merit, commonly the overall power cycle efficiency. As the largest deployment to date of a complete power cycle is on the order of 10 MW, specifically designed for the “waste” heat application [1], a considerable amount of field work will be required to bring the associated technologies to commercial readiness for large scale concentrated solar, nuclear, or fossil energy applications. During this process it is likely that power cycle configurations now anticipated to be “optimal” will need to be modified to conform to real-world, commercial constraints.

Engineering and economic assessments of proposed power plant configurations is a fundamental activity undertaken to identify those configurations and deployments that can maximize efficiency, minimize cost of power, or provide some other desirable feature. These assessments are often used to compare alternate configurations or deployments. In order to be useful, the costs and performance of alternative configurations must be controlled to a common basis. Features of the power plants proposed which are not directly associated with the technology alternatives should be as similar as possible in order to make the comparisons useful and meaningful.

The work described here identifies assumptions and constraints for steady state modeling of sCO_2 Brayton Power Cycles which are commonly used by the National Energy Technology Laboratory (NETL) and Electric Power Research Institute (EPRI) when undertaking engineering/economic assessments. These are recommended to sCO_2 power cycle designers and researchers as design constraints that allow more meaningful results for end-users and more meaningful comparisons of results between researchers.

General Technical Assumptions

Techno-economic analyses generally include a plant operating at a reference site with specific conditions, as well as a thermal resource with certain characteristics. Options for defining these characteristics in a standardized format are discussed below.

Ambient Environment Operating Constraints

In general, a nominal plant location should be specified to ensure that environmental conditions which may affect the plant's performance are based on historical observations. This is particularly important for solar and geothermal sCO_2 plants, which are confined to regions with appropriate resources to be utilized in powering the cycle. Table 1 below lists potential locations that have been used by NETL and EPRI in many of their techno-economic analyses. Utilization of the site ambient conditions also allows for easy comparison of sCO_2 plant analyses with competing technologies analyzed using the same location. Western United States locations have been used in studies utilizing Powder River Basin coals, but may also be useful for geothermal sCO_2 cycle studies. Generic plant locations useful for fossil- or nuclear-fueled sCO_2 plants in the U.S. are typically based in the Midwest, and utilize either ISO conditions (NETL) or actual conditions for Kenosha, Wisconsin (EPRI). Techno-economic analyses for specific sites should report the site conditions listed in Table 1 for the site chosen.

Table 1: Plant Site Conditions

Site Conditions	**Montana** [2]	**Midwest ISO** [2]	**Kenosha, WI** [3]
Elevation, m (ft)	1,036 (3,400)	0 (0)	184 (604)
Barometric Pressure, MPa (psia)	0.090 (13.0)	0.101 (14.7)	0.0993 (14.4)
Average Ambient Dry Bulb Temperature, °C (°F)	5.6 (42)	15 (59)	15 (59)
Average Ambient Wet Bulb Temperature, °C (°F)	2.8 (37)	10.8 (51.5)	13 (55)
Design Ambient Relative Humidity, %	62	60	60
Cooling Water Temperature, °C (°F)	8.9 (48)	15.6 (60)	--

Fuel/Heat Source Specifications

The specific characteristics of the cycle heat source should be specified such that future or alternate analyses can be compared on a consistent basis. In fossil-fuel applications, this involves specifying the coal or natural gas characteristics, as well as the oxidant's specifications. NETL's Quality Guidelines for Energy Systems Studies (QGESS) includes specifications for several types of coals [2]. For consistency and ease of comparison against IGCC and pulverized coal (PC) NETL baseline studies, sCO_2 power cycle analyses can use one of the coals detailed in Table 2. Illinois #6 is a bituminous coal used in NETL's Bituminous Baseline studies [4, 5], while Rosebud and Beulah-Zap are sub-bituminous (Powder River Basin) and lignite coals, respectively, used in NETL's Low Rank Coal Baseline studies [6]. If the solid fuel requires drying to a specified moisture content for the sCO_2 cycle under consideration, ensure that the thermal requirements this process are included in the plant's energy balance.

For gas-fired systems, the average U.S. natural gas composition used in NETL systems studies is listed in Table 3. Most notably, this includes several higher hydrocarbons that affect the fuel's heating value relative to pure methane, as well as nitrogen that affects CO_2 purity and compression power requirements in direct-fired sCO_2 systems.

Table 2: Specifications for Common Reference Coals [2]

Coal	Beulah-Zap		Rosebud PRB		Illinois #6	
Location	Freedom, ND		Montana		Franklin Co., IL	
Rank	Lignite		Sub-bituminous		HV Bituminous	
	As Rec'd.	Dry	As Rec'd.	Dry	As Rec'd.	Dry
Proximate Analysis (weight %)						
Moisture	36.08	0	25.77	0	11.12	0
Ash	9.86	15.43	8.19	11.04	9.70	10.91
Volatile Matter	26.52	41.48	30.34	40.87	34.99	39.37
Fixed Carbon	27.54	43.09	35.70	48.09	44.19	49.72
HHV (kJ/kg)	15,391	24,254	19,920	26,787	27,113	30,506
LHV (kJ/kg)	14,804	23,335	19,195	25,810	26,151	29,444
Ultimate Analysis (weight %)						
Carbon	39.55	61.88	50.07	67.45	63.75	71.72
Hydrogen	2.74	4.29	3.38	4.56	4.50	5.06
Nitrogen	0.63	0.98	0.71	0.96	1.25	1.41
Chlorine	0.00	0.00	0.01	0.01	0.29	0.33
Sulfur	0.63	0.98	0.73	0.98	2.51	2.82
Oxygen	10.51	16.44	11.14	15.01	6.88	7.75

For waste heat applications, the composition, temperature, pressure, and flow of the waste heat stream being utilized should be specified. These are typically application specific, so reference conditions cannot be listed here. However, if a specific waste heat stream has not been targeted, such as exhaust from a particular gas turbine, the heat source specifications can be drawn from existing natural gas combined cycle (NGCC), or other, system analyses available in the literature [4].

For concentrated solar power (CSP) applications, in concert with the location specification, the average hours of sunlight at a particular location are required, as well as daily or seasonal incident solar radiation variation for detailed techno-economic analyses. NREL has developed a Solar Prospector tool that can be used to assist with this process [7].

Table 3: Reference Natural Gas Composition [2]

Component	Volume Percentage	
Methane, CH_4	93.1	
Ethane, C_2H_6	3.2	
Propane, C_3H_8	0.7	
n-Butane, C_4H_{10}	0.4	
Carbon Dioxide, CO_2	1.0	
Nitrogen, N_2	1.6	
Total	100.0	
	LHV	**HHV**
MJ/scm (Btu/scf)	34.71 (932)	38.46 (1033)
kJ/kg (Btu/lb)	47,454 (20,419)	52,581 (22,625)

Closed Brayton Power Cycle Modeling

Turbine

Inlet Temperature

The increase in sCO_2 power cycle efficiency with increasing turbine inlet temperatures has been well documented, and drives a trend towards higher temperatures to improve the attractiveness of the sCO_2 cycle from an efficiency perspective. These temperatures are limited by the materials of construction for the sCO_2 turbine and the upstream heat exchanger which transfers heat into the cycle. The current state of the art is directly analogous to ultra-supercritical (USC) steam power plants, which operate at turbine inlet temperatures of 600-620 °C (1112–1148 °F) with lower cost ferritic and austenitic steels. A U.S. Department of Energy program is currently underway which aims to demonstrate turbine inlet temperatures of 700 °C in a 10 MW_e sCO_2 pilot plant [8]. This is the recommended turbine inlet temperature assumption for near-term indirect sCO_2 cycle studies.

Recently, industry and government consortiums have been identifying, fabricating, and testing advanced nickel alloys for use to 760 °C and 34.5 MPa (5000 psia) in Advanced Ultra-supercritical (AUSC) steam boilers and turbines [9]. This has led to a completed ASME design code case for Inconel 740H, and another in progress for Haynes 282, which will allow design of high pressure components with a maximum use temperature of 800 °C (1472 °F). Allowing for a safety margin, turbine inlet temperatures up to 760 °C (1400 °F) can be assumed for longer-term indirect-sCO_2 systems studies as use of these materials becomes more widespread.

For direct-fired sCO_2 systems, thermal input occurs internally via combustion of fuel and oxygen in a dilute sCO_2 environment, eliminating the constraint on transferring heat into the system across a large pressure boundary, as in a boiler or CSP receiver. Here, the combustor walls can be insulated or cooled to maintain a safe wall temperature, and the internal combustion products can pass to the turbine in much the same manner as a gas turbine. Modern gas turbine inlet temperatures are approaching 1700 °C through the use of ceramic blades and advanced internal blade cooling strategies, though temperatures in direct-fired sCO_2 turbines are currently 1100 – 1200 °C (2012-2192 °F), which also require blade cooling. This is limited by the need to keep turbine exit/recuperator inlet temperatures no greater than 760 °C (1400 °F) per the above material limitations. For sCO_2 turbine pressure ratios of about 10, which are considered in most direct-sCO_2 studies, this yields a turbine inlet temperature of about 1150 °C (2102 °F), which is the recommended assumption for direct sCO_2 cycle analyses. Little benefit has been shown in considering higher temperatures in full sCO_2 cycle models [10].

Inlet Pressure

Similar to the turbine inlet temperature, inlet pressures are dictated by material constraints at high temperatures, though these can be mitigated to some extent with double shell turbine casings, similar to those used in steam turbine casing designs. Turbine inlet pressures up to 34.5 MPa (5000 psi) can be specified as an upper limit, similar to those considered for AUSC steam turbine designs. However, the effect of pressure on overall system cost should be considered, where thicker tubes, pipes, and turbine casings are required for higher pressure, which may raise the plant's capital cost significantly if nickel alloys are required.

Isentropic Efficiency

Turbine efficiencies are primarily a function of scale and speed. As turbine sizes increase, their optimal speed at which peak efficiency can be achieved decreases. Utility scale turbines are axial flow machines, and 90% is a standard and accepted isentropic efficiency assumption. Turbine efficiencies gradually decrease as power output decreases, and below about 30 MW, radial turbines are more efficient [11], with design isentropic efficiencies up to 85%. These are based on industry standard design practices utilizing dimensionless turbine speed and tip diameters. Recommended turbine efficiencies are 90% for axial turbines above 30 MW, and 85% for radial turbines below 30 MW.

Part Load Performance

While typically not used in steady state cycle analyses, part load turbine efficiencies are useful to have when considering startup, shutdown, and off-design operation. This requires that a turbine geometry be specified, from which performance at off-design flow rates can be determined. Examples are found in the literature [1, 12].

Outlet Conditions

As noted above, turbine outlet temperatures in direct-fired sCO_2 systems should not exceed 760 °C (1400 °F), the maximum temperature for materials which will convey the exhaust to recuperators and for construction of the hot end of the recuperator. Otherwise, this temperature is primarily dictated by the turbine inlet temperature, isentropic efficiency, and pressure ratio. Note that very low turbine outlet pressures may increase system capital costs by increasing the size of the turbine and the low pressure flow passages in the recuperator due to reduced sCO_2 densities at low pressure.

Compressor

Inlet Temperature

Compressor power requirements are lower with increasing fluid inlet densities, therefore, it is generally beneficial to operate compressors with the minimum temperature that can be achieved with the selected cooling method and sCO_2 cooler design. Details on these items can be found in the appropriate sections below, though in general, compressor inlet temperatures below 20 °C are rarely achievable in practice without the addition of refrigeration processes.

Cycle designers have generally avoided inlet conditions in the region near CO_2's critical point at 31 °C and 7.37 MPa (88 °F and 1071 psia) due to the unknown erosive damage effects that near-critical or partially liquefied CO_2 might have on the compressor. With additional experimentation and operational experience in this region [13, 14], this is less of a concern than in years past. Given the large variations in CO_2 physical properties near the critical point, however, design state points should avoid this region, as small variations or perturbations in cooling water temperature, for instance, could have a large impact on the overall cycle performance and control stability.

Inlet Pressure

The compressor inlet pressure is one of the primary variables at the cycle designer's disposal for optimizing the performance of the cycle. Restrictions on this pressure are limited to consideration of sub-critical pressures for condensing CO_2 cycles, to supercritical pressures for non-condensing cycles. Until system operability demonstrations at the critical point have been performed, inlet pressures in the immediate vicinity of the critical pressure (7.17-7.57 MPa, 1040-1098 psia)) should be avoided to minimize large variations in CO_2 physical properties from affecting overall cycle performance.

Isentropic Efficiency

As with sCO_2 turbines, compressor isentropic efficiencies are a function of compressor type (centrifugal or axial), size and speed. Well-designed centrifugal compressors are capable of efficiencies of 75-87% and axial compressors are capable of 80-91% [15]. The main compressor in a recompression sCO_2 cycle is expected to be a centrifugal compressor at all cycle sizes, as this type of compressor is very adept at handling large variations in fluid density that may occur near the CO_2 critical point. The recompression compressor is expected to be of centrifugal design at low cycle power levels, transitioning to an axial compressor above cycle power outputs of 100 MWe [11]. In most studies, 85% is an accepted isentropic efficiency for either the main or recompression compressors, and is the recommended assumption.

Inter-cooled Temperature/Pressure Drop

Intercooled compressors can typically achieve the same low sCO_2 temperature as produced at the exit of the sCO_2 cooler, with fairly low pressure drop. This is discussed in detail in the corresponding section below.

Part Load Performance

As with sCO_2 turbines, part load compressor performance maps help in modeling cycle startup, shutdown, and part-load operation, and require a basic compressor design speeds and geometries. Specific examples are available in the literature [1].

Recuperator(s)

Effectiveness/Approach Temperature

sCO_2 Brayton power cycle efficiency is critically dependent on the recuperative heat exchangers used to transfer residual in the turbine exhaust to the high pressure CO_2 before it enters the primary heater. The model performance of these heat exchangers will generally be a function of either an effectiveness or approach temperature (cold/warm temperature difference at the hot end of the heat exchanger). Either parameter can be calculated from the other.

While it is tempting to specify very high heat exchanger effectiveness in order to achieve high cycle efficiency, the cost of recuperative heat exchangers will rise dramatically with effectiveness. Figure 1 suggests that compact recuperator cost roughly doubles as specified effectiveness is increased from 80% to about 93%, and then increases very dramatically at higher specified effectiveness. It is likely that it will be challenging to design shell and tube recuperators that achieve an effectiveness of 90% [16].

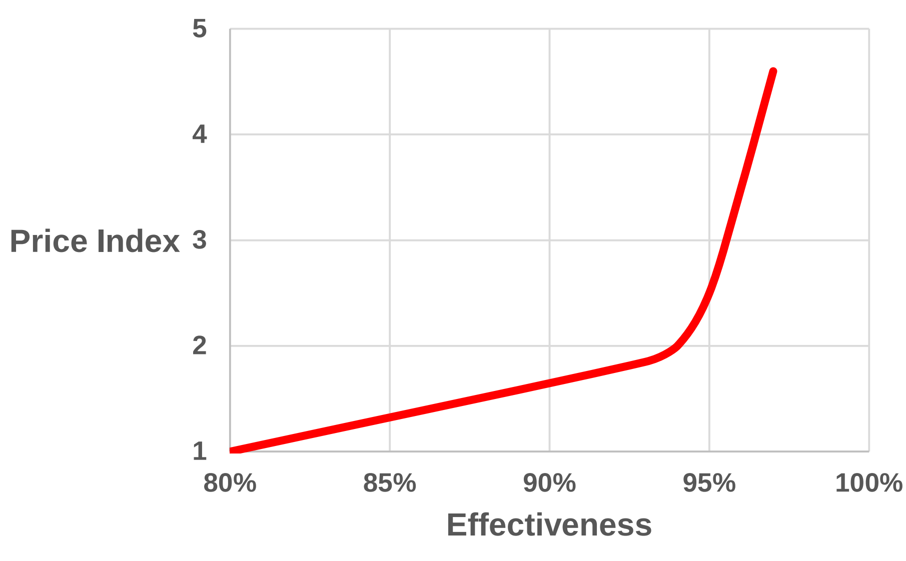

Figure 1: Influence of Specified Effectiveness on Compact Recuperator Cost [17]

Unless a detailed recuperator model is included in a sCO_2 Brayton power cycle modelling effort, model recuperator effectiveness greater than about 93% is not recommended due to expectations of unrealistic cost. Indeed, recuperator effectiveness greater than about 90% may not be economically justified.

No recuperator temperature should exceed 760 °C (1400 °F), the maximum temperature for which ASME code-approved materials are available with which to build the pressure-containing components. The use of Inconel 740H has recently been approved for use to 800 °C (1472 °F), with a maximum recommended use temperature of 760 °C (1400 °F) providing a margin of safety, though efforts are underway to extend the temperature-stress curve in the ASME code case to 825 °C (1517 °F). The ASME code case for Haynes 282 is currently in process, and is expected to be valid for similar temperatures and pressures.

Pressure Drop

Multiple literature sources suggest that recuperator cold side (high pressure) pressure drop of approximately 140 kPa (20 psid) and a hot side (low pressure) pressure drop of 280 kPa (40 psid) can be reasonably used. Note that the choice of this parameter also has a large impact on the size and cost of the recuperator.

Compressor Inlet Cooler/Intercooler

The temperature (and pressure) of the compressor inlet flow has an extra-ordinary effect on overall cycle performance as it is near the critical point where the properties of CO_2 change dramatically with temperature. For this reason, care must be exercised in deciding how to model the compressor inlet cooler; theoretical cycle performance will generally improve as this temperature approaches the critical temperature due to reduced compressor work. On the other hand, practical heat rejection constraints must be observed.

Process cooling for a sCO_2 Brayton power cycle will be provided in the same way cooling is provided to steam-electric power plants. Candidate cooling technologies are listed in Table 3. Note that wet (evaporative) cooling systems are generally designed to supply cooling water at an overall design temperature (based on design wet bulb/dry bulb temperatures), and, usually, summer day wet bulb/dry bulb temperatures. For operational reasons, they will generally have a minimum cooling water supply temperature to avoid problems associated with freezing. Once-through and dry water-cooling systems will generally operate with a minimum cooling water supply temperature near 5 °C (41 °F), again to avoid challenges associated with freezing.

Compressor inlet cooler/intercooler coolant temperatures will be determined by the cooling system selected, local ambient wet bulb/dry bulb temperature, and, for once-through systems, the local water temperature. In the absence of specific data, the coolant temperatures (conservatively high) indicated in Table 3 might be used.

Note that all cooling systems will impose one or both of the following cold-side auxiliary power demands:

- Coolant pressure drop through the cooler
- Air/water pressure drop/lift in the cooling tower

If the cooling system is not specifically modeled, aggregate cold-side auxiliary power loads indicated in Table 3 might be used.

Table 3: Compressor Inlet Cooler/Intercooler Cooling Systems [18]

System	Generic Coolant supply temperature	Generic CO_2 cooler approach temperature (CO_2 outlet temperature less cooling inlet temperature)	Estimated Aggregate Cold-side Auxiliary Power use (per MWth cooler duty)
Water-cooling with wet (evaporative) cooling tower	32 °C (90 °F)	8 °C (13 °F)	Mechanical Draft: 16 kWe Natural Draft: 10 kWe
Water-cooling with dry cooling tower	43 °C (110 °F)	8 °C (13 °F)	Mechanical Draft: 16 kWe Natural Draft: 10 kWe
Hybrid wet/dry cooling systems	37 °C (100 °F)	8 °C (13 °F)	Mechanical Draft: 16 kWe Natural Draft: 10 kWe
Once-through water cooling	21 °C (70 °F) North Sea: 5°C (39°F)	8 °C (13 °F)	9 kWe
Direct, air-cooling with mechanical draft (radiator)	Ambient dry bulb	15 °C (28 °F)	30 kWe

Effectiveness/Approach Temperature

As with recuperators (see above) selection of a CO_2 cooler effectiveness or approach temperature for a field deployment will be heavily influenced by dramatically increasing costs as effectiveness rises above about 90% (approach temperature drops below about 5 °C [8 °F]). Table 3 lists generic approach temperatures that are commonly used for condensers in steam cycle studies and which are reasonable compromises between cost and cycle efficiency for steam power plants. Given the large effect that cooling temperatures have on sCO_2 power cycle efficiency, further study will likely shift this balance towards higher-cost cooling systems that can deliver lower cooling temperatures for increased plant efficiency [10].

Pressure Drop

CO_2–side pressure drop in the inlet cooler and any compressor intercoolers employed is likely to be modest. Intercooler pressure drops of 7-15 kPa (1-2 psid) are often used. For condensing cycles, the CO_2 condensation process occurs at constant pressure conditions, thus the CO_2 cooler pressure drop can be neglected. In non-condensing cycles, operating [13] and design [19] experience shows that sCO_2 cooler pressure drops are low, and can be assumed to be 15 kPa (2 psid). Cooler design for low pressure drop is necessary, though, where sCO_2 cooling in a baffled shell can lead to 5-10 psid pressure drop [20].

If a direct, air-cooled system is used for plant modeling, the pressure drop delivering the sCO_2 to the air-cooler and returning it to the compressor block should be included.

Primary Heater

The primary heater models will vary greatly, depending on the heat source being exploited. For the nuclear and solar thermal heat sources, comparable heat exchanges are not in common service. The primary heaters which will be deployed for “waste” heat utilization will be comparable to HRSGs used for combustion turbine combined cycles. Primary heaters deployed for fuel-fired service will be comparable to their counterparts in fuel-fired steam cycle power plants.

Effectiveness/Approach Temperature

In the solar application, the heat flux to the primary heat exchanger will generally be specified along with the sCO_2 temperature leaving the absorber. There is no need to assume an effectiveness or approach temperature.

The nuclear application will incorporate a reactor coolant to CO_2 heat exchanger. This heat exchanger will be comparable to that of recuperators (see above). Lacking a specific primary heater model, an effectiveness of no more than 90%-93% should be used for cycle modeling purposes.

Modeling sCO_2 Brayton power cycles for the "waste" heat and fuel-fired applications will require a fairly detailed combustion/gas-side model, as much of the heat available on the gas-side will be at a temperature lower than the turbine inlet temperature, in the temperature range in which recuperation also occurs. The use of combustion air pre-heaters will also complicate the low-temperature heat balance. It is generally the case that coal combustion products are cooled to no less than approximately 120-175 °C (250–350 °F) to avoid sulfuric acid mist formation and associated corrosion in flue gas handling components. For natural gas (and other very low-sulfur fuels) the flue gas exit temperature might be reliably reduced to 120 °C (250 °F), constrained by the need for buoyancy to disperse the flue gas plume. Approach temperatures no lower than 28 °C (50 °F) and commonly near 55 °C (100 °F) are used in design of water-cooled flue gas heat exchangers.

Pressure Drop

The CO_2-side pressure drop in nuclear and solar primary CO_2 heaters and in "waste" heat applications is unlikely to be substantially greater than 200 kPa (30 psid).

Pressure drop in fuel-fired applications has not been studied in detail. It is common for water-side pressure drop in fired steam generators to be near 3,400 kPa (500 psid). Pressure drops this high are employed to give boiler designers flexibility in precisely controlling flows to different parts of the boiler, allowing them to maximize heat flux (minimizing furnace size) without inordinate risk of overheating the boiler tubes. The low power required to increase the pressure of incompressible water does not impose a significant auxiliary power penalty associated with this pressure drop. The (power) cost to compress CO_2 is significantly greater and it is unlikely that a 3,400 kPa (500 psid) pressure drop in the fired heater will be acceptable. The problem is exacerbated by the fact that sCO_2 mass flow through the fired sCO_2 heater is 5-10 times water flow through a steam generator for the same heat absorbed. A pressure drop of 3,400 kPa (500 psid) would be a conservative approach. It is possible that further design work could halve this requirement but it is difficult to see how the pressure drop would fall to significantly less than 700 kPa (100 psid).

Interconnecting Pipes

No pipe temperature should exceed 760 °C (1400 °F), the maximum temperature for which ASME code-approved materials are available with which to build the pressure-containing components. If higher fluid temperatures are to be modeled, the pressure-boundary piping components must be cooled to below this level. (Note that this is <u>not</u> common practice in power plant design and should be specifically justified and the effects included in the model.)

Temperature Drop

Temperature drop in piping is largely a function of the insulation applied and the length of the pipe run. If specific piping designs are developed, temperature drop can be modeled. Lacking a model for piping heat loss, a temperature loss between components of 3 °C (5 °F) is easily justified.

Pressure Drop

Piping pressure drop is largely a function of fluid properties, fluid velocity, and length of the pipe run. If specific piping designs are developed, pressure drop can be modeled. Lacking a model for piping pressure drop, a value between 34 kPa and 345 kPa (5 psid and 50 psid) should be used, depending on the anticipated length of the piping run in a range 3 m to 60 m (10 ft to 200 ft).

Net Plant Performance

Beyond the performance of the sCO_2 cycle, its integration into an overall power production plant must be considered in order to accurately assess its performance and cost of electricity relative to other power production methods. In particular, auxiliary power loads from ancillary balance of plant equipment can be large, depending on the particular application, and can significantly affect the plant's overall performance and cost of electricity metrics.

Primary heater efficiency

Primary heater efficiency is roughly defined as the quantity of heat transferred to the sCO_2 cycle divided by the quantity of heat available in the heat source. Specification of this value is very application specific, and requires at least a basic design for the thermal integration of the heat source with the sCO_2 cycle in the primary heater, including heat losses to the environment.

Generator & Gearbox efficiency

Conversion of sCO_2 turbine shaft power to electricity requires the use of a generator. The recommended generator efficiency for this conversion is 98.5% [21].

In large power plants, the turbine operates at synchronous generator speeds matched to the frequency of the standard line voltage employed in that country or region (e.g., 60 Hz in the U.S., 50 Hz in Europe). As plant size decreases below about 100 MWe, increases in turbine shaft speed are required to maintain high turbine efficiency, and a gearbox is required to transfer the turbine shaft power to a synchronous generator. The recommended gearbox efficiency is 99% [11].

Drive motor efficiency

The compressors in a sCO_2 plant may either be turbine- or motor-driven. Motor-driven compressors allow for independent operation and improved cycle control. Recommended motor efficiencies are 95% for power < 1 MW, 96.5% between 1 and 10 MW, and 97% for sizes over 10 MW [21].

While operation of the compressor(s) on the same shaft as the turbine eliminates motor losses, it also limits the shaft speed to that of the generator, and may result in non-optimal compressor shaft speeds and reduced compressor efficiency. For large plants, it is typically beneficial from an efficiency and control perspective to operate a power turbine at synchronous generator speeds, in parallel with a separate compressor turbine that operates at optimal compressor efficiencies [11].

Balance of Plant

Specific assumptions required for modeling the balance of plant is application specific, and will not be covered in detail here. As a starting point, NETL's Quality Guidelines for Energy Systems Studies covers many of the assumptions required for modeling air handlers, flue gas cleanup processes, cooling water systems, and miscellaneous plant equipment needed in fossil-fueled applications [21].

Economic Modeling

Modeling costs for new technology is a challenging activity. Cost modeling often relies on past experience which is inherently limited for new technologies. Major cost categories to be included in economic modeling include:

- Capital costs and the associated time value of capital expenditures
- Fixed non-fuel operation and maintenance
- Variable non-fuel operation and maintenance
- Fuel costs (generally a function of plant efficiency and duty cycle)

AACE International publishes a general approach to estimating capital costs for large projects [22]. AACE International identifies five levels of cost estimating efforts as indicated in Table 4. It is generally the case that uncertainty in cost estimates is reduced as more effort is undertaken (more $ spent) in the cost estimate. Any cost estimates reported should identify the level of effort undertaken. The AACE Taxonomy indicated in Table 4 is generally adequate for such purposes.

Table 4: AACE Cost Estimate Classifications

Estimate Class	Primary Characteristic	Secondary Characteristic			
	Level of Project Definition % of complete project definition	**End Usage** Typical Purpose of the Estimate	**Methodology** Typical Estimating Method	**Expected Accuracy Range** Typical variation in low and high ranges	**Preparation Effort** Typical degree of effort relative to least cost index of 1(*)
Class 5	0% - 2%	Concept screening	Capacity Factored, Parametric Models, Judgment, of Analogy	L: -20% to -50% H: +30% to +100%	1
Class 4	1% - 15%	Study or Feasibility	Equipment Factored or Parametric Models	L: -15% to -30% H: +20% to +50%	2 to 4
Class 3	10% - 40%	Budget, Authorization, or Control	Semi-Detailed Unit Costs with Assembly Level Line Items	L: -10% to -20% H: +% to +30%	3 to 10
Class 2	30% - 70%	Control or Bid/Tender	Detailed Unit Cost with Forced Detailed Take-Off	L: -5% to -15% H: +5% to +20%	4 to 20
Class 1	50% - 100%	Check Estimate of Bid/Tender	Detailed Unit Cost with Detailed Take-Off	L: -3% to -10% H: +3% to +15%	5 to 100

Economic modeling work should include capital cost estimates meeting, at a minimum, AACE Class 5 requirements. The cost of conducting an AACE class 4 capital cost estimate for a full scale base load electric power generating plant is likely to be in excess of $200,000.

EPRI and NETL have published jointly-developed procedures for constructing economic modeling efforts and uniform reporting [23]. Table 5 lists capital cost categories and definitions of accumulated capital costs used by these organizations. The calendar year of the capital cost estimate should be reported in order to inflate the reported capital costs over time.

Economic modeling should conform to the procedures published by EPRI/NETL in order to provide useful, unambiguous results. (Alternatively, comparable detail on the procedures used, and differences from EPRI/NETL procedures, should be reported.) NETL has also published a set of supporting guidelines for conducting engineering and economic assessments which provide useful information on cost scaling, retrofit costs, first of a kind vs. Nth of a kind, etc. [24]

Note that economic modeling results can be particularly affected by the following factors (which should be reported along with results):

- Purely economic assumptions (debt/equity, time cost of money, etc.)
- Contingencies (representing incidental project costs in deployment of new technologies at the required scale.)

Table 5: Capital Cost Nomenclature

Capital Cost Categories	Notes
A. Bare Erected Cost	Total constructed costs of all on-site processing and power production units and facilities that directly support production, to the battery limits.
B. EPC Cost	Engineering and home office costs, overhead, and fees.
C. Contingencies	Costs associated with the uncertainty in general project costs and scale-up
D. Owner's Cost	Pre-paid royalties, land costs, financing costs, initial inventory (fuel, chemicals, catalysts, spares, etc.), pre-production (start-up).
E. IDC/AFUDC, escalation	Cost of financing progress payments to vendors and contractors during construction and increases in costs due to escalation during the construction period.
Capital Cost Accumulations	
Total Plant Cost	A + B + C
Total Overnight Cost	A + B + C + D
Total Plant Investment	A + B + C + E
Total Capital Required (Total As-Spent Capital)	A + B + C + D + E

Conclusions

In modeling sCO_2 power systems, consistency in applying modeling assumptions is crucial to putting forth reasonable cycle designs, and in enabling a fair and meaningful comparison of researchers' results. In an effort to standardize sCO_2 systems analyses, a set of recommended modeling assumptions is proposed in this work. These rely heavily on the standard modeling practices employed by NETL and EPRI in performing steady state techno-economic analyses, as well as sCO_2-specific assumptions derived from literature and industry experience.

While these recommendations are in no way binding or all-inclusive, they serve as a starting point from which a more comprehensive set of assumptions can be constructed. As such, specific feedback on these modeling assumptions is sought, particularly from industry and experimental research programs, so that revised and more accurate recommendations for sCO_2 modeling assumptions can be proposed in the future.

REFERENCES

[1] Held, T.J., "Initial Test Results of a Megawatt-Class Supercritical CO_2 Heat Engine," *The 4th International Symposium - Supercritical CO_2 Power Cycles*, September 9-10, 2014, Pittsburgh, Pennsylvania.

[2] *Quality Guidelines for Energy Systems Studies: Specifications for Selected Feedstocks*. DOE/NETL-341/011812. January 2012.

[3] *Performance and Economic Evaluation of Supercritical CO_2 Power Cycle Coal Gasification Plant*. EPRI, Palo Alto, CA: 2014. 3002003734.

[4] *Cost and Performance Baseline for Fossil Energy Plants Volume 1a: Bituminous Coal (PC) and Natural Gas to Electricity Revision 3*, DOE/NETL-2015/1723. July 2015.

[5] *Cost and Performance Baseline for Fossil Energy Plants Volume 1b: Bituminous Coal (IGCC) to Electricity Revision 2b – Year Dollar Update*, DOE/NETL-2015/1727. July 2015.

[6] *Cost and Performance Baseline for Fossil Energy Plants - Volume 3 Executive Summary: Low Rank Coal to Electricity*. DOE/NETL-2010/1399. September 2011.

[7] http://maps.nrel.gov/node/10

[8] Dennis, R., "U.S. DOE R&D Program on Supercritical CO_2 Power Cycles," EPRI International Conference on Corrosion in Power Plants; San Diego, CA, Oct. 2015.

[9] Weitzel, P.S. *et al.*, "Advanced Ultra-Supercritical Power Plant (700 to 760C) Design for Indian Coal," Technical Paper BR-1884, *Power-Gen Asia*, Bangkok, Thailand, October 3-5, 2012.

[10] IEAGHG, *Oxy-combustion Turbine Power Plants*, Report: 2015/05, August, 2015.

[11] Sienicki, J.J., *et. al*, "Scale Dependencies of SCO_2 Brayton Cycle Technologies and the Optimal Size for a Next-Step SCO_2 Cycle Demonstration," SCO_2 Power Cycle Symposium, Boulder, Colorado, May 24-25, 2011.

[12] *Modified Brayton Cycle for use in Coal-Fired Power Plants*, EPRI, Palo Alto, CA: 2015. 1026811.

[13] *Modeling and Experimental Results for Condensing Supercritical CO_2 Power Cycles*, Sandia Report SAND2010-8840, January 2011.

[14] Lettieri, C., *et al.*, "An Investigation of Condensation Effects in Supercritical Carbon Dioxide Compressors," *The 4th International Symposium - Supercritical CO_2 Power Cycles*, September 9-10, 2014, Pittsburgh, Pennsylvania.

[15] *Perry's Chemical Engineers' Handbook,* 8th Edition, p. 10-45. McGraw-Hill: New York, 2008.

[16] Performance and Cost Targets for sCO_2 Heat Exchangers. T. Held. EPRI/NETL Workshop on Heat Exchangers for Supercritical CO_2 Power Cycles. San Diego, CA. 15 October 2015.

[17] Technology Development Needs for sCO_2 Heat Exchangers. R. LePierres. EPRI/NETL Workshop on Heat Exchangers for Supercritical CO_2 Power Cycles. San Diego, CA. 15 October 2015.

[18] *Power Plant Cooling System Overview: Guidance for Researchers and Technology Developers.* EPRI, Palo Alto, CA: 2015. 3002007194.

[19] *Closed Brayton Power Cycles Using Supercritical Carbon Dioxide As the Working Fluid:Technology Resume and Prospects for Bulk Power Generation*. EPRI, Palo Alto, CA: 2014. 3002004596.

[20] Clementoni, Eric, *Private Communication re: Integrated System Test Loop*, Feb. 16, 2016.

[21] Quality Guidelines for Energy Systems Studies: Process Modeling Design Parameters. DOE/NETL-341/051314. May 2014.

[22] Cost Estimate Classification System – As Applied in Engineering, Procurement, and Construction for the Process Industries. AACE International Recommended Practice 18R-97. TCM Framework: 7.3 – Cost Estimating and Budgeting.

[23] *Toward a Common Method of Cost Estimation for CO_2 Capture and Storage at Fossil Fuel Power Plants.* EPRI, Palo Alto, CA: 2013. 3002000176.

[24] Quality Guidelines for Energy Systems Studies. http://www.netl.doe.gov/research/energy-analysis/quality-guidelines