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Overview

As a higher efficiency replacement to steam-Rankine cycles, indirectly heated sCO₂ power cycles warrant in-depth consideration for use in fossil-fueled power plants. In this configuration, the heat of combustion from an external coal or natural gas combustion process is transferred into the supercritical CO₂ (sCO₂) cycle via a primary heat exchanger, in a similar manner to a boiler in a steam-Rankine power plant. In addition to its typically large-scale, fossil-fueled power plants offer additional challenges to the application of sCO₂ power cycles, most notably in the range of temperatures required for making efficient use of the flue gas thermal energy for power generation. Options for sCO₂ power cycles and efficiently coupling them to primary heat exchangers are considered in this chapter, including recovery of lower temperature flue gas thermal energy, which is critical for effective thermal resource utilization and minimizing the size and cost of the sCO₂ cycle. Performance and cost of fossil-fueled indirect sCO₂ cycles from the literature to date are compared, as well as particular sCO₂ cycle component challenges that are unique to fossil-fueled power plants.

In addition, open cycle, direct-fired sCO₂ cycles have also been proposed that are more analogous to gas turbines than steam-Rankine cycles. In addition to the efficiency advantages of sCO₂ power cycles, these cycles also allow for extraction of CO₂ at inherently high purity and high pressure, making them very valuable for carbon capture and storage consideration. These cycles employ stoichiometric fuel combustion with high-purity oxygen in a heavily diluted sCO₂ environment, such that the working fluid also contains water vapor, incomplete combustion products, and other impurities from the fuel and oxidizer streams. In addition to discussing these challenges, this chapter also discusses oxycombustion issues, turbine cooling needs, and other component-specific challenges. As with the indirect sCO₂ cycles, the performance and costs of direct sCO₂ cycles from the literature are also compared, as well as prospects for future utilization.

Key Terms

Coal, Direct-fired sCO₂, Fossil fuels, Indirect sCO₂, Natural gas, Open cycle sCO₂, Oxycombustion, Syngas.

12.1 Introduction

At similar turbine inlet temperatures, recompression Brayton cycles using supercritical carbon dioxide (sCO₂) as the working fluid offer the potential for higher thermodynamic cycle efficiencies than steam-based Rankine cycles, and are therefore of significant interest for power generation applications. This chapter presents a discussion of technology for indirectly heated recompression Brayton cycles and the directly heated semiclosed Brayton cycle, where both these cycles use sCO₂ as the working fluid. The indirectly fired cycle, as discussed here, is relevant to coal-fueled furnaces (or boilers), while the semiclosed directly heated cycle is applicable to gaseous hydrocarbon fuels, such as coal-derived synthesis gas and natural gas, which create the turbine working fluid through fuel and pure oxygen combustion. The directly heated cycle would be particularly suited for fossil fuel power plants that capture carbon dioxide (CO₂) for geologic storage or use.

A key question in the investment and development of power cycles based on sCO₂ is *can these cycles be configured in a power plant and deliver a higher plant efficiency that translates into a lower cost of electricity (COE)?* For fossil energy sCO₂ power cycle applications there are in general two competing incumbents. One is the coal-fueled boiler with a steam-based Rankine cycle compared with the indirectly heated recompression Brayton cycle (and variants), and the second is the coal-derived synthesis gas or natural gas-fueled combined cycle with either pre- or postcombustion CO₂ capture compared to the directly heated sCO₂ power cycle. For comparison purposes, it is important to consider the state of the art of these incumbent power cycles.

Steam-based Rankine cycles have been the mainstay of power generation since the early 1700s. Over the decades, in fact hundreds of years, the steam-Rankine cycle has progressed in operating conditions and performance to include American Electric Power's (AEP) Philo Unit 6 (Pawliger, 2003), operating at 31.0 MPa and 621°C, per unit, with plant efficiencies in the 39–40% range [higher heating value (HHV)] (SWEPCO, 2016), and the AEP Turk plant (Peltier, 2013) with a steam-Rankine cycle operating at 26.2 MPa and 600°C, per unit. The reported efficiency of the Turk plant is 40% (HHV) (Santoian, 2015). Forward thinking advanced ultrasupercritical (AUSC) steam conditions operating at as high as 34.4 MPa and 760°C, per unit, have been envisioned with plant efficiencies in the range of 43.7–44.1% based on HHV (Weiland and Shelton, 2016). This suggests a future steam-based Rankine cycle operating at 34.4 MPa and 760°C with a cycle efficiency as high as 52.0%. Assuming boiler efficiencies as high as 89% and current plant efficiencies today in the 40% range, steam AUSC Rankine plants are likely targeting thermodynamic efficiencies in the mid-40s and slightly higher. Thermodynamic efficiencies of indirectly heated sCO₂ power cycles, assuming similar costs, will need to target efficiencies in the 50–55% range (HHV) to compete with power plants with state-of-the-art and future steam cycles.

While the direct-fired sCO₂ cycle can be applied to coal gasification applications, where the comparative incumbent would be a coal-based Integrated Gasification Combined Cycle (IGCC) with precombustion carbon capture, the most likely near-term incumbent to surpass is the natural gas-fueled combined cycle (NGCC) with

post-combustion CO₂ capture. It has been shown that post-combustion CO₂ capture on NGCC power plants will have an efficiency reduction of approximately six percentage points and a nominal 44 plus percent increase in cost of electricity (NETL, 2015a). Specifically an NGCC plant using state-of-the-art 2013 F-Class gas turbines (two on one configuration) is estimated to have an efficiency of 57.0% [lower heating value (LHV)], based on lower heating valves and a COE of \$57.60/MWh. When this same plant is built with postcombustion capture (90%), the efficiency drops to 50.61% (LHV) and the COE increases to \$83.30/MWh. This represents a 45% increase in COE due to CO₂ capture. Similarly for an H-class turbine NGCC system, the efficiency drops from 59.5 to 52.2% (LHV) with a 42% increase in COE to \$76.2/MWh. The coal-based incumbent is not discussed here; however, it can be assumed that the directly fired cycle that competes with the NGCC with capture can be used in the gasification application with equal success.

This discussion of incumbent technologies is far from complete; however, certain performance targets can be inferred that set the stage for technology development. These targets include: (1) the development of an indirectly heated sCO₂ power cycle with a thermodynamic cycle efficiency that exceeds 50% (HHV) and (2) the development of fossil-fueled power plants based on directly heated sCO₂ power cycles that capture CO₂ and generate electricity at a cost less than \$75–85/MWh.

The chapter is divided into two sections relevant to the two main types of sCO₂ power cycles, indirect and direct sCO₂ cycles, for fossil fuel applications. The presentation will include the major components required for each cycle, related status, and technical issues. Although not overtly discussed, resolution of technology issues and optimization of components and cycles are being driven to meet performance targets established for these cycles with the intent to surpass the performance of the incumbent competing cycles and successfully deploy in the market place.

12.2 Indirect supercritical CO₂ cycles

Most sCO₂ power cycles considered in this book are indirect sCO₂ cycles in that the heat source is external to the closed sCO₂ cycle [e.g., concentrated solar power (CSP), nuclear power]. The primary factors affecting the particular application of the sCO₂ cycle are its size and the temperature of the heat source. CSP and nuclear power applications are unique in that the heat source is at a relatively constant temperature, which favors the use of recompression sCO₂ cycle as discussed later. Waste heat recovery applications, as well as most fossil energy applications, are concerned with transferring as much heat as possible from a combustion process or other industrial process into the sCO₂ cycle. Doing so effectively requires that heat be transferred from a hot fluid (e.g., flue gas) to the sCO₂ cycle over a wide range of temperatures as the fluid cools down. This has significant implications for the primary heat exchangers (PHXs) and sCO₂ cycle designs in these applications.

To illustrate this point, consider the temperature–heat transfer (T-Q) diagram of typical coal-fired heat sources in Fig. 12.1(a). Pulverized coal (PC) combustion generates high-temperature flue gas from which heat must be recovered down to its

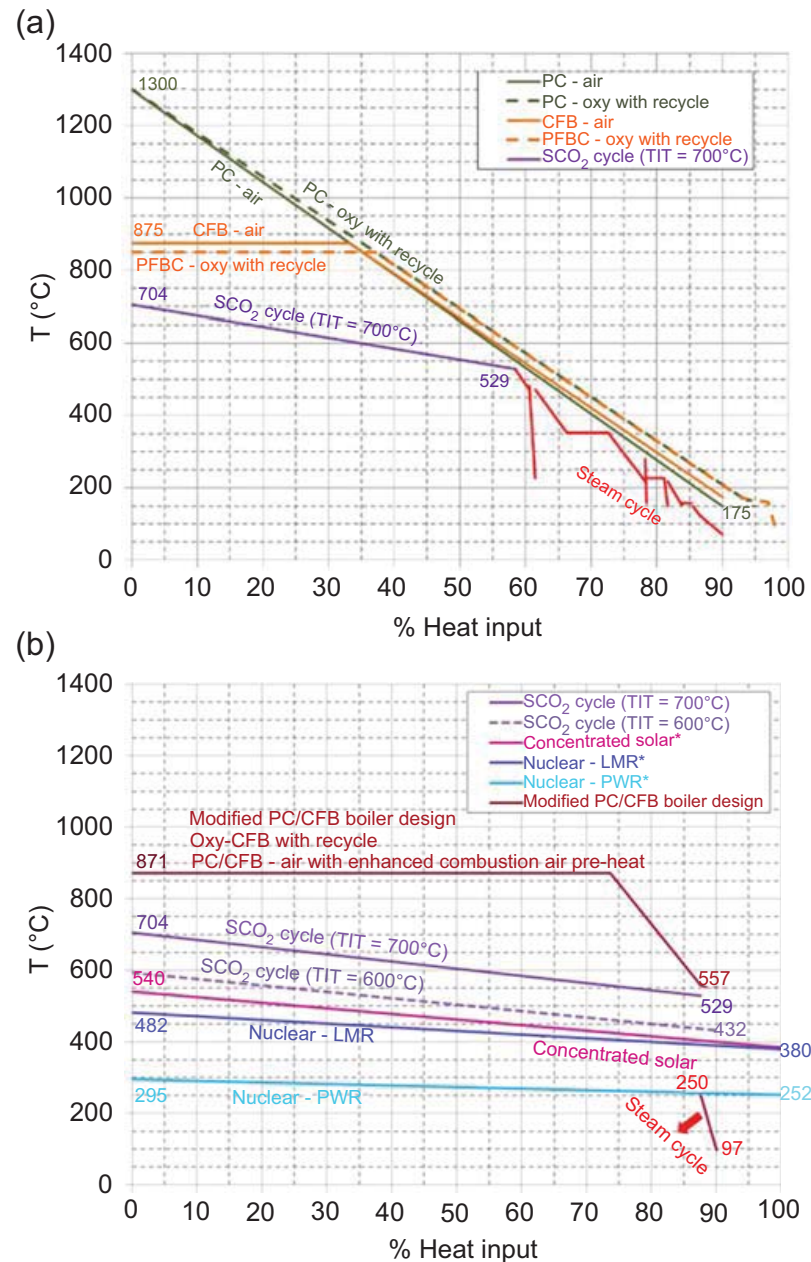


Figure 12.1 Temperature–heat transfer diagrams for (a) typical coal-fired boilers and (b) modified coal boilers, nuclear and concentrated solar. CFB, circulating fluidized bed; PC, pulverized coal; PFBC, pressurized fluidized bed combustor.

boiler exhaust temperature. A recompression sCO₂ cycle has a narrow temperature window for heat addition, such that only 55–60% of the flue gas thermal energy can be utilized in the sCO₂ cycle in this example, requiring the use of a bottoming cycle (steam, as shown in this case) to recover the remaining useful energy. Oxycombustion, circulating fluidized bed (CFB) combustion, and pressurized fluidized bed combustor (PFBC) technologies result in similar difficulties in integrating with highly recuperated sCO₂ cycles.

One possibility for better accommodating a recompression sCO₂ cycle is to increase the amount of flue gas recycle to the combustion process, as represented by the modified

PC/CFB boiler design curve in Fig. 12.1(b). This extends the heat addition window to better suit a recompression sCO_2 cycle, allowing for $\sim 80\%$ heat addition. Note that the nuclear and concentrated solar heat sources are more constant temperature processes, as shown in Fig. 12.1(b), improving their applicability to recompression sCO_2 power cycles.

12.2.1 Indirect supercritical CO_2 cycle architectures for fossil energy

There are a variety of thermodynamic cycles that can be used for efficient power production using sCO_2 working fluid, most of which are variants of the closed Brayton cycle. These cycles are based in large part on the early work of Angelino (1968), who studied condensing versions of these cycles for coal-fired power generation. A few of the cycles most appropriate for utilization with fossil energy sources are covered in later subsections.

One option for improving the performance of any of the cycles below is to condense CO_2 within the cycle by operating the cooler at temperatures and pressures below the CO_2 critical point of 31.0°C and 7.38 MPa . Doing so significantly increases the density of the CO_2 exiting the cooler, thereby reducing the compression power required to pump it to the maximum cycle pressure. The efficiency benefits and feasibility of operating an sCO_2 cycle in condensing mode has been studied by Wright et al. (2011), where compressor operation near the sCO_2 critical point has been shown to present no problems to steady-state cycle operation for a centrifugal compressor.

12.2.1.1 Recuperated cycle

In the simple recuperated cycle (Fig. 12.2), there is a single recuperator (R) that transfers heat from the turbine (T) thermal exhaust to preheat the sCO_2 entering the

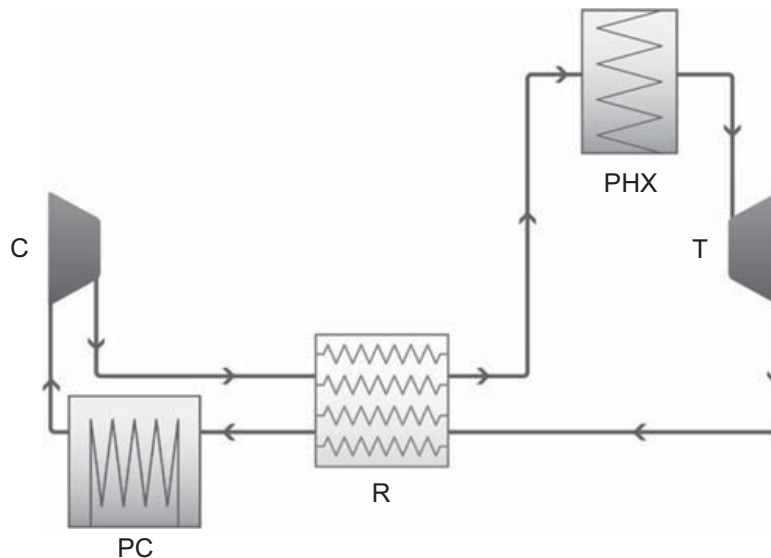


Figure 12.2 Simple recuperated cycle. *C*, compressor; *PC*, primary cooler; *PHX*, primary heat exchanger; *R*, recuperator; *T*, turbine.

PHX. Following recuperation of the turbine's thermal exhaust, the CO₂ is cooled in the primary cooler (PC) and pumped to high pressure in the compressor (C) for preheating in the recuperator. Due to the increased heat capacity of the lower pressure CO₂ near the critical point, equal sCO₂ mass flow on each side of the recuperator will lead to a temperature imbalance at the hot end of the recuperator, leading to inefficiency. One way to compensate is with a recompression cycle, as described in the next section, which leads to improved heat transfer and increased cycle efficiency. Alternatively, a portion of the high-pressure sCO₂ stream could be heated with waste heat from the heat source or some other process. Similar to the recompression cycle, this requires a low-temperature recuperator (LTR) and a high-temperature recuperator (HTR) in series, similar to the cycle shown in Fig. 12.3.

12.2.1.2 Recompression cycle

One option for improving the performance of the simple recuperated cycle is to split the recuperator into two components and bypass a portion of the high-pressure sCO₂ flow around the LTR to balance its heat duty and sCO₂ temperatures. The bypass portion is compressed in a recycle compressor (RC), as shown in Fig. 12.3, and joins the flow exiting the high-pressure side of the LTR, which has passed through the primary cooler and main compressor (MC). Typically, the bypass flow fraction is set to balance the heat duty across the LTR, allowing the minimum approach temperature to be met at both ends of the LTR, and at the cold end of the HTR.

This cycle has been identified as one of the highest performing sCO₂ cycles, and is used in CSP and nuclear applications to maximize efficiency. However, its application to fossil energy systems is complicated by the small temperature window used for heat addition. In applications utilizing the recompression cycle for fossil-fueled power plants, additional consideration must be given to making effective use of the significant thermal energy remaining in the combustion flue gas after primary heat transfer to the sCO₂ recompression cycle. Some of these strategies are discussed in Section 12.2.4.

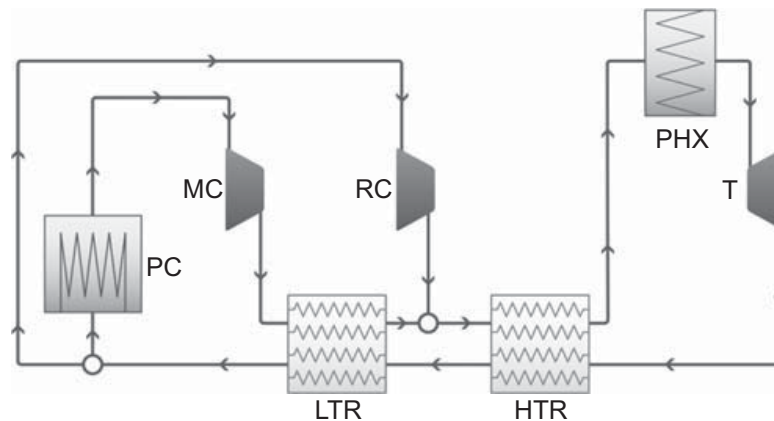


Figure 12.3 Recompression cycle. *HTR*, high-temperature recuperator; *LTR*, low-temperature recuperator; *MC*, main compressor; *PC*, primary cooler; *PHX*, primary heat exchanger; *RC*, recycle compressor; *T*, turbine.

Operation of a recompression cycle in condensing mode not only reduces the power requirements of the MC but also lowers the temperature of the high-pressure CO₂ entering the LTR, enabling additional recuperation from the hot side. For a fixed recuperator approach temperature, this has the end effect of reducing the temperature entering the RC as well, such that the increased sCO₂ density also reduces the RC power requirements. Intercooling of the main compressor has a similar effect.

12.2.1.3 Precompression

The precompression cycle, as shown in Fig. 12.4, utilizes a boost compressor (BC) stage between the HTR and LTR. The benefit is an increase in the amount of thermal energy that can be transferred from the hot side of the LTR, helping to alleviate the temperature pinch point problem in the LTR. In addition, this cycle allows for higher pressure ratio and power output across the turbine (T), which in turn leads to lower turbine exit temperatures, hot side HTR temperatures, and a lower temperature entering the PHX. This expands the temperature window for heat addition in the PHX. The difficulty is that precompression at high temperatures is inefficient, negatively impacting the overall cycle efficiency (Kulhanek and Dostal, 2009).

12.2.1.4 Partial cooling cycle

The partial cooling cycle is similar to the recompression cycle in that a portion of the sCO₂ flow bypasses the primary cooler and MC to better balance the LTR, as shown in Fig. 12.5. In contrast, a higher pressure ratio across the turbine is used, and following recuperation of the heat through the HTR and LTR, some cooling of the total flow is performed, followed by a stage or two of compression in a BC to the main cooler pressure, which is typically close to the CO₂ critical pressure. The flow splits at this point to the main cooler, and to a RC, as with the recompression cycle.

A major advantage with this cycle is the increased power output from the turbine due to the higher pressure ratio. This has the secondary effect of increasing the

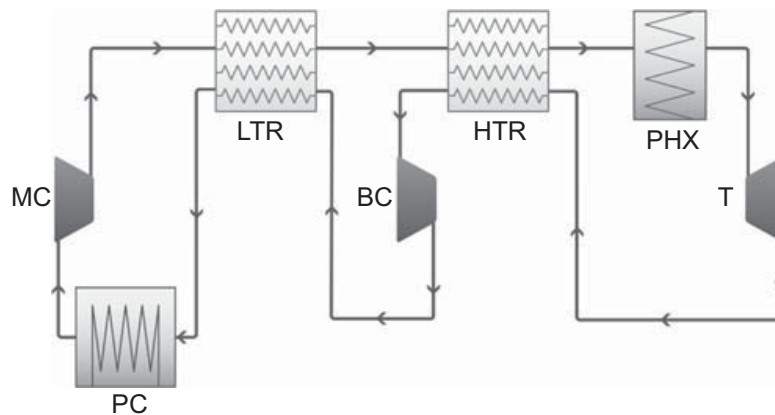


Figure 12.4 Precompression cycle. *BC*, boost compressor; *HTR*, high-temperature recuperator; *LTR*, low-temperature recuperator; *MC*, main compressor; *PC*, primary cooler; *PHX*, primary heat exchanger; *T*, turbine.

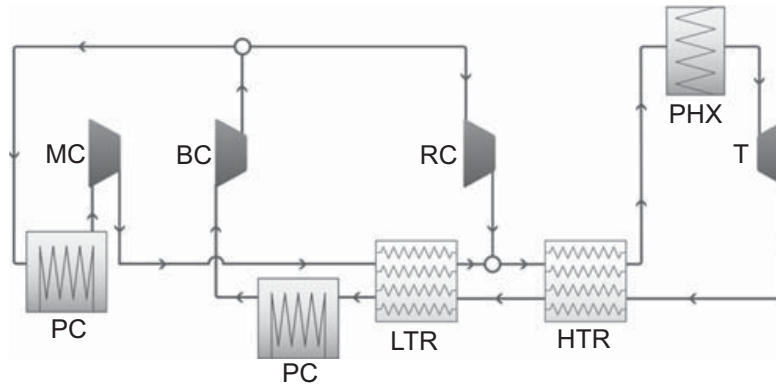


Figure 12.5 Partial cooling cycle. *BC*, boost compressor; *HTR*, high-temperature recuperator; *LTR*, low-temperature recuperator; *MC*, main compressor; *PC*, primary cooler; *PHX*, primary heat exchanger; *T*, turbine.

temperature window for primary heat transfer to the cycle, similar to the precompression cycle, which is a major benefit to fossil-fueled cycles in general. A major disadvantage is in the additional hardware that is required (and additional cooler and compressor compared to the recompression cycle), although some of the compressors can likely be combined onto a single shaft, possibly with a single casing.

In Kulhanek's comparison of power cycles at 550°C turbine inlet temperature for nuclear applications, the partial cooling cycle outperforms the recompression cycle at turbine inlet pressures below 20 MPa (Kulhanek and Dostal, 2009). In another study, the partial cooling cycle outperforms the recompression cycle above about 600°C turbine inlet temperature (Kulhanek and Dostal, 2011). One reason for this improvement is that auxiliary power requirements for the first-stage compressor and bypass compressor are lower due to the precooling, improving overall cycle efficiency. The same study further concluded that the partial cooling cycle could better handle deviations in pressure ratio for operation at part-load conditions relative to the recompression cycle. Coupling its high efficiency with increased temperature window for heat addition, the partial cooling cycle warrants serious consideration for use with fossil-fueled power plants, although no studies to date have considered this option.

12.2.1.5 Cascade cycle

This cycle utilizes split flow out of the MC, with about half of the flow going to the PHX and high-temperature turbine (HT). Recuperation of the HT's exhaust occurs in an HTR to drive a second, lower temperature turbine (LT), as shown in Fig. 12.6. Low-pressure exhaust from both turbines is then combined to provide additional recuperation of the sCO₂ going to the HTR for the low-temperature turbine.

This cycle has the advantage of a large temperature window for heat addition, in contrast to the recompression cycle. For this reason, this basic cycle, and its variations, are favored for applications involving waste heat recovery due its ability to extract the maximum amount of heat from flue gas or another waste heat source. This is covered in more detail in Section 12.2.4.2.

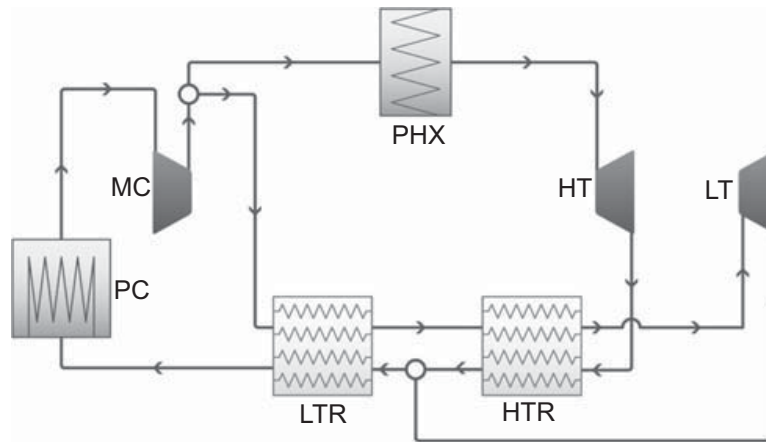


Figure 12.6 Cascade cycle. *HT*, high-temperature turbine; *HTR*, high-temperature recuperator; *LT*, lower temperature turbine; *LTR*, low-temperature recuperator; *MC*, main compressor; *PC*, primary cooler; *PHX*, primary heat exchanger.

12.2.2 Plant sizing and operational requirements

One of the major challenges with implementing $s\text{CO}_2$ cycles for fossil energy is the size of such plants relative to the state of the art for concentrated solar or waste heat recovery systems. While $s\text{CO}_2$ turbomachinery may be more compact than comparably sized steam turbines, such equipment does not yet exist, and conceptual designs for large scale turbines are just now in the process of being created (EPRI, 2013; Bidkar, 2016a,b). Likewise, as of 2016, $s\text{CO}_2$ recuperators have thus far only been built at the 10 MW level, though a 47 MW recuperator is in the process of being designed for a 10 MWe $s\text{CO}_2$ demo plant (Chordia, 2015). Newer full scale coal-fired plants typically range in size from 400 to 800 MW, which would be expected to have recuperator heat duties on the order of 1500–3000 MW.

In addition to size, the operational profile of the plant should be considered as well. In the recent past, coal-fired power plants were utilized for baseload power, and could therefore be assumed to operate at full capacity for a majority of the time. More recently, renewable power production from wind and solar systems has led to the need for cycling operations of fossil-fueled plants to offset the variable power production from these systems due to variations in the availability of these renewable resources. Furthermore, the increase in highly efficient natural gas combined cycle (NGCC) plants and reduction in natural gas prices has enabled NGCC plants to compete with coal-fired plants on a COE basis. In today's power markets, few fossil-fueled plants have the luxury of operating as baseload units, thus some amount of daily load following is to be expected at the plant (EIA, 2016). As a result, the part-load performance of the $s\text{CO}_2$ cycles must also be considered, as their performance under these conditions will significantly impact the economic feasibility of these plants. In addition, fossil-fueled power is often tasked with compensating for more rapid fluctuations in renewable power output, on the order of minutes, due to intermittent cloud cover or winds. Given the high pressures utilized in $s\text{CO}_2$ power cycles, the thick walls required for pressure containment preclude rapid temperature transients for

power variation, although mass flow or sliding pressure control could be used at relatively constant temperatures for more rapid turndown. Therefore, for fossil-fueled plants, designs and control strategies to increase ramp rate for rapid load following should also be considered.

12.2.3 Heat source integration

Successful integration of an sCO₂ cycle with a fossil-fueled heat source requires careful consideration. The incumbent technology to be replaced is the steam-Rankine cycle, which has the advantage of over a century of development and refinement. Although much of this development work can be leveraged for fossil-fueled sCO₂ primary heater designs, there are important differences that are worth noting.

First, for the same level of heat input, required mass flow rates of sCO₂ are 8–12 times higher than for a steam cycle. This occurs not only because the sCO₂ turbine pressure ratio is lower than in steam cycles but also because the specific heat of steam, on a mass basis, is 2–4 times higher than that of CO₂ over the temperature and pressure ranges considered in heater/boiler designs. Ultimately, this has significant implications for heater tube sizing, which leads to the second major difference: the acceptable pressure drop. In steam cycles, pumping of the liquid water to high pressures consumes very little power, such that high pressure drops in steam boiler tube piping are fairly commonplace. Compression of sCO₂, by comparison, is much more energy intensive, and piping design to minimize pressure drop in the primary heater is beneficial to cycle and plant performance. Finally, boiler and steam power plant designs have been optimized around a particular temperature input window that does not necessarily match that of the sCO₂ cycle. The particular effects of this difference, as well as the other differences noted here, are described for the relevant fuel sources in the following sections.

12.2.3.1 Conventional coal-fired heaters

Coal-fired sCO₂ primary heaters will differ in many respects from their steam cycle counterparts, although much of the combustion processes (coal handling, low NO_x burners, ash handling, soot blowing, etc.) will remain largely the same. In general, conventional coal boilers in steam power plants contain a combustion section with a central flame, where the boiler walls are lined internally with water or steam-filled tubes to protect the boiler casing and structure. Heat transfer to the water or steam in this “membrane” wall occurs radiatively, and in the case of subcritical steam operation, results in boiling of the liquid water to steam. Beyond the combustion section, the flue gas moves downstream where heat is transferred to the steam in tube banks, primarily via convective heat transfer. In this section, the steam is raised to its highest temperature before entering the steam turbine. The flue gas is cooled through the tube banks to 350–370°C, where it enters a final heat exchanger to preheat the incoming air to the combustion process.

In 2014, Electric Power Research Institute (EPRI) collaborated with Babcock & Wilcox (B&W) to develop a scoping study for a coal-fired sCO₂ primary heater

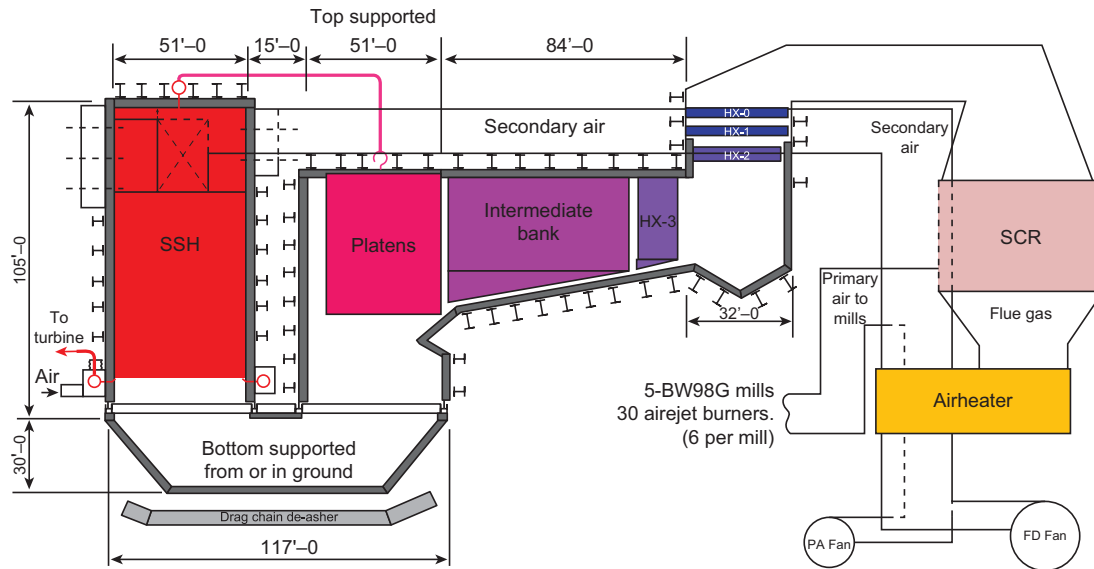


Figure 12.7 Inverted-tower boiler design for $s\text{CO}_2$ (EPRI, 2014a).

(EPRI, 2014a). This study assumed that an $s\text{CO}_2$ recompression cycle would be used, and that an inverted tower boiler design developed for higher temperature steam conditions could be modified to accommodate the $s\text{CO}_2$ cycle as shown in Fig. 12.7. In noting the challenges associated with increased $s\text{CO}_2$ mass flows while simultaneously minimizing pressure drop, the study points out that reduced pressure drop is typically attained with larger tube diameters, which will have a significant impact on boiler cost if high-strength nickel alloy tubes are required for temperatures above about 620°C . The scoping study did not attempt to resolve the $s\text{CO}_2$ pressure drop/mass flow problem, although planned follow-on studies will investigate this issue in more detail.

The EPRI/B&W study also noted other difficulties with a $s\text{CO}_2$ primary heater design. Due to the recompression cycle design assumed, the smaller temperature window for heat addition results in a flue gas temperature of about 566°C , which greatly exceeds the capabilities of modern flue gas air preheaters, with inlet temperatures typically around 370°C . As a result, the study employed a small cascade-type $s\text{CO}_2$ bottoming cycle to reduce the flue gas temperature to about 370°C , as discussed in Section 12.2.4.2 below. In addition, the elevated temperature of the $s\text{CO}_2$ entering the primary heater in the recompression cycle (about 530°C) is too high to effectively cool the membrane walls of the boiler's radiant section, requiring the use of refractory-lined heater walls behind the membrane wall. This has been done on smaller scale industrial boiler designs, but not at the scale considered for commercial power generation (EPRI, 2014a). Other changes required for $s\text{CO}_2$ utilization in the inverted tower boiler design are noted, but the authors also indicate that $s\text{CO}_2$ cycle designs for this application require further development.

Some of these challenges are highlighted in a conceptual study by Moullec (2013), who developed a rough coal-fired primary heater design for use with a recompression $s\text{CO}_2$ cycle. While the "shortcut method" employed in the study does not result in a

commercially feasible heater design, the calculations provide the most detailed analysis of a coal-fired sCO₂ heater to date. The overall plant design uses a double reheat with a steam turbine inlet temperature of 620°C, and is thermally integrated with a monoethanolamine (MEA)-based carbon capture and storage (CCS) process.

The calculations of this study yield 7.65, 4.81, and 5.88 bar pressure drops through the main heat, first reheat, and second reheat, respectively, for an overall pressure drop of 18.34 bar through the primary heater. Most of this pressure drop (59%) occurs in the radiant section membrane walls, with another 14% occurring in the main heat economizer section, and the remainder fairly evenly distributed through the other six superheater tube banks. The primary heater pressure drops are noted as a primary area for process improvements, as net plant thermal efficiency (LHV) could be 1.6% points higher if pressure drops were negligible.

Also noted in the Moullec study is the high flue gas temperature (540°C) resulting from the narrow temperature window for heat addition to the recompression sCO₂ cycle chosen (Moullec, 2013). The study assumes that a high-temperature air preheater can be developed to handle these elevated flue gas temperatures. The required thermal duty of the air preheater is less than that contained in the high-temperature flue gas, thus a parallel economizer is used to heat a portion of the high-pressure sCO₂ flow coming from the MC. On the sCO₂ cycle side, this economizer is in parallel to the sCO₂ recuperators, and its heat duty, combined with that of the air preheater, is sufficient to lower the flue gas to 110°C for cleanup (Moullec, 2013). A follow-up study by Mecheri and Moullec (2015) examines alternative configurations for handling this high flue gas temperature problem, but ultimately recommends the approach taken in the original study (Moullec, 2013), assuming that a high-temperature air preheater can be constructed. It is notable that both the Moullec (2013) study and its follow-up study (Mecheri and Moullec, 2015) show 2.4–4.9% points net LHV plant efficiency improvements with sCO₂ relative to comparable steam cycles under the conditions studied.

12.2.3.2 *Nonconventional coal-fired heaters*

Apart from the conventional coal-fired heater designs, there are other steam boiler designs that may also work well with sCO₂ power cycles. In particular, there are several forms of coal bed combustion that have been utilized in steam boiler designs. In these systems, a bed of coal particles is burned as air is passed upward through the bed. Different designs result depending on the flow rates of air through the bed. As air flow increases, regimes pass from fixed bed to fluidized bed, bubbling fluidized bed (BFB), and a CFB, where coal particles are entrained in the air/flue gas flow. For commercial-scale power production, BFB boilers have been designed for use with biomass and other specialty application, although CFB designs are most prevalent for coal (Kitto and Stultz, 2005).

One of the primary advantages to CFB combustion is the ability to capture sulfur from the coal within the combustor by adding limestone to the coal feed. The limestone first reacts with oxygen to create lime, which further reacts with sulfur to create calcium sulfates (gypsum) that can be removed with the coal ash. In-bed sulfur

removal eliminates the need for downstream sulfur cleaning, and generally provides a cost and efficiency benefit. To enable the sulfur capture reactions, the CFB must operate between 800 and 900°C, which also helps to eliminate thermal NO_x production and its associated emissions reduction equipment. The combustion temperature is controlled by adjusting the fuel and air feed rates and the rate of heat removal to the bottoming cycle (Kitto and Stultz, 2005). The fairly constant CFB combustion temperatures may provide a benefit to sCO₂ cycles by enabling a smaller temperature window for heat addition. The low end of the temperature window could be extended through the use of flue gas recirculation, to better match the recompression sCO₂ cycle heat addition temperature window as shown in Fig. 12.1(b). To do so, however, a significant boiler design effort would need to be undertaken to accommodate the change in oxygen content in the fluidizing gas, as well as the change from steam to sCO₂ heat transfer, particularly for the relatively larger sCO₂ flow rates relative to steam.

Any of the coal boiler types can be operated in either air- or oxycombustion modes without substantial modifications. Oxycombustion typically entails mixing oxygen from an air separation unit (ASU) with recycled flue gas to provide an oxidizing medium that is similar to air. The combustion products are primarily carbon dioxide and water, where the latter can be readily condensed from the flue gas to provide a relatively pure CO₂ stream for easy CCS. The penalty incurred in this method is in the ASU, where a significant auxiliary power load is associated with separation of oxygen from air.

A recent study by the National Energy Technology Laboratory (NETL) has investigated the integration of an sCO₂ recompression cycle into an oxycoal-fired CFB boiler, with incorporation of CCS (Shelton et al., 2016). The studied system utilizes the heat remaining in the flue gas to preheat recycled flue gas to the CFB after water knockout, and also provides additional heat to the sCO₂ cycle in parallel with the LTR to reduce the bypass flow of the recompression cycle. The study investigated several cycles at 620°C turbine inlet temperature: a base cycle, and additions of compressor intercooling, reheat, and both reheat and intercooling. Comparison to a comparable CFB Rankine cycle shows a 2% point efficiency benefit for utilizing the sCO₂ cycle with reheat and intercooling. In addition, a 760°C turbine inlet temperature sCO₂ recompression cycle case with reheat and intercooling was studied, resulting in a 39.3% net plant thermal efficiency (HHV) with CCS.

PFBCs have also been developed to reduce the size of the boiler and introduce the possibility of additional power production via expansion of the flue gas through a turbine. Pratt & Whitney Rocketdyne designed a Zero Emissions Power and Steam (ZEPS) system that utilizes an oxy-fired PFBC primary heater for use with a recompression sCO₂ cycle, as shown in Fig. 12.8 (Johnson et al., 2012; Vega et al., 2014). The PFBC utilizes a BFB that operates at 0.83 MPa and 871°C, using near-stoichiometric oxygen mixed with recycled CO₂. The lower section of the PFBC has in-bed heat exchange tubes, followed by convective heat exchange above the bed (Subbaraman et al., 2011). Utilization of an sCO₂ recompression cycle was calculated to yield 2.4–3.7% improvement in net HHV plant efficiency compared to a comparable steam-Rankine cycle (Subbaraman et al., 2011). Development of this technology has been transferred to Aerojet Rocketdyne, and more recently, the Gas Technology Institute

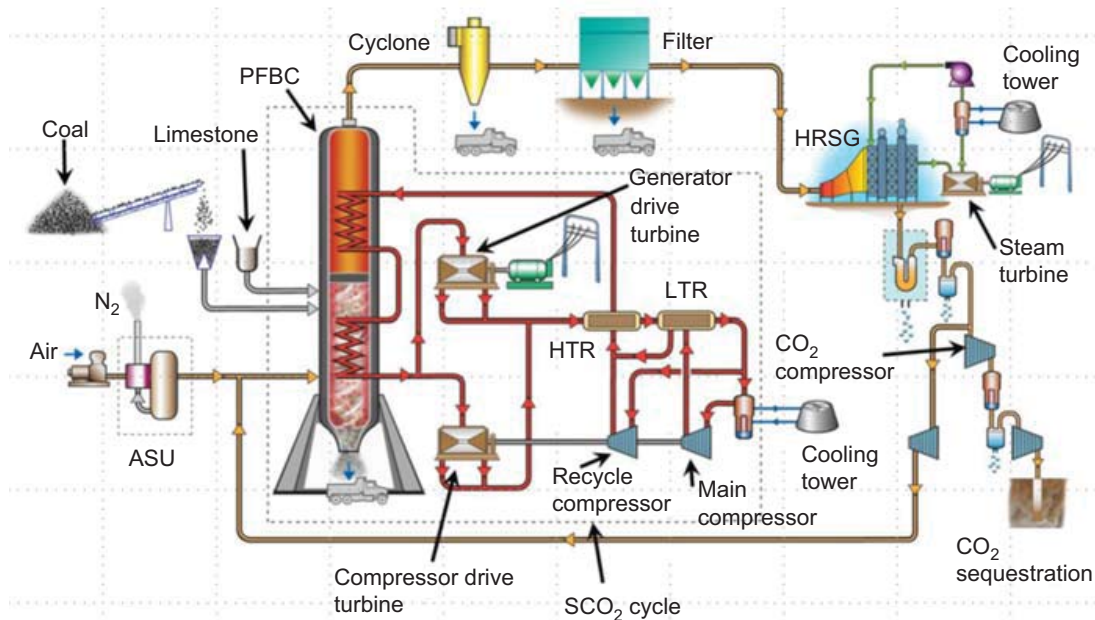


Figure 12.8 The Zero Emissions Power and Steam (ZEPS) system (Vega et al., 2014). ASU, air separation unit; HRSG, heat recovery steam generator; HTR, high-temperature recuperator; LTR, low-temperature recuperator; PFBC, pressurized fluidized bed combustors.

(GTI). GTI is pursuing a pilot-scale PFBC capable of utilizing sCO₂ heat exchange, which is in the process of being constructed and tested in collaboration with CANMET (Follett and Fitzsimmons, 2015).

Oxycoal combustion in an sCO₂ environment at 10 MPa has also been suggested for coupling to an indirect sCO₂ cycle (McClung et al., 2014). This concept has the potential to produce a very cost-effective coal-fired sCO₂ system; however, its implementation incurs additional challenges. In particular, high-temperature and high-pressure processes must be developed for coal combustion in sCO₂, cyclonic ash particulate removal, and particulate-tolerant recuperators (McClung et al., 2014).

While several coal-fired primary heaters have been conceptually investigated for indirect sCO₂ cycle use, most utilize the recompression cycle, which may be better suited to concentrated solar and nuclear power cycles. Additional research is needed to determine the most appropriate sCO₂ cycle for coupling to coal combustion. The primary goals of this research should be to effectively utilize low-grade thermal energy in the flue gas for the sCO₂ cycle, minimize sCO₂ pressure drops through the PHX, and improve the power density of the sCO₂ cycle to reduce its size. Partial cooling cycles and careful consideration of condensing cycle operation may help achieve some of these goals.

12.2.3.3 Natural gas-fired heaters

For commercial-scale power generation, indirect sCO₂ cycles powered by natural gas are not expected to be able to compete with NGCC plants in terms of efficiency or COE. The combined cycle efficiency of NGCC plants currently exceeds 60% LHV (Yuri et al., 2013), due in large part to high gas turbine firing temperatures. Direct

sCO₂ cycles operating on natural gas may be comparable in efficiency to NGCC cycles with CCS, which is covered in [Section 12.3](#). Opportunities to use sCO₂ bottoming cycles in NGCC plants is covered in [Section 12.2.4.2](#), as well as in Chapter 10.

These issues aside, there may be niche applications where a natural gas-fired primary heater is desired in an sCO₂ cycle. For instance, early sCO₂ demonstrations or smaller scale power plants are likely to utilize natural gas for its simplicity relative to coal ([EPRI, 2014a](#)). In these instances, off-the-shelf natural gas-fired package boilers utilized in petroleum refining may be well suited to the task, provided that the increased mass flows and reduced sCO₂ heat capacities relative to steam are considered by the boiler manufacturers, as well as the relevant temperatures and sCO₂ material compatibility.

12.2.3.4 Reheating

A common method of increasing cycle efficiency is to add a reheat stage or two to the cycle, a method that also applies to indirect sCO₂ power cycles. This has the effect of increasing the overall temperature of heat addition, approximating the constant-temperature heat addition process of the ideal Carnot cycle. The addition of one reheat stage generally adds 1.2–1.7% points to the cycle efficiency, with diminishing returns for additional reheat stages ([Shelton et al., 2016](#)). The major tradeoff for adding reheating to an sCO₂ cycle is in cost, where an additional turbine, primary heater tube banks, and sCO₂ transfer piping must be added. For temperatures above approximately 620°C, nickel alloy heater tubes, transfer piping, and turbine components must be used, which are typically much more expensive than austenitic steels used at 620°C and below. In these cases, the added capital cost may negate the fuel savings gained via the reheat stage's efficiency benefits. The economics of these cases should be carefully considered to weigh the efficiency benefits of reheating against its added capital cost. For turbine inlet temperatures at 620°C and below, added capital costs are reasonable, and one or two stages of reheat may be economical ([Moulllec, 2013](#)).

12.2.4 Recovery of low-grade heat

As noted earlier, one of the challenges with the use of a recompression or other high-efficiency sCO₂ cycle is the narrow temperature window for heat addition. This results in a higher temperature flue gas (550°C) than might otherwise be seen in a comparable steam cycle, which extracts heat from the flue gas in an economizer down to about 370°C ([EPRI, 2014a](#)), at which point an air preheater can be used to recover the remaining low-grade heat in the flue gas. The following section details a variety of options for efficient utilization of the remaining heat in the flue gas following primary heat addition to the sCO₂ cycle.

12.2.4.1 Advanced air preheaters

Current air preheater technology is limited to flue gas inlet temperatures up to 370°C due to material and mechanical limitations ([EPRI, 2014a](#)). Air preheaters are available in recuperative and regenerative styles. Recuperative air preheaters employ

tube-in-shell or plate heat exchanger configurations, where the physical separation of the flue gas from the incoming air results in very little leakage. These preheaters also have no moving parts, but are very large and heavy due to the need for large surface area for low-temperature gas-to-gas heat exchange (Kitto and Stultz, 2005). More common in electric utilities are the regenerative style air preheaters, which are more compact. In these preheaters, hot flue gas passes through and heats part of a rotating solid matrix, which preheats incoming air passing through on the other half of the device. Heat transfer is very effective in these devices, although their design does allow for 5–15% leakage, both through the sealing faces of the rotating matrix, and due to alternating air and flue gas flows through the matrix passages (Kitto and Stultz, 2005).

Increasing the inlet flue gas temperatures for higher temperature air preheating will present several design challenges. For recuperative air preheating, the already large heat exchangers will become even larger for effective air preheating, thus the capital cost must be weighed against the fuel savings from the efficiency benefits. Regenerative air preheaters would also need to be larger, which presents additional problems. Since flue gas and air typically enter the preheater from opposite sides, a temperature gradient exists along the axis of the preheater such that thermal expansion causes the rotating solid matrix to bow, opening up gaps in the seals between stationary ductwork and the rotating matrix. Increasing the upper operating temperature may exacerbate this seal leakage problem through increased solid matrix deformation. Furthermore, the increased inlet temperature would require longer flow passages for more effective heat transfer, which will increase expected pressure drops across the preheater, as well as introduce additional flow leakage upon air/flue gas flow reversal due to increased volume within the flow passages. Other problems that typically occur in regenerative air preheaters that may be complicated by high temperatures include erosion by entrained ash particles and preheater fires that sometimes occur if the boiler is started on fuel oil (Kitto and Stultz, 2005).

12.2.4.2 Supercritical CO₂ bottoming cycles

In 2013, the energy-generating capacity of the United States (Fig. 12.9) was approximately 1000 GW with 30% coming from coal, 19% coming from combined cycle gas turbine power plants, and 14% coming from simple cycle gas turbines (EIA, 2014). Of those sources, the most efficient source of electricity, comes from NGCC power plants as depicted in Fig. 12.10. A combined cycle refers to a gas turbine “topping cycle” combined with a steam-Rankine “bottoming cycle.” The bottoming cycle utilizes the exhaust from the gas turbine topping cycle to provide heat to a steam-Rankine power cycle. The highest efficiencies reported by an NGCC configuration was achieved by a Mitsubishi Heavy Industries gas turbine that achieved a turbine inlet temperature exceeding 1600°C and a combined cycle efficiency of 61.5% LHV (Yuri et al., 2013). Having exhaust gas temperatures in the 500–600°C range, utility gas turbines can be conveniently coupled with a steam-Rankine cycle; however, recent studies have shown some indication that some sCO₂ power cycles may have an efficiency and power output benefit over conventional steam-Rankine bottoming cycles.

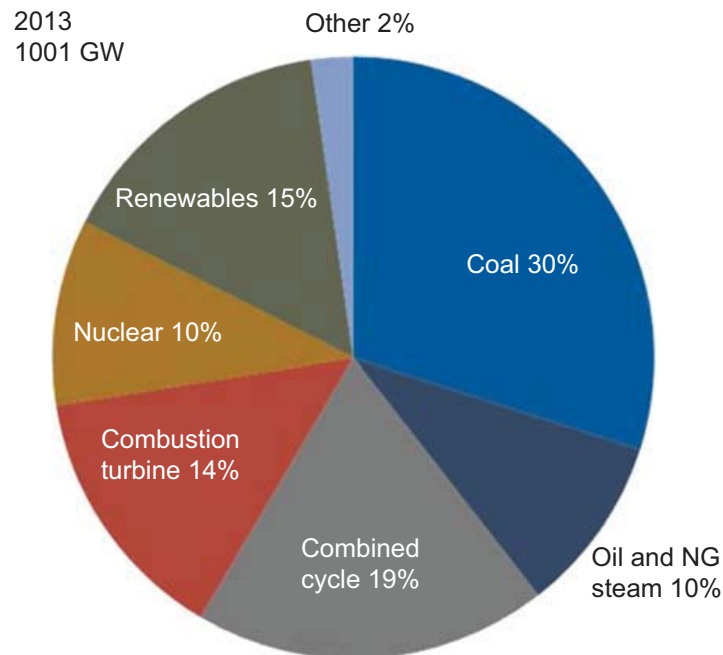


Figure 12.9 The 2013 energy generating capacity in the United States (EIA, 2014).

A theoretical study was performed by Kimzey (2012) to compare a series of $s\text{CO}_2$ bottoming cycles, based on a cascade cycle architecture as discussed in Section 12.2.1.5, to a conventional steam-Rankine bottoming cycle when coupled to a utility scale Siemens H-class gas turbine. The study determined that slightly higher bottoming cycle efficiency could be obtained by the $s\text{CO}_2$ bottoming cycle; however, none of

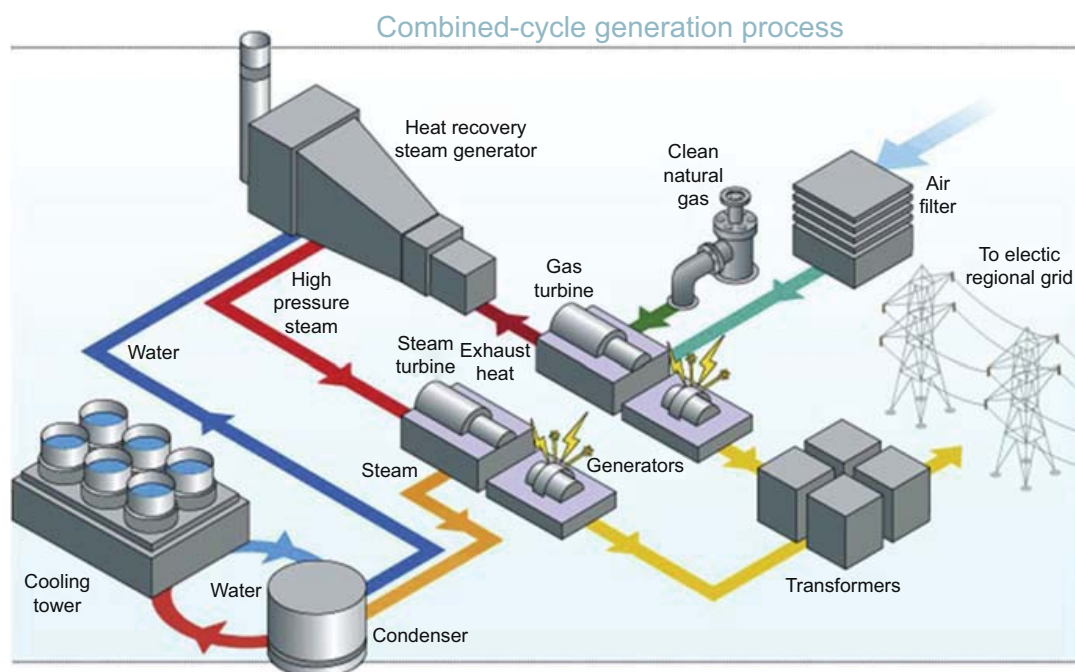


Figure 12.10 Natural gas combined cycle power plant (<http://www.tenaskawestmoreland.com/how-it-works/>).

the sCO₂ cycles evaluated could achieve the same bottoming cycle power output as the conventional steam bottoming cycle. The study concluded that a high cycle efficiency does not necessarily mean that the cycle is well suited to a bottoming cycle application. This conclusion was also made in a presentation by [Echogen Power Systems \(2012\)](#). A consequence of recovering heat from a sensible heat source, such as a gas turbine exhaust stream, is that only a small portion of the available waste heat is recoverable at high temperatures. Because of this, special care must be taken to design bottoming cycles to utilize as much of the waste heat as possible through the entire range of temperatures that the heat is available.

[Cho et al. \(2015\)](#) performed a theoretical study comparing various sCO₂ bottoming cycles to a conventional steam-Rankine bottoming cycle. They concluded that existing well-known sCO₂ layouts are unsuitable for the purpose of waste heat recovery when compared against a conventional steam-Rankine cycle. More complex cascade cycles, on the other hand, did show some promise in superiority over a conventional steam bottoming cycle. When compared against the Siemens SCC5-2204°C combined cycle, having an LHV efficiency of 58.5%, the “Cascade Cycle 3” originally from Kimzey, shown in [Fig. 12.11](#), produced 3% greater power output at a combined cycle LHV efficiency of 59.1%.

Authors from the studies noted earlier acknowledge that there is still work to be done to evaluate the practical aspects of integrating sCO₂ power cycles as bottoming cycles in an NGCC. Although efficiency benefits seem small compared to the potential additional capital cost of a cascade sCO₂ cycle, there also has not been an optimization study performed to determine the optimal sCO₂ bottoming cycle for minimizing COE. In fact, the high-power density of sCO₂ cycles may lend themselves to low-capital-cost bottoming cycles that can improve upon COE relative to traditional NGCC systems based on steam. The aforementioned studies also employ gas turbine topping cycles that have been optimized for a steam-Rankine bottoming cycle and not a cascade sCO₂ bottoming cycle. For instance, designing a topping cycle for slightly higher exhaust gas temperature (lower topping cycle pressure ratio) would reduce topping cycle power output, but could have benefits to bottoming cycle efficiency and power output. These types of optimization studies are needed to better predict the applicability of sCO₂ as an option for utility scale NGCC bottoming cycles.

Cascade cycles similar to those investigated by [Cho et al. \(2015\)](#) and those discussed in [Section 12.2.1.5](#) have also been investigated for heat recovery from coal boilers. A report by [EPRI \(2014a\)](#), discussed in [Section 12.2.3.1](#), described a theoretical scoping study undertaken by B&W to perform a preliminary design of a coal-fired sCO₂ heater for comparison against a similarly sized AUSC steam power cycle. A 750-MWe sCO₂ recompression closed Brayton cycle (RCBC) was integrated with the coal boiler, and a second cascade sCO₂ cycle was added to capture some of the lower grade heat and reduce the flue gas temperature to allowable limits for a conventional air preheater. The total power output and power cycle efficiency of the sCO₂ configuration was 823 MWe and 41.6% (HHV), while the AUSC steam cycle had a power output and efficiency of 750 MWe and 43.2% (HHV). While the sCO₂ power cycle efficiency is slightly lower than the AUSC efficiency, the authors stated that the sCO₂ cycles had not been optimized to achieve these results. Results from the few existing studies show promise and highlight needs,

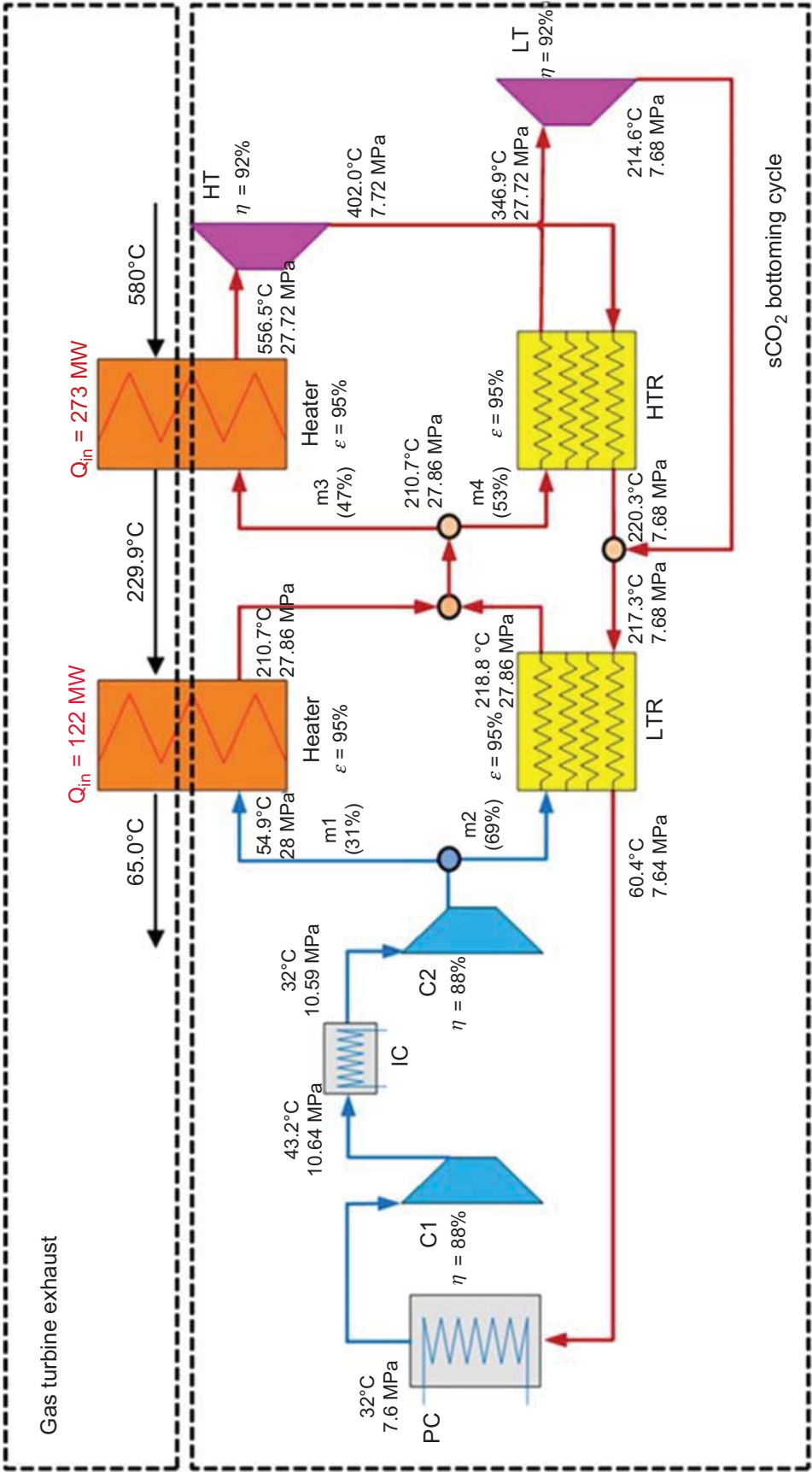


Figure 12.11 Brayton Cascade-3 bottoming cycle for gas turbine waste heat recovery (Cho et al., 2015).

but optimization studies are needed to further refine the integration of sCO₂ bottoming cycles with coal boilers for utility scale power generation.

12.2.4.3 Cogeneration and combined heat and power

Another option for the recovery of low-grade heat is cogeneration, or combined heat and power (CHP). Cogeneration combines electricity production along with production of useful heat from one energy source. CHP plants are typically installed for a customer at the point of use and sized for the thermal needs of the customer. CHP plants offer potential for improved efficiency. Examples of CHP systems are for use in providing district heating and for industrial customers where the recovered heat is used internally at the industrial site. The use of sCO₂ power cycles with CHP systems has not been widely investigated; however several potential configurations have been evaluated (Moroz et al., 2014a,b). Two types of CHP plants were considered. The first is a steam-Rankine cycle CHP plant using a bottoming sCO₂ cycle. The steam-Rankine cycle is used to produce electricity. The generated steam is split, heating the sCO₂ bottoming cycle to generate additional electricity, and to heat water for consumers. The steam turbine electrical output is kept constant, and depending on the heat load, the steam is either increased to the water heater to increase heat production or increased to the bottoming cycle to increase electricity production of the CHP plant.

The second configuration is a CHP plant using sCO₂ as the working fluid. Fuel is burned to heat the sCO₂ working fluid, which is split into a cascade of high-temperature and low-temperature sCO₂ cycles to generate electricity. In both of the cycles, after expansion and recuperation of heat, the sCO₂ is further cooled in a water heater, heating the water for consumer use. Since the high-temperature cycle accounts for the majority of the electrical power in this configuration, an additional simplified configuration was considered. In this case, the low-temperature sCO₂ cycle is replaced with a water heater. The single sCO₂ cycle generates all of the electricity, and water heating is provided by water heaters in the high-temperature sCO₂ cycle as well as the new water heater located after the sCO₂ heater. Calculated performance of the various plant configurations were compared to the base case for a traditional steam-based CHP plant. The cascaded sCO₂ CHP plant had the best electrical efficiency, and each configuration had better electrical efficiency than the base steam case.

12.2.4.4 Steam-Rankine bottoming cycles

Another option for recovering low-grade heat after primary heat exchange with the sCO₂ cycle is to add a steam bottoming cycle. This was proposed by Pratt & Whitney Rocketdyne in ZEPS (see Fig. 12.8), where the exhaust of the PFBC is cleaned and used to raise steam for a low-pressure turbine (Subbaraman et al., 2011; Johnson et al., 2012). In this case, the power addition is less than 2% of the sCO₂ turbine power output (Johnson et al., 2012), so the added efficiency benefit may not be worth the increased capital expense.

Steam bottoming cycles have also been proposed in a study by EPRI (2013), although this implementation is unique in that sCO₂ recuperation following the

sCO₂ turbine has been replaced with steam generation for the bottoming cycle. Since sCO₂ is not preheated via recuperation prior to entering the primary heater, heat addition to the sCO₂ cycle can occur over the full range of the primary heat source temperature. The study proposed two variations on this cycle, with one applicable to a new-build power plant, and a second to retrofit repowering of an existing steam power plant. The new-build cycle performed slightly better than the comparable steam-only cycle, but worse than the sCO₂-only cases without a steam bottoming cycle. This is due in part to temperature mismatches in the sCO₂–steam heat exchangers, as well as the lack of a cooler on the sCO₂ cycle, which drastically increases the power required for sCO₂ compression. The repowering case also shows an improvement over a comparable AUSC topping cycle addition, but suffers from the same compressor inlet temperature problem as the new-build cycle implementation (EPRI, 2013). With cycle improvements, variations on this idea may be suitable for repowering existing subcritical steam power plants at reduced capital cost.

12.2.5 Component challenges

This section provides a discussion on the component challenges that must be addressed for indirectly heated sCO₂ power cycles to become viable for fossil energy utility scale power generation. The section is split into three main categories: compressors, heat exchangers, and turboexpanders. Although the section may touch upon challenges common to all heat source applications, the focus of the section will be on utility scale power generation using a fossil heat source.

12.2.5.1 Turbine

sCO₂ fluid behavior in the turboexpander is more predictable than in the compressors, because fluid state points are well into the supercritical regime. In comparison to the MC operating conditions, predictable fluid properties and flow characteristics place the sCO₂ turboexpander closer to the realm of familiarity for turbomachinery designers. While the aerodynamic design may be less of a hurdle for the turboexpander, an sCO₂ turbine does not come without its host of challenges.

Even though the flowfield may be predictable well into the supercritical regime, these turboexpanders will have 10 times the power density of a steam turbine in a Rankine cycle (Ahn et al., 2015). The result is larger forces and fluctuating loads per unit blade volume. For utility scale fossil energy applications, turbine inlet temperatures will likely be required to exceed 700°C to achieve a significant efficiency benefit over existing steam-Rankine cycles (Moulllec, 2013).

By nature, CO₂ as a fluid in the supercritical regime has high density and low viscosity. These properties make seal design for turbomachinery a challenge. Dry gas seals for turboexpanders at the 10 MWe scale are commercially available, but fossil energy applications will require turboexpanders in the 500 MWe scale. Dry gas seals for these large-scale machines do not exist at the time of this writing. As depicted in Fig. 12.12, while steam leakage from a Rankine cycle can be condensed and pumped back to pressure as a liquid, sCO₂ must be recompressed as a vapor

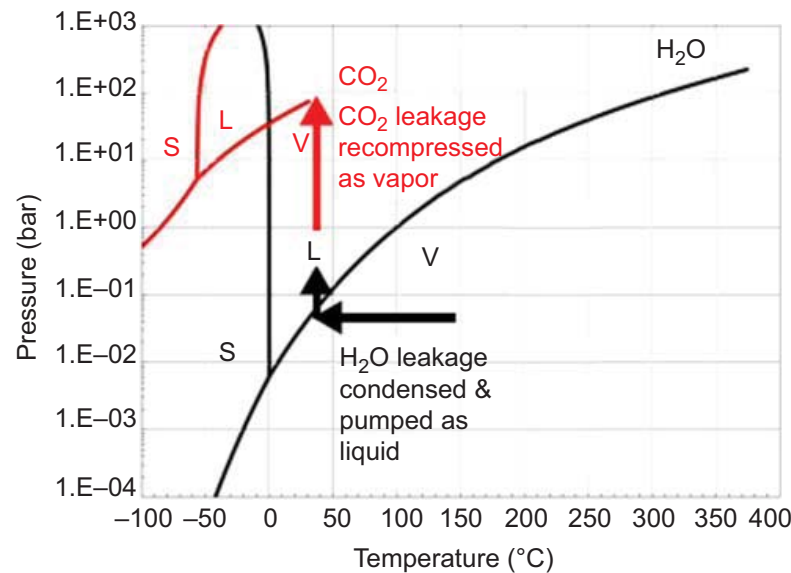


Figure 12.12 Thermodynamic differences in recovery of turbine and seal leakage (Bidkar et al., 2016c).

from near-atmospheric pressure conditions to the lower cycle pressure of the closed loop sCO₂ power cycle. This results in large auxiliary compression load. For a utility scale sCO₂ cycle, developing dry gas seals to reduce leakage can save multiple efficiency points on the power cycle (Bidkar et al., 2016c).

Bearings also pose a significant challenge to the design of turboexpanders for sCO₂ power cycles. As shaft speeds and operating temperatures increase, seal design becomes increasingly complex to maintain separation of conventional oil-lubricated bearings from the working fluid. Gas-foil bearings have been employed in sCO₂ test facilities, and issues have been reported (Sienicki et al., 2011; Fuller, 2007). It was noted by Iverson et al. (2013) that bearing design is a significant challenge that needs to be addressed before industrial use of sCO₂ power cycles. Preuss (2016) provides guidelines to aid in sizing hydrostatic journal and thrust bearings for sCO₂, and Chapman (2016) discusses a new gas foil bearing for use within the turbine cavity that uses sCO₂ as the lubricating fluid. While this concept has been proved at a conceptual level, significant development efforts will be required to advance these concepts and design guidelines to utility scale.

Another challenge associated with increasing turbine inlet temperatures is the turbine throttle control valve and turbine stop valve. For small-scale demonstration purposes, the throttle control valve can be placed upstream of the recuperator at a low temperature state point in the cycle. For large-scale power cycles, however, more precise control of the turbomachinery will require the throttle control valve to be placed immediately upstream of the turbine inlet at the state point in the cycle having the highest temperature. Turbine stop valves are also required at this location for turbine overspeed protection in the event of loss of load if the turbine is not also connected to a compressor or other load to help protect against this failure mode. Multimillion dollar development efforts are necessary to improve throttle control and stop valve capabilities for turbine inlet temperatures to reach 700°C, although significant progress toward this goal has

been achieved through the Materials for Advanced Ultrasupercritical (AUSC) Steam Turbines consortium (Purgert et al., 2015).

Erosion in turbomachinery has been observed in small-scale sCO₂ test loops (Clementoni and Cox, 2014; Fleming et al., 2014). Fleming et al. (2014) believed that the erosion (as shown in Fig. 12.13) was caused by small particulates that originated from different materials around the test loop. While the exact source of particulates has not been determined, the effects clearly identify erosion as a threat to safe, reliable, long-duration operation of a utility scale fossil energy sourced sCO₂ turboexpander. While erosion (as well as deposition and resulting spallation of thermal barrier coatings) is a well-understood challenge for air breathing gas turbines (Hamed and Tabakoff, 2006; Richards et al., 1992; Wenglarz and Wright, 2003), air inlet particulate filters can be used to mitigate the negative effects of erosion. For indirectly heated closed loop sCO₂ power cycles on the other hand, erosion is a much more difficult challenge to address, as cycle efficiency is sensitive to pressure losses that filters might present. It will require parallel efforts in materials development and turbomachinery design to adequately solve the problem of erosion for indirectly heated sCO₂ power cycles.

12.2.5.2 Heat exchangers

Depending on the power cycle configuration, there can be several types of heat exchangers in the system: a PHX, one or more recuperators, and sCO₂ coolers. PHXs are covered in Section 12.2.3. The following discussion focuses on recuperators, although many of the same challenges can be extended to sCO₂ coolers by considering an alternate fluid on the cold side of the heat exchanger. For the purposes of this general discussion on heat exchanger challenges, it is assumed that either a simple recuperated Brayton cycle or an RCBC would be utilized.

To provide a sense for recuperative heat duty requirements in a utility scale sCO₂ cycle, a 550-MWe RCBC with a turbine inlet temperature of 700°C would require approximately 4000 MW of recuperative heat duty (Johnson et al., 2012). The sheer



Figure 12.13 Turbine nozzle undercut by erosion at Sandia National Labs (Fleming et al., 2014).

volume of these heat exchangers and consequently the material cost requirements make recuperator cost a significant challenge for utility scale fossil energy application of indirectly heated sCO₂ power cycles. Even though optimum pressure ratios for recuperated cycles are relatively low, the high pressure of the CO₂ critical point means that pressure differentials in the range of 15–20 MPa would be experienced between flow passages of the recuperator. This high-pressure differential makes design for low stress a challenge. As turbine inlet temperatures exceed 700°C (as they likely would for fossil energy applications to be viable), turbine exit temperatures will approach 600°C. Fig. 12.14 shows that as metal temperatures increase, allowable stresses for materials decrease exponentially. Recuperator design is an optimization challenge weighing allowable stresses against material volume requirements. A heat exchanger can be designed to incorporate less expensive stainless steels; however, more material would be required to reduce stresses. Increasing material volume per unit area in the heat exchanger increases thermal resistance and ultimately leads to lower recuperator effectiveness (Hesselgreaves, 2001). Decreasing the recuperator effectiveness would then have a detrimental effect on cycle efficiency (Dostal et al., 2005). On the other hand, that same recuperator could be designed to utilize high-strength nickel alloys requiring less material and perhaps allowing for higher effectiveness due to lower thermal resistance between flow paths, but nickel alloys are much more expensive than stainless steels.

A similar optimization exists when weighing recuperator effectiveness against recuperator size and power cycle efficiency. Generally, heat duty for a given recuperator module increases as the temperature difference between the high-pressure and low-pressure fluid streams increases. On the other hand, as the temperature difference between gas stream increases, the recuperator effectiveness

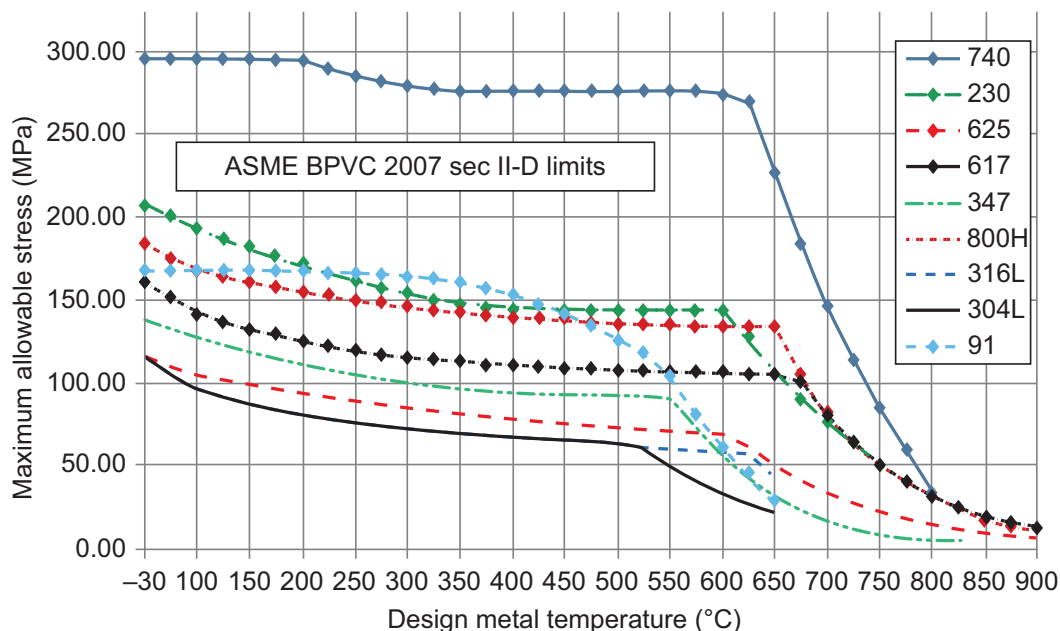


Figure 12.14 Curves of maximum allowable stress vs. design metal temperature for several candidate heat exchanger alloys (Carlson et al., 2014). BPVC, boiler pressure vessel code.

decreases, thus decreasing the efficiency of the power cycle. This means that less material is required for a given recuperative heat duty as the temperature difference increases. Modularity also needs to be considered when sizing recuperators. Several factors, such as performance, manufacturability (i.e., freight, brazing furnace size), and serviceability, will influence the most cost-effective module size, as single gigawatt-scale recuperators are very unlikely to be practical or cost effective. Ultimately, the parameter that captures everything is the COE produced by the plant, which takes into account power cycle efficiency and plant capital cost, including that of the recuperators. Because recuperators play such a major role in indirectly heated sCO₂ power cycles, the determination of effectiveness, operating temperatures, allowable stress, modularity, and material requirements must be optimized to achieve minimum COE for the plant. As sCO₂ power cycles emerge as a player in the market, availability of materials will need to increase and costs will therefore decrease. Breaking into the market with current materials costs makes recuperator design for low cost a major challenge for indirectly heated sCO₂ power cycles. Additional details on recuperators for sCO₂ power cycles can be found in Chapter 8.

Unmentioned in the earlier discussion are the challenges associated with materials exposure to sCO₂ at temperatures exceeding 600°C and pressures exceeding 25 MPa. Materials exposure tests have been conducted for short durations on the order of 1000 h (Mahaffey et al., 2014; Pint and Keiser, 2014; Saari et al., 2014), but there is still uncertainty associated with materials exposure to sCO₂ for long durations (see Chapter 4). As stated previously, utility scale fossil energy power plants would need to be safe, reliable, and capable of operating efficiently. This means heat exchangers must operate effectively for thousands of hours between maintenance shutdowns. Long-duration testing will be required of heat exchangers before sCO₂ power cycles can be adopted by the utility scale fossil energy sector.

12.2.5.3 Compressors/pumps

Perhaps one of the biggest technical barriers to predictable, reliable, and efficient operation of sCO₂ power cycles lies in the design of the MC. High mass flow requirements lead to a high shaft power requirement. Although pressure ratios are much lower than compressor sections in utility scale gas turbines, these sCO₂ compressors will be required to operate at much higher overall pressures than turbomachinery manufacturers are familiar with designing. Fortunately, much work has been done in recent years on CO₂ compressors for enhanced oil recovery (EOR) and CCS applications, such that several utility scale commercial compressor offerings are available at the desired sCO₂ outlet conditions. These tend to be of lower efficiency, particularly those designed for EOR, and compressor efficiency improvements are sought to help increase sCO₂ cycle efficiency.

Furthermore, commercial CO₂ compressors typically have inlet conditions near atmospheric pressure, whereas those for sCO₂ cycles will be much closer to the CO₂ critical point, creating uncertainty associated with multiphase flows that may exist locally. Condensation on rotating components may degrade performance or even damage the impeller. To make matters worse, acoustic velocity near the critical point

is low, which places the flow in a transonic regime. The combination of transonic flow with potential local phase changes makes the fluid behavior, and thus the compressor operation, unpredictable. Existing numerical prediction tools for this type of fluid behavior in a compressor are incapable of capturing important gas dynamic effects (Munroe et al., 2009; Lettieri et al., 2014), which requires compressor designers to be conservative, sacrificing precious efficiency points. While all the challenges noted in the previous discussion may be common to all sCO₂ power cycle applications (nuclear, CSP, waste heat recover, utility scale fossil), they are more critical for the large-scale MCs that would be required in a utility scale fossil power plant. To compete with tried and tested steam-Rankine cycles accepted and matured by the fossil energy sector for decades, indirectly heated sCO₂ power cycles must be designed for optimum efficiency and reliability. This will require improved prediction tools and design methods for these large-scale turbomachines.

The high-pressure compression of sCO₂ could result in significant heat of compression. As less energy is required to boost the pressure of a cool, dense gas, both upstream and interstage cooling may be desirable for compressor design. The number of compression stages and intercooling between stages will affect the overall power required for the compression process (Moore et al., 2007). Additionally, intercooling design (internal vs. external) must consider the additional pressure drops, potential leakage challenges, and piping system requirements associated with the compressor configuration. Intercooling approaches could include typical air-to-gas heat exchangers between compressor sections, or isothermal compression, which requires cooling the gas with ambient air between each stage of compression (requiring integrally geared or multistage centrifugal compressor with internal cooling). The isothermal process allows for more cooling than the traditional process but is still limited by the temperature of the ambient air. Subambient cooling (water cooling) upstream of the inlet CO₂ gas or in the internal diaphragms can provide additional improvements to compressor efficiency and power requirement (Moore et al., 2007).

Several studies have performed analysis on compressor selection for utility scale sCO₂ power plants. These studies have predicted that the MC of an sCO₂ Brayton cycle would likely be a radial unit for most power levels (at least for the first stage) to assure more robust operations near the critical point. The recompressor (RC) would transition to an axial unit above 100 MWe (Sienicki et al., 2011). Centrifugal compressors can maintain adequate performance, both in terms of overall efficiency and ability to operate without significant penalty, with inlet conditions close to critical pressure and temperature (EPRI, 2013).

12.2.6 Overall plant performance and costs

Performance for sCO₂ cycles is expected to be higher in terms of efficiency than for comparable steam cycles, as covered in Chapter 3. Power cycle efficiencies alone do not tell the whole story, however. As discussed earlier, the recompression cycle can be more efficient than a steam cycle at comparable turbine inlet temperatures, but this is due in part to the higher average temperature of heat addition to the cycle, since significant sCO₂ preheating occurs in the recuperator prior to entering the PHX.

From a fossil-fired power plant perspective, effective utilization of all the heat of combustion factors heavily into the overall plant performance, thus effective strategies for recovering low-grade thermal energy, must be included as discussed in [Section 12.2.4](#), if a recompression sCO₂ cycle is to be used. This point is illustrated in EPRI's studies, where a 2013 study lists recompression and reheated recompression sCO₂ cycles as having net HHV *cycle* thermal efficiencies of 52.1% and 53.1%, respectively, which outperforms the AUSC steam cycle thermal efficiency of 48.8%. In a later study ([EPRI, 2014a](#)), inclusion of a conceptual primary heater design, as well as an sCO₂ bottoming cycle for low-grade heat recovery, resulted in a *plant* HHV thermal efficiency of 41.6%, compared to 43.2% for a comparable steam plant. While the authors note that no attempts were made to optimize the sCO₂ cycle design to improve efficiency, these studies highlight the challenges associated with overall plant design to maximize use of the fossil-fueled thermal energy source. In this regard, an overall "boiler efficiency," as used in steam power plant design, is also a useful metric for measuring the performance of an sCO₂ power plant. This is the amount of heat transferred into the sCO₂ (or steam) cycle, divided by the fuel's thermal input.

As discussed in [Section 12.2.3.1](#), a study by [Moullec \(2013\)](#) employed a recompression sCO₂ cycle with two reheats and a turbine inlet temperature of 620°C to extract power from an air-fired coal combustion process. The base plant without carbon capture had a net thermal efficiency of 44.8% (HHV), compared to 40.5% (HHV) for the comparable steam plant. When integrated with an MEA CO₂ capture process, the sCO₂ plant had an HHV thermal efficiency of 36.9%, compared to 32.5% for the competing steam plant. The sCO₂ recompression plant designs included an sCO₂ economizer and a 500°C air preheater to recover the low-grade combustion thermal energy.

Studies of oxy-fired coal combustion sCO₂ systems have been done for CCS applications, and have also been shown to yield higher efficiencies than steam-based systems ([Shelton et al., 2016](#)). For an atmospheric pressure, oxy-fired CFB primary heater and a 620°C turbine inlet temperature, recompression cycles with a base configuration, reheat, intercooling, and intercooling + reheat yield plant HHV thermal efficiencies of 32.9%, 34.4%, 34.0%, and 35.2%, respectively, compared to 33.2% for the comparable Rankine cycle plant. Additional sCO₂ cycle improvements are also possible to improve the efficiency of these designs ([Shelton et al., 2016](#)). The ZEPS plant described in [Section 12.2.3.2](#), which includes an oxy-fired PFBC system designed for sCO₂ operation and CCS, yields a plant thermal efficiency of 37.3% (HHV), a 4 percentage point increase over the steam-based ZEPS plant ([Vega et al., 2014](#)). An indirect sCO₂ cycle powered by a separate sCO₂ oxy-combustion loop has been studied by [McClung et al. \(2014, 2015\)](#), and has a reported HHV plant efficiency of 37.5% with a carbon capture rate of 99%. The projected COE from this cycle is estimated at \$121/MWe.

In general, costs for utility scale fossil-fired sCO₂ plants are highly speculative, since none of the sCO₂ cycle components, with the possible exception of the compressors, have been built to this scale to date. [Moullec \(2013\)](#) offers a plant capital cost and levelized cost of electricity (LCOE) comparison, with sCO₂ turbine costs scaled from high-pressure steam turbine costs using the turbine volume as a scaling factor. This

analysis suggests a 15% lower capital cost for the sCO₂ plant with CCS, relative to the steam plant with CCS, largely due to reductions in the size of the turbine island. Coupled to fuel usage reductions from the efficiency benefit, this study projects a 16.6% decrease in LCOE for the sCO₂ plant (Moullec, 2013).

NETL's ongoing analysis of sCO₂ plant costs has thus far revealed that the large mass flow of sCO₂ relative to a steam plant of comparable net power output is a significant factor in the overall capital cost. This plays significant role in the costing of high-temperature (>620°C) sCO₂ cycles, where high-cost nickel alloys are required in significant quantities for the PHX tubing and sCO₂ transfer lines between the turbine and PHX. Given the significant role of sCO₂ mass flows in cycle capital costs, the specific power of the cycle, equal to the cycle net power output divided by the total sCO₂ mass flow rate, can be used as a surrogate for sCO₂ power cycle costs during the initial design stages. The specific power often follows the net cycle efficiency, but in some cases (i.e., compressor intercooling) the two may differ.

12.3 Direct supercritical CO₂ cycles

With recent advances in cryogenic oxygen production approaching 160 kWh/ton of O₂ (Tranier et al., 2011), direct sCO₂ power cycles, which employ oxy-combustion, have recently received interest as a potentially lower cost, fossil-fueled power cycle with inherent amenability to carbon capture. These cycles differ from the indirect sCO₂ cycles discussed throughout this book, since combustion is used to heat the sCO₂ internally within the cycle, allowing for higher turbine inlet temperatures and thus higher thermal efficiencies. Compared to conventional fossil-fueled power generation technologies, direct sCO₂ cycles are more akin to gas turbines, whereas most indirect sCO₂ cycles are similar to steam-Rankine cycles.

One characteristic feature of direct sCO₂ cycles is that the working fluid is not purely CO₂, but contains a mixture of diluents resulting from the internal combustion process. Combustion products are typically CO₂ and water, although additional diluents may come from the fuel or the oxidizer streams. The CO₂ recycle rate and CO₂ mass fraction are both very high (~90–95%), although provision must be made for the removal of diluents along with excess CO₂ to avoid buildup within the cycle.

12.3.1 Cycle overview and architecture

In the open, or direct-fired, sCO₂ system, natural gas or syngas fuel is burned with oxygen in a combustor and recycled CO₂ is used as a diluent to control the turbine inlet temperature. Various film and convection cooling approaches are used with the combustor and turbine components, similar to what is done in gas turbine cycles. The maximum firing temperature of these systems is typically limited by metallurgical constraints of the HTR, currently about 760°C for modern nickel-based alloys. With the relatively low pressure ratio at which these systems optimize ($Pr \approx 3-10$), the turbine inlet temperature and pressure end up around 1150°C and 30 MPa, respectively. This leads to a high power density cycle with a reduced footprint relative to

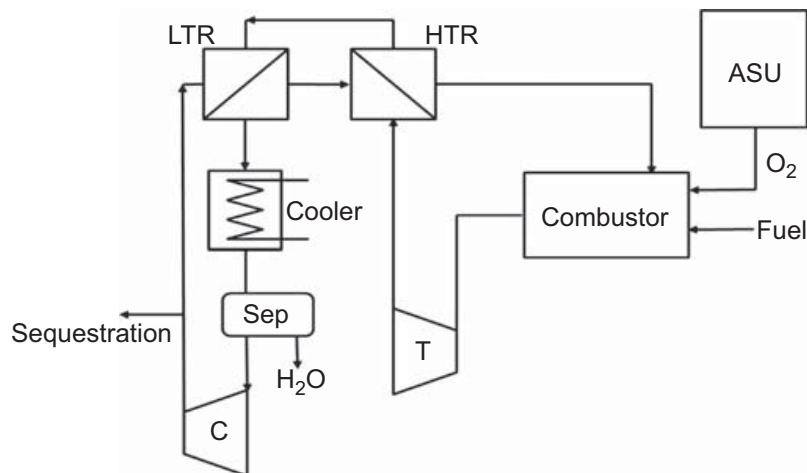


Figure 12.15 Direct-fired simple supercritical CO₂ Brayton cycle (Strakey et al., 2014). ASU, air separation unit; C, compressor; HTR, high-temperature recuperator; LTR, low-temperature recuperator; T, turbine.

conventional power generation technologies. Resulting capital cost reductions are somewhat offset by the need to contain the high pressures, but combined with the high efficiencies, direct-fired sCO₂ power plants are expected to be comparable to, or better than, conventional combined cycle plants with CCS on a COE basis.

A simple, direct-fired Brayton cycle with oxyfuel combustion and carbon sequestration is shown in Fig. 12.15 (Strakey et al., 2014). In this cycle, there is no recompression loop as all the flow leaving the LTR is cooled to near-ambient temperatures to remove the water from the system. The benefit of this approach is that the output of the compressor is a nearly pure stream of CO₂ that can be recycled to the combustor for temperature control with the remainder being ready for purification and storage. In this sense, the cycle permits carbon capture at no “additional” cost. A drawback of the simple cycle approach is that the flow split used in a recompression cycle cannot be used to overcome the pinch point issue in the recuperator. One way to deal with this is to integrate the waste heat from the ASU into the recuperator (Allam et al., 2013, 2014a).

12.3.1.1 Natural gas-fired cycles

Natural gas is an attractive fuel because it allows for the development and demonstration of various aspects of the technology with a clean burning fuel as opposed to coal-derived syngas. The expected thermal efficiency is also high (in excess of 58%) based on the LHV of natural gas (Allam et al., 2013). In a system analysis carried out by Allam et al. (2013), the turbine was assumed to operate at an inlet pressure of about 30 MPa with a pressure ratio of 10 and an inlet temperature of 1150°C. The turbine exit temperature was roughly 750°C. At the bottom end of the cycle the working fluid is cooled to near-ambient temperature, and the pressure is roughly 3 MPa. After water separation, the working fluid is compressed back up to 30 MPa in a multistage, intercooled compressor train. As noted in their study, the recuperator pinch point problem is overcome through the integration of waste heat from the ASU, which is

introduced into the 30-MPa working fluid in the recuperator. The pinch point is a result of the fact that the specific heat capacity of sCO₂ has a pressure dependence, increasing from 1.06 to 1.47 kJ/kg-K as pressure is increased from 3 to 30 MPa around 225°C. As was noted earlier, one of the main benefits of this cycle configuration for power production from natural gas is that the cycle naturally allows for CO₂ sequestration at pipeline ready pressures (about 15 MPa) without any additional compression work (Fig. 12.15). CO₂ separation and compression in conventional air-fired Brayton cycle gas turbine combined cycle power plants typically incurs a thermal efficiency penalty as much as 6 percentage points, along with a large capital cost increase (NETL, 2015a).

12.3.1.2 Coal-fired cycles

Coal can also be considered as a fossil fuel for direct-fired cycles; however, due to its inherent ash content, direct combustion of coal within the sCO₂ cycle is technically difficult due to the need for complete particulate removal at high temperature and pressure. Typically, the coal is first gasified and cleaned of its ash content, with combustion of the resulting syngas occurring within the sCO₂ combustor. Coal also contains other impurities such as sulfur, nitrogen, and chlorine, which may also be cleaned from the syngas before entering the sCO₂ cycle using conventional gasification technologies, although an efficiency benefit may result if these can be cleaned within the sCO₂ cycle (Lu, 2014). This is discussed in more detail in Section 12.3.2.1.

NET Power has also developed a coal-fired version of their direct sCO₂ cycle, in which coal is first gasified and cleaned before syngas is burned in the sCO₂ cycle combustor (Allam et al., 2013). In the baseline system, an entrained flow, dry-fed, slagging gasifier is used with a water quench, which produces a claimed net plant thermal efficiency of 47.8% (HHV) on bituminous coal (Lu, 2014). Variations in coal type, gasifier type, and heat recovery processes yield a range of HHV efficiencies from 43.3% to 49.7% (Lu, 2014).

The EPRI has also studied a syngas-fired direct sCO₂ power plant based on coal gasification in a slagging, entrained flow gasifier (EPRI, 2014b; Hume, 2016). The study includes Shell's dry-fed gasifier design (including a steam bottoming cycle powered by the syngas cooler's thermal output) and investigates the effects of oxygen purity and coal feed carrier gas on the purity of the CO₂ in the power cycle's turbomachinery. The study concludes that high oxygen purity (99.5%) and CO₂ coal feed carrier gas are required to produce a storage-ready stream with sufficient CO₂ purity (98.1%) for permanent sequestration (EPRI, 2014b).

A similar study was undertaken by NETL, also utilizing a Shell gasifier and gas cleanup train prior to introduction of syngas to the sCO₂ combustor (Weiland et al., 2016). The syngas cooler is used to raise steam for the coal drying, ASU, and sulfur removal processes, but is not used to power a steam bottoming cycle, as with the EPRI study (2014b). Additional heat is harvested in the syngas cooler by preheating compressed syngas prior to introduction to the sCO₂ combustor and to provide additional preheating of recycle sCO₂ for the oxycombustor. Several parameter sensitivity studies were undertaken, and at the time of this writing, the results are being used to refine the system model (Weiland et al., 2016).

12.3.1.3 *Condensing option*

While most of the proposed CO₂ power cycles operate solely in the supercritical regime, there are variations that employ condensation of the CO₂ before compression. At a pressure of 6 MPa, the condensation point of CO₂ is 20°C. The benefit of condensation is that the higher density of the CO₂ requires less compression power. In steam cycles, the condenser temperature does not significantly affect the cycle efficiency, although the opposite is true in sCO₂ cycles, since compression power can be significantly reduced by reducing the pump inlet temperature, thereby increasing cycle efficiency. In general, any effort to reduce the cold temperature of the sCO₂ cycle, short of adding a refrigeration system, will more than pay for itself through improved cycle efficiency (IEAGHG, 2015).

Also, condensing cycles offer the potential for reduced recuperation demand. This occurs because lower CO₂ temperatures require lower saturation pressures at the pump inlet, raising the turbine pressure ratio and lowering the turbine outlet/recuperator inlet temperature. The drawback of condensing cycles is that at temperature below about 100°C, the differences in specific heat between the low-pressure and high-pressure sides of the recuperator increases significantly, which aggravates the pinch point problem and introduces large irreversibilities (Kim et al., 2012).

Another issue for condensing cycles is the need for very cold cooling water (10.15°C) and small temperature approaches in the sCO₂ cooler. Cooling towers are another alternative to once-through cooling, where mechanical draft cooling towers are preferred to natural draft towers due to their reduced approach temperature (4°C vs. 7°C for natural draft) and smaller size. This allows for lower sCO₂ temperatures and higher densities at the compressor inlet, such that the reduction in compression power more than offsets the fan power requirements of the mechanical draft cooling towers (IEAGHG, 2015). The pre or recompression approaches typically employed in indirect cycles are not really appropriate for direct-fired cycles, which limits the applicability of condensation for direct-fired cycles.

12.3.1.4 *Thermal integration options*

In the natural gas-fired direct sCO₂ systems, thermal integration of the sCO₂ cycle is possible with the ASU and CO₂ Purification Unit (CPU), particularly with their compressors. NetPower's cycle requires thermal integration with the ASU compressor to help balance the heat duties on either side of the recuperator. To increase the amount of heat available for integration, the ASU main air compressor is not intercooled. Although this increases the ASU power requirement, its effect is offset by improved sCO₂ cycle recuperation, leading to higher overall plant efficiency (Allam et al., 2013).

In coal-fired systems, there are additional thermal integration opportunities with the gasification train. In particular, there is a considerable amount of heat that can be recovered from the syngas exiting the gasifier, which must be cooled prior to ash removal and syngas cleaning. In typical gasification systems, this heat is recovered in a syngas cooler, which raises steam for power and thermal duty elsewhere in the gasifier train. This can also be used for syngas preheating or additional CO₂ preheating

prior to combustion, although preheating oxygen-containing streams is discouraged for safety reasons (Weiland et al., 2016). The sulfur recovery plant and syngas compressor also provide sources of thermal energy for integration with the sCO₂ cycle, although this is a topic that has not yet been explored in current studies.

12.3.2 Working fluid

One of the unique challenges for direct-fired cycles is that the working fluid is no longer pure CO₂ as it is in indirect cycles. Depending on whether the cycle uses natural gas or syngas as fuel, the working fluid composition passing through the turbine will be different, as shown in Table 12.1. Furthermore, the working fluid impurities are a function of the temperature rise required in the combustor, thereby dictating the fuel and oxidizer requirements. This in turn is a function of the turbine pressure ratio, and the approach temperature at the hot end of the recuperator, which both determine the temperature of the recycle CO₂ entering the combustor.

Impurities in the working fluid as defined by anything other than CO₂ can have a detrimental effect on the cycle performance as shown in a system study by Hume (2016). For a compressor operating near the critical point of CO₂, the compressor power was found to increase by roughly 6% as the working fluid purity dropped from 100% CO₂ to 95.6% CO₂ with the balance being composed of O₂, N₂, H₂O, and Ar. A further drop to 90.9% CO₂ purity resulted in an increase in compressor power of roughly 34% over the pure CO₂ case. The increase in compressor power is a result of the decreased fluid density arising from gaseous impurities.

12.3.2.1 Sources of supercritical CO₂ impurities

Impurities can enter the sCO₂ cycle through the fuel, oxidizer, or other internal processes. Oxidizer impurities are either nitrogen or argon, the quantities of which depend upon the level of effort expended to separate oxygen from air in the ASU. Furthermore, use of more oxygen than is needed for combustion can result in oxygen impurities downstream of the combustor. Fuel impurities derived from natural gas include nitrogen at 0.9–1.6 volume %, depending on location, and should be accounted

Table 12.1 Turbine inlet fluid compositions (volume %)

Component	Natural gas (IEAGHG, 2015)	Coal/syngas (EPRI, 2014b)
CO ₂	91.80	95.61
H ₂ O	6.36	2.68
O ₂	0.20	0.57
N ₂	1.11	0.66
Ar	0.53	0.47

for in systems studies (NETL, 2012a; IEAGHG, 2015). In syngas-fired combustors, impurities are a function of the coal gasification process and syngas cleanup train. Nitrogen from the coal may be passed along to the syngas, as well as other sulfur-, chlorine-, and nitrogen-containing species that are not fully cleaned from the syngas. The syngas itself is mostly composed of CO and H₂, which are converted to CO₂ and water in the combustor. In general, water is typically condensed from the sCO₂ cycle after the recuperator, although its concentration in the sCO₂ can affect its condensation temperature, potentially affecting the recuperator heat balance. This can be mitigated by minimizing water content, thus a lower H₂/CO ratio in the syngas may be preferred. Even after a condensation-based water removal system, however, water vapor will always be present to some degree in the system.

The water content is an example of a process-dependent impurity, since it is generated from the internal combustion process. This water may also combine with CO₂ or other gaseous impurities to generate carbonic, sulfuric, and other acids. Other process impurities may come from coal transport fluids in the gasifier (EPRI, 2014b), syngas quench water, and incomplete combustion products. Furthermore, since the combustion process is occurring at near-stoichiometric conditions, CO can be expected to form in significant quantities. Camou (2014) measured CO concentration at the exit of a 10 MPa oxycombustor on the order of 150 ppm. Much of the CO formed in the combustor may shift to CO₂ as the combustion gasses cool in the turbine and recuperators, but this is very much dependent on the rate of expansion and cooling. For a very rapid expansion, the CO concentration may essentially be “frozen,” and not able to reach chemical equilibrium. Since these factors are system dependent, it is not clear how much CO can be expected at the low-temperature end of the cycle.

12.3.2.2 *Water and pollutant removal*

Aside from the immediate issues any of these pollutants may raise for the cycle efficiency or hardware life and reliability, there is also the need to reduce pollutants to acceptable levels before the exhaust gases can be released to the atmosphere or sequestered. Oxycombustion of syngas will produce many of the contaminants present in the products from a coal oxycombustion process including oxygen, nitrogen, and argon from the ASU, as well as acid gases such as SO₃, SO₂, HCl, and NO_x (Murciano et al., 2011). Both precombustion and postcombustion sulfur removal techniques have been considered for direct sCO₂ cycles utilizing syngas from coal gasification (Lu et al., 2016). McClung et al. (2014) concluded that desulfurization using limestone slurry with recovery of gypsum by-product is the preferred flue gas desulfurization method. On the other hand, postcombustion removal techniques may have efficiency benefits for direct cycles in theory (Lu et al., 2016). One factor important to consider for the postcombustion sulfur removal option is that hot corrosion from sulfur in the syngas combustion products can have detrimental effects on turbine and recuperator materials (Lai, 2007). Acid gasses such as SO_x and NO_x will be formed in the combustion process and are likely to condense out in either the LTR or the water removal system. These corrosive effects can critically shorten the life of nickel alloy components that will be required for use in the turboexpander as well as the

recuperator. The capital cost as well as maintenance costs of the recuperators could prove to be a limiting factor in the realization of large-scale commercial application (Strakey et al., 2014). Mercury is also a concern but is expected to fall out with the nitric acid (Allam et al., 2013).

Water vapor in the combustion exhaust remains in the working fluid through the turboexpander and recuperator, but must be removed prior to CO₂ compression. For a direct-fired cycle with a low-pressure side operating at 3 MPa, the condensation point for water is about 17°C. As the working fluid cools through the recuperator, it begins to condense. After exiting the recuperator, cooling water is used to condense the remaining water vapor, then liquid water is removed in a water separator (Allam et al., 2014a). Where liquid water exists in the presence of carbon dioxide, from the recuperator through the water separator, materials must be resistant to carbonic acid corrosion. Therefore, corrosion-resistant stainless steels must be used (Allam et al., 2014a).

12.3.2.3 Carbon capture and storage

One of the primary benefits of direct-fired sCO₂ cycles is its ease of CO₂ capture, which is relatively pure and at higher pressures. Underground storage of CO₂ requires that the CO₂ be pumped to high pressures for transport underground and to drive dispersion of the CO₂ into the bedrock or dissolution of the CO₂ into the brine formation. The pressure for storage is typically 2200 psig (15.08 MPa) for most systems studies (NETL, 2013).

In addition, existing CO₂ pipelines that are used for EOR have purity specifications for CO₂ that are in place to protect the pipeline hardware. These specs are listed in an NETL's Quality Guidelines for Energy Systems Studies (NETL, 2013), and in Table 12.2. Meeting these specs typically requires the use of a CPU, which utilizes

Table 12.2 CO₂ purity specifications per end use (NETL, 2013)

Impurity	Units	Carbon steel pipeline, saline reservoir storage	Enhanced oil recovery	Venting concerns
H ₂ O	ppmv	500	500	N
N ₂	vol%	4	1	N
O ₂	vol%	0.001	0.001	N
Ar	vol%	4	1	N
CH ₄	vol%	4	1	Y
H ₂	vol%	4	1	Y
CO	Ppmv	35	35	Y
H ₂ S	vol%	0.01	0.01	Y
SO ₂	Ppmv	100	100	Y
NO _x	Ppmv	100	100	Y

cryogenic distillation columns to reject impurities from the CO₂ storage stream (Jin et al., 2015).

A study by the IEAGHG (2015) compares the performance of a base sCO₂ system with a distillation column CPU (90% CO₂ capture rate at 99.8% purity) against a case without the use of a CPU (100% CO₂ capture at 98% purity), and against a case utilizing an additional membrane to improve capture rate in the base case CPU (98% CO₂ capture at 99.8% purity). The study found that in the case of no CPU, resulting in low CO₂ storage purity, lower auxiliary power consumption leads to an increase of about 0.2% in the plant's net thermal efficiency over the base case. For the high-purity, high-capture rate case including the membrane in the CPU, a higher compression power requirement reduces the plant's efficiency 0.4% relative to the base case. This underscores the need to consider CO₂ capture rates and CO₂ purity when comparing direct-fired sCO₂ cycles against one another, or against competing technologies.

12.3.3 Component challenges

Discussed in Section 12.2.5 of this chapter were the component challenges associated with indirectly heated sCO₂ cycles for utility scale fossil energy applications. While many of these challenges exist for direct cycles, there are a host of additional challenges associated with directly heated sCO₂ power cycles for fossil energy applications. This section discusses those component challenges unique to direct cycles.

12.3.3.1 Oxycombustor

At pressures on the order of 30 MPa, a direct-fired sCO₂ combustor is more likely to resemble a rocket engine than any type of conventional gas turbine combustor. At these very high pressures and energy release densities, issues such as injector design, wall heat transfer, and combustion dynamics may play a challenging role in combustor design. At these conditions, combustor design is an area where there is very little experience (Strakey et al., 2014).

Also, since these pressures are well beyond current design experience, computational fluid dynamics (CFD) modeling will not only be useful but also may be a necessity in the design process. CFD modeling to predict combustion dynamics or just overall mixing and performance requires validated chemical kinetic mechanism for the governing reactions. Typically, for natural gas combustion, GRI-Mech (2014) is used, but this mechanism has only been developed and validated for pressure below about 3 MPa, which is an order of magnitude below the typical 30 MPa of direct-fired systems. Reduced mechanisms, as well as simple global reaction mechanisms are usually derived from the more detailed mechanisms like GRI-Mech and as such are only valid for combustion below about 3 MPa. At higher pressures, 3-body recombination reactions deplete the pool of combustion radicals such as O, H, and OH, as shown in the equilibrium calculation in Fig. 12.16 (Strakey et al., 2014), which is for a stoichiometric CH₄/O₂/CO₂ mixture with a nominal temperature of about 2000K. Also note the increase in temperature as pressure is increased due to

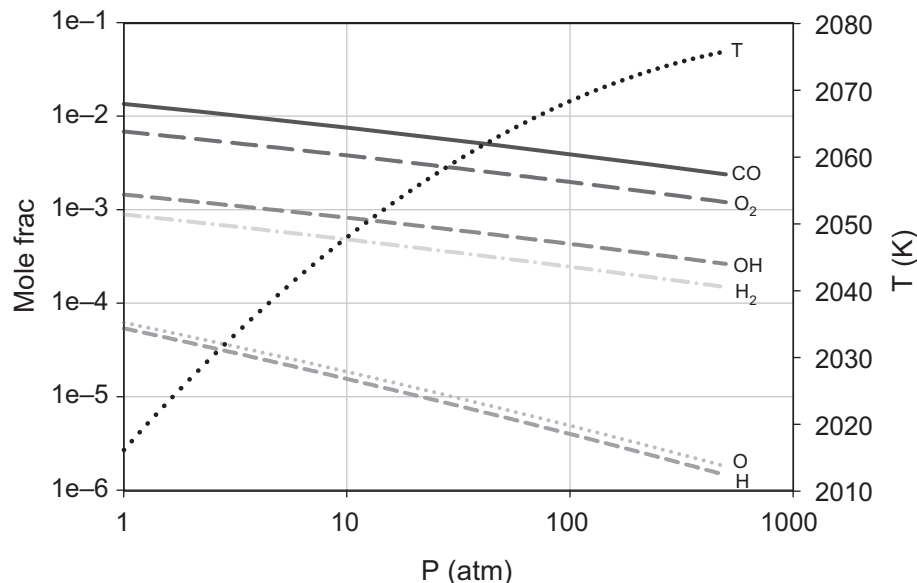


Figure 12.16 Equilibrium calculation for stoichiometric methane–oxygen mixture with carbon dioxide diluent (Strakey et al., 2014).

the decrease in radical concentrations and subsequent increase in product gas formation (CO₂ and H₂O).

The extrapolation of low-pressure kinetic mechanisms to the very high pressures of sCO₂ cycles introduces a large uncertainty in reaction rates and heat release. Confidence in CFD and other modeling predictions require validation of candidate mechanisms using shock tube, combustion bomb, and flame speed measurements at high pressure. For coal-based systems operating on coal syngas, even more data is needed for the prediction of high-pressure syngas chemistry with potential contaminants.

For oxy-fired combustion, NO_x formation is not really an issue as it is in gas turbines, so the preferred approach is diffusion flame combustion where the fuel and oxidizer are mixed and burnt in the combustion chamber. It is well known in the gas turbine community that diffusion flame combustors are inherently more resistant to combustion dynamics than premixed combustors. However, it is also well known that rocket engine combustors, which are all diffusion based, have a long history of combustion dynamics problems due to the very large pressure and large energy release density, which tends to couple with the resonant acoustic modes of the combustion chamber. Problems with combustion dynamics (Strakey, 2016) are typically discovered and solved with full-scale hardware testing through the use of Helmholtz resonators, baffles, and injector design modifications. Some of the most damaging coupling occurs in the 1–5 kHz range due to combustor tangential acoustic modes and is known as screech in the rocket engine community. Recently, some promise has been shown with CFD modeling using compressible large eddy simulations to predict combustion dynamics, although this is computationally expensive and far from maturity.

As of now, there is only one known high-pressure oxyfuel combustion rig in that has been operated. A 5-MW, 30-MPa combustor was developed and tested by Parametric Solutions Inc. (PSI) for Toshiba/NetPower (Allam et al., 2013). The PSI

combustor was designed mainly for demonstration purposes, although some data are publicly available (Iwai et al., 2015; Camou, 2014). A review of the pertinent literature can be found in Strakey et al. (2014).

12.3.3.2 Turbine

There are a number of design challenges for sCO₂ turboexpanders, many of which were discussed in Section 12.2.5 of this chapter including seals, bearings, throttle control valve durability, and erosion in turbomachinery components, among others. For direct cycles those challenges all still exist, but are even further from being resolved because of the higher operating temperatures expected in a direct cycle. A natural gas-fueled power plant based on the Allam cycle would need to have a turbine inlet temperature of 1150°C and pressure of 30 MPa to achieve an efficiency comparable with NGCC power plants (Allam et al., 2013). While sCO₂ properties are predictable at these temperatures and pressures, making aerodynamic design less of a challenge, structural design still remains a challenge because of the large blade loading expected from such a dense fluid. With turbine inlet temperatures exceeding 1150°C, turbines will not only require internal cooling but also may require film cooling. Cooling techniques matured through the development of air breathing gas turbines (Han et al., 2000) should be adaptable for application in sCO₂ direct cycle turboexpanders. A preliminary assessment of sCO₂ turbine blade cooling strategies has been performed by Aerojet Rocketdyne (EPRI, 2015), and the effect of blade cooling flows has been included in an external report on the natural-gas-fired Allam cycle configuration (IEAGHG, 2015).

The high direct sCO₂ cycle temperatures also preclude the use of turbine throttling valves upstream of the turbine, as would be used in indirect sCO₂ cycles, requiring alternative strategies for turbine throttle control. Similar to gas turbines, this is more easily accomplished via fuel and oxidizer flow control, which can provide rapid modulation of turbine inlet temperature, although the use of cold, high-pressure sCO₂ could also be used for hot sCO₂ attemperation, albeit at a penalty to cycle efficiency. Furthermore, the turbine shaft should also drive one of the cycle's compressors to provide a braking effect in the event of loss of electrical load.

Another complication not present in indirect cycles is the presence of impurities in the hot gas path of the turbomachinery. Downstream of the oxycombustor, the primary constituents will be CO₂ and H₂O, but depending on the fuel, other impurities such as SO₃, SO₂, HCl, as well as environmental particulates will be present in the hot gas path making erosion, corrosion, and deposition all possible issues for design consideration. Several studies exist in the open literature to report the effects of erosion, corrosion, and deposition in air breathing gas turbines (Hamed and Tabakoff, 2006; Richards et al., 1992; Wenglarz and Wright, 2003), but little is known about these effects at the high pressures that will exist in direct cycle turboexpanders. Some research has been conducted to better understand the effects of corrosion on stainless steels and nickel alloys in sCO₂ environments (Mahaffey et al., 2014; Pint and Keiser, 2014; Saari et al., 2014), but little is known about sCO₂ corrosion effects at temperatures exceeding 1150°C in the presence of water, sulfur, and other impurities, particularly

for the long-duration exposure times that utility scale turboexpanders will be expected to operate between maintenance shutdowns.

12.3.3.3 *Recuperator*

Recuperative heat duty in direct sCO₂ cycles is approximately twice as high as the electrical power output of the cycle (Allam et al., 2014a). While this recuperative heat duty is not as significant as in indirectly heated sCO₂ cycles, recuperator cost will still be a considerable factor in plant design. All the recuperator challenges discussed in Section 12.2.5 still exist for the recuperators in direct cycles, and direct cycle recuperators will be required to operate at much higher temperatures than even the HTR in an indirect cycle. In fact, as the highest temperature component in the cycle that cannot utilize cooling, the high-temperature and pressure capabilities of the recuperator are the limiting factor that prevents the cycle from operating at more severe, and more efficient, conditions. As turbine inlet temperature in the direct cycle exceeds 1150°C, turbine exhaust (recuperator inlet) temperature approaches 800°C (Allam et al., 2014a). To make matters more challenging, the direct cycle turboexpanders are expected to operate at a higher pressure ratio than indirect turboexpanders, making the direct cycle recuperator subject to higher pressure differentials than indirect cycle recuperators. These pressure differentials approaching 27 MPa will result in higher stress levels in the recuperator, as well as increased recuperator sizes or pressure drops to accommodate the larger volume flows at the low-pressure side of the cycle, relative to indirect sCO₂ cycles.

Ceramic heat exchangers will likely be required for the highest temperature recuperative duty to reach the full potential of these directly heated sCO₂ power cycles. Lewinsohn et al. (2016) are currently developing a ceramic heat exchanger for sCO₂ applications with focus on validation of design tradeoffs, verifying reliability, and verifying manufacturing costs which may approach \$200/kWt. Compact ceramic heat exchangers are immature and costly as compared to their metal counterparts and little is known regarding the effects that exhaust gas species at sCO₂ conditions may have on the ceramics, but ongoing R&D is expected to address these challenges.

There is no PHX in a directly heated sCO₂ power cycle so the challenges associated with design of that component do not exist for direct cycles; however, impurities in the hot gas path from the combustion of hydrocarbon fuel could cause fouling in the recuperator. This means that the challenges associated with fouling in the PHX of an indirect cycle transfer to the recuperator in the direct cycle. These particle fouling effects are discussed in Section 12.2.5.

12.3.3.4 *Compressor*

The design challenges for sCO₂ compressors discussed in Section 12.2.5.3 of this chapter (multiphase flows, condensation, aerodynamic losses, etc.) will likely apply to compressors utilized in direct cycle applications.

Diluents (consisting of mainly nitrogen and argon from the syngas and 95 vol% pure oxygen streams) entering the sCO₂ working fluid during combustion will likely be discharged throughout the cycle. Thus studies have shown an increase in the

specific power needed to compress the recycled CO₂ containing the impurities vs. a pure sCO₂ case as discussed in [Section 12.3.2 \(EPRI, 2014b\)](#). Optimization efforts to ensure purity of the CO₂ working fluid will be required to combat large power requirements and reduce cycle efficiency penalties.

Additionally, as discussed in [Section 12.3.3.2](#), combustion of coal-derived syngas will produce impurities of SO₂ and NO/NO₂. As these impurities will be exposed to condensed liquid water and excess oxygen at later points in the cycle, namely, within the heat exchanger, acid conversion (H₂SO₄ and HNO₃) is expected to occur ([Allam et al., 2013](#)). These impurities present corrosion challenges to exposed turbomachinery, including the direct cycle's CO₂ compressor, depending on the cleanup and separation processes included within the specific cycle. While cycle specifics may vary, careful design selection of corrosion-resistant turbomachinery will likely require consideration.

12.3.4 Cycle performance, cost, and prospects

The significant recent interest in direct sCO₂ power cycles comes largely from their claims to high efficiency with CCS included. A summary of the results from the direct sCO₂ systems analyses in the literature is shown in [Table 12.3](#).

The results in [Table 12.3](#) show that natural gas-powered direct sCO₂ cycles generally perform better than coal-fired cycles, which is expected due to the added requirement of a gasification train for coal use. The IEAGHG analysis of the Allam cycle in [Table 12.3 \(IEAGHG, 2015\)](#) results in about 3 percentage point lower efficiency than that reported by [Allam et al. \(2013\)](#); however, NetPower commentary contained in the IEAGHG report explains that these differences result from improved thermal integration, cooling, and overall process optimization relative to the

Table 12.3 Direct-fired sCO₂ plant performance results from selected studies

Source	Fuel	Turbine inlet		Recuperator inlet	Plant thermal efficiency
		°C	bar	°C	HHV
McClung et al. (2015)	Methane	1200	200	850	46.5%
IEAGHG (2015)	Natural gas	1150	300	740	49.9%
Allam et al. (2013)	Methane	1150	300	775	53.1%
Allam et al. (2013)	Illinois#6 coal	1150	300	775	48.9%
EPRI (2014b)	PRB coal	1123	300	760	39.6%
Weiland et al. (2016)	Illinois#6 coal	1149	300	760	38.1%

[IEAGHG study \(2015\)](#). The studies by McClung et al., report efficiencies similar to that of the IEAGHG study at similar conditions (2015). Furthermore, they achieved higher thermal efficiencies in condensing recompression cycles in a direct-fired configuration, although the turbine outlet temperatures were well beyond the capabilities of existing HTRs.

For coal-fired systems, the high thermal efficiency of 48.9% (HHV) reported by [Allam et al. \(2013\)](#) falls within the range of thermal efficiencies reported by [Lu \(2014\)](#) (43.3–49.7%), which cover several variations in coal type, gasifier type, and heat recovery processes, although turbine inlet and other cycle conditions are not reported. Part of the reason for higher efficiencies than reported in the other coal-fired studies in [Table 12.3](#) is the use of SO_x and NO_x removal processes that are internal to the sCO₂ cycle, as discussed in [Section 12.3.2.1](#). This may improve the plant HHV thermal efficiency by about 3 percentage points ([Lu et al., 2016](#)), although the impact of acids in the sCO₂ working fluid will need to be considered from a material compatibility perspective. Further efficiency improvements are reflective of significant thermal integration between the gasifier and sCO₂ cycle, elimination of steam generation requirements for conventional syngas cleanup, and high-pressure (8.5 MPa) gasifier operation ([Allam et al., 2014b](#)).

As for plant costs and COE, very little information is available, and what has been published is highly speculative, given that many of the required components have never been built and are still in development. Further, caution should be applied in the comparison of COE or LCOE estimates between studies, as these metrics include operating and financial assumptions (e.g., capacity factors, inflation rates, financing structures, etc.) that reach well beyond the core technologies considered.

For the natural-gas-fired case, [IEAGHG \(2015\)](#) estimates a LCOE of \$105.3/MWh, while Net Power reports an LCOE of \$92.9/MWh. In the coal-fired case, EPRI's study estimates the LCOE at \$133/MWh, although significant uncertainty in the sCO₂ capital cost was noted ([EPRI, 2014b](#)).

The primary competing technology for natural-gas-fired direct sCO₂ systems are NGCC plants with CCS. NETL's baseline studies list the HHV thermal efficiency of an NGCC plant using a state-of-the-art 2013 F-class turbine at 45.7%, with a COE of \$83.3/MWh, excluding carbon transport and storage (T&S) costs (\$87.3/MWh including T&S) ([NETL, 2015a](#)). Direct sCO₂ systems are therefore competitive in terms of thermal efficiency and cost, accounting for differences in fuel prices and other economic factors between the studies.

Similarly, IGCC systems are the conventional coal-fired technology with CCS that directly competes with coal-fired direct sCO₂ cycles. NETL's baseline study of a GE radiant gasifier with CCS yields a plant HHV efficiency of 32.6%, with a COE of \$135.4/MWh (\$144.7/MWh with T&S costs) ([NETL, 2015b](#)). Direct, coal-fired sCO₂ systems are therefore very competitive with IGCC systems; however, additional competition may come from indirect sCO₂ plants with CCS, with air-fired HHV efficiencies of 36.9% ([Moulec, 2013](#)) and oxy-fired HHV efficiencies of 35.2–39.8% ([Shelton et al., 2016](#); [Subbaraman et al., 2011](#)), depending on turbine inlet temperature and cycle configuration.

12.4 Conclusions

Application of indirect $s\text{CO}_2$ power cycles to fossil-fueled applications has been considered, including discussion of relevant power cycle configurations, integration with various types of primary heaters, and techniques for recovering low-grade heat in the combustion flue gas. With appropriate consideration given to these issues, efficient and cost-effective coal-fired indirect $s\text{CO}_2$ cycles are achievable, although additional thought must also be given to the operational load profile of the plant, and to size-related component difficulties that must be worked through as the cycles are scaled up from demonstration to commercialization.

Unfortunately, near-term prospects for coal-fired indirect $s\text{CO}_2$ power plants in the United States are marginal, due largely to the EPA's Carbon Pollution Standards Final Rule, which cap carbon emissions from new coal-fired power plants at 1400 lbs CO_2 per megawatt of gross power generation (EPA, 2015). In addition, recent natural gas prices have allowed NGCC power plants to become cost-competitive with coal plants. Coupled to the lower capital cost of NGCC plants, little coal generation has been proposed or is in construction in recent years. However, it may be possible to reach this emissions limit with an air-fired $s\text{CO}_2$ cycle operating at AUSC conditions; analyses for such a case has not yet been performed as of this writing. If possible, this may boost interest in coal-fired $s\text{CO}_2$ cycle development in the United States. Fossil-fired $s\text{CO}_2$ cycles may have improved prospects internationally; however, research and development interest in $s\text{CO}_2$ cycles has only recently increased internationally and may reflect low interest. The 10-MWe demonstration $s\text{CO}_2$ plant under planning and design at the time of this writing may spark interest both domestically and internationally, if successful.

Demonstration of direct-fired $s\text{CO}_2$ power plants is also proceeding through the development and construction of the Net Power/8 Rivers 25-MWe power plant in La Porte, Texas. These cycles offer high efficiency with an inherent ability to capture CO_2 at relatively high purity and pressure, with natural gas combustion occurring within the cycle, and coal utilization made possible by gasification of the coal and combustion of the resulting syngas. Specific challenges in these cycles, as discussed earlier, are related to impurities in the predominantly $s\text{CO}_2$ working fluid, the higher temperatures in the cycle that require combustor and turbine cooling, and specific oxy-combustion challenges at pressures comparable to those in rocket engine combustion.

Future interest in direct $s\text{CO}_2$ cycles will forever be tied to the desire to capture and store/utilize CO_2 emissions from fossil-fueled power generation. Given that greenhouse gas reductions are likely to be the major thrust of power generation research in the coming decades, prospects for continuation of development of direct-fired $s\text{CO}_2$ power cycles seems likely to be strong for the foreseeable future.

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