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Perspective of S–CO_2 power cycles

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ABSTRACT

In this perspective paper, the research and development of S-CO_2 cycles are analyzed from two aspects: (i) the system design and analysis and (ii) energy transfer/conversion mechanisms and key components development. Based on the analysis, barriers for further promotion of S-CO_2 cycles are summarized, including the lack of system design and analysis methodology, not well understood mechanisms of energy transfer/conversion, and technic barriers such as seal, leakage and roto-dynamics stability of key components. To overcome these issues, perspectives on three aspects are proposed. First, S-CO_2 cycle adapting to the distinct characteristic of a heat source should be optimized to promote the global system efficiency. Second, new theoretical/numerical works are suggested emphasizing the real gas effect of S-CO_2 to improve the accuracy, convergence and stability of numerical simulations, fine experiments should be expanded to verify the correctness of the numerical simulations. Third, the integrated solution strategies for key components such as intermediate heat exchanger, recuperator heat exchanger and turbomachines should be developed and verified in large-scale test loop for reliable and efficient operation. This critical review is hopefully to present readers a clue to promote the development of S-CO_2 cycles driven by nuclear energy, renewable energy and fossil energy.

1. Introduction

Since 1822 when Baron Charles Cagniard de la Tour discovered supercritical fluids while conducting experiments involving the discontinuities of sound in a sealed cannon barrel filled with various fluids at high temperature, supercritical fluids have been investigated extensively for food preparation and ingredients, pharmaceuticals, cosmetics, polymers, powders, biofuels and functional materials [1]. Currently, supercritical water-steam Rankine cycles are widely used for electricity generation driven by primary energy of coal/natural gas, nuclear and solar. The water-steam Rankine cycle power plants have some limitations. The cycle efficiency can be improved when increasing the turbine inlet temperatures. However, this improvement is limited by metallic materials. For high steam temperature ~700 °C, metallic corrosion is serious [2]. Besides, Rankine cycles operate some facilities in high pressure but others in vacuum pressure, resulting in large component/system size.

Carbon dioxide is non-poisonous, non-flammable, colorless and tasteless. S-CO_2 cycle uses CO_2 as the working fluid to recover heat from heat source, generating CO_2 vapor to drive turbine. S-CO_2 cycle has following benefits. First, supercritical cycle can be easily reached due to low critical parameters (304.13 K/7.377 MPa). Second, it is possible to further increase turbine inlet vapor temperature to improve cycle efficiency, due to more inert nature of CO_2 with metallic material than water-steam. Third, S-CO_2 cycle is compact to be attractive for many applications. Fig. 1 shows physical properties of water and CO_2.

S-CO_2 cycle was first proposed by Sulzer in 1950 [3]. The thermal efficiency of a S-CO_2 cycle is higher than a water-steam Rankine cycle for higher vapor temperature than 550 °C [4]. S-CO_2 cycle was

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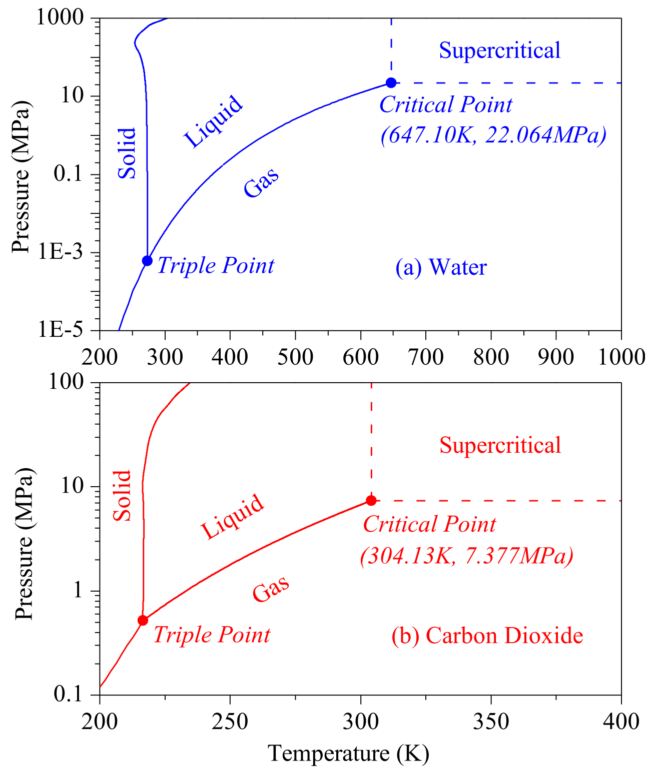

Fig. 1. The three-phase pattern for water and CO_2 (a: water, b: CO_2).

not paid much attention since its concept proposal. Recently, many countries put great efforts in the research and development of S-CO_2 cycles (see Fig. 2 for the trend of publications). This hot research area is related to the green energy development. The electric supply demands the reduction consummation of fossil energy but the increased applications renewable energy utilization [5]. The electric grid will be operating in a mixing mode to include energies of fossil, renewable and nuclear. The fossil energy power plants slowly respond to load variations. The mismatch of time constants between fossil energy and renewable energy power generations accounts for the “wind and solar curtailment” phenomenon [6]. S-CO_2 cycle balances various power generations, creating a robust and smart grid. Due to better efficiency and

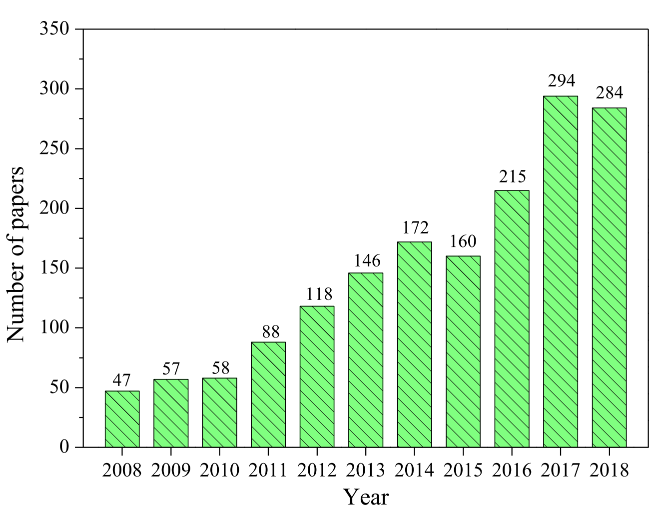

Fig. 2. Number of papers regarding S-CO_2 cycles on Web of Science from 2008 to 2018.

compact size, S-CO_2 cycles also have the potential to be applied in distributed or portable energy systems.

2. Current options and barriers

2.1. Heat source options

S-CO_2 cycle can be driven by nuclear energy, solar energy, fossil energy and waste heat. Direct S-CO_2 cycle uses CO_2 as the coolant to recover the heat source energy, but the indirect S-CO_2 cycle extracts the heat source energy via an intermediate heat exchanger. Since the early 2000s, many countries have attempted to develop Generation IV nuclear reactors, targeting at the efficiency improvement and emission reduction. Dostal et al. [4] compared the S-CO_2 cycle with helium-gas and water-steam based cycles, concluding the recompression S-CO_2 cycle being suitable if the outlet temperature of a reactor core is higher than 550 °C. Due to the narrow range of heat source temperature, S-CO_2 cycle makes the convenient optimization and control of the system. For concentrated solar power (CSP) applications, the direct S-CO_2 cycle has higher global efficiency [7,8]. The benefit is accompanied by the increased cost of thermal storage [8]. On the contrary, the indirect S-CO_2 cycle has lower efficiency due to various heat losses. The oscillating solar energy challenge the fine control and the system operation.

The fossil energy S-CO_2 cycle is classified as air and oxygen combustion system. The former is developed by EDF (France), EPRI (USA) and China [9–11]. The oxygen combustion system includes the systems of Direct-Oxy-Fuel-Combustion and Indirect-Oxy-Fuel-Combustion [10,12]. The Allam cycle belongs to the direct combustion system [12]. Natural gas is burned with the mixture of O_2 and CO_2 in a combustor to drive the turbine. The exhaust gas releases heat in a recuperator heat exchanger and is further cooled. Water is separated from the mixture and the high purity CO_2 returns to the cycle for heat-power conversion. The Allam cycle has thermal efficiencies of 59% for natural gas and 52% for coal [12]. The cycle processes carbon capture easily. The high-pressure-combustion ~30 MPa is a challenge for cycle operation. The 300 MWe natural gas fired Allam cycle is in the design stage and will be put into operation in 2021 [13,14]. In an indirect combustion system, the two CO_2 fluids in furnace side and boiler side are separated by tube walls.

S-CO_2 cycle can recover the waste heat of a gas turbine. The temperature of the waste heat is in the range of 500–600 °C [15–17]. The waste heat S-CO_2 cycle shall compete with the water-steam cycle and organic Rankine cycle. Echogen (USA) has fabricated a 7.3MWe waste heat power plant [15]. Kim et al. [17] found that the recompression S-CO_2 cycle is not suitable for low temperature heat source. The partial heating CO_2 cycle is more suitable to have simple system layout, while the split CO_2 cycle generates more power. The transcritical CO_2 cycle is an alternative option to recovery the waste heat. Shu et al. [18] proposed a transcritical CO_2 cycle including a pre-heater and a recuperator heat exchanger. It is shown that such a cycle increases the net power output by 50% compared with a conventional transcritical cycle. Kim et al. [19] noted the importance of the temperature match between heat source and working fluid of the cycle on power generation.

2.2. Barriers of S-CO_2 cycle

There are various heat sources to drive S-CO_2 cycle. There does not exist a fixed S-CO_2 cycle that can be suitable for all the heat sources. For instance, the recompression cycle is suitable for nuclear or solar energy, but is not good for waste heat [17]. For fossil energy, S-CO_2 cycle shall recover the flue gas heat over a very wide temperature range, which is one of the difficult tasks. Besides, the development of S−CO_2 cycles presents many scientific and technical issues to be addressed. At present, the commercial operation of a S-CO_2 cycle power plant is not available. Targeting at higher cycle efficiency, better economic performance and stronger feasibility, we highlight following barriers that prevent from commercial and wide applications of S-CO_2 cycles.

(1) There is lack of methodology for system design and analysis for S-CO_2 cycles driven by different heat sources, under design/off-design condition and transient operation.
(2) S-CO_2 heat transfer and heat-power conversion mechanisms are not fully understood, preventing key components from accurate design and evaluation.
(3) There are various technic barriers of fabrication, seal, leakage and roto-dynamics stability for various heat exchangers and turbomachinery.

The detailed reasons why we propose these barriers and the corresponding perspectives will be presented in next section.

3. Future perspectives

3.1. Material compatibility with S-CO_2

Chemical reaction is important to decide the maximum CO_2 vapor temperature that we can have in the cycle. The material compatibility in pure CO_2 and CO_2 mixture with impurities of O_2 and H_2O are reported in Refs. [20−23], respectively. The alloy with higher content of chrome, nickel, titanium or aluminium has lower reaction rate in high pressure/temperature CO_2 environment. CO_2 temperature is more important than pressure to influence the corrosion rate. Drawbacks on this topic are: (i) ultra-purity (>99.99%) CO_2 was used [24,25], deviating from practical cycle operation environment; (ii) some measurements may not last enough time [20−23,25]; and (iii) reaction data was characterized by the weight increment method [20−25], it is more valuable to use the weight loss method. The database of metal compatibility with CO_2 and its mixture should be expanded using the standard method.

3.2. S−CO_2 cycle analysis

Recompression (RC), recompression + intercooling (RC + IC) and recompression + reheating/double-reheating (RC + RH/DRH) are the basic S-CO_2 cycles (see Fig. 3). Other S-CO_2 cycles can be considered as the extended versions. RC separates the total CO_2 flow rate into two streams, one stream dissipates extra heat to environment and then is compressed, but the other stream is directly compressed. RC + IC cycle introduces an intercooler to cool CO_2 during compression to reduce the compression work. RC + RH/DRH has one or two additional heaters to heat CO_2, thus expansion occurs in several turbines in a higher temperature level. RC + IC has a slight efficiency improvement compared to RC, but the effect of reheating is obvious [11]. Two recuperator heat exchangers (LTR and HTR) are included in the cycle. S-CO_2 cycle has lower pressure ratio and temperature drop across the turbine inlet and outlet compared with water-steam Rankine cycle. The turbine outlet temperature is high such as ~550 °C [9,11]. Direct heat dissipation to environment obviously worsens the cycle efficiency. Recuperator heat exchanger recycles part of heat in the cycle to decrease CO_2 temperature entering cooler, improving the cycle efficiency.

Because different heat sources have different characteristics, it is hard to fix a specific cycle that is suitable for all heat sources. The first task is to optimize suitable S-CO_2 cycles for specific heat source to reach maximum global efficiency. RC is suitable for nuclear/solar energy within narrow temperature range, but is not good for waste heat [17]. Noting that the global efficiency includes various efficiency components, the S-CO_2 cycle efficiency is only one of these efficiency components. Sometimes, a higher cycle efficiency may not mean an optimal global efficiency. Even though many articles deal with renewable and nuclear energy S-CO_2 cycles, these studies focused on the analysis of the cycle alone, not coupling with the heat source [26]. For coal fired power plant, the flue gas covers a wide temperature range 1500−120 °C. A cascade energy utilization strategy including a top cycle, a bottom cycle and an air preheater extracts the flue gas energy over an entire temperature range [27,28]. The global analysis of S-CO_2 cycle coupling with heat source is strongly welcome. Thermodynamics analysis coupling with thermal-hydraulics is also recommended.

The second task is to determine reasonable key component efficiencies for system design, which are overestimated by many investigators. For instance, isentropic efficiencies are set as 0.9 for compressor and 0.93 for turbine [9,29], and they are 0.92 for both compressor and turbine [30]. Power capacity strongly influences compressor and turbine efficiencies. Small capacity chooses radial flow machines but large-scale needs axial flow machines. The high efficiencies of ~0.9 are difficult to be reached for small capacity applications (<10 MW). The efficiency data for large-scale axial-flow compressors and turbines are not found in references. The extended studies are welcome in this field to have reliable system analysis.

3.3. S-CO_2 heat transfer and heat exchangers

3.3.1. Fundamental studies of S-CO_2 heat transfer

S−CO_2 heat transfer takes place in various components such as intermediate heat exchanger, recuperator heat exchanger and cooler. The available S-CO_2 heat transfer data are limited to small diameter tubes and low parameters of pressures, temperatures and heat fluxes [31,32]. For practical operation, the pressure should be > 20 MPa (~3 times of the critical pressure), the heat flux can be > 100 kW/m^2. The available experimental data focuses on the near-critical pressure ~8 MPa [32]. Many authors reported numerical simulations of S-CO_2 heat transfer, using RNG k-ε model and other low Reynolds number k-ε models of LS, YS and AKN, which are embedded in ANSYS software [33,34]. The sharp variations of physical properties and buoyancy force and acceleration effects are paid attention to influence the S-CO_2 heat transfer [35,36].

There are no general correlations of heat transfer coefficients suitable for wide experimental data ranges [31,32]. The empirical heat transfer correlations are suitable for ones' own data, but are difficult to be extended beyond the parameters range. The heat transfer deterioration HTD is key to ensure the safety operation, but the mechanism is still an open question. Classically, supercritical fluid is believed to have the homogeneous single-phase structure. Zhu et al. [37] used the "boiling" concept to deal with S-CO_2 heat transfer. The core idea is to decouple the added heat into two parts, one part for temperature rise, and the other part for "boiling". The analogy between subcritical heat transfer and supercritical heat transfer results in a new non-dimensional parameter (supercritical-boiling-number *SBO*) to dominate the HTD occurrence. Covering wide parameters range, sudden changes between normal heat transfer and heat transfer deterioration are discovered when crossing a critical *SBO* of 5.126×10^{-4}. The finding supports the heterogeneous structure of supercritical fluids.

We suggest the following S-CO_2 heat transfer studies: (i) expanding experimental database covering wide parameters range including pressures, temperatures, heat fluxes, tube diameters, circular/non-circular channels, uniform/non-uniform heating conditions; (ii) improving the prediction accuracies of numerical

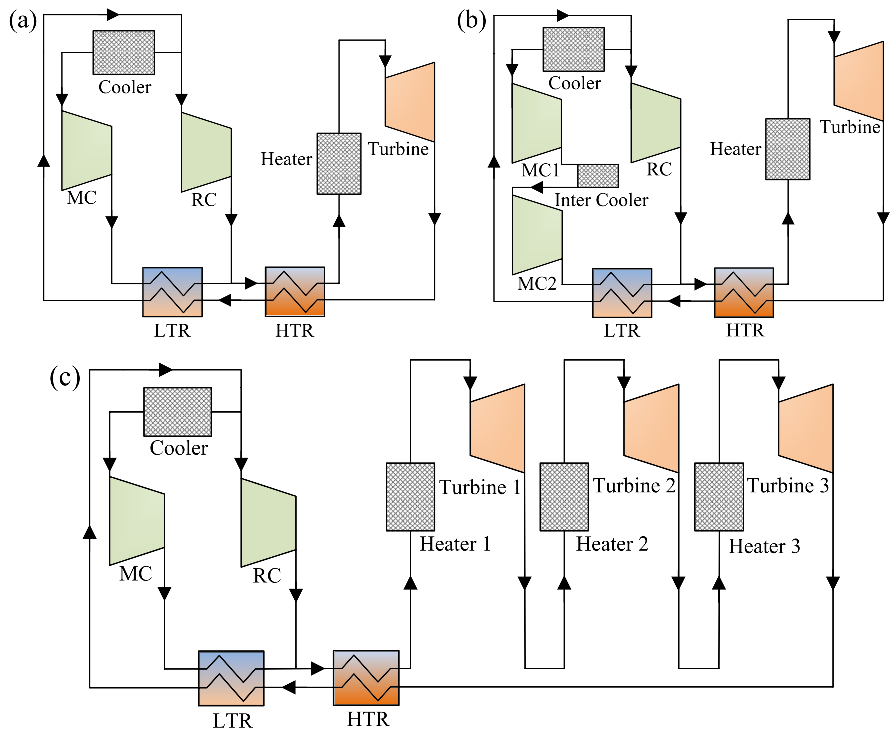

Fig. 3. S-CO_2 layouts (a: RC cycle, b: RC + IC cycle, c: RC + DRH cycle; MC-main compressor, RC- recompression compressor, LTR-low temperature recuperator, HTR-high temperature recuperator).

simulations; (iii) development of general heat transfer correlations; (iv) the heterogeneous structure of supercritical fluids and its effect on supercritical heat transfer.

3.3.2. *Heaters for direct/indirect S-CO_2 cycles*

The generation IV nuclear reactor uses direct/indirect S-CO_2 cycle in gas cooled reactors or indirect S-CO_2 cycle in all reactors for heat-power conversion. Indirect S-CO_2 cycles make reheat possible [38]. The review on nuclear reactor is beyond the scope of this paper, but intermediate heat exchanger (IHX) is paid attention here. IHX couples the primary loop of nuclear reactor and the secondary loop of S-CO_2 cycle. The pressure drop of IHX affects the cycle efficiency, thus gas cooled reactor with gas-gas IHX performed worse than liquid metal cooled reactor with lead alloy-gas IHX [38]. In addition, for liquid metal cooled reactor, pool reactor configuration is more widely recognized than loop configurations [39,40]. Therefore, the design methodology coupling liquid metal and CO_2 should be optimized.

Concentrated Solar Power (CSP) S-CO_2 cycles are classified as direct and indirect ones. Direct cycle behaves higher global efficiency than indirect cycle. In direct cycle, S-CO_2 flows in solar receiver tube to extract solar energy. The high heat flux irradiation and strong non-uniform circumference heat flux distribution present severe heat transfer issue and strong thermal stress to threaten the solar receiver safety [41]. Numerical simulation and experiment studies should be enhanced to support the solar receiver design and operation. In indirect S-CO_2 cycle, molten salt can be the heat carrier fluid to extract solar energy, which is similar to indirect water-steam Rankine cycle. Current investigations on the thermal coupling between molten salt and CO_2 are not sufficient [42], but should be paid great attention. Technical issues such as corrosions, leakage and blockage due to the use of molten salt should be overcome before commercial operation.

For coal or natural gas fired power plant, the direct S-CO_2 cycle contains heaters in different boiler parts (cooling wall tubes, preheaters, reheaters and superheaters), presenting challenges on the design and operation. First, the S-CO_2 heat transfer coefficient is in the range of 3–5 kW/m^{2}K, which is not as high as one imagines. At the heat flux of 200–300 kW/m^2, the temperature difference between CO_2 and inner tube wall can reach 40–100 K. Second, required by the S-CO_2 cycle, the CO_2 temperature entering the heater tubes reaches ~520 °C, which is more than 100 °C higher than a water-steam boiler [11]. Third, CO_2 is heated with non-uniform heat flux boundary condition. Heat transfer deterioration may occur at the extreme-heating conditions. Recent progresses can be found in Refs. [35,37]. Investigations should be continued to guarantee the heaters safety.

For waste heat S-CO_2 cycle, a heat exchanger extracts heat from waste heat source. Shell-tube heat exchanger contains fins on outer tube walls to enhance heat transfer. CO_2 flows in tubes. Plate heat exchanger involves flue gas flowing in one side and CO_2 flowing in opposite side. Because the heat exchanger operates in dirty environment, particle deposition and corrosion are series problems to be addressed. To raise the heat extraction efficiency, the match between flue gas and CO_2 should be minimized to have a small global temperature difference [43].

3.3.3. *Printed circuit heat exchangers (PCHE)*

PCHE was initially developed by Heatric (UK). Due to the high power-density and compact size, PCHE has been applied in offshore oil/gas processing, floating liquefied natural gas devices and nuclear power systems [44]. For a S-CO_2 cycle, the recycling heat can be 3–4 times of the net power output [11]. Reducing the recuperator heat exchanger size is important to keep the whole system compact and fast response to load variations. PCHE has been demonstrated to be effective in small-scale S-CO_2 cycles [45]. NET Power (USA) has integrated PCHE into a 50 MWth natural gas demonstration power plant [14]. Zigzag is the conventional flow channel configuration. Due to ~1 mm scale hydraulic diameter of channels, the zigzag configuration generates larger pressure drops. Recent progress focused on the optimization of channel configuration and size to decrease pressure drop while enhancing heat transfer. The S-shape PCHE reduces pressure drop to one-fifth of the zigzag PCHE [46]. Alternatively, the airfoil channel PCHE decreases pressure drops to one-twentieth of the zigzag channel PCHE [47]. Jiang et al. [48] indicated that PCHEs have fast responses due to small metal masses and high heat transfer coefficients compared with conventional shell-tube heat exchangers. New methods shall be developed to clean impurities in curved channels, decrease the thermal stress and fabrication cost, which are important for large scale utilization of PCHEs.

3.4. Turbomachines for S-CO_2 cycles

Different from water-steam Rankine cycle, turbomachines in S-CO_2 cycle behave distinct characteristics [49,50]: (i) low pressure ratio due to high pressure operation (>7.38 MPa); (ii) serious bearing, sealing and rotor-dynamics issues due to high pressure operation and large axial force; (iii) ultra-high rotating speed (20,000 rpm for 10 MWe turbine). Radial flow and axial flow turbomachines are suggested for small-scale and large-scale capacities, respectively.

Fig. 4 shows *T-P* curves with inlet/outlet state parameters cited from Ref. [28]. The dotted lines are possible compression and expansion processes. A supercritical pressure corresponds to a pseudo-critical temperature T_{pc} at which specific heat is maximum. The turbine expansion deviates from pseudo-critical point, but the compression approaches pseudo-critical point especially at the inlet and outlet. The numerical simulation based on ideal gas assumption causes obvious deviations from measurements, regarded as the real gas effect due to sharp variations of physical properties when crossing pseudo-critical point [51]. The singular behavior near pseudo-critical point prevents numerical simulations from capturing important gas dynamic effects. Currently, various numerical simulations are based on commercial software, which may not capture the real gas effect [52]. Numerical simulations should be improved regarding the accuracy, convergence and stability.

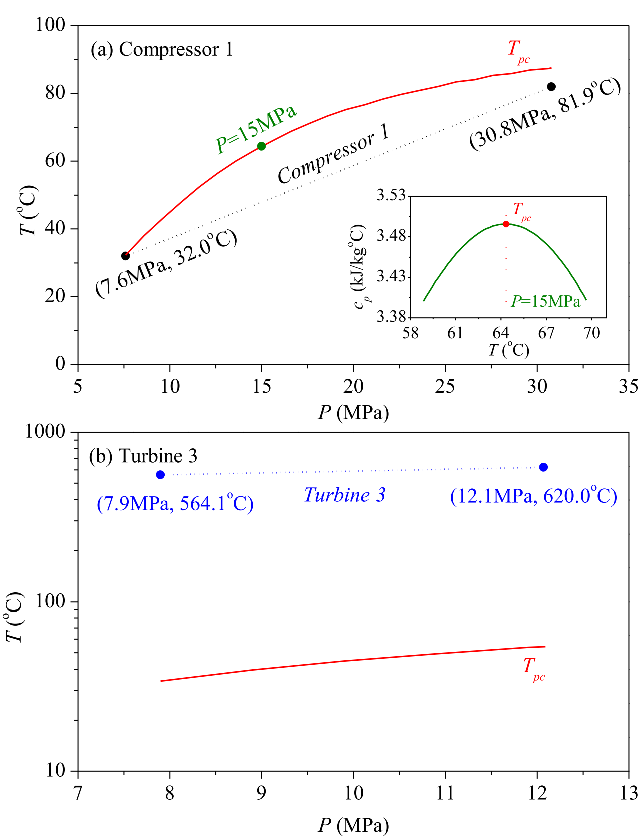

Fig. 4. Typical compression and expansion process for S-CO_2 power cycles.

USA (Sandia National Lab, Southwest Research Institute, Echogen, GE, Net Power), Japan (TIT), Korea (KIER) and China (North China Electric Power Univ.) have built several S-CO_2 loops, see Table 1 for cycle layout, parameters range, uniqueness and comments. Great attention has been paid to the bearing, sealing and rotor-dynamics stability issues. KAIST found that the ideal gas assumption introduces errors for internal loss estimation, and the external loss model should be corrected to match the measurements [51]. KIER built test loop with two types of turbo-generator with a conventional carbon mechanical seal and oil-lubricated tilting bearings, resolving bearing failure problems reported by other research groups [58]. Their tests showed that the CO_2 leakage is one of the key challenges to reach high S-CO_2 cycle performance. Recently, SwRI and GE are developing a new film-riding face seal design, restricting CO_2 leakage from supercritical pressure to near-atmospheric pressure with shaft speeds up to 3600 rpm [50,60,61]. SwRI and GE are also developing 10 MWe class turbomachines [50,60,61]. Scale-up of the Sunshot turbine beyond 50 MWe is limited by the availability of long, large diameter rotor forgings and requires a change to an assembled rotor design. A clean-sheet conceptual design of a 450 MWe assembled turbine rotor is presented [61]. It appears possible to package this reheat turbine rotor into a single casing at 3600 rpm and still maintain rotor-dynamic stability.

Conclusions and lessons are summarized as follows (see Table 1): (i) Successful demonstration of S-CO_2 cycles has been achieved, most of them are for small-scale (10 kWe−1 MWe) radial flow turbomachines. (ii) Small-scale test loops have lower efficiencies, sometimes the output parameters are lower than the designed values. (iii) The CO_2 leakage is serious to deteriorate the system performance. (iv) Problems related to small-scale radial flow turbomachines may not occur for large-scale axial flow turbomachines. The research directions are recommended as: (i) Real gas effect shall be incorporated into numerical models to improve the design accuracy. (ii) Technic issues of bearing, sealing and rotodynamic stability shall be thoroughly solved. (iii) A package design of turbine-rotor is to be expected, especially for axial flow turbomachines. (iv) The maximum isentropic efficiencies of compressors and turbines shall be presented based on reliable numerical/experimental works.

3.5. Off-design and transient operation of S-CO_2 cycle

Off-design operation is defined by a change in boundary conditions of heat exchangers, affecting the mass flow rate and temperature on one or both sides of heat exchangers. Various heat exchangers are subjected to mass flow rate variations or temperature difference changes across the two sides of heat exchangers, thus to change the heat transfer rate and pressure/temperature parameters across the cycle. The pressure ratio in compressor and turbine is then changed to yield power generation and cycle efficiency deviating from the on-design values. Compressor and recuperator heat exchanger should be paid more attention due to the strong real gas effect. The off-design operation even switches a supercritical cycle to transcritical cycle operation [62]. For nuclear reactor or fossil energy fired power plant, the off-design shall consider the varied heat sink temperature. Floyd et al. [63] showed

Table 1
The test loops for S-CO_2 power cycles.

Team	cycle layout	design conditions	Contribution and uniqueness	experience and comments
SwRI, GE, Thar, Bechtel, KAPL, Aramco, EPRI (USA) [53]	simple recuperated cycle, axial flow turbine	Capacity: ~1 MWe; Turbine: 715 °C/25 MPa/ 21,000 rpm; Flow rate: 8.3 kg/s	characterize the mechanical and aerodynamic performance of turbine and compact recuperator; the first MWe-scale S-CO_2 cycle demonstration at 700 °C; will develop and fabricate a 5-MWt recuperator heat exchanger.	The turbine has achieved its initial test at 550 °C/ 18MPa/21,000 rpm. An external heater provides high temperature CO_2 and a pump provides high pressure CO_2.
Sandia National Lab (USA) [54,55]	recompression cycle; 2 turbo-alternator-compressors (TAC)	Capacity: 250 kWe; Turbine: 715 °C/ 13.5MPa/75000 rpm; Flow rate: 5.7 kg/s; Pressure ratio: 1.75	Investigate key issues for this power cycle and confirm model estimates of system performance; one of the first S-CO2 power-producing cycles operating; the first S-CO_2 recompression cycle; provide insights and rich data about the operating principles of closed cycles.	the rated speed 75,000 rpm had not been reached; the bearing and seal problems may not occur in large-scale units.
KAPL, BAPL (USA) [56]	simple recuperated cycle; 1 TAC and 1 turbine	Capacity: 100 kWe; Turbine: 299 °C/ 16.5MPa/75000 rpm; Pressure ratio: 1.8; Cycle efficiency: 12.5%	demonstrate operating, control and performance of an S-CO_2 cycle over a wide range of conditions; explore basic laws of control and operation.	The analysis and conclusions are dependent on small turbomachinery machines, which may not apply to larger scale applications.
Echogen (USA) [15]	simple recuperated cycle; turbo-generator and turbo-pump	Capacity: ~7 MWe; Turbine: <550 °C; Turbine-generator: 30,000 rpm; Turbo-pump: <24,000–36,000 rpm; Turbo-pump: 2.7 MW	develop the first MW-scale, commercial-scale S-CO2 cycle for waste heat recovery.	2.4 MWe power was generated.
TIT (Japan) [57]	simple regenerated cycle; 1 TAC	Capacity: 10 kW; Turbine: 277 °C/ 11.9MPa/69,000 rpm; Pressure ratio: 1.45; Flow rate: 1.2 kg/s	one of the earliest S-CO_2 cycle for power generation; providing date and experience.	net power output 110 W, much less than the theoretical developed value, maybe due to the windage losses in the rotor, compressor inlet pressure is 8.23 MPa.
KIER (Korea) [58]	transcritical cycle; axial-type turbine –generator and a liquid pump	Capacity: 25.7 kW; Turbine: 200 °C/13.5 MPa/45000 rpm; Pump inlet pressure: 5.7 MPa; Flow rate: 1.6 kg/s	develop an S-CO_2 power generation system with an axial-type turbine resolving bearing failure and other problems of radial-type turbine; because axial-type turbines, oil lubricated bearings and mechanical seal technologies can be applied to MW-scale turbo-generator.	a 11 kW of electric power was obtained under 205 °C and 100 bar turbine inlet conditions; high rotational speed may not apply to large scale applications.
KIER (Korea) [58]	recuperated transcritical cycle; radial-type turbine –generator and a liquid pump	Capacity: 2~5 kW; Turbine: 500 °C/ 13.0 MPa; Pump inlet pressure: 5.73 MPa; Flow rate: 0.07 kg/s	experience high temperature turbomachinery and the cycle characteristics.	A 287W of electric power and maximum of 401 °C and 112 bar turbine inlet conditions were reached; early stage of system operation and hard to maintain stable operation due to leakage and make-up balance.
KAERI, KAIST POSTECH (Korea) [45,59]	recuperated cycle; 1-TAC, 1-turbine, 1-compressor	Capacity: 300 kW; Turbine: 500 °C/14 –20 MPa; Pressure ratio: 2.67/1.8; Flow rate: 6.4 kg/s	accumulate operating experience, understand fundamental physical phenomena and establish the control logic for S-CO_2 cycle.	Ongoing project, now is in Phase 2
Net Power (USA) [12,13]	Allam cycle natural gas demonstration power plant	Capacity: ~25MWe	prove out operation of the cycle and validate performance, control methodology, operational targets and component durability; the first S-CO_2 cycle demonstration power plant to promote technology development.	Ongoing project
North China Electric Power University (China) [37]	A forced convection S-CO_2 experimental setup	Capacity: 120 kW; Pressure: 26 MPa; Temperature: 550 °C; Flow rate: 0.2 kg/s	Focus on the investigation on S-CO_2 heat transfer under various conditions such as uniform/non-uniform heating, various flow channels and inclination angles, expanding the data range for S-CO_2 heat transfer.	Based on the expanded database of S-CO_2 heat transfer, new theory is being developed regarding the heat transfer deterioration.

Note: SwRI: Southwest Research Institute, USA; Thar: Thar Energy LLC; KAPL: Knolls Atomic Power Lab; Aramco: Aramco Services Co.; EPRI: Electric Power Research Institute, USA; GE: General Electric Company; BAPL: Bettis Atomic Power Laboratory; KIER: Korea Institute of Energy Research; KAIST: Korea Advanced Institute of Science and Technology; POSTECH: Pohang University of Science and Technology; TIT: Tokyo Institute of Technology.

the deteriorated power generation and efficiency when heat sink temperatures are increased from 21 °C to 40 °C, which are caused by a fall of pressure ratio as a result of the non-ideal gas effect. To achieve a constant thermal power and efficiency at the elevated heat sink temperature, a degree-of-freedom in compressor is necessary. For solar powered S-CO_2 cycle, both variations of solar irradiation heat flux and heat sink temperature should be considered. These two factors shall be coupled to analyze the off-design performance.

Perspective off-design studies are: (i) The model development including component and system are being developed by many groups [62–66]. Reliable models reflecting true phenomenon such as real gas effect is expected for S-CO_2 cycle driven by different heat sources. (ii) The available experimental data for off-design test comes from Argonne and Sandia National Laboratories (USA) [62–65]. Expanded test database is necessary to verify the correctness of the developed models.

The objective of the transient analysis and control for S-CO_2 cycles is to ensure the safety operation of various components and supercritical pressure operation under inevitable disturbance. S-CO_2 cycle has faster response to external variations compared with water-steam Rankine cycle. The Plant Dynamics Code (PDC) has been developed at ANL (USA) for system level transient analysis. By comparing with small scale (100 kWe) S-CO_2 cycle experiment, the code is verified to be success [67]. The transient analysis and control for Generation IV nuclear reactor driven S-CO_2 cycle were reported in Refs. [59,67,68]. Two control strategies of CO_2 mass inventory control (IC) and flexible recompression control (FRC) are proposed to stabilize the turbine inlet temperature when the net solar power drops below the designed value [69]. Flexible temperature mode (FTM) and constant temperature mode (CTM) were proposed to adapt the solar S-CO_2 system with the variations of solar irradiation flux and heat sink temperature. Both schemes can deal with the fluctuations of system [70]. The strategy consisting of four consecutive phases was proposed to bring the solar S-CO_2 cycle from cold state to the full load condition, which is regarded as a smart start-up scheme. Due to the wide range of heat source temperature, fossil energy driven S-CO_2 cycle may have combined cycle configuration, the transient analysis and control are expected to be done in the near future. In summary, due to many technical barriers, the S-CO_2 cycle is still in R&D phase and has never been deployed or tested, even though Sandia National Laboratory has proposed to deliver the first 10 MWe S-CO_2 demonstration plant in 2019.

3.6. Comparison of S-CO_2 cycle with water-steam rankine cycle

High expectation has been paid on S-CO_2 cycles regarding the efficiency improvement and flexible operation mode. For nuclear or solar energy utilization, the recompression cycle is sufficient, having higher cycle efficiency than water-steam Rankine cycle when the turbine inlet temperature is above 550 °C [4]. Recent progress has been made on fossil energy utilizations. The coal fired S-CO_2 cycle theoretically offers interesting performances, of 47.8%-LHV (low heating value) efficiency, with existing materials at current operating conditions in a relatively near timeframe [71]. The oxy-combustion indirect S-CO_2 cycle is assessed by comparing with water-steam Rankine cycle [10]. It is shown that S-CO_2 cycles are 2–4% points more efficient than comparable steam-Rankine cycle, but do not show a cost advantage either in capital costs or LCOE (levelized cost of electricity). The promising efficiency improvement by using S-CO_2 cycle for fossil energy is also reported in Refs. [30,72].

In our opinion, the benefit of S-CO_2 cycle is over-estimated due to following reasons: (i) The oxygen production and CCS (carbon capture and storage) are energy-consuming processes. When these processes are integrated with S-CO_2 cycle, the whole system efficiency should be carefully evaluated with reliable data supporting. (ii) Water-steam Rankine cycle system is mature after long term evolution. The benefit of S-CO_2 cycle can only be verified by long term commercial operation of the system. Currently, S-CO_2 cycle for large-scale utilization is in the infancy R&D stage. The life cycle assessment should be performed for S-CO_2 cycles driven by various heat sources including nuclear, solar, waste heat or fossil energy. A thorough comparison between S-CO_2 cycles and mature water-steam Rankine cycle is recommended, which is useful to evaluate if the S–CO_2 cycle can replace, or partially replace, the water-steam Rankine cycle.

4. Conclusions

S-CO_2 cycle has the potential to be applied in various situations. Currently, the development of S-CO_2 cycle faces three problems: (1) the lack of system design and analysis methodology for steady/transient operations; (2) the S-CO_2 energy transfer and conversion mechanisms are not well understood; (3) there exist bottlenecks such as seal, leakage and roto-dynamics stability of key components. Perspectives to solve these issues are.

(1) S-CO_2 cycle adapting to the distinct characteristic of a heat source should be optimized to promote the global system efficiency.
(2) Numerical/experiment investigations should be expanded to improve the design accuracy of key components.
(3) Package solutions shall be proposed to solve the technic barriers for key components, which are to be verified in practical system for reliable and efficient operation.

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