Review Paper

Review of supercritical carbon dioxide (sCO_2) technologies for high-grade waste heat to power conversion

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Abstract

In the European Industry, 275 TWh of thermal energy is rejected into the environment at temperatures beyond 300 °C. To recover some of this wasted energy, bottoming thermodynamic cycles using supercritical carbon dioxide (sCO_2) as working fluid are a promising technology for the conversion of the waste heat into power. CO_2 is a non-flammable and thermally stable compound, and due to its favourable thermo-physical properties in the supercritical state, can lead to high cycle efficiencies and a substantial reduction in size compared to alternative heat to power conversion technologies. In this work, a brief overview of the sCO_2 power cycle technology is presented. The main concepts behind this technology are highlighted, including key technological challenges with the major components such as turbomachinery and heat exchangers. The discussion focuses on heat to power conversion applications and benefits of the experience gained from the design and construction of a 50 kWe sCO_2 test facility at Brunel University London. A comparison between sCO_2 power cycles and conventional heat to power conversion systems is also provided. In particular, the operating ranges of sCO_2 and other heat to power systems are reported as a function of the waste heat source temperature and available thermal power. The resulting map provides insights for the preliminary selection of the most suitable heat to power conversion technology for a given industrial waste heat stream.

Keywords Supercritical CO_2 power cycle · Waste heat recovery · High temperature heat to power conversion · Heat exchangers · High temperature and pressure materials · Heat exchangers · Turbomachinery

Abbreviations

BMPC	Bechtel Marine Propulsion Corporation
BUL	Brunel University London
CVR	Research Centre Rez
ENO	Enogia
FPHE	Formed Plate Heat Exchanger
GE	General electric
HT	High temperature
IAE	Institute for the Atomic Energy
KAERI	Korean Atomic Energy Research Institute
KAPL	Knolls Atomic Power Laboratories
KIER	Korean Institute for Energy Research
LT	Low temperature
MT	Medium temperature
ORC	Organic Rankine Cycle
PCHE	Printed Circuit Heat Exchanger
PHE	Plate Heat Exchanger
PMHE	Plate Mesh Heat Exchanger
sCO_2	Supercritical carbon dioxide
sCO_2 HeRo	Supercritical CO_2 heat removal system
SNL	Sandia National Laboratories
SWRI	South West Research Institute
TAC	Turbine Alternator Compressor
TFC	Trilateral Flash Cycle
TIT	Tokyo Institute of Technology
WHR	Waste heat recovery
WMHE	Wire-mesh heat exchanger

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1 Introduction

The more stringent national and international regulations on greenhouse gas emissions as well as the increasing environmental concerns are driving academia and industry to seek new sustainable solutions to meet the growing energy demand. Apart from enhancing a higher penetration of renewables in the energy mix, the energy efficiency improvement of existing industrial and power generation facilities is considered as essential to achieve the aforementioned targets.

One of the approaches to lower the carbon footprint of the current industrial sector is represented by the recovery and re-utilization of the heat lost after combustion and heat transfer processes, whose relevance has been estimated in the 63% of the global primary energy consumption in industry [1]. This amount of waste thermal energy includes the heat rejected into the environment as effluents and exhausts. On the other hand, the thermal energy dissipated in irreversible processes such as friction losses, electrical resistance and transmission is unlikely to be recoverable.

The exploitation of potential is hence crucial and, depending on the characteristics of the waste heat source, it involves technical challenges. For instance, different technologies must be considered if the waste heat source is available at low (< 100 °C), medium (100–300 °C) or high temperature levels (> 300 °C) [2]. Other factors to take into account are the temporal availability of the waste source, the composition of the heat carrier, the intensity or modality of supply but also the economic and financial feasibility of the retrofit and the utilization of the recovered energy [2].

In particular, two main approaches can be pursued, the direct re-use of the waste heat recovered or its conversion into electric power. Unlike the direct use of the recovered heat that requires a heat demand in the industrial site or in the nearby ones, an electrical energy recovery is more favorable in terms of energy management, since the surplus of electricity can be dumped to the electrical grid [3]. In addition, electricity is considered more valuable from an economic perspective [2].

The conversion of waste heat into electrical power has been conventionally addressed in the last decades through power units based on steam or Organic Rankine cycles (ORC). These technologies are similar in the underpinning thermodynamics but present significant technical and economic differences at applied level. In general terms, ORC units are more suitable at small-scale, i.e. tens or hundreds of electrical kilowatts, while steam power cycles are more suitable at megawatt scale [4]. Both technologies however show some shortcomings and limitations in terms of temperature and efficiency which are addressed in this paper. Supercritical carbon dioxide (sCO_2) power cycles represent a promising solution thanks to their compactness, high efficiency and operational flexibility [5].

Supercritical CO_2 power systems have been investigated and reviewed for nuclear, concentrated solar and advanced power generation applications [6–8]. In this work, instead, an overview on sCO_2 power technology is presented, with a particular focus on the potential use of these systems for waste heat to power conversion applications. Firstly, a map comparing the different operating ranges of the available heat to power conversion technologies as a function of the main parameters identifying the waste heat source is showed in Sect. 2. In this analysis, the operating range of sCO_2 power technology is specified, showing that these systems can represent a breakthrough for the sector. Afterwards, the main concept behind the technology is presented in Sect. 3 and an up to date review of progresses as well as the technical challenges in the main component's development is detailed in Sect. 4. This review includes also novel information on the design and selection of heat exchangers and materials acquired by the authors during the construction of a 50 kW sCO_2 testing facility for waste heat recovery (WHR) applications at Brunel University London [9].

2 Heat to power conversion applications

Originally studied for the nuclear sector because of the lower chemical reactivity of CO_2 with molten sodium [10] (most promising candidate as primary heat carrier in next generation nuclear reactors), sCO_2 systems have been also investigated for fossil fueled power plants, because of their high operational flexibility and ease of implementation of carbon capture and storage systems [11–13].

Supercritical CO_2 power cycles are also being considered in the renewable energy sector, particularly for Concentrated Solar Power (CSP) and geothermal applications. In the latter case, the use of CO_2 allows the exploitation of geothermal sources at a higher depth and temperature levels, increasing the maximum efficiency and power output obtainable with more conventional technologies (as for instance Organic Rankine Cycle systems) [14]. In the CSP sector, the benefits are instead more related to the consistent reduction of the production cost of electricity thanks to the possibility of directly recovering the solar radiation without using any intermediate energy carrier, although this poses challenges in the design of high pressure solar collectors and receivers, as well as energy storage systems [7, 15, 16].

2.1 High-grade waste heat potential

Equally promising can be the use of the technology for waste heat recovery (WHR) applications. As showed by Fig. 1, the potential energy recovery only in the European industrial sector, which accounts of the 25.9% of the primary energy consumptions (Fig. 1a), has been estimated to be around 918 TWh (Fig. 1c). This potential energy waste is rejected into the environment through exhausts and effluents and it is widely distributed across the main European sites, being supplied at low (< 100 °C), medium (100–270 °C), and high (> 270 °C) temperature levels.

As it is shown in Fig. 1c, the potential heat recovery from the low temperature waste heat sources represents the higher share (51%), However if the Carnot efficiency is considered, the high temperature waste heat sources give the higher contribution, which is equal to almost 154 TWh (55% of the total WHR potential as showed in Fig. 1d). The lack of available technologies for recovery and power conversion of high temperature waste heat makes sCO_2 power cycles a promising candidate to harvest this energy waste at a competitive levelized cost of electricity.

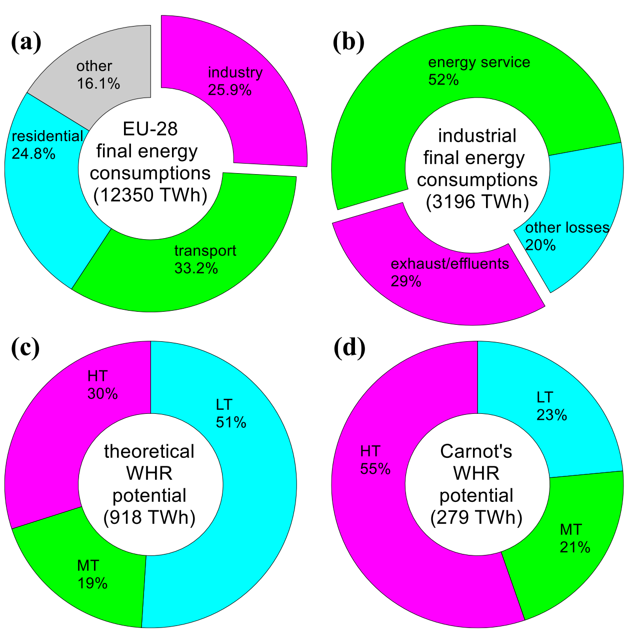

Fig. 1 Theoretical and Carnot's waste heat recovery potentials in EU28 industrial sector: overall EU28 energy consumptions (**a**), overall industrial consumptions (**b**), theoretical waste heat recovery potential divided by temperature levels (**c**) and Carnot's WHR potential (**d**) [3]

2.2 Benchmark of supercritical CO_2 heat to power cycles

Figure 2 displays the operating ranges of conventional and innovative waste heat to power conversion technologies as a function of the heat source temperature level and capacity, i.e. including type and mass flow rate of the waste exhaust or effluent. Although Organic Rankine Cycle (ORC) systems proved to be a successful technical solution especially for large scale applications [17], the use of this technology is limited in a range of temperatures of the waste heat source that goes from 100 °C up to 400 °C [18]. The upper limit is imposed by the flammability and low chemical stability of the organic fluids at high temperatures, while the lower one by their vapour pressure which, in turn, limits the efficiency and the output of ORC units at extremely low temperatures.

For waste source capacities from 10 kW up to 200 kW, ORC systems equipped with positive displacement machines are preferred to the ones with axial or centrifugal turbines [19] (Fig. 2). In fact, in this power range, volumetric machines can achieve higher efficiencies compared to dynamic ones, whose reduced size leads to losses. Furthermore, positive displacement machines benefit from a reduced installation and maintenance complexity due to lower revolution speeds, reduced vibration levels and wider range of optimal operating conditions [20].

For power scales between 200 kW and 15 MW (the larger ORC installation to date [21]), turbines are instead adopted since their size can be increased with consequent benefits in efficiency [22].

Steam Rankine cycles are usually preferred to ORC to exploit waste heat sources with higher thermal capacities (from 10 MW up to hundreds of MW) [23], because of the

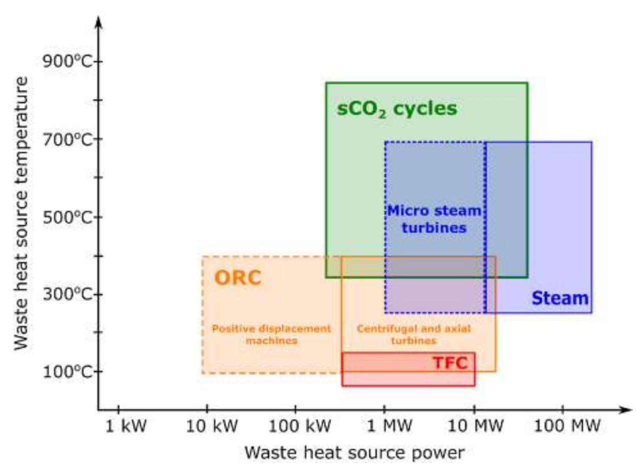

Fig. 2 Comparison of different operating range of heat to power conversion technologies based on bottoming thermodynamic cycles for WHR applications

higher efficiency and the lower capital cost due to more standard components [23]. The operating range of this technology can be further extended to power scales lower than 10 MW using micro steam turbines (Fig. 2) which are however characterised by lower performance than large machines due to high tip leakage losses [23].

The temperature range at which the Steam Rankine technology is usually employed goes from 250 °C up to 700 °C (Fig. 2) [23, 24]. The lower limit is given by the low vapour pressure of water, while the upper one from material and technological constraints. More advanced units, as the ultra-supercritical steam power systems, can also exploit heat sources beyond 620 °C, but they require significant additional investment costs [23].

Waste heat source available at temperature levels lower than 100 °C can be still exploited by adopting the Trilateral Flash Cycle (TFC) technology (Fig. 2). In these kind of systems, the organic working fluid is heateduntil the saturated liquid conditions) and undergoes to a two-phase expansion [25]. For these reasons, these units are suitable for ultra-low temperature WHR applications, from 200 °C down to 70 °C

Furthermore, because of the two-phase expansion, volumetric machines are usually adopted since they guarantee a higher adiabatic efficiency. The size of these machines, however, limits the maximum thermal capacity of the waste heat source exploitable, which can go up to 5 MW [26, 27]. For capacities lower than 1 MW, ORC systems are more competitive [26].

Hence, it is possible to notice that sCO_2 systems fill an important gap in industrial WHR applications (Fig. 2). For waste heat source temperatures higher than 700 °C, sCO_2 power cycles are the only option available and can thus constitute a breakthrough in the sector (Fig. 2). The high chemical stability of CO_2 allows to directly recover and convert heat at temperatures up to 850 °C (Fig. 2), which is a limit posed by current materials [28]. The lower limit is instead set at 350 °C considering a simple regenerated layout, which represents the most convenient option for WHR applications [29]. For such low cost systems, characterized by a low cycle pressure ratio, the achievement of higher temperatures at the turbine inlet to obtain a positive net electric output is required [29].

From a power scale perspective, several technological challenges and the high investment costs set the lowest feasible unit capacity at 50 kWe [9], which correspond to a waste source thermal power of 300 kW assuming a 20% system thermal efficiency [29]. Among the technical limitations, the main ones arise from the reduced size of the turbomachines. Typical wheels diameters range from 30 mm to 50 mm, with consequent issues of leakage, high vibrations level and friction due to the elevated revolution speeds (over 60,000 RPM) [9, 30]. On the other side, it is possible to scale up sCO_2 systems until tens of MW (Fig. 2), as it has been proved by the SunShot program and Echogen units [31, 32]. In this case, rather than on the turbomachinery side, which can benefit from the knowledge acquired in the gas turbine and steam power plants sectors, technological limitations arise on the scaling up of heat exchangers.

3 sCO_2 power cycles

Supercritical CO_2 power cycles use carbon dioxide in the supercritical phase as working fluid to convert the heat received by a given thermal source into electricity. The thermodynamic cycle usually performed is the Joule–Brayton one [33], even if some references to the Rankine cycle can be found in the literature when the condensation of the CO_2 takes place during the heat rejection phase [34, 35].

The simplest configuration of a sCO_2 power cycle is the simple regenerated layout presented in Fig. 3. This system is composed of three heat exchangers and two turbomachines, namely compressor and turbine. The low, medium and high temperature heat exchangers pursue different functions and, in turn, they are commonly referred to as gas cooler, recuperator and primary heater respectively. With reference to the temperature-entropy (T–s) diagram of Fig. 3b, the CO_2 is pressurized in the compressor (1–2) and heated up first in the cold side of the recuperator (2–3) and then in the primary heater (3–4), where the actual heat recovery from the industrial topping process takes place (red line). Afterwards, the high enthalpy CO_2 flow is expanded in the turbine (line 4–5) until a pressure close to the critical one. The CO_2 temperature at the turbine outlet is further cooler down at the hot side recuperator (5–6). Finally, the initial cycle thermodynamic conditions are restored by rejecting the residual heat to a cooling source through the gas cooler (6–1).

The main advantages of this technology come from the particular properties that CO_2 assumes after the critical transition. In the supercritical state, the CO_2 presents a higher density which enables to downsize the equipment with respect to more conventional technologies as gas turbines or steam power plants [36]. The reduced size of the system components allows to reduce the investment and maintenance costs as well as the footprint of the system itself, which can be beneficial for WHR applications [37, 38]. A further benefit derives from the fluid properties near the critical point, which occurs at a temperature and pressure of 31.1 °C and 73.9 bar respectively. At these thermodynamic conditions, the carbon dioxide experiences high values for specific heat at constant pressure and isothermal compressibility [33]. These properties, together with

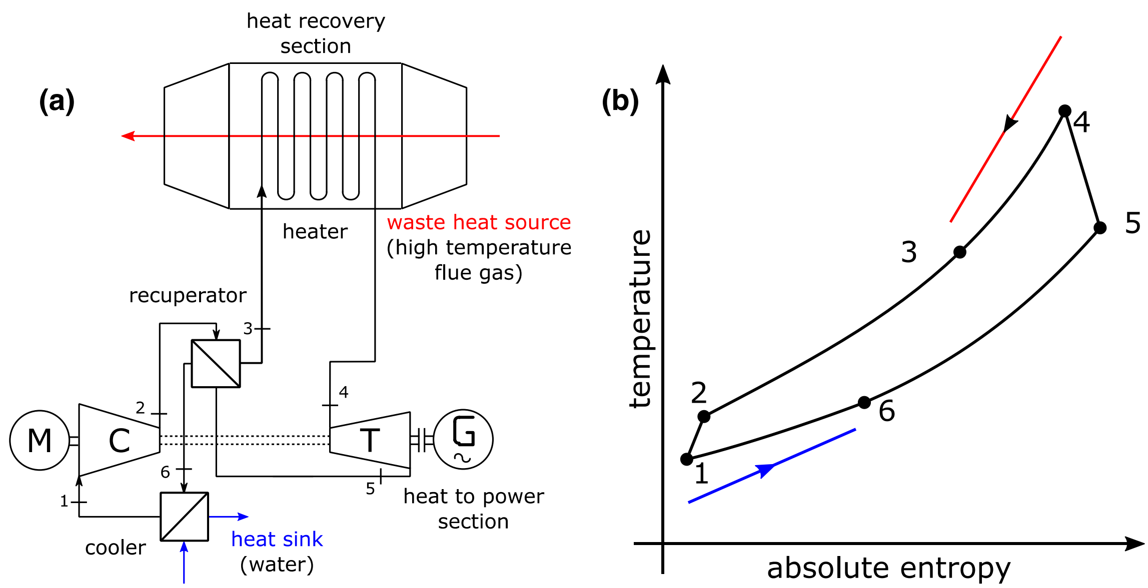

Fig. 3 Simple regenerated CO_2 Brayton cycle system: layout scheme (**a**) and absolute T-s diagram (**b**)

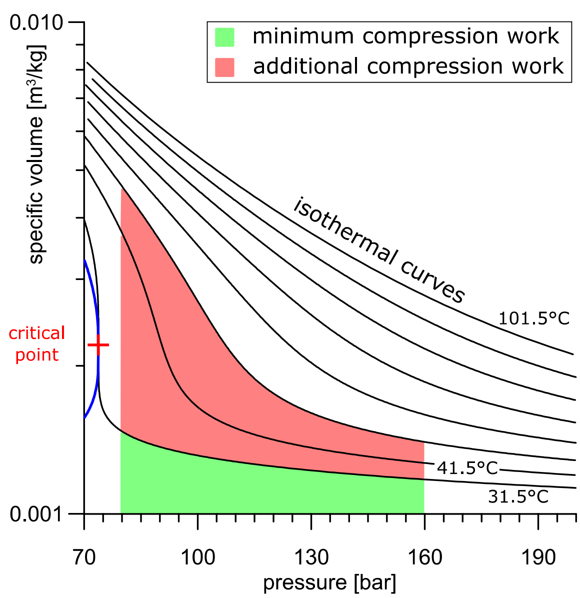

Fig. 4 P–v diagram of the CO_2 compression at different temperature levels

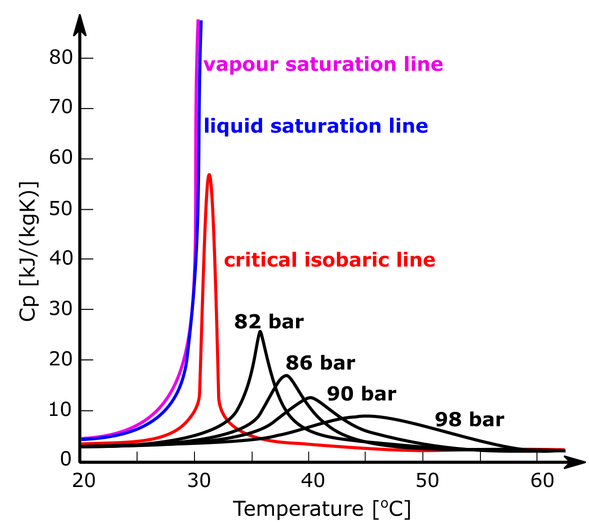

Fig. 5 Specific heat of CO_2 as a function of pressure and temperature near the critical point

a liquid-like density, ensure a reduction of the mechanical work required to pressurize the fluid, with consequent advantages in terms of cycle efficiency and power output.

The beneficial effect of a mechanical compression near the critical point is displayed in Fig. 4 through the CO_2 pressure versus specific volume diagram. Assuming an ideal isothermal compression of a unitary mass of CO_2 between two given pressure levels (from 75 bar to 160 bar), the green area refers to the work needed when the temperature of the fluid at the compressor inlet is equal to the critical one (31.5 °C), while the red one indicates the additional work required when the temperature is increased up to 41.5 °C.

On the other hand, the closer fluid is to the critical point and the more its thermo-physical properties change with temperature and pressure. As an example, Fig. 5 reports the variation at different pressure levels of the of the CO_2 specific heat at constant pressure as a function of temperature. This poses a challenge in terms of system regulation, since accurate instrumentation is required to control the compressor inlet temperature.

The CO_2 critical point also limits the maximum pressure ratio achievable in the cycle. In fact, since the minimum pressure of the cycle is fixed by the critical transition (75 bar) and the maximum one by technological constraints (typically around 250–300 bar), only a maximum pressure ratio of 4 can be achieved (which is extremely low compared to the value of 200 in Rankine steam technology [39]). To overcome these limitations, research is being carried out to develop new materials for harsh operating conditions [40] and in the field of CO_2 doping, which consists in the mixing of carbon dioxide with compounds able to lower the CO_2 critical pressure [41, 42].

The low cycle pressure ratio achieved leads to elevated temperatures at the end of the expansion and then a high level of recuperation is needed to obtain reasonable performance [43]. Even though this fact guarantees high thermal efficiency, at the same time it limits the system net power output, and hence high temperatures at the turbine inlet are needed to generate electric power (especially considering the slow divergence of the CO_2 isobaric lines). However, at these high temperature levels, the CO_2 assumes a strong corrosive behaviour, requiring the development of innovative materials to withstand at such harsh operating conditions.

4 Progress in sCO_2 power cycle technology

The high potential behind the sCO_2 technology has led to extensive academic and industrial research activities in the last decade. Despite these efforts, there are still a number of technological challenges that need to be addressed in order to advance the technology readiness level of such systems. The main ones relates to the development of turbomachines and heat exchangers (especially if considering very large or small power scales), as well as materials and manufacturing techniques. In this work, a particular focus has been given on heat exchangers and materials, which are seen as the key bottlenecks to the improvement of the sCO_2 power cycles performance and economic feasibility. In fact, literature studies estimated that nearly 80% of the investment costs in a sCO_2 power plant relates to heat transfer equipment [44].

4.1 Turbomachinery

The higher density of CO_2 allows to downsize turbines and compressors and thus to decrease the overall capital and operating expenditures of the system. However, it also poses additional technological challenges in terms of machine design [45, 46], single or multiple shaft configurations [47], bearings [44, 48–51], seals [52–55], rotor dynamics and pressure containment [44]. Many of these issues arise when small-scale systems are considered and leakage problems become relevant. For larger sizes, technical solutions are available from other technological fields such as gas turbines or ultra-supercritical steam power plants.

While this holds especially for turbines, different considerations must be made for compressors. Despite the high benefits achievable in terms of efficiency and system power output, operating the compressor close to the CO_2 critical point may cause the partial condensation of the working fluid at the inlet of the impeller [56, 57], with the consequent challenges deriving from a wet compression.

To overcome this limitation Poerner et al. in [58] proposed to adopt the gas ejection technology, widely diffused in the oil and gas wet compressors. A series of holes are placed at the blade inlet of the compressor impeller and used to eject a dry gas to break the liquid film formed on the airfoil. The result is the restored aerodynamic performance of the compressor and an improved lifetime of the component [58].

A further challenge derives from the management and control of the compressor. Since close to the critical point a slight change of pressure and temperature translates in a dramatic variation of the fluid thermo-physical properties (i.e. density, thermal conductivity or thermal capacity), very accurate sensors are needed to be able to regulate effectively the device and thus the overall system [59]. The characteristics of the first prototypes of turbines, compressors and Turbine-Alternator-Compressor (TAC) units designed by the different industrial and academic institutions worldwide are summarized in Tables 1 and 2 respectively.

4.2 Heat exchangers

Heat transfer equipment for sCO_2 applications is fundamental to enhance the efficiency and the economic viability of this technology. While a simple regenerated sCO_2 system one would require only three heat exchangers, up to five devices would be needed for more complex and performant cycle architectures [10]. These devices also represent the largest components in the system and must withstand high temperatures and pressures.

Furthermore, to avoid the excessive erosion of the already limited cycle pressure ratio, sCO_2 heat exchangers must be designed to minimize the pressure drops, which is not trivial especially considering that a trade-off exists between the minimization of pressure drops, maximization of the heat duty and reduction of costs [68]. Reducing the pressure drops across heat exchangers leads to a reduction of their wet surface, which limits their effectiveness and heat transfer capabilities. Increasing the heat duty however, allows to increase cycle

Table 1 Technical features of the first prototypes of sCO_2 turbines and compressors commissioned and operating in the different academic and industrial organisations involved in research on sCO_2 power cycles

Institution	Type	Rotational speed (RPM)	Diameter (mm)	Power (kW)	Design point (°C/bar/kg/s)	Bearings type
Turbines						
BMPC [60]	Radial	55,000	45	100	282/141/2.1	Gas foil
SWRI/GE [61]	Axial	n.a.	n.a.	1000	700/250/8.4	Tilting pad
Echogen [32]	Radial	30,000	n.a.	8000	275/n.a./n.a.	Tilting pad
KIER [62]	Axial	45,000	73	93	216/123/1.5	Tilting pad
KAIST [63]	Radial	80,000	325	n.a.	435/125/5.0	n.a.
Compressors						
KAIST [63]	Radial	35,000	272	100	33/78/6.4	n.a.

n.a. information not available

Table 2 Technical features of the first prototypes of sCO_2 Turbine-Alternator-Compressor (TAC) units commissioned and operating in the different academic and industrial partners involved in the research on sCO_2 power cycles

Institution	Rotational speed (RPM)	Mass flow rate (kg/s)	Compressor inlet (°C/bar)	Turbine inlet (°C/bar)	Net power (kW)	Bearing type	Lubrication
KIER [64]	70,000	3.1	36/79	180/130	13	Gas foil	CO_2
BUL/ENO [65]	60,000	2.1	35/75	435/127	50	Rolling	Oil
SNL [43]	75,000	3.5	33/77	477/105	150	Gas foil	CO_2
sCO_2 HeRo [66]	50,000	0.6	33/74	185/117	4	Ball	Grease
TIT [67]	69,000	1.1	31/75	260/106	0.11	Gas foil	CO_2

efficiency and power output, but requires also greater heat transfer area, thus increasing pressure drops and costs.

Additional design specifics must also be fulfilled depending on the type of heat exchanger considered. For the gas cooler, despite the lower operating pressure and temperatures, the heat duty requirements and the type of heat sink considered (air or water) can considerably affect the component design and optimization [69]. The recuperators are instead more challenging because of the higher pressures and temperatures involved. In this case also long term creep and fatigue resistance are required because of the wide range of temperatures occurring across the different heat exchanger sections and the limited operation maintenance due to the extreme compactness of these devices [44]. Even more ambitious is the design of the primary heater, which not only is the component exposed to the highest temperature and pressure of the cycle, but depending on the nature of the heat source, it may have to operate also in an extremely corrosive environment (i.e. nuclear or WHR applications). A further requirement could be the minimization of the pressure drops on both the working fluid and the heat source side, especially if exhausts are considered as heat source. Hence, considering all these aspects different technologies have been considered and investigated for sCO_2 heat exchangers.

4.2.1 Primary heater technologies

Due to the aforementioned design and operational requirements, when the heat source is in a gaseous form, shell and tube (S&T) heat exchangers are usually adopted a primary CO_2 heaters. The sCO_2 usually flows inside the tubes, while the heat source/sink flows along the shell. Plate baffles are embedded to enhance the heat transfer, but they also lead to increased pressure drops. The main drawbacks of this heat technology are the low compactness and the high heat transfer surface required to achieve an effectiveness at least higher of 0.85 for sCO_2 power cycle applications [70].

To increase the compactness of these heat exchangers and further enhance the heat transfer performance, the tube size can be reduced (tube diameter lower than 1 mm), obtaining a so-called micro-tube heat exchanger. The enhanced heat transfer coefficient obtainable thanks to the smaller tubes allows to avoid baffles and thus to decrease the pressure drops on the flue gas side, even if this leads to an increase of the pressure drops and a reduced flow velocity on the CO_2 side. Further advantages of micro-tube heat exchangers with respect to the conventional shell and tube ones are a greater scalability and modularity as well as a better resistance to harsh operating conditions; on the other hand, a drawback is represented by the higher manufacturing cost due to the special welding operations required to assemble the tubes in the headers [70]. Should the heat source be an effluent, Printed Circuit Heat Exchangers (PCHEs) also become a viable option.

4.2.2 Recuperator technologies

PCHEs and Formed Plate Heat Exchangers (FPHEs) are always preferred for sCO_2 recuperators, thanks to their extremely high heat duty per unit volume, compactness, creep and fatigue resistance as well their capability to withstand high pressure and temperatures. The reduced material shrinkage due to the additive manufacturing techniques can make FPHE cheaper than PCHE [71]. On the other hand, PCHE can be operated at higher pressures (up to 1000 bar) compared to the 250 bar achievable by the FPHE technology [72].

To further reduce the cost of these components, industrial and academic organizations are investigating new technical solutions. Among the different ongoing alternatives, the most promising heat exchanger technologies are the Plate-Matrix and Wire-Mesh Heat Exchangers (PMHE and WMHE) [72, 73], which guarantee even higher compactness, heat duty and lower costs due to a lower material use.

4.2.3 Gas coolers technologies

The shell and tube technology can be adopted as gas coolers if air is used as cooling medium [43, 70]. If a liquid coolant is considered, PCHE provide a compact yet pricey technological solution. At small power scales or when the footprint is not a major issue for the sCO_2 system, components available from the CO_2 refrigeration sector, such as Plate Heat Exchangers (PHE), can be used instead of PCHE with an extreme reduction of capital and operational costs [9]. Table 3 summarizes the heat exchanger technologies available for each of the heat exchanger typology in sCO_2 power cycles as well as the average cost per kW/K of the devices.

4.3 Materials

SCO_2 systems performance strongly depend on the maximum temperature achieved in the cycle. Hence, the development of suitable materials is crucial for the establishment of the technology. The challenge is even emphasized by the high cycle pressures and the strong corrosive behavior that CO_2 shows at temperatures higher than 500 °C. Among the several forms of corrosion, the main relevant mechanisms that can occur in a CO_2 environment are high temperature oxidation, carburization and metal dusting [76].

While high temperature oxidation is common also for more conventional working fluids (i.e. steam or air), carburization and metal dusting assume a particular relevance for CO_2 applications. Both these corrosion mechanisms combined with the high temperature oxidation phenomena can lead to catastrophic failure of the system components [77]. In particular, the carburization reaction leads to the depletion of ferrous ions present in the alloys which, reacting with CO_2, form carbon atoms [78]. The carbon atoms increase the rate of interstitial carbides formation along the grain boundaries of the alloy, reducing its strength and creep resistance.

The formation of these interstitial carbides also leads to the depletion of all that elements which prevent the oxidation of the alloy (i.e. chromium), so accentuating the high oxidation temperature corrosion. When metal dusting occurs, i.e. a more widespread carburization on the alloy surface, the size of the interstitial carbides increases exponentially until the carbide zone becomes super-saturated and graphite nucleation takes place. As this nucleation begins, the graphite nuclei starts to grow and break the oxide layer previously formed [76].

To overcome these issues, usually nickel, vanadium and titanium are used as alloying elements. Nickel, thanks to its poor solubility with carbon, offers good resistance to carburization since it mitigates the carbon migration towards the alloy surface. Vanadium and Titanium, on the contrary, rapidly react with carbon because of their strong chemical affinity with the compound and prevent the formation of chromium carbides. For these reasons, for the components of the sCO_2 system operating at high temperature

Table 3 Cost per UA unit ($/(kW/K)) of the different heat exchangers used in sCO_2 power cycles (gas cooler, recuperator and primary heater) grouped by technology (green colour when the heat source/sink is in gaseous state and light blue when it is in liquid form)

Heat exchangers	S&T	Micro-tube	PCHE	FPHE	PHE	PMHE	WMHE
Gas cooler	1700 [70]		>2500 [74]	2000 [44]	50 [74]		
Recuperator			>2500 [74]	>2000 [44]		FOAK [75]	FOAK [75]
Primary heater	>5000 [44]	>5000 [9]	>5000 [44]				

FOAK First of a kind component

(i.e. turbine and primary heater) the most suitable material candidates are nickel rich alloys [79, 80], while stainless steel can be used for components withstanding lower temperatures (i.e. recuperators operating below 500 °C) [40].

For components subjected to less harsh operating conditions (i.e. compressor and gas cooler) aluminum alloys could also be adopted [31]. Table 4 recalls the main alloys investigated and tested for sCO_2 applications classified by to their maximum operating temperature and the most suitable component.

4.4 Integral testing facilities

To prove the concept of sCO_2 power cycles, different industrial and academic research institutes developed prototypes of sCO_2 heat to power conversion systems. Sandia National Laboratories (SNL) and the Knolls Atomic Power Laboratories (KAPL) were the first research centers to develop two integral sCO_2 power cycle test facilities for nuclear applications. Afterwards, many institutes started their own research activities, both in the United States (e.g. Argonne National Laboratories, ANL, South West Research Institute, SWRI, Bechtel Marine Propulsion Laboratories, BMPL) and in Asia, especially in Japan (Tokyo Institute of Technology, TIT) and South Korea (Korea Advanced Institute of Science and Technology, KAIST, and Korea Institute of Energy Research, KIER). In the latest years, European institutions also developed laboratory scale facilities. Table 5 summarizes the experimental test rigs currently commissioned and their main characteristics.

Table 4 Tested materials for sCO_2 power cycles applications at a pressure of 200 bar for a minimum duration time of 3000 h (LT = Low Temperature, HT = High Temperature)

Temperature	Component	Alloy	Type
T ≤ 250 °C	Compressor, gas cooler	304ss, P91, T22 [31]	Low cost austenitic or ferritic alloys
T ≤ 400 °C	LT recuperator	347ss [31]; 310ss and 316ss [80]	Austenitic alloys recommended
T ≤ 550 °C	HT recuperator, LT primary heater, LT turbine	347ss [31]; 310ss [81]; 316L [76].	Austenitic steels with a lower level of Ni, Cr and Co (316)
T ≤ 650 °C	HT turbine, HT primary heater	Haynes 230 [80]; IN-617 [81]; 800H [78].	Higher Ni/Cr alloys are recommended
T > 650 °C	Very high temperature applications	Haynes 282 [31]; IN-713 [82]; IN-718 and IN-738 [40]; IN-690, IN-693, IN-725 and IN740 [79]; EP823 [83]	Little testing completed.

Table 5 Integral sCO_2 heat to power conversion testing facilities

Institution	Cycle layout	Net power (kW)	Turbomachinery	Heat source	Heat source capacity (kW)
United States					
Echogen [32]	Pre-heating/split expansion	8000	1 compressor 2 turbines	Flue gas	33,000
SNL [43]	Recompression	300	2 TAC	Electric	780
KAPL [84]	Simple regenerated	100	1 TAC 1 turbine	Electric	835
South Korea and Japan					
KAIST/KAERI [63]	Simple regenerated	300	1 TAC 1 turbine 1 compressor	Electric	1300
KIER [85]	Simple regenerated	80	1 compressor 2 turbines	Flue gas	611
TIT/IAE [67]	Simple regenerated	10	1 TAC	Electric	160
European union					
sCO_2 HeRo [86]	Simple regenerated	9	1 TAC	Steam/electric	6/195
BUL [9]	Simple regenerated	50	1 TAC	Flue gas	700
CVR [87]	Simple regenerated	n.a.	1 pump 1 expansion valve	Electric	110

n.a. information not available

5 Conclusions and future research

In this work, a review of sCO_2 power cycle technology has been presented. The use of CO_2 as working fluid in heat to power conversion systems can provide several benefits in terms of efficiency, compactness and operational flexibility compared with other more conventional technologies. Despite the extensive research carried out in the field, the technology readiness level is still quite low.

Additional research and development is required for the scaling up of compact high temperature and pressure heat exchangers at reasonable cost, to reduce leakage and mechanical losses in turbomachines, and to ensure optimal regulation and safe operating conditions of the compressor close to the CO_2 critical point. Additional research is also needed to improve the properties and reduce the cost of materials for operation at temperatures beyond 650 °C in order to enhance the overall system performance and operating lifetime.

Research programs are being carried out to evaluate the performance of sCO_2 power cycles in different applications such as geothermal, concentrated solar power and nuclear power applications. Another promising application is high temperature waste heat recovery to power conversion, in particular for waste heat temperatures above 400 °C and thermal capacities in the range between 300 kW and 30 MW. At high waste heat temperatures, Organic Rankine Cycle systems have limitations due to the poor thermal stability of organic fluids and steam Rankine technology has lower efficiencies than sCO_2 cycles. Further research can extend both the range of operation and power output of sCO_2 systems.

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Compliance with ethical standards

Conflict of interest On behalf of all authors, the corresponding author states that there is no conflict of interest.

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