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Review

Supercritical Carbon Dioxide(s-CO_2) Power Cycle for Waste Heat Recovery: A Review from Thermodynamic Perspective

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Abstract: Supercritical CO_2 power cycles have been deeply investigated in recent years. However, their potential in waste heat recovery is still largely unexplored. This paper presents a critical review of engineering background, technical challenges, and current advances of the s-CO_2 cycle for waste heat recovery. Firstly, common barriers for the further promotion of waste heat recovery technology are discussed. Afterwards, the technical advantages of the s-CO_2 cycle in solving the abovementioned problems are outlined by comparing several state-of-the-art thermodynamic cycles. On this basis, current research results in this field are reviewed for three main applications, namely the fuel cell, internal combustion engine, and gas turbine. For low temperature applications, the transcritical CO_2 cycles can compete with other existing technologies, while supercritical CO_2 cycles are more attractive for medium- and high temperature sources to replace steam Rankine cycles. Moreover, simple and regenerative configurations are more suitable for transcritical cycles, whereas various complex configurations have advantages for medium- and high temperature heat sources to form cogeneration system. Finally, from the viewpoints of in-depth research and engineering applications, several future development directions are put forward. This review hopes to promote the development of s-CO_2 cycles for waste heat recovery.

Keywords: supercritical carbon dioxide power cycle; waste heat recovery; thermodynamic cycle

1. Introduction

Efficient conversion and utilization of energy is an effective pathway to address energy shortage and environmental problems faced by mankind. However, several losses occur during the energy conversion process, starting from the primary energy carrier to the end-user, mainly in the form of waste heat. The waste heat discharged to the environment has high exergy value and also contains large quantities of pollutants, thus, contributing to global warming. The recovery of this waste heat for targeted use can significantly raise the process efficiency and reduce the primary energy consumption [1,2]. Such an energy-saving potential is particularly significant for industrial processes, thermal engines, and mechanical devices [3–5]. Take China as an example, the annual energy consumption of the industrial sector accounts for more than 70% of the total national energy consumption, of which at least 50% is converted to waste heat at different temperatures and with different carriers. The estimated waste heat in the cement, iron/steel, and glass industries in China, in 2016, were, 41.0 GWth, 2.9 GWth, and 1.8 GWth, respectively [3].

Generally speaking, there are many kinds and forms of waste heat resources. According to the investigation by Galanis et al. [6], waste energy is released from industrial processes to the environment via four main states of matter; namely, liquid streams at temperatures between 50 °C and 300 °C, exhaust at temperatures ranging between 150 °C to 800 °C, steam at temperatures ranging from 100 °C to 250 °C, as well as the process gases and vapors within a temperature range of 80 °C to 500 °C. Furthermore, low-temperature (<350 °C) waste heat accounts for the majority in most end-use energy sectors. As shown in Figure 1, evaluations reveal that 63% of the waste heat streams arise at temperatures below 100 °C, and the largest proportion of which is produced by the electricity sector, followed by the commercial and transportation sector [1].

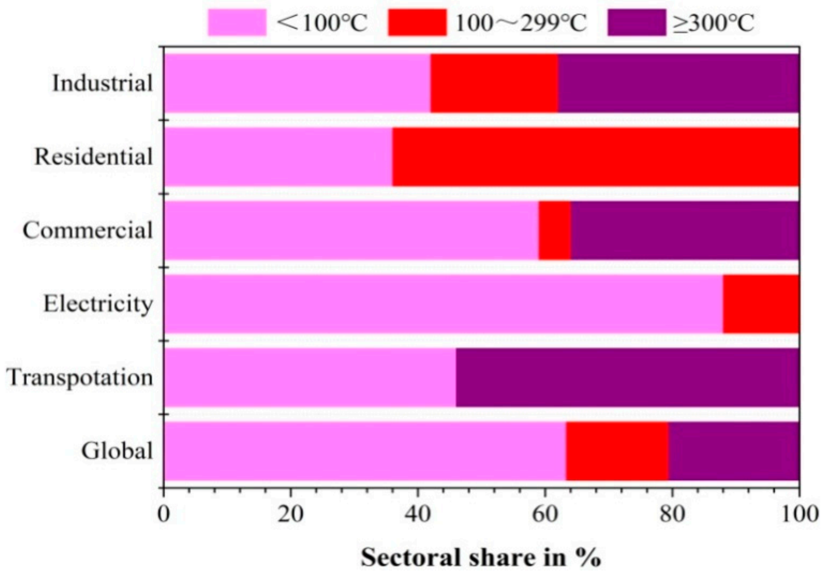


Figure 1. Sectoral shares of worldwide waste heat distribution [1].

From a thermodynamic point of view, energy from waste heat can be recovered in various ways. Direct heat exchange can take place between waste heat and other fluids, namely the heat transfer fluids for the heating and cooling process [7,8]. Conversion of waste heat into useful power is done using a thermodynamic cycle. The waste heat can be used as a heat source, i.e., the high-temperature side, which is used to heat the working fluid to obtain a gas phase of certain temperature and pressure. This working fluid, or the waste heat fluid itself, can be used to perform expansion work [9,10]. Another method is to raise the temperature of the waste heat to a required value using a heat pump for special applications such as distributed energy systems [11,12]. These diverse pathways face challenges, such as the low-grade and fluctuation and intermittency of heat sources, inefficiencies in the energy conversion process, and cascade utilization of different energy sources.

2. Barriers to Waste Heat Recovery

Using waste heat as an energy source is based on many aspects, which are elaborated as follows: Many factors make waste heat recovery very difficult, such as operating principle of heat recovery facilities, user demand, and characteristics of the source of the waste heat. Each method of waste heat recovery is posed with different problems, thus, the technologies face a number of barriers. A graphical representation of the various energy conversion pathways for waste heat is shown in Figure 2. Waste heat is known to mainly originate from two types of sources, i.e., fossil fuel and renewable energy. Most of the waste heat from fossil fuel involves industrial processes while renewable energy can be used directly through an air pre-heater, waste heat boiler, and economizer, and a small part of them need to go through a thermal power cycle before being used [13]. This is one of the main reasons for the temperature grade diversification of waste heat resources. Meanwhile, as mentioned above, the diverse forms of waste heat recovery technologies depend on the specific energy form needed by the end-user. At this point, the limitation of space for equipment as well as the economic and environmental boundaries should be taken into account. These abovementioned issues make an efficient waste heat recovery challenging.

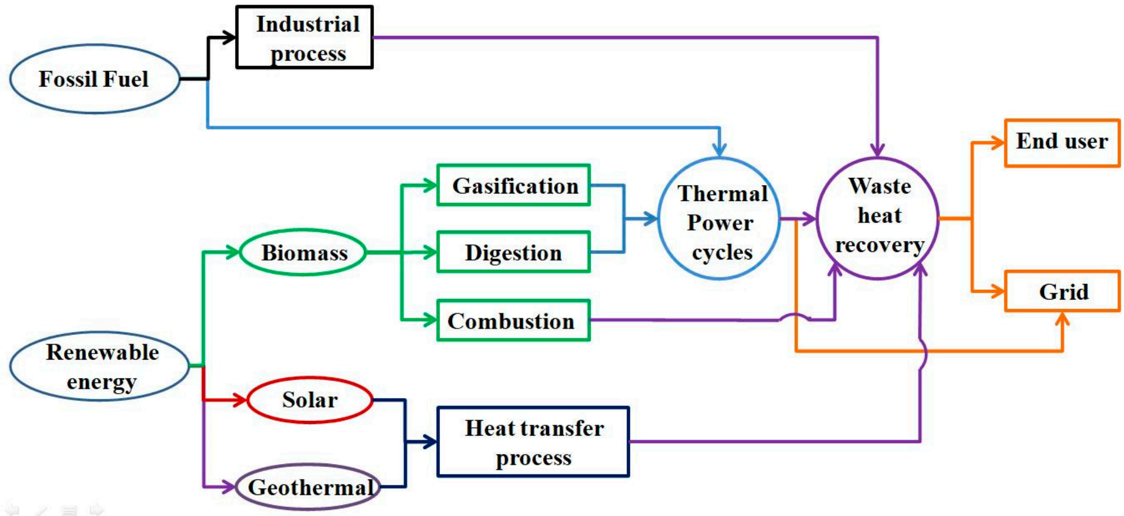


Figure 2. Waste heat recovery from various heat sources [13].

From a technical perspective, the following aspects need to be highlighted. First of all, waste heat sources are prone to more fluctuation and intermittence than traditional heat sources, severely affecting the operation stability of the recovery system. This is especially challenging in context to an evaporation process where the energy transfer takes place directly between the heat source and the working fluid. The abrupt fluctuation of the temperature of the heat source during a transient scan cause the variation in physical and chemical properties of the working fluid [14]. The indirect evaporation with an intermediate heat transfer fluid between the fluctuating waste heat stream and the working fluid can solve such problem effectively. Moreover, the development of a proper control strategy can also avoid large fluctuations in the performance of the system [15,16].

A second technical aspect involves systems that have more than one heat source, such as the waste heat from internal combustion engines (ICE) and from other complex industrial processes. For such a case, a waste heat recovery system must be designed and optimized for the multiple heat sources. In ICEs, the engine coolant is low-grade waste heat with temperature below 100 °C which is used as a preheat source, and the exhaust gas is high-grade waste heat with temperatures ranging from 500 °C to 700 °C [17]. Technical solutions involve matching the characteristic temperature drop of each source of waste heat.

Thirdly, due to the variety and complexity of waste heat resources, where the waste heat is present, the chemical composition of the waste heat carrier is important when considering recovery from steam generation [18,19]. Furthermore, the waste heat sources are usually dispersed geographically, which makes it challenging to integrate the recovery system with the original industrial process. In these cases, additional problems like the pressure drop of the flue gases in the heat absorber [20] should be taken into account.

To realize an economical and efficient waste heat recovery process, the barriers to the wide application of recovery technology for low-temperature waste heat have been described as follows: (1) The mismatch of energy grade of the waste heat resource and user demand in terms of time and space, (2) the intermittent of waste heat sources, and (3) the lack of a comprehensive methodology for global energy transfer and transformation optimization at the system level [21–23], especially for larger zones.

3. State-of-the-Art Thermodynamic Cycles for Waste Heat Recovery

Various implementation pathways can be adopted with respect to the quality of waste heat sources [2]. The main approach is to utilize the available waste heat as heat source to drive the thermodynamic cycle and convert the thermal energy into useful power. One of the most important indicators for choosing a proper cycle is the operation temperature. Figure 3 depicts the possible thermodynamic cycles and the range of their relevant operation temperatures. The most suitable waste heat temperature range for the Steam Rankine Cycle (SRC) is medium-high temperature, at about more than 300 °C. Systems for lower temperature heat sources are much less cost-effective and may lead to surface corrosion problem. Furthermore, the Organic Rankine Cycle (ORC) which uses lower boiling point organic fluids, has been also extensive investigated in last few decades. A suitable waste heat temperature to obtain competitive efficiency of the ORC ranges from 90 °C to 250 °C. Additionally, the Kalina Cycle uses a mixture of ammonia and water as working fluid to closely meet the temperature profile of waste heat sources during the phase change heat transfer process, between 100 °C and 450 °C.

Recent years, the use of CO_2 power cycles for waste heat recovery are gained more and more attentions. Such cycles are usually operated under trans-critical or supercritical conditions, due to the relatively low critical point of carbon dioxide. Its main advantages can be summarized as follows. First of all, the CO_2 cycles are suitable for the recovery of waste heat sources with a wide range of temperatures, while the steam based Rankine cycle is only efficientto recovery waste heat at high-temperature. Therefore, the main propose to investigate the CO_2 power cycles was not only to replace the SRC [24] but also as a more wide range to waste heat recovery temperatures [25]. The special physical properties of CO_2 can reduce the heat transfer loss between working fluid and heat source [26]. Furthermore, the CO_2 is a non-combustible and nontoxic refrigerant which frees up space for the cycle operation temperature to rise [27,28]. In addition, the CO_2 power cycles occupy a smaller space as compared with SRC, which has a large working fluid steam volume and therefore shows great potential for system downsizing and lightweight. Accordingly, the chemistry and condensingprocesses are also simpler [28,29].

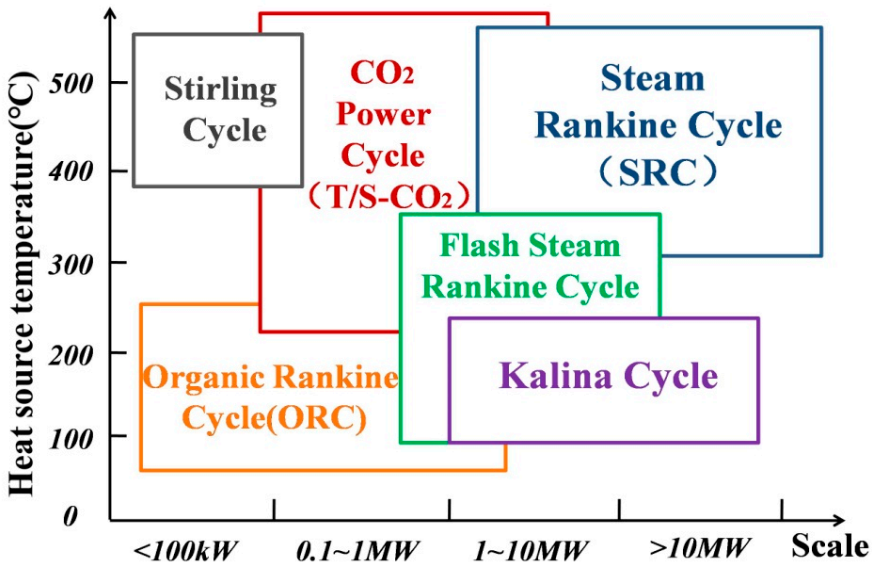


Figure 3. Thermodynamic cycles for waste heat recovery at various temperature-levels and scales.

Besides to thermodynamic cycles, other choices is to use thermoelectric (TE) materialsfor power conversion of waste heat. Relevant investigations focus on the thermoelectric generator [30], electrochemical systems [31], thermogalvanic cells [32] and pyroelectric energy conversion [33], which are still under study and has no large scale market application.

In summary, with respect to conventional thermodynamic cycles, supercritical carbon dioxide(s-CO_2) cycles offer a wide range of operating temperatures and potentially higher efficiencies, and a much smaller environmental footprint [34–36]. Power cycles running on s-CO_2 have received wide attention, and the number of publications has risen exponentially in recent years. However, the great potential for this technology in waste heat recovery remains to be further explored, which makes this comprehensive review of recent advances in s-CO_2 cycles for waste heat recovery highly relevant.

4. Advances in s-CO_2 Power Cycles for Waste Heat Recovery

In general, a cogeneration system can be consisted of a topping- and a bottoming cycle on the basis of the sequence of energy use. In the topping cycle, the input primary energy is used to first produce power and thermal energy, whilein a bottoming cycle the waste heat rejected from the topping cycle is further used to generate power through a recovery heat exchanger and a turbine machine. The bottoming cycles are suitable for recovery the low-grade waste heat produced by industrial processes to realizing the cascade utilization of energy.

In this section, studies on different applications of CO_2-based bottoming cycles for waste heat recovery have been summarized and discussed in detail. It should be noted that the investigations on CO_2-based power cycles are primarily focused on solar energy [36,37] and carbon capture systems [38,39]. Nevertheless, this section will only focus on those studies that utilize CO_2 power cycles for industrial waste heat recovery applications. Figure 4 illustrates a roadmap for the progress of research of the s-CO_2 cycle for different industrial waste heat recovery applications in the last ten years. Research has been mainly concentrated on three aspects, i.e., recovering waste heat from fuel cells, internal combustion engines (ICE), and gas turbine. Moreover, waste heat recovery from nuclear power plants and landfills has been also carried out.

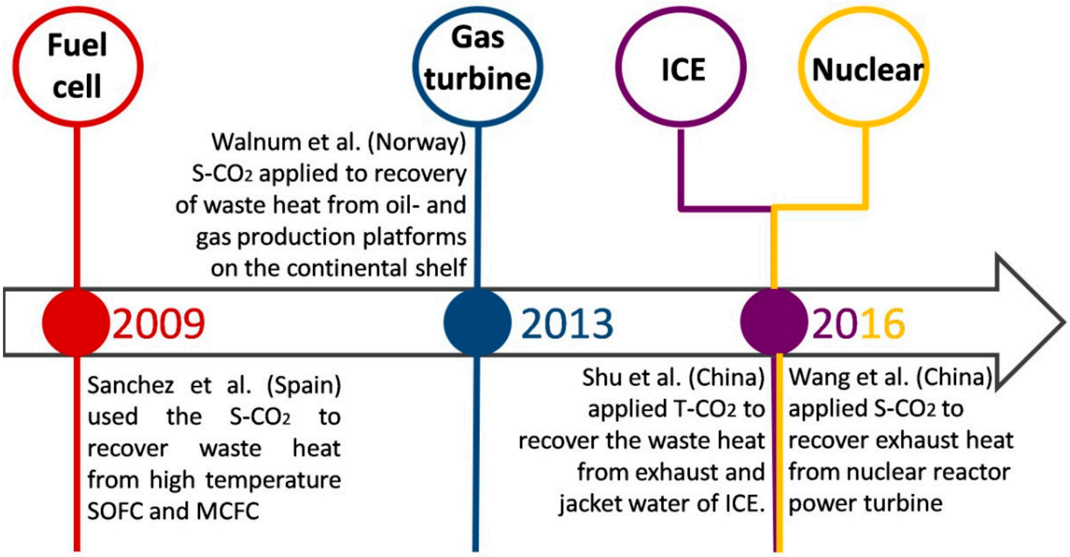


Figure 4. Roadmap of s-CO_2 investigation in different industrial waste heat recovery applications.

4.1. Fuel Cell

The research in relevant region begins with the employment of CO_2-based bottoming cycles for waste heat recovery in high temperature fuel cells. The pioneering work was reported in 2009 [40], and it applied a regenerative s-CO_2 cycle to recover flue gas waste heat from high-temperature solid oxide fuel cell (SOFC) and molten carbonate fuel cell (MCFC). The total system efficiency was increased by 4.4%, while the total net output power was increased by 583.6 kW. The study then compared the combinations of six different configurations of fuel cell and s-CO_2 cycles. The results indicated that the required power consumption of the compressor for the s-CO_2 bottoming cycle was far lower than the air bottoming cycle, and the operation performance of the bottoming cycle was less affected by the fuel cell operating temperature [41]. Bae et al. [42] compared to the thermodynamic performance of four different configurations of the s-CO_2 bottoming cycle to recover waste heat from the MCFC flue gas and compared it with the regenerative air Rankine cycle. The results showed that the total efficiency of the system could be improved by nearly 11% by using the cascade cycle, which was much higher than that of the system using the air Rankine cycle as the bottoming cycle. Moreover, the investigation from Baronci et al. [43] showed that the adoption of the s-CO_2 bottoming cycle under optimal conditions could enhance the total energy efficiency of the system by 8.15%. Meanwhile, through a performance comparison, it was found that the total energy efficiency of the system using s-CO_2 as the bottoming cycle could reach 55.3%, while total energy efficiency of the system using ORC (cyclohexane as the working fluid) as the bottoming cycle was only 53.3%.

Moreover, Ahmadi et al. [44] proposed a combined cycle of proton exchange membrane fuel cell (PEMFC) and s-CO_2, which used s-CO_2 fluid to replace the cooling water of conventional fuel cells. It also reused the gasification cooling energy of liquefied natural gas (LNG) to reduce the condensation temperature of the combined cycle to improve the cycle efficiency. Through a sensitivity analysis of the system operation parameters, the study demonstrated that the total energy efficiency of the system would decrease with the increase of operating temperature of the fuel cell and the increase of the pinch point temperature difference in the pre-heater of the bottoming cycle. The total energy efficiency of the system could be improved with the increase of the turbine inlet temperature of the bottoming cycle and the decrease of the pinch point temperature difference of the condenser. This study also showed that using s-CO_2 as the bottoming cycle could increase the net output power of the cycle by 39.56%, and the total energy efficiency could reach up to 72.36%. Furthermore, Mahmoudi et al. [45] put forward the MCFC/s-CO_2/ORC cascade system to create a combined supply of cooling, heating, and power, and optimized the system with multiple objectives to maximize the exergic efficiency and minimize the initial investment of the system. The results showed that the largest exergy loss of the system came from the fuel cell cycle, and the operating temperature of the fuel cell was positively correlated with the exergy efficiency and initial investment of the system. The latest research by Ryu et al. [46], compared the thermodynamic performance of the MCFC cycle using three different configurations of the s-CO_2 bottoming cycle. The results showed that the total energy efficiency of the system could be increased by 3.41–4.6% by adding the bottoming cycle, and the back pressure of the compressor was the key parameter that affected the bottom cycle performance. In addition, the economic analysis of the system showed that the combined cycle had obvious economic advantages over the traditional thermal and power cogeneration system, i.e., the heating cost was less than \$28/Gcal, and the cost of printed circuit board heat exchanger (PCHE) was lower than \$100/kW.

4.2. Internal Combustion Engine

Another research hotspot in this field is the comprehensive recovery and utilization of waste heat of internal combustion engines (ICE) using the CO_2-based bottoming cycle. The most extensive research in this area has been conducted by Shu et al., which have been summarized herein. In context to theoretical research, the system performance of different forms of CO_2 or ORC bottoming cycles for the cascade recovery of the waste heat of exhaust and jacket water, of a four-cylinder four-stroke water-cooled internal combustion engine, was analyzed. The results showed that the combined use of preheating and the regenerative CO_2 cycle could increase the total net output power of the system by 9.0 kW at the highest, and the corresponding thermal and exergic efficiencies of the system were increased by 184% and 227%, respectively [47]. Moreover, after further comparison of four different configurations of CO_2 bottoming cycles, it was found that the contribution of the pre-heater to the recovery of waste heat from the jacket water and the improvement of the net output power of the system due to the set of the regenerator, were 5.5 kW, and 7.0 kW, respectively. The relevant overall exergic efficiency of the system could reach 48% by using the pre-heater with the regenerator CO_2 bottoming cycle. At the same time, the system economy studies shown that setting the regenerator is conducive to improving the system economy, while adding the pre-heater is not as useful [48]. In addition, by establishing a dynamic cycle model, the influence of different operating parameters on the performance of the CO_2 bottoming cycle under partial load condition of the ICE was studied. Accordingly, a system operation control strategy with the mass flow rate of the working fluid as the regulation target was proposed [49].

In terms of experimental research, the CO_2 bottoming cycle performance under typical operating conditions of the ICE was tested with different cycle pressure ratios [50]. Meanwhile, the dynamic performance of the CO_2 bottoming cycle of three different configurations under given operating conditions was compared to a study of the influence of different working fluid mass flow rates and cycle pressure ratios. The time constant of the dynamic system performance was obtained [51,52]. Furthermore, aimedat the special ICE operation conditions, such as start, idling, and emergency stop, the operating performance of the CO_2 bottoming cycle with the pre-heater, was studied. The results showed that the preheating effectively prevented the pressure surge at the inlet of the expander, so as to ensure the stable and safe operation of CO_2 bottoming cycle under special working conditions. It also improved the energy efficiency of the overall system under partial load conditions [53]. The latest investigation by Shu et al., pertains to the development of an ICE-CO_2 cold-power cogeneration system, which consists of theoretical analyses on the operating performances of the system under various operation modes. The results showed that compared with the traditional system, the proposed system could reduce fuel consumption of the ICE by 2.9% under the refrigeration mode, and increase the total net output power by 4.8%. Meanwhile, under the ice-making mode, the fuel consumption could be reduced by 3.4% and the total net output power of the system was increased by 1.6% [54]. In addition, Shu et al. proposed to adjust the condensation temperature of the CO_2 bottoming cycle by using mixed working fluid, and to simulate the dynamics of the system with different mixed working fluid by using the finite volume method and a moving boundary model. The results of the off-design modelling of the proposed systems showed that under the same operating conditions, with the increase of CO_2 concentration in the mixed working fluid, the dynamic response speed of the system became faster, while the thermal efficiency and net output power of the system decreased slightly. Moreover, the maximum value of net output power appeared at operation conditions with high working fluid pump speed [55]. Moreover, the performance of the system with the CO_2/R134a mixture as the working fluid was experimentally analyzed. The influence of different concentrations of the mixed working fluid on the energy efficiency of the system was investigated. The results indicated that the energy efficiency and the net output work showed a trend of first rising and then falling as the mass fraction of R134a increased [56].

Relevant studies in this area were also conducted by Choi et al. [57] who proposed to use the temperature difference between the jacket water of a marine engine and seawater to drive the two-stage reheat CO_2 power cycle. Thermodynamic analyses showed that the maximum net output power of the CO_2 bottom cycle was 383 kW, the highest thermal efficiency of the system was 7.87%, and the highest exergic efficiency was 5.96%. Moreover, Sharma et al. [58] carried out thermodynamic analyses on the regenerative and recompressed s-CO_2 Brayton cycle used to recover waste heat from flue gas of marine engines. The influence of several key operation parameters, such as the inlet temperature of the turbine and compressor, as well as the equipment pressure drop on the overall performance of the combined cycle, was investigated. The results showed that the s-CO_2 bottoming cycle improved the overall cycle efficiency, and the net output power by 10%, and 25%, respectively. In addition, the exhaust composition and exhaust temperature of the gas turbine in the topping cycle had a significant effect on the performance of the s-CO_2 bottoming cycle. Hou et al. [59] proposed a tri-generation system by recovering the waste heat from the marine engine based on the recompression s-CO_2 cycle. They carried out a thermal-economic optimization for the proposed system through a genetic algorithm. The study showed that the high-temperature regenerator and evaporator of the refrigeration system were the key components that affected the thermal economy of the proposed system. Manjunath et al. [60] presented the energetic and exergetic performance analyses of a supercritical/transcritical CO_2

based bottoming cycle for marine engine. It was found that under the optimal operating conditions, the enhancement of the power output by the proposed system is nearly 18% and provide cooling of 892 TR having the COP (coefficient of performance) of 2.75. Liang et al. [61] proposed the s-CO_2/ORC combined cycle for waste heat recovery of the ICE, which increased the net output power of the overall system by 6.78%. Feng et al. [62] proposed to adopt s-CO_2/Kalina combined cycle to recover waste heat of marine engines. The influence of inlet temperature and pressure of the compressor and turbine on the combined cycle performance has been investigated. Multi-objective optimization was conducted to optimize the thermal-economic performance of the system, and the annual fuel consumption of the engine was reduced by 16.62%. Liang et al. [63] investigatedan engine waste heat powered thermal-power cogeneration system whichcombined thes-CO_2 power cycle with a transcritical CO_2 refrigeration cycle. This configuration was used to replace the conventional absorption cooling cycle. The results indicated that the proposed configurationreduce the size and weight of the system and is therefore proper on-board application. Pan et al. [64] proposed a cogeneration cycle which combined the s-CO_2 power cycle and ejector expansion refrigeration cycleas the bottoming cycle to recovery the waste heat from ICE. Working fluid in both sub-cycles is CO_2-based zeotropic mixture. The effects of the important operating parameters on system performance wereinvestigated. Zhang et al. [65] developed a novel s-CO_2 power cycle based on recompression cycle configuration to recover the waste heat from ICE. The influence of main operation parameters have beencomprehensively studied and a genetic algorithm was used to maximize the system net output power. The results indicated that for the intermediate pressure the maximum system net output power can reach to 39.49 kW. Song et al. [66] proposed a two-stage bottoming cycle for ICE waste-heat recovery. Thes-CO_2 cycle was coupled with an ORC to further recover the heat rejected from the s-CO_2 cycle. The proposedcycle can contribute a maximum net power output of 215 kW, which accounts for ca. 18% of rated power of ICE.

4.3. Gas Turbine

Besides the ICE, there are several studies that proposed to use the CO_2 power cycles for recovery waste heat from exhaust of gas turbines. Walnum et al. [67] conducted thermodynamic analyses on the applications of regenerative and two-stage CO_2 cycles for the recovery of waste heat from flue gas of offshore oil- and gas platforms. The operation performance of bottoming cycles under partial load conditions of the gas turbine was studied. The results showed that single-stage cycles could increase the total net output power and the overall system efficiency of the oil- and gas platform by about 27.6%, and 10.6%, respectively, while the improvement of the double-stage cycle was more significant. Moroz et al. [68] compared the thermodynamic performance of various combined s-CO_2 cycles for recovery the waste heat from a 53 MW gas turbine. The results indicated that the simplest cascade cycle provided a power output of 16.13 MW, while value of the cycle with most complicated configuration is 17.05 MW, which represented 32% from the power output of topping cycle. The output power bythe single regenerative s-CO_2 bottoming cycle or a recompression s-CO_2 bottoming cycle is 12.94 MW, and 11.85 MW, respectively. Cho et al. [69] investigated cascade systems which consist of a recompression (or pre-compression) cycles and a partial heating cycle for recovering waste heat from a 288 MW gas turbine. The minimum cycle pressure and temperature, and isentropic efficiency of compressor and turbines have been discussed. They found that the cascade systems are uncompetitive due to their complex configuration and lower power output. Huck et al. [70] carried out a thermodynamic performance comparison of different dual flow splitting s-CO_2 bottoming cycles and steam bottoming cycles to recovery the waste heat from heavy-duty and aero-derivative gas turbine combined cycles. The waste heat recovery efficiency and thermal efficiency were calculated in detail. It was found that the efficiency improvement of s-CO_2 bottoming cycle was not significant when the system operated under more realistic assumptions. Wright et al. [71] compared three typical configurations of s-CO_2 cycles for recovering waste heat from a 25 MW gas turbine. The results showed that the total heat recovery efficiencies of the considered s-CO_2 cycles, ranged from 20.3% to 21.2%, which were 4% higher than the concerned baseline cycle. This was due to the higher recovery of waste heat offset the decrease in system thermal efficiency. Moreover, the single regenerative power cycle shows the best economy. Kim et al. [72] compared the thermodynamic performance of the waste heat recovery of gas turbines in a landfill plant with nine different configurations of the s-CO_2 bottoming cycle. The study showed that the recompressing cycle was not suitable for waste heat recovery, and the two-stage split-flow cycle had a significant effect on the improvement of the net output power of the overall system, but its structure was too complex. Khadse et al. [73] carried out an investigation of a simple construction of a s-CO_2 bottoming cycle to recovery the waste heat recovery from a gas turbine. The results indicated that a maximum improvement of 22.9% can be gained by the use of recompression configuration. Cao et al. [74] propose a cascade configuration which composed of a s-CO_2 Brayton cycle and a transcritical CO_2 Rankine cycle to recovery the waste heat from a gas turbine. Both cycles were based on simple configurations, and the CO_2 was condensed by using the cold energy of LNG (liquid nature gas). The results indicated that the power output by cascade cycles contributed nearly 28.9–39.1% to the power output of the whole system. Moreover, the investigation from Gao et al. [75] indicated that the partial heating cycle provided the highest power output compared to the single regenerative cycle due to its good waste heat absorption performance. Tozlu et al. [76] carried out a bi-objective optimization of a single regenerative s-CO_2 cycle for waste heat recovery from the exhaust gas of gas turbine. It was found that the s-CO_2 bottoming cycle showed a potential to increase the net power output of the turbines by 19.3%. Zhang et al. [77] proposed an improved cascade s-CO_2 bottoming cycle for recovering the waste heat from flue gas of the offshore oil- and gas platform and adopted an artificial bee colony algorithm to carry out the multi-objective optimization of the bottoming cycle design parameters. The results showed that the s-CO_2 bottoming cycle could improve the net output power of the overall system by 30% under rated conditions. Meanwhile, the high-temperature part of the cascade cycle had a greater impact on the thermal performance of the overall system, while the low-temperature part had a greater impact on the economic performance of the overall system. Sánchez et al. [78] usea partial heating s-CO_2 bottoming cycle to recover the waste heat from high temperature exhaust of a gas turbine. The results showed that, compared to conventional steam bottoming cycle, the proposed partial heating s-CO_2 bottoming cycle reached a high thermal efficiencyand reduced the system initial investment by a quarter. Zhou et al. [79] developed a novel supercritical-/transcritical-CO_2 combined cycle system for recovering waste heat from offshore gas turbines. Comprehensive parametric analysis was conducted to simultaneously optimize the net output work and net present value (NPV) under different conditions. Recently, Tao et al. [80] proposed applying the two-stage reheat and recompression split s-CO_2 cycle to recover the waste heat of gas turbines in distributed energy system. The preliminary thermodynamic analysis results showed that the total thermal efficiency of the system could reach up to 48% under the optimal split ratio.

4.4. Others

Other studies in this field have been summarized in this section. Wang et al. [81] adopted a genetic algorithm to optimize the exergy-economy of a s-CO_2 bottoming cycle for recovering waste heat from combustion engines in nuclear reactors, which increased the total thermal efficiency and the net output power of the overall system by 7.92%, and 13.7 MW, respectively. Astolfi et al. [82] compared the performance of the dual regeneratives-CO_2 bottoming cycle against three traditional cycle layouts. The results indicated that the dual regenerative layout was found to be the best choice, if the minimum heat source temperature has been not constrained. Olumayegun et al. [83] studied the dynamic performance of the recompression s-CO_2 cycle for recovering waste heat from cement plants. Those results indicated that the inlet temperature of the main compressor could be controlled by adjusting the mass flow of cooling water, and the inlet pressure of the compressor could be kept constant through the throttle valve to improve the dynamic performance of the whole cascade system. Luo et al. [84] proposed a multi-generation system which combined s-CO_2 cycle and transcritical CO_2 refrigeration cycles using waste heat as power source. Exergoeconomic evaluation and optimization has been conducted under different operating conditions. It was found that the refrigeration cost is the highest, while the cost of the power is the lowest of total system operating cost.

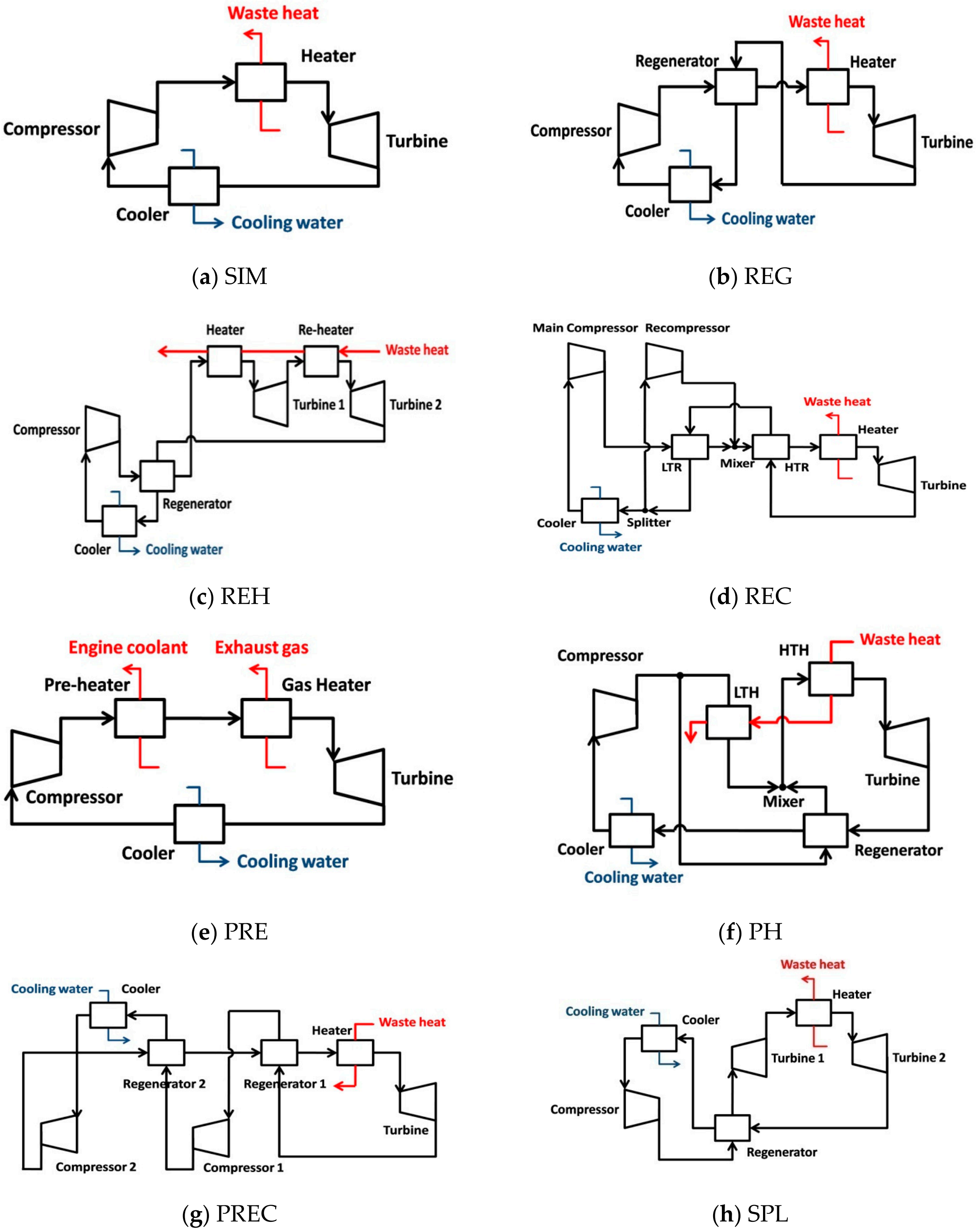
For a clearer and more intuitive comparison, Table 1 summarizes the main information in the above-mentioned literature, and eight typical configurations of s-CO_2 bottoming-cycle used for waste heat recoveryare illustrated by Figure 5. It can be found from this table that research on waste heat recovery from gas turbine accounts for half of the total listed literatures, while the amount of investigation on waste heat recovery from fuel cell and ICE are quite similar. Meanwhile, research on ICE and turbine has been a hot topic in this region in recent five years.

Table 1. Most relevantstudiesin chronological order.

Year	Author	Application	T_{HS} (°C)	Cycle Layouts	$W_{CO2,net.max}$ (kW)
2009	Sanchez et al. [40]	Fuel cell	709	REG	583.6
2011	Sanchez et al. [41]	Fuel cell	650	REG	540.4
2013	Walnum et al. [67]	Turbine	532	REG/ two stage REG	41,100 42,000
2014	Bae et al. [42]	Fuel cell	709	REC/REG/ two stage SIM	600.8/582.8/ 603.8
2015	Baronci et al. [43]	Fuel cell	398	REG	2800
2015	Moroz et al. [68]	Turbine	425–700	REG/REC/PREC/ cascade	17.1
2015	Cho et al. [69]	Turbine	580	cascade/PH/PRE	>100,000
2016	Ahmadi et al. [44]	Fuel cell	343	s-CO_2+LNG	276.1
2016	Shu et al. [47]	ICE	777	SIM/PRE+REG	3.6/4.4
2016	Choi et al. [57]	ICE	354	two stage REH	383
2016	Wang et al. [81]	Nuclear	850	two stage SIM	22,000
2016	Huck et al. [80]	Turbine	650–750	SPL	>100,000
2016	Wright et al. [71]	Turbine	538	REG/PH/SPL	8500
2016	Kim et al. [72]	Turbine	520	RE/REC/PH/PREC/ various cascade cycles	2180/2200/2750/2230 3230
2017	Shu et al. [48]	ICE	777	SIM/PRE/REG/ PRE+REG	3.7/5.5/4.6/ 9.1
2017	Sharma et al. [58]	ICE	368	REG+REC	5.6
2017	Khadse et al. [73]	Turbine	630	REG/REC	110,000
2017	Cao et al. [74]	Turbine	440–543	REG+ORC	21,000
2017	Gao et al. [75]	Turbine	538	REG/REC/PH/cascade	38,000
2018	Tozlu et al. [76]	Turbine	567	REG	>1000
2018	Zhang et al. [77]	Turbine	490	two stage SIM	6170
2018	Hou et al. [59]	ICE	466	REC+REF	5000
2018	Manjunath et al. [60]	ICE	550	two stage s-CO_2/t-CO_2	3694
2018	Astolfi et al. [82]	Generic WHR	200–600	SIM/REG/REC	15,000
2019	Liang et al. [61]	ICE	423–488	REG+ORC(R1233zdE)	40.9
2019	Olumayegun et al. [83]	Turbine	380	REG+REC	5000
2019	Luo et al. [84]	Turbine	500	SIM+REF	---
2019	Tao et al. [80]	Turbine	550	REG+REC	300
2019	Sanchez et al. [78]	Turbine	598	PH	74,000
2020	Ryu et al. [46]	Fuel cell	360	REC/REC+REH	281.6/282.6
2020	Feng et al. [62]	ICE	268	SIM+Kalina	242.6
2020	Liang et al. [63]	ICE	380	SIM+REF	16.5
2020	Pan et al. [64]	ICE	557	SIM+REF	20.8
2020	Zhang et al. [65]	ICE	450	REC	39.5
2020	Song et al. [66]	ICE	460	SIM+ORC	215
2020	Zhou et al. [79]	Turbine	435	two stage s-CO_2/t-CO_2	55,000

SIM: simple. REG: regenerative. REH: reheat. REC: recompression. PRE: preheat.REF: refrigeration. ORC: organic Rankine cycle. LNG: liquid nature gas. PH: partial heating. PREC: pre-compression. SPL: flow split.

Figure 6 presents the heat source temperature and maximum power output of the s-CO_2 bottoming cycle with various configurations. The maximum power output is presented on a logarithmic scale to clearly identify the system operation maps. It can be found that system size below 1 MW and above 1 MW are equally divided in the observed research data. Among them, power output of the s-CO_2 cycles used to recover waste heat of the fuel cell are distributed around 1 MW, while most of the systems used to recover waste heat from gas turbines are mostly larger than 10 MW. Research of small-scale system has mainly focused on waste heat recovery from the ICE. This is due to the flexible operating range and multiple application scenarios of the internal combustion engines. It can be also observed that temperature range of waste heat sources studied in this field is between 400–700 °C and the positive correlation between heat source temperature and system output work can obviously be found by the fitting curve.



(**a**) SIM (**b**) REG (**c**) REH (**d**) REC (**e**) PRE (**f**) PH (**g**) PREC (**h**) SPL

Figure 5. Typicalconfigurations of s-CO_2 bottoming-cycle used for waste heat recovery.

In summary, for low and medium temperature waste heat sources, the transcritical CO_2 cycles were mainly considered, and their performance as bottoming cycles were evaluated against ORC or Kalina cycles. Whereas, the supercritical CO_2 cycles were competitive technology for medium- and high temperature sources to replace the air- or steam Rankine cycles. It can be found that the selection of proper configurations of s-CO_2 cycles for waste heat sources with different temperature level was the main research content of this field. Based on the results of the existing literatures, the simple and regenerative configuration were more suitable for the transcritical cycles. The reason is that application scenarios with low- and medium temperature waste heat sources often have limited device space, so that the layout simplicity and compactness should be taken seriously. However, for the waste heat of medium- and high temperature, two-stage or cascade configurations have been more studied to form a thermal-power cogeneration system.

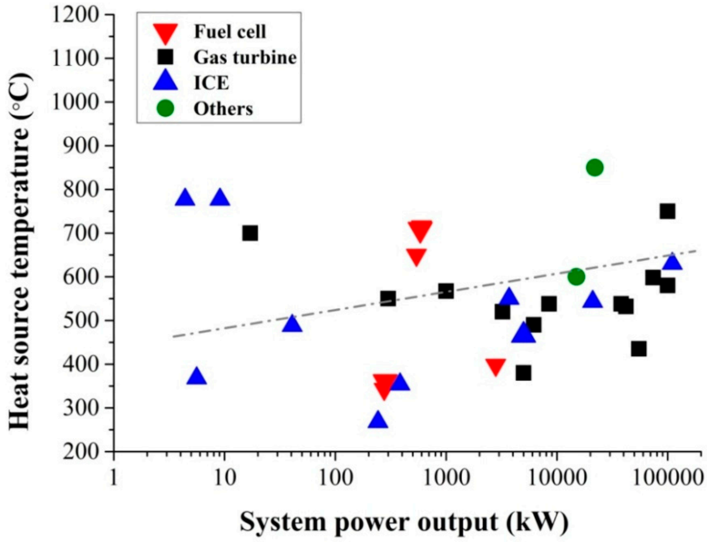


Figure 6. Heat source temperature vs. system power output (s-CO_2 bottoming cycle) based on database in Table 1.

5. Further Perspectives

Generally speaking, many technical characteristics of s-CO_2 power cycles reflect the advantage of its application in waste heat recovery. First of all, the smaller and compact system size makes the flexible and on-site arrangement for recovering the fragmented waste heat sources possible and reduces the investment cost. Secondly, the relative higher cycle efficiency leads to increased power production for lower thermal energy input. Moreover, the wide range of applicable heat source temperatures makes it possible to cascade the recovered waste heat resources at different temperatures grades and in different forms. Thirdly, a reduction in water consumption and greenhouse gas emissions brings more environmental benefits and enhances the market competitiveness of such waste heat recovery units simultaneously. However, the application of s-CO_2 cycles for waste heat recovery still faces several challenges which are presented in the following paragraphs.

First, while, the expansion turbine and heat exchangers of s-CO_2 cycles are theoretically smaller and more compact than other competitive thermal cycles with similar size, such components do not yet exist at the commercial level. A component design of the s-CO_2 Rankine cycle is a very crucial area of research, and relevant investigations are still in progress [85–88]. Another component design-related problem is the piping pressure drop. Since the working fluid mass flow rate of the CO_2 cycle is obviously higher than that of a conventional steam based Rankine cycle by the same plant scale, the compression process of supercritical carbon dioxide fluid is much more energy consuming. Optimal design of construction of waste heat exchanger, as well as the pipeline to reduce the pressure drop is also crucialfor improving the performance of the entire system [89].

Secondly, further investigations are needed to consider the large scale power plants. Experimental data is still scarce, especially for megawatt-level systems. Furthermore, much larger scale industrial process plants are taken into account, a heat exchanger network (HEN) is typically constructed for recovery heat between different streams [90,91]. Understanding how to integrate waste heat recovery units, such as s-CO_2 and ORC into the HEN for further enhancement of energy utilization efficiency, is still an open question.

The third challenge to be addressed is the off-design, as well as the transient performance of the plant. This is due to fluctuation in the availability of waste heat resources leading to the variability of output power and a mismatch between the energy source and user demand in terms of time and spatial dimensions [2]. Therefore, the off-design performance of the s-CO_2 cycles should be studied further. As the part-load performance plays a decisive role in the stable and economical operation of the whole combined power plant, in this context, energy storage will be necessary to maintain stable input and sufficient output [92]. Research in this area is on the rise.

With reference to completed and ongoing engineering projects, the pioneering megawatt-level commercial s-CO_2 cycle for waste heat recovery was developed by Echogen (USA) [93]. A simple recuperating cycle with a turbine-generator was built for recovering waste heat from the exhaust of a gas turbine (500–600 °C, 60–75 kg/s) to generate nearly 2.4 MW of power. Moreover, in a gas compressor station owned by the Canadian energy company, TC Energy, Siemens Energy is currently working to install a system for converting waste heat to power with turbines driven by s-CO_2. The s-CO_2 turbine is expected to go on the grid in 2021, and it will supply electricity for more than 10,000 households [94]. Owing to the above-mentioned challenges, as well as some uncertainties in technological advancement, performance, and component costs, further research and development is needed to better understand the applications of s-CO_2 power cycles for waste heat recovery. The potential market of s-CO_2 power cycles will continue to depend upon several technical, economic, environmental, and social factors.

6. Conclusions

The s-CO_2 power cycle has the technical and economic potential to be applied in waste heat recovery. The comprehensive analysis presented herein gives an extensive overview of the background, technical barriers, and current advances of s-CO_2 cycles for waste heat recovery. It aims to help researchers who are trying to solve problems related to waste heat recovery with s-CO_2 cycles, and its main aspects have been summarized as follows:

(1) Unlike the stand-alone s-CO_2 power cycle, whose objective is to solely achieve a high thermodynamic efficiency, the aim of the s-CO_2 bottoming cycle for waste heat recovery is to achieve both the high cycle thermodynamic efficiency and larger waste heat regenerative amount. The improvement in total heat recovery efficiency is mainly a consequence of more effective heat extraction from the waste heat source and a higher cycle thermal efficiency resulting from the cycle optimization. Moreover, great attention should be paid to study to a unique characteristic of a waste heat source, which is the common problem faced by all waste heat recovery technologies.

(2) Although the cascade configuration could theoretically be more efficient, the optimization cannot leave out of consideration of simple constructions when system economics and dynamic performance are taken into account.

(3) Unlike the organic Rankine cycle whose commercial modules are widely available, operation data from experimental investigations or commercial proto types of the s-CO_2 power cycle are still limited.

The perspectives for solving the above-mentioned issues are provided as follows:

(1) A comprehensive evaluation system shall be proposed to solve the technical barriers through the trade-off between the characteristics of the heat source, the thermodynamic performance of the system, and economics.

(2) It is necessary to carry out transient performance analysis and exploit dynamic control strategies for both the s-CO_2 bottoming cycle and the entire cascade system to ensure the stable and safe operation of the system.

(3) Extensive experimental research and engineering practices are needed to improve the design accuracy of key components and verify the mechanism of the cycle performance. In addition to the

prototypes, broader scales of the s-CO_2 cycle should be further developed to fill gaps in the current commercial market.

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