

Selected Issues of the Structures and Parameters of Supercritical CO₂ Gas Turbine Cycles

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The paper presents a variant analysis of the structures of closed gas turbines using supercritical carbon dioxide (super-CO₂) as a working fluid. Several configurations covered in the available literature were collected, commented on and compared. The parameters of the cycles, such as operating temperature and heat supply are noted and commented on. There are three main configurations considered in the available literature: the pre-compression cycle, partial cooling cycle, and recompression cycle.

Keywords: waste heat recovery, supercritical CO₂, Brayton cycle, gas turbine.

1. Introduction

In the 1960s, Feher [1] studied the properties of various gases with a view to determining the most suitable one for a supercritical thermodynamic cycle. Carbon dioxide was proposed as a working fluid for several reasons. First, its physical properties e.g. critical pressure, which is significantly lower than water, allows lower operating pressures. Second, the thermodynamic and transport properties of CO₂ are well known, hence cycle analysis is based on reasonably firm data. Finally, carbon dioxide is abundant, non-toxic and is relatively low cost. The analysis proved that the CO₂ supercritical cycle offers several desirable features: high thermal efficiency (the investigated cycle achieved thermal efficiency of 55% under ideal conditions), low volume-to-power ratio and no blade corrosion or cavitation. The paper suggests using it for electric power generation (both terrestrial and space) or to provide shaft power for marine propulsion.

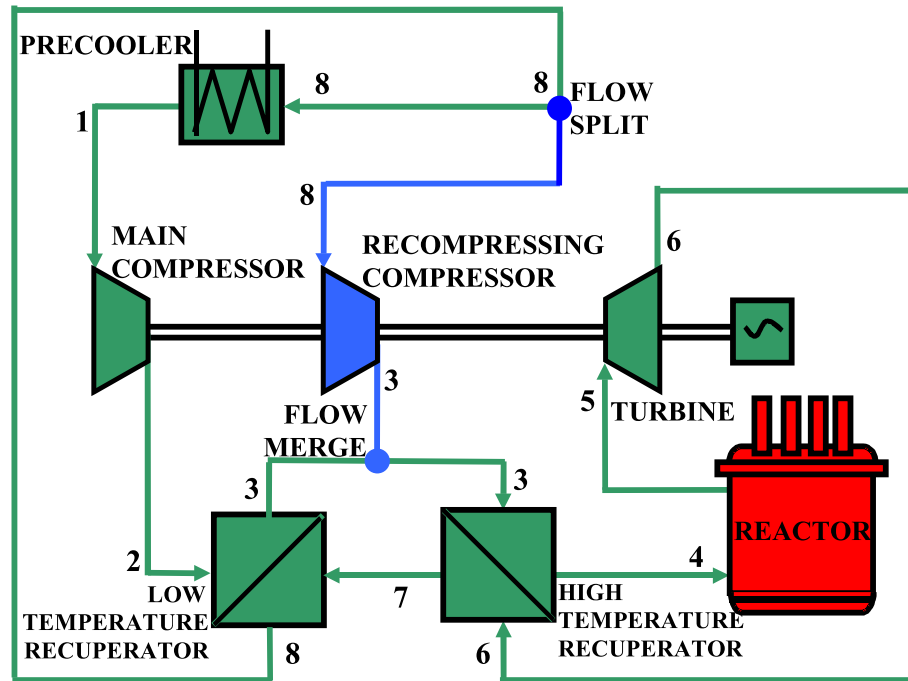
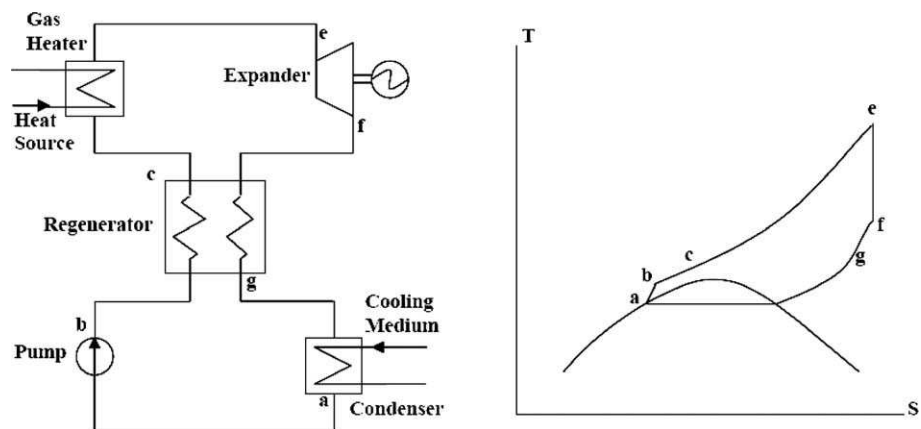


Figure 1 Recompression Brayton cycle layout [2]

Figure 2 CO₂ transcritical system layout and cycle T-S chart [4]

Research on the supercritical CO₂ power cycles was resumed by Dostal four decades later. In 2004 [2] he performed a systematic, detailed major component and system design evaluation and multi-parameter optimization of the family of supercritical CO₂ Brayton power cycles for application to advanced nuclear reactors. His analysis showed that the recompression cycle shown in Fig. 1 was the best performing cycle layout due to its simplicity, compactness, cost and thermal efficiency. Three direct cycle designs of this layout were selected for further investigations. They achieved thermal efficiencies of 45.3%, 50% and 53%, assuming turbine inlet temperatures 550°C, 650°C and 700°C respectively. According to the analysis the turbomachinery is highly compact the 600 MWth/246 MWe power plant is fitted with a turbine of 1.2 m in diameter and 0.55 m long, which translates into power density of 395 MWe/m³.

Later, Driscoll [3] presented a report on cost projections for the supercritical CO₂ Brayton indirect power cycle as applied to GEN-IV advanced reactors. In order to evaluate economical competitiveness of the cycle a cost comparison procedure was adopted in which projections were made on the basis of published cost estimates for related reactor systems coupled with the direct or indirect helium Brayton cycle or to the conventional indirect Rankine cycle. A preliminary analysis showed savings at least of the order of 10%, with the dominant parameters being cycle thermodynamic efficiency and turbomachinery capital cost.

These two papers have spawned a number of further studies of the CO₂ supercritical cycle in the field of parameters and layout optimization, possible applications and modeling of critical cycle components. Chen [4] evaluated transcritical CO₂ as a working fluid in low-grade waste heat recovery cycles by comparing it to R123 Organic Rankine Cycle (ORC). Fig. 2 shows the basic ORC system layout and the ORC schematic cycle in a TS chart. The results of the comparison showed that, when utilizing a low-grade heat source with equal thermodynamic mean rejection temperature, the CO₂ transcritical cycle has slightly higher power output than ORC and is more compact as well. On the other hand, further research provided by Vidhi et al. in 2011 [5] showed that although CO₂ has the advantages of being available in abundance, environmentally safe and economically favorable, its performance in a transcritical power cycle is not as efficient as R32 based organic Rankine cycle over the range of source temperatures from 140°C to 200°C. A comparative analysis of a recompression CO₂ Brayton cycle combined with ORC and a single recompression cycle was also performed. It showed that the exergy efficiency of the combined cycle could be higher than that of the single recompression cycle by up to 11.7% and total product unit cost lower by up to 5.7% [6].

Parametric optimization performed by Wang et al. [7] using a genetic algorithm and artificial neural network showed that the key thermodynamic parameters, such as turbine inlet pressure, turbine inlet temperature and environment temperature have a significant effect on the performance of a supercritical CO₂ power cycle and exergy destruction in each component.

Kulhanek and Dostal [8] found that among various cycle layouts shown in Fig. 3, a recompression Brayton cycle achieves the highest efficiency in the range of turbine inlet temperatures between 500 and 600°C, whereas partial cooling cycle is better at higher temperatures. On the other hand, Bryant [9] proved that, indeed, the recompression cycle will always be more efficient than a simple cycle provided that

the two cycles have the same precooler inlet temperature, but in order to satisfy this condition the recompression cycle will always require more total recuperator area. The paper demonstrated that when two cycles are compared on the basis of equal total recuperator area, the efficiency advantage of the recompression cycle is substantially reduced or even disappears altogether.

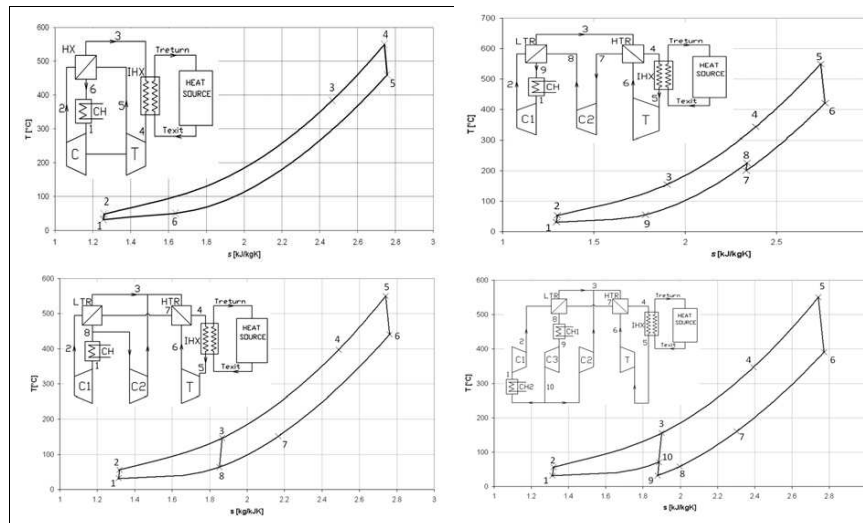


Figure 3 Simple Brayton, Precompression, Recompression and Partial Cooling cycle [8]

Kim and Favrat in 2012 [10] presented a novel transcritical Rankine cycle using both low and high temperature heat sources to maximize the power output of the CO₂ power cycle with a given high temperature source for use in applications such as nuclear power, concentrating solar power and combustion. The analysis showed the large internal irreversibility in the recuperator related to the higher specific heat of the high-pressure side than that of the low pressure side. Additional low temperature heat provided to the recuperator in the proposed cycle mitigates the specific heat difference, and thus makes it possible to achieve higher recuperator CO₂ outlet temperatures. This feature in conjunction with reduced compression work and exergy losses makes the low-high temperature Rankine cycle even more effective than the recompression Brayton cycle.

Application of supercritical CO₂ cycles in a cogeneration power plant was considered by Moroz in 2014 [11]. The performance of several stand-alone supercritical CO₂ cycles and combined steam/supercritical CO₂ cycles was compared with typical steam cogeneration cycles. The cascaded supercritical CO₂ recompression Brayton cycle achieved the best electrical efficiency of 39.4% at turbine inlet temperature of 540°C, which beat the ordinary steam CHP unit.

In 2009 Moiseyev [13] examined alternative supercritical CO₂ Brayton cycle layouts, which were presumed to perform better than the recompression Brayton cycle when coupled with Sodium Fast Reactors. This assumption was based on the

fact that SFRs operate at lower temperatures (core outlet temperature of 510°C) than the temperature for which a satisfactory recompression cycle performance had been proved. Even though a double recompression cycle, intercooling between compressor stages and reheating between high and low pressure turbine were analyzed, the recompression cycle demonstrated the highest efficiency. Later, Perez-Pichel [12] conducted a similar analysis in which he compared a wide range of configurations, from the simplest one to combined cycles (with organic Rankine cycles, ORC). As a result, he discovered that the most basic layouts (such as the recompression cycle and basic combined ORC cycle) could reach thermal efficiency as high as 43.3%, which is comparable to efficiencies obtained through supercritical steam Rankine cycles. The simplest combined cycle, which achieved the highest efficiency, is presented in Fig. 4.

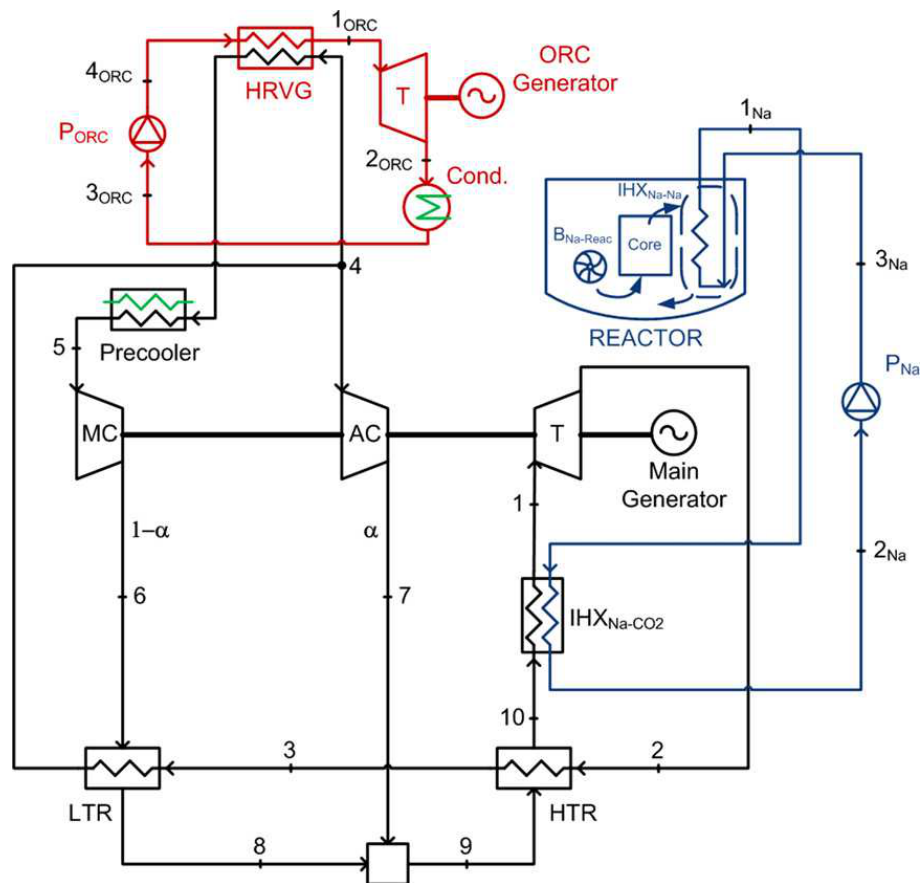


Figure 4 Simple sCO₂ORC cycle [12]

Harvego and McKellar [15] performed a comparative study of the direct and indirect recompression Brayton cycle coupled to a nuclear reactor. Both layouts were examined in the same conditions, i.e., operating Brayton cycle pressure of 20 MPa and reactor outlet temperature between 550°C and 850°C. The results of the analysis showed that, for the direct supercritical CO₂ power plant cycle, thermal efficiencies in the range of 40 to 50% could be achieved over the assumed reactor coolant outlet temperature. For the indirect supercritical power plant cycle, thermal efficiencies were approximately 11-13% lower than those obtained for the direct cycle over the same core outlet temperature range. In 2012 Halimi [14] conducted a computational analysis of the supercritical CO₂ Brayton cycle power conversion system for fusion reactor application. The analysis results showed that thermal efficiency of 42.44% was achievable for a recompression cycle. Additional 0.69% benefits can be obtained by adopting the reheating concept shown in Fig 5. Yoon et al. [16] suggested coupling a supercritical CO₂ cycle with small and medium sized water cooled nuclear reactors (SMR). According to the cycle evaluation, the maximum cycle efficiency at a temperature of 310°C and compressor outlet pressure of 22 MPa is 30.05%, which is comparable to the efficiency of current steam Rankine cycles. Moreover, the total volume of turbomachinery which can service 330 MWth of SMR is less than 1.4 m³ excluding the casing.

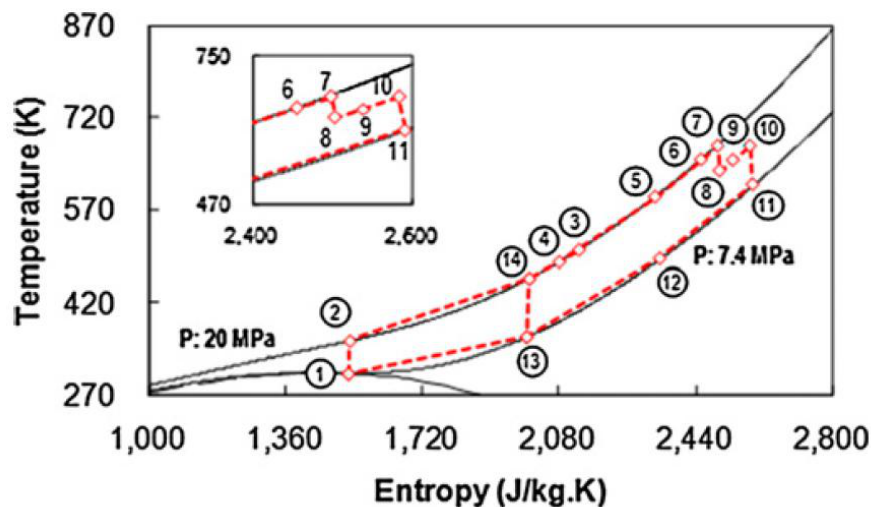


Figure 5 T-s diagram of the recompression cycle with reheating [14]

Besides the studies of the supercritical CO₂ cycle as a nuclear application, a number of analyses of these novel cycles coupled with Concentration Solar Power have been performed. Zhang and Yamaguchi conducted three successive semi-experimental studies using a real Rankine cycle with a relief valve as a counterpart of a turbine. They accomplished maximum CO₂ temperature of 165°C at the collector outlet [18],

which achieved theoretical electric output efficiencies of 11.4% [19] and 11.6% [20] in the next study. These efficiencies were slightly higher than those obtained by the solar cell used in the experiment for the purpose of comparison.

Fig. 6 shows a new type of solar energy based power generation using supercritical CO_2 and heat storage. The calculations performed showed that the supercritical CO_2 Rankine cycle not only achieves higher energy conversion efficiency than conventional water-based systems, but also overcomes the intermittent nature of solar energy. The paper also proved that the efficiency of the expander and the performance of the heat storage/regenerator have significant effects on the systems overall performance, while the pump is relatively unimportant [17].

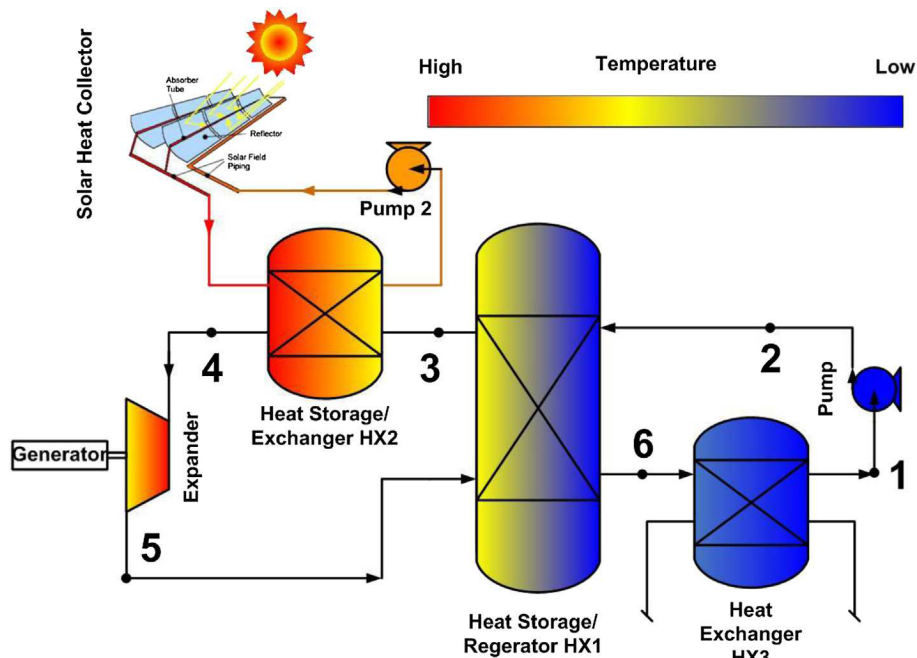


Figure 6 Schematic diagram of a solar energy storage and power generation system based on CO_2 [17]

Iverson and Cowboy in [21] supported the statement above, emphasizing good cycle efficiency especially over 600°C . They used an experimental loop installed in Sandia National Laboratories, which was the split flow supercritical CO_2 Brayton cycle shown in Fig. 7. The experiment showed good cycle behavior as a response to intermittent heat supply. Measurements of the system indicated an overall efficiency of approximately 5% for the operating conditions used in the experiment. However, the authors expected this efficiency to increase to 15% at design conditions and to approximately 24% with minor modification to improve insulation.

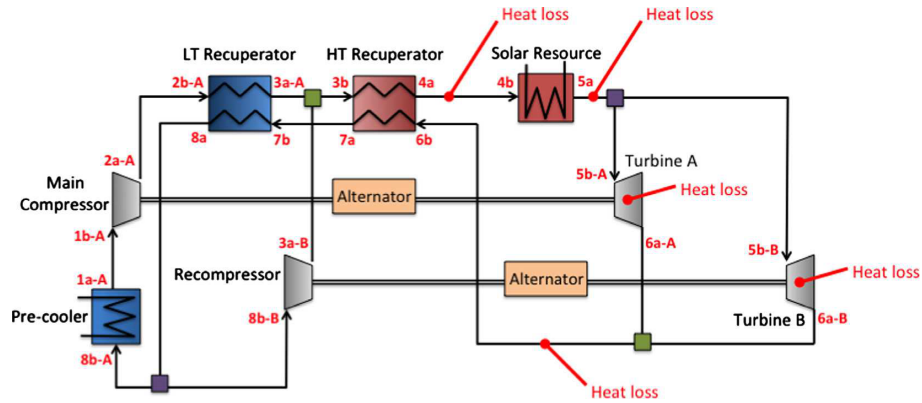


Figure 7 Layout of split-flow recompression Brayton cycle components [21]

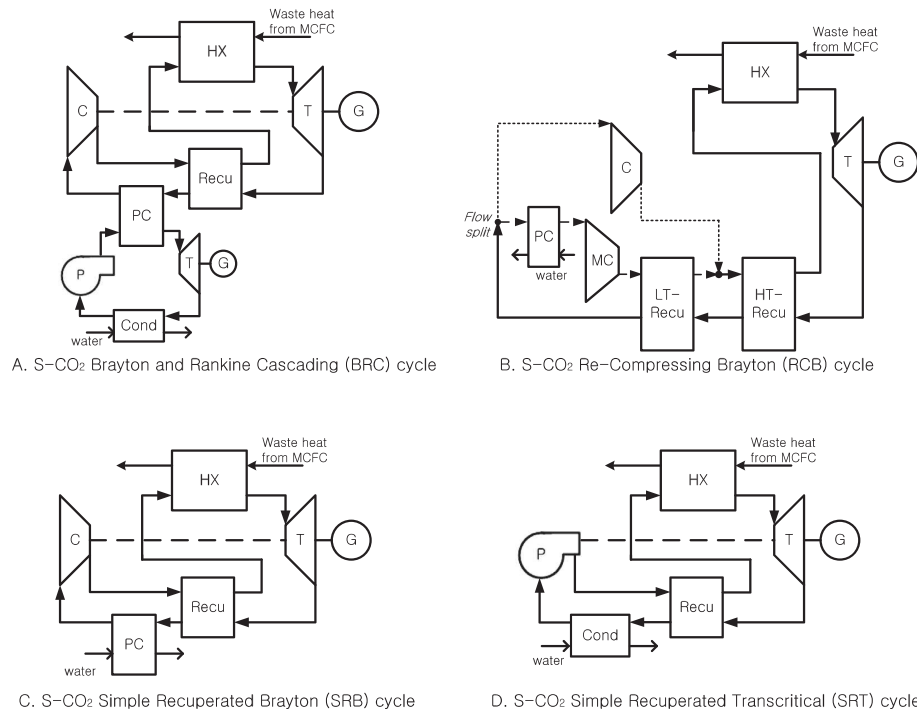


Figure 8 Various sCO₂ cycle layouts studied evaluated as a bottoming cycle for the MCFC hybrid system [27]

In 2015 Padilla et al. [22] analyzed the effect of turbine inlet temperatures and the cycle configuration on the thermal performance and exergy destruction of a supercritical CO₂ cycle within a CSP central receiver application. They found that the thermal efficiency of the supercritical Brayton cycle increases monotonically with the temperature of the cycle. The recompression cycle with main compressor inter-cooling achieved the best thermal performance (55.2% at 850°C). However, Cheang et al. [23] in their study of the same year argued that although the supercritical CO₂ cycle looks attractive, it is still both less efficient and less cost competitive than a superheated steam Rankine cycle.

The next area in which research has been made is a supercritical CO₂ cycle application as a bottoming cycle within fuel cells [24] systems. Snchez et al. in 2009 [25] expected a CO₂ cycle to perform better for intermediate temperature heat recovery applications than an air cycle. Their paper showed that, even though the new cycle is coupled with an atmospheric fuel cell, it is still able to achieve the same overall system efficiency and rated power than the best conventional cycles currently being considered. Furthermore, under certain operating conditions, the performance of the new hybrid systems beats that of existing pressurized fuel cell hybrid systems with conventional gas turbines. Calculations carried out by Muoz de Escalona [26] proved that an indirect supercritical CO₂ Brayton cycle coupled to a Molten Carbonate Fuel Cell (MCFC) can achieve thermal efficiency of almost 40%, which enables the whole system to approach overall efficiency of 60%. In addition, the supercritical CO₂ cycle performs better at part load than existing hybrid systems.

Bae et al. compared various cycle layouts presented in Fig. 8 in terms of application as an MCFC bottoming cycle. The results showed that all of the analyzed sCO₂ Brayton cycle layouts perform better than the air Brayton cycle [28], in particular the recompression Brayton, the cascading Brayton and the Rankine cycles can increase net hybrid system efficiency by over 10% more than the single MCFC system [27].

Another part of research concerns use of a supercritical CO₂ cycle in coal applications. Moullec [29] adopted a supercritical CO₂ Brayton cycle to the coal-fired boiler thermal output shown in Fig. 9. An energetic evaluation of the overall power plant indicated net power plant efficiency of 41.3% with carbon capture [30], and CO₂ compression to 110 bar. Moreover, a technical-economic analysis of a designed power plant showed a levelized cost of electricity (LCOE) reduction of 15% compared to a reference supercritical coal-fired power plant equipped with a standard carbon capture process. A further study showed that the oxy-combustion cycle seems the best fitted for the supercritical CO₂ Brayton cycle due to the simpler thermal integration and the CO₂ purification devices already integrated in the CO₂ processing unit. However, the main technological challenges were also identified, namely, the very large exchanger needed in the cycle in order to achieve high power cycle efficiency, and the development of a supercritical CO₂ turbine, which differs significantly from steam or gas turbines especially due to the very large effort on the wheel and the small size of the equipment [31].

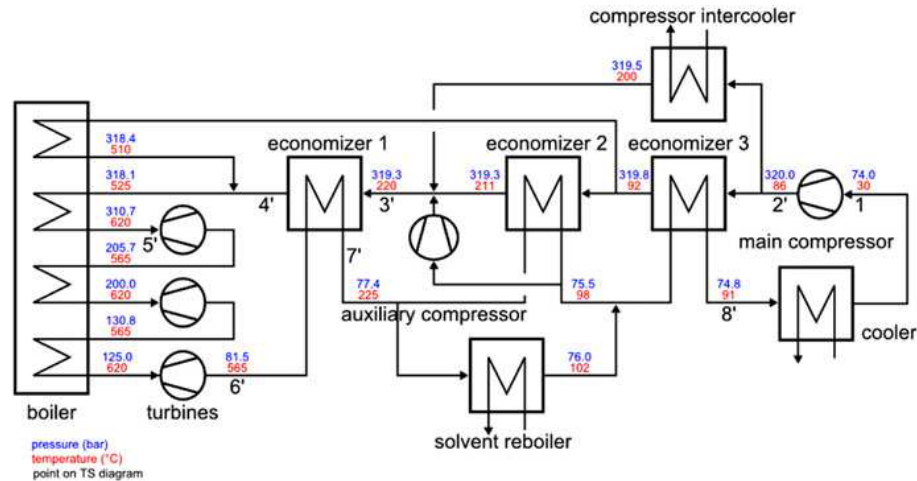


Figure 9 Supercritical Brayton CO₂ power cycle adapted for a coal-fired boiler with carbon capture [29]

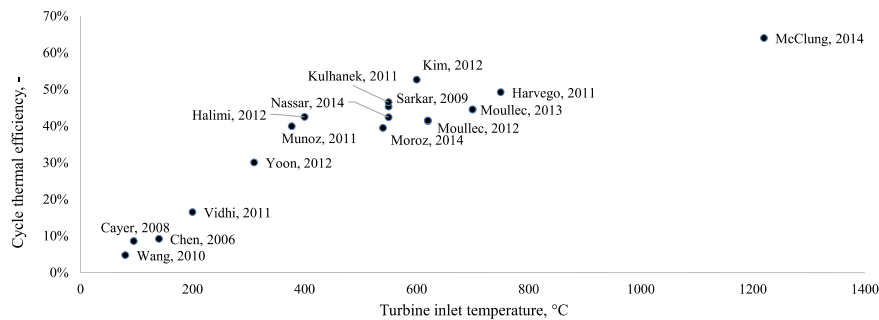


Figure 10 General correlation between cycle efficiency and turbine inlet temperature throughout different studies [2–23, 25–27, 29, 31–33]

Although some supercritical CO₂ cycles, such as the recompression cycle, exhibit high efficiency, they utilize a high degree of recuperation leading to a narrow change across the thermal input device. This narrow window may be acceptable for waste heat and nuclear applications, but it is not suitable for a traditional coal or natural gas fired system. McClung [32] proposed two cycles: Cryogenic Pressurized Oxy-Combustion (CPOC) and Advanced Supercritical Oxy-Combustion (ASOC). The calculations performed showed that, for both direct cycles, turbine inlet temperature of 1,220°C enables power block thermal efficiencies of near to 64% and overall power plant efficiency exceeding 52%. However, the CPOC cycle seems to be more attractive due to the wider thermal input window, which leads to simpler combustor designs and more efficient usage of fossil based thermal input.

Table 1 Brief performance review of supercritical CO₂ cycles [2–23, 25–27, 29, 31–33]

Year	Author	Efficiency, %	TIT, °C	Max P, MPa	Application
2006	Chen	9.2	140	16	waste heat
2007	Zhang	16.5	180		solar
2008	Cayer	8.6	95	13.6	waste heat
2009	Sarkar	45.3	550	20	general
2010	Wang	4.75	80	10.8	waste heat
2011	Munoz	40	377	21.6	MCFC
2011	Vidhi	16.5	200		waste heat
2011	Kulhanek	46.5	550		general
2011	Harvego	49.2	750	20	nuclear
2012	Yoon	30.1	310	22	nuclear
2012	Kim	52.6	600		general
2012	Moullec	41.3	620	30	coal with CCS
2012	Moullec	44.5	700		coal with CCS
2012	Halimi	42.4	400	20	nuclear
2013	Moullec	41.5	620		coal with CCS
2013	Moullec	44.5	700		coal with CCS
2014	Moroz	39.4	540	21	CHP
2014	McClung	64	1220	29	coal with CCS
2014	Bae	45			MCFC
2014	Bae	46			MCFC
2014	Nassar	42.4	550		general

The results of the most significant studies referenced above, are plotted in a co-ordinate system presented in Fig. 10, where the x and y axes correspond to turbine inlet temperature and cycle efficiency respectively. A positive correlation between these two parameters can be seen in the chart. More detailed information about the studies is presented in Table 1.

2. Supercritical CO₂ cycles classification

2.1. Operating temperature

In general all the supercritical carbon dioxide cycles proposed in the literature can be divided into two groups in terms of the level of operating temperature. The first group is represented by high-temperature cycles, which are usually designed for heat source temperatures of over 300°C. The second group consists of low-temperature cycles, designed for use mostly in waste heat applications, where the heat source temperature is under 300°C.

High temperature cycles are based on the Brayton supercritical carbon dioxide cycle, in which the working fluid operates entirely in the supercritical region. The loop consists of a compressor, a turbine, an intermediate heat exchanger as a heat source, a recuperator and a cooler used as a heat sink. Owing to higher operating temperatures, the cycle achieves good thermal efficiencies, ranging from 30 up to 64%. However, due to higher temperatures and pressures the loop components are more sophisticated and more expensive to manufacture.

The high temperature cycles are proposed mainly for nuclear and coal applications, and as a bottoming cycle within high temperature fuel cells [34].

The low temperature cycle works on the basis of the transcritical Rankine cycle in which the working fluid parameters are under the critical point in some part of the cycle. The loop consists of a turbine, an intermediate heat exchanger as a heat source, a recuperator, a pump and condenser instead of a compressor and cooler, which are used in the high temperature cycle. This type of cycle achieves lower efficiency (up to 18 %) [5] than the high temperature cycles; nonetheless, it is expected to be profitable as part of a concentrating solar power or waste heat application where large amounts of cheap heat are available.

2.2. Heat supply

The second possible division of carbon dioxide cycles can be made on the basis of the manner of heat supply. Both direct and indirect cycles performance has been evaluated in the literature.

In most proposed cycles heat is transferred to the cycle indirectly, through an intermediate heat exchanger. However, this manner of heat supply entails some negative consequences, such as exergy and energy losses in the intermediate heat exchanger. Thus, large and expensive devices have to be used in order to avoid losses, but there is still some efficiency potential resulting from the temperature drop across the exchanger [15].

A direct supercritical CO₂ cycle makes direct use of the heat produced in the heat source. Although the cycle in this concept is simplified, because it does not employ an intermediate heat exchanger, several other challenges emerge. In the case of fossil fuel applications the heat produced in the heat source is transported to a cycle by the combustion products, which also function as a working fluid. This requires the use of a special flue gas cleaning installation to remove particles and other impurities from combustion products in order to avoid turbine blade erosion and congestion of heat exchanger fine channels. In addition, when using coal as a fuel, a high pressure supercritical oxy-combustor and a high pressure coal feed are required [32].

2.3. Main cycle layouts

The most basic and compact supercritical CO₂ cycle is a simple Brayton cycle. It is simple and offers relatively good efficiency. However, there is still potential to improve its performance. The biggest reduction in efficiency of the supercritical Brayton cycle comes from the large irreversibility in the recuperator [2]. So called compound cycles have been introduced to overcome this problem and, as shown later on in this paper, these cycles perform significantly better than the regular supercritical Brayton cycle.

2.3.1. Pre-compression cycle

The pre-compression Brayton cycle is one of the ways to increase generation within the cycle and reduce the pinch-point problem. As shown in Fig. 11 the cycle is similar to the normal Brayton cycle with a small modification. First, the working fluid is compressed and then heated in the high temperature recuperator (1) using

exhaust heat from the turbine. The fluid passes to a heat source (2), where heat is added, and then expands in the turbine (3). The remaining exhaust heat is extracted from the fluid in the high temperature recuperator (1). The difference from the normal Brayton cycle is that in the middle of the recuperation process, when the hot fluid temperature approaches the heated fluid temperature, a compressor (5) is introduced that compresses the fluid to a higher pressure. As the fluid pressure rises, so does its temperature and specific heat. Thus, the regeneration process can continue and more available heat is returned to the heated fluid. This extra heat reduces the average temperature at which heat is rejected from the cycle, and increases the average temperature at which heat is added to the cycle. This results in an efficiency improvement of 6% over a Brayton cycle that would otherwise suffer from the pinch point problem [35].

2.3.2. *Partial cooling cycle*

Another cycle layout that aims at reducing Brayton cycle drawbacks is the partial cooling cycle presented in Fig. 12. In general, its operation differs from the previously described cycle in terms of two adjustments. The first is that only a fraction of the working fluid is compressed in the low temperature compressor (pump). The rest is compressed in the recompression compressor that is introduced before the pre-cooler and after the pre-compression compressor. The second difference is the introduction of another pre-cooler before the pre-compression compressor.

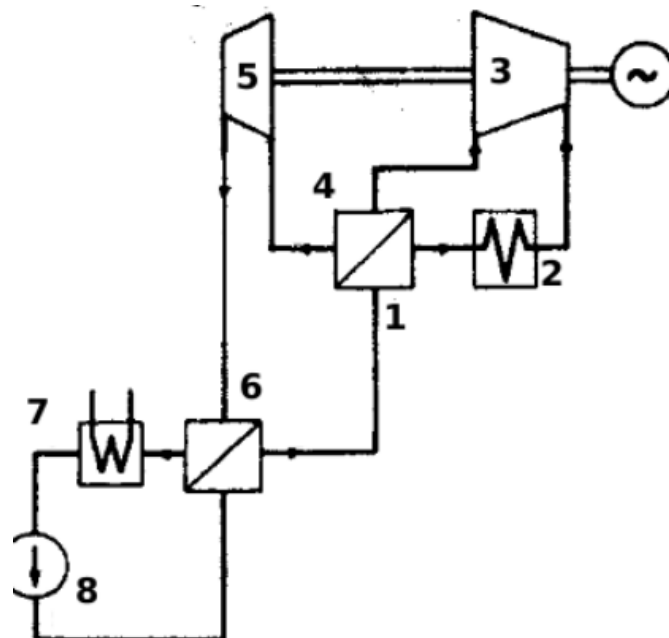


Figure 11 Schematic diagram of the pre-compression Brayton cycle [2]

This way, similar to the pre-compression cycle, more heat is available for the regeneration process.

After compression in the main compressor (1), a fraction of the working fluid is heated in the low temperature recuperator (2) and merged with the flow from the re-compressing compressors, which is at the same conditions. The fluid is then heated in the high temperature recuperator (3) and in the heat source (4) in turn and then enters the turbine (5). After the expansion process the fluid returns its heat in the high and low temperature recuperator (2,3). Then it passes to the pre-cooler (6) where it is cooled to the pre-compressor inlet temperature, and subsequently compressed in the pre-compressor (7). A part of the pre-compressed fluid is sent to the pre-cooler (8) and the main compressor. The rest is recompressed in the second recompressing compressor (9) to the high temperature recuperator inlet conditions, and then is merged with the stream from the main compressor. This move eliminates the pinch point problem, since due to the lower mass flow rate on the high pressure side of the low temperature recuperator, the mass flow weighted heat capacity of the streams is about equal and a pinch point does not occur.

The cycle improves its efficiency by reducing the average temperature of heat rejection so that the efficiency improvement is bigger than that for the pre-compression cycle.

2.3.3. *Recompression cycle*

Although the partial cooling cycle looks attractive due to its efficiency benefits, the complication of the cycle layout may prove detrimental to the economic outcome. Therefore, another cycle is introduced, a recompression cycle, which is simpler than both the partial cooling and pre-compression cycle. The general layout of the cycle is shown in Fig. 13.

The advantage of this cycle is that it completely eliminates one precooler and pre-compressing compressor from the cycle. After the regeneration process in the high temperature recuperator (3) the fluid is heated in the heat source (1) and passes to the turbine (2). Then it enters successively the high and low temperature recuperators (3,4) and returns its heat to the fluid on the high pressure side. The fluid flow is then split into two streams. The first is sent directly to the recompression compressor, where it is compressed to the same pressure conditions as the CO₂ leaving the main compressor and merged with it in the high pressure recuperator. The second flow is cooled in the precooler (5), compressed in the main compressor (6) and heated in the recuperators.

The effect of recompression is sufficient to overcome a pinch point problem. Owing to the decreased mass flow rate on the high pressure side of the low temperature recuperator, the mass flow weighted heat capacity of the streams is about equal on both sides and a pinch point does not occur.

The recompression cycle is, along with the pre-compression cycle, the simplest among the surveyed cycles. In addition, at the desired operating conditions of turbine inlet pressures and temperatures (20 MPa and 550°C), it achieves the highest efficiency of all examined cycles [2, 8].

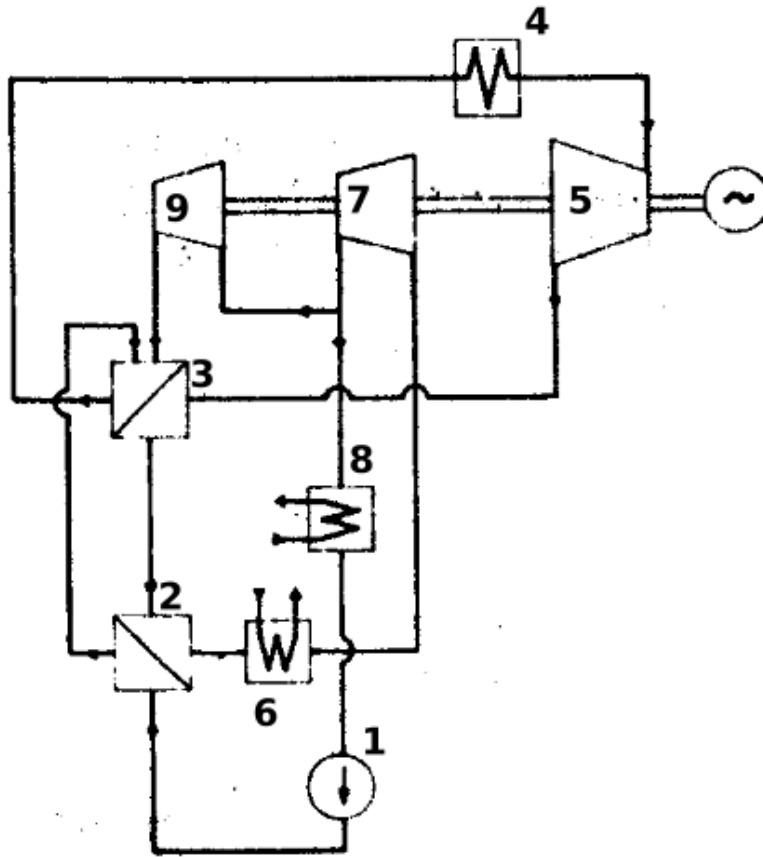


Figure 12 Schematic of the partial cooling Brayton cycle [2]

Therefore, the recompression cycle is usually selected as the best-suited cycle and investigated with respect to various applications in the literature.

3. Conclusions

This paper presents a short review of supercritical carbon dioxide based gas turbine cycles. Several configurations covered in the available literature were presented, commented on and compared. The parameters of the cycles, such as operating temperature (80800°C), pressure (74290 bar), and heat supply are noted and commented. Based on the analysis presented, ultra high efficiency can be expected (60% with TIT = 1,220°C). The reported rotary equipment efficiency is quite high for the small size of the turbomachinery, reaching 87% for the expander and 70% for the compressor. The small size of the turbomachinery requires elevated rotary speeds of up to 69,000 rpm.

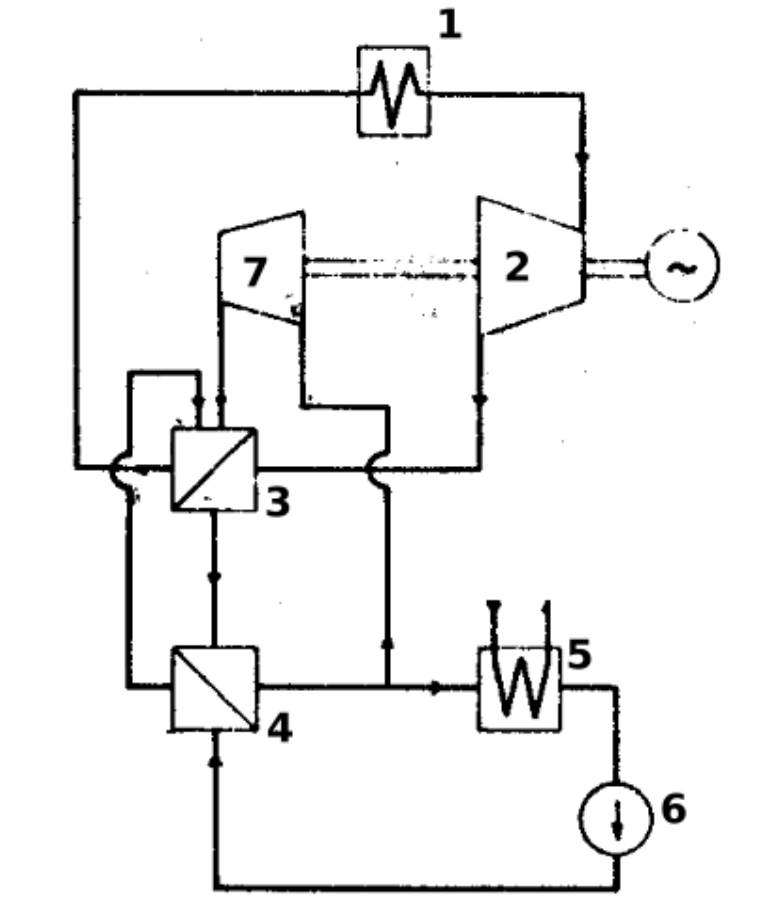


Figure 13 Schematic diagram of the recompression Brayton cycle [2]

There are three main configuration considered in the available literature: the pre-compression cycle, partial cooling cycle, and recompression cycle. Future applications of those systems are to be expected mainly in nuclear power plants and concentrated solar power generation applications.

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