Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 123 (2017) 305–312

Energy Procedia

www.elsevier.com/locate/procedia

1st International Conference on Sustainable Energy and Resource Use in Food Chains, ICSEF 2017, 19-20 April 2017, Berkshire, UK

# Techno-economic comparison of different cycle architectures for high temperature waste heat to power conversion systems using $CO_2$ in supercritical phase

Matteo Marchionni, Giuseppe Bianchi*, Konstantinos M. Tsamos, Savvas A. Tassou

*Brunel University London, Institute of Energy Futures, Center for Sustainable Energy Use in Food Chains, Uxbridge, Middlesex UB8 3PH, UK*

**Abstract**

Bottoming thermodynamic systems based on supercritical carbon dioxide as working fluid ($sCO_2$) are a promising technology to tackle the waste heat to power conversion at high temperature levels and that might outperform the conventional power units based on Organic Rankine Cycles. In fact, $CO_2$ is an inexpensive, non-toxic, non-flammable, thermally stable and eco-friendly compound. Moreover, $CO_2$ in its supercritical state shows an extreme increase in density that allows turbomachinery downsizing and a high cycle efficiency due to the reduced work required by the compression stage. In addition, supercritical $CO_2$ permits a better temperature glide matching within the heat source which increases the overall efficiency of waste heat utilization. With the aim of identifying pro and cons of different $sCO_2$ cycle layouts, this paper investigated four design Joule-Brayton configurations at increasing complexity: simple regenerative, with recompression, with reheating and with recompression and reheating. The research methodology is based on 1$^{st}$ and 2$^{nd}$ laws thermodynamic analyses and includes correlations to estimate the investment costs of the equipment. With reference to a high temperature industrial waste heat source, performance, costs and exergy losses in the different cycle layouts are compared. Furthermore, a parametric analysis regarding the effects of the cycle pressure ratio on net power output and back work ratio is carried out.



*Keywords:* supercritical $CO_2$; waste heat recovery; power generation; thermodynamic analysis; exergy analysis

* Corresponding author. Tel.: +44-1895-267707; fax: +44-1895-269777.
*E-mail address:* giuseppe.bianchi@brunel.ac.uk

1876-6102 
10.1016/j.egypro.2017.07.253

## 1. Introduction

Among the Waste Heat Recovery (WHR) technologies, the heat to power conversion systems appear as one of the most promising solutions since they allow a flexible reuse of the recovered heat and they are economically convenient in several industrial scenarios. However, the temperature range at which the heat is rejected affects the choice of the heat to power conversion technology. For mid-grade heat, in a range of temperatures between 100 and 370 °C, it has been proven that the Organic Rankine Cycle (ORC) systems are the most suitable solution [1], and they are already available on the market. On the other hand, at higher temperature levels (370 up to 900 °C) the ORC technology is still an option, but it is less attractive because of the lower chemical stability and high flammability of the working fluids to employ in such applications [2]. Moreover, conventional Rankine Cycle power plants achieve poor efficiencies for hot source temperatures lower than 700 °C [3] and generally they present high capital and maintenance costs. Therefore, in this temperature range, a promising alternative is represented by the use of the supercritical carbon dioxide ($sCO_2$) as a working fluid to convert heat into power by realizing a Joule-Brayton cycle.

The $sCO_2$ Brayton power cycle was proposed for the first time in [4] and reintroduced for nuclear power generation applications in [5]. The main benefit of this technology derives from the particular chemical and physical properties that $CO_2$ assumes in the supercritical state, and in particular near the critical point (30.98 °C, 7.38 MPa), in which $CO_2$ presents a very high density, isobaric thermal capacity and isothermal compressibility. These properties allow to reduce substantially the mechanical compression work supplied to the fluid, and consequently, to increase the net power output and then the overall thermal cycle efficiency [6].

As concerns WHR applications, the attractiveness of this technology, as before mentioned, increases when the heat is rejected at temperatures which are above the 400 °C. In fact, in this range the $sCO_2$ Brayton cycle presents numerous benefits respect to the ORC or the conventional Rankine Cycle systems. The $CO_2$ is in fact is not flammable and more chemically stable at higher temperatures compared with the organic fluids used in ORC power unit [7]. Moreover, it is less expensive and more eco-friendly, since is a non-toxic compound and has a lower Global Warming and Ozone Depletion Potentials than organic fluids. Compared with the Rankine Cycle power cycle instead, the $sCO_2$ Brayton power cycle allows to achieve better efficiency at lower temperatures [8] and it presents lower CApital and OPerational EXpenditures (CAPEX and OPEX) since the use of supercritical carbon dioxide, extremely denser than steam, allows to have consistently downsized and simpler components which also require less maintenance [9]. Furthermore, with respect to both technologies, the $sCO_2$ Brayton cycle allows a better thermal matching between the working fluid and the hot source and consequently a higher 2$^{nd}$ law (exergy) efficiency [10].

In recent years the $sCO_2$ technology has been intensively studied for several purposes and applications, such as Concentrated Solar Power (CSP) [11], geothermal [12], fossil sources [13], WHR [14] and especially the nuclear sector [15], [16]. Several works have also been carried out in the optimization of the power cycle [17], in the study and dynamic modelling of each component [18], in the analysis of performance of the system coupled with some other bottoming power unit [19], in the characterization of the cycle performance when mixture of $CO_2$ and other fluids were used [20]. Another relevant topic, which has not been fully addressed yet, is the analysis of the optimal $sCO_2$ Brayton power cycle scheme for WHR applications. In [21] the authors analyzed the recompression configuration or Feher cycle and how the introduction of reheating and intercooling affects its efficiency and net power output. In [22] the simple Brayton cycle has been compared to the recompression, pre-compression, split expansion, partial cooling and partial cooling with improved regeneration cycle. From the results did not emerge a privileged scheme, since the performance of each configuration are too much affected by the particular operating conditions. Even more complex configurations are reported in [23] and in [24] several schemes are presented to maximize the heat recovered and to achieve the best thermal matching inside the $sCO_2$ regenerators.

The current research work presents an assessment of the theoretical capabilities of four $sCO_2$ cycle layouts as well as a preliminary estimation of the investment costs using literature correlations for heat exchangers and turbomachinery. The design configurations are compared with reference to the same thermal power recovery and in terms of performance, exergy losses and investment costs. Furthermore, the effect of cycle pressure ratio on the net power output in every cycle layout is investigated.

## 2. Methodology

### 2.1. Thermodynamic analysis

Four Joule-Brayton cycle configurations at increasing complexity have been investigated to assess the best performing one. The cycle layouts and the corresponding entropy diagrams are reported in Figure 1. With respect to the simple regenerative configuration (SR – Fig. 1.a), the one with recompression (RC – Fig. 1.b) splits the flow upstream the condenser and then compresses the two contributions in two different machines. On the other hand, in the re-heating one (RH – Fig. 1.c), after the first expansion the working fluid is heated again with the hot source and eventually expanded up to the lowest pressure of the cycle. The configuration shown Fig. 1.d eventually embeds both the recompression and reheating solutions (RCRH).

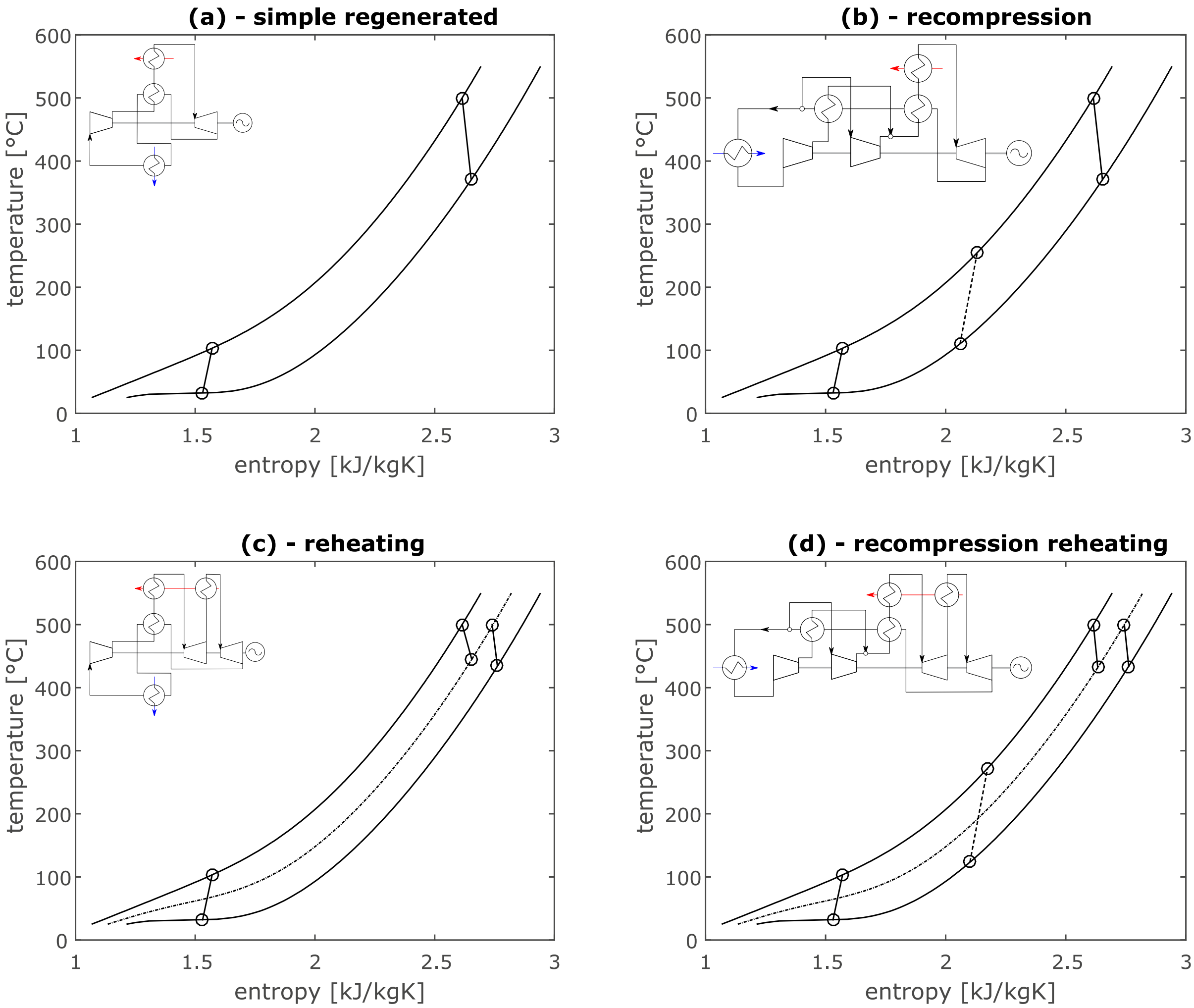


Fig. 1. – Investigated $sCO_2$ cycle configurations

The different cycle architectures were investigated using the same methodology. In particular, steady state energy and exergy analysis were carried out using the software CycleTempo. Thermodynamic properties were calculated using the NIST database coupled with the thermodynamic solver. Parametric analysis were eventually carried out through a coupling of the CycleTempo models with Matlab.

Regardless of the cycle layout considered, the governing equations for a steady state analysis are mass and energy balances. Hence, with reference to a linear system composed of these kinds of equations, CycleTempo solves them to provide the values of mass flow rate in each pipe of the system, whose number is equal to the number of components. Afterwards pressure, enthalpy and temperature values are calculated while exergy analysis is the latest step of the procedure.

### 2.2. Cost analysis

After the thermodynamic analysis, the several layouts proposed have been investigated also from an economic perspective. The aim of the study was to calculate the CAPEX per $kW_e$ of each $sCO_2$ cycle layout, in order to define the best architecture in terms of payback time and return of investment for WHR applications. Regarding the turbomachines, their cost has been estimated as a function of the $sCO_2$ mass flow processed ($m$), their isentropic efficiency ($\eta$)and the cycle pressure ratio ($\beta$). The correlations used have been proposed by [25] and they are reported in the Eqns. (1) and (2).

$$c_T = 479.34m\left(\frac{1}{0.93-\eta_T}\right)\ln(\beta)\left(1+\exp\left(0.036T_{in}-54.4\right)\right) \tag{1}$$

$$c_C = 71.10m\left(\frac{1}{0.92-\eta_C}\right)\beta\ln(\beta) \tag{2}$$

in which the mass flow rates are expressed in kg/s, while the inlet turbine temperature ($T_{in}$) is in Celsius degrees.

As concerns the heat exchangers, their cost has been estimated as a function of their that has been estimated from the thermodynamic results and the global heat transfer coefficients reported in Table 1. These orders of magnitudes derive from the know-how gained by the Authors during the ongoing design of a $sCO_2$ test rig at Brunel University. For the same reasons, with reference to real quotations for Printed Circuit Heat Exchangers (PCHE) and for flue gas to $sCO_2$ heaters, the correlations proposed by [25], [26], have been corrected using the coefficients reported in Table 1. In particular, Eqn. 4 is valid for flue gas to $sCO_2$ heaters while Eqn. 3 refers to PCHE working as recuperator and cooler. In both cases the heat transfer surface ($A$) is expressed in $m^2$.

$$c_{HX} = k\;2681A^{0.59} \tag{3}$$

$$c_{HX} = k\;130\left(\frac{A}{0.093}\right)^{0.78} \tag{4}$$

Table 1. Additional calculation parameters

| | Global heat transfer coefficient ($W/m^2K$) | Corrective coefficient k |
|---|---|---|
| Heater (flue gas-$CO_2$) | 100 | 7.0 |
| Recuperator ($CO_2$-$CO_2$) | 1700 | 1.8 |
| Cooler($CO_2$-water) | 2900 | 8.0 |

## 3. Results and discussion

To properly compare the techno-economic performances of the different $sCO_2$ cycle layouts, a reference operating point was considered. As concerns the hot and the cold sources, inlet and outlet temperatures have been set according to typical values for Medium to High Grade WHR applications. Regarding the turbomachines, isentropic efficiency and inlet temperature have been imposed. In particular, the inlet pressure and temperature of the compressor have been set slightly above the fluid critical point, in order to benefit of the advantageous physical properties of $sCO_2$ at these conditions avoiding the dynamic instabilities occurring during the fluid critical transition [27]. On the other hand, the turbine inlet temperature and pressure have been chosen according to the limits imposed by material strength and corrosion resistance [28]. Finally, pinch point temperatures of 5K have been set in the recuperators. Quantitative values are reported in Tables 2 and 3. Reference thermodynamic conditions for exergy calculations were 1 atm and 25°C.

Table 2. Hot and cold source input data.

| | Inlet Temperature [°C] | Outlet Temperature [°C] | Mass flow rate [kg/s] |
|---|---|---|---|
| Hot source – flue gas | 900 | 500 | 1 |
| Cold source – water | 15 | 45 | not fixed |

Table 3. Turbomachinery input data.

| | Compressor | Turbine |
|---|---|---|
| Inlet temperature [°C] | 32 | 500 |
| Inlet pressure [bar] | 75 | 250 |
| Isentropic efficiency | 0.70 | 0.85 |
| Mechanical efficiency | 0.98 | |
| Electric efficiency | 0.95 | |

The results of the comparison are presented in Figure 2. In particular, Figure 2.a and 2.b report the electric net power output and the thermal efficiency for each different cycle layout respectively. These charts show that the reheating (RH) and the recompression reheating (RCRH) configurations can reach the highest power output and thermal efficiency: 167.7 kWe with the 35% of efficiency for the RH layout and 174.9 kWe with an efficiency of nearly 38% for the RCRH configuration. These layouts also achieve the highest exergy efficiency (Figure 2.d), which in this case is slightly higher for the RH configuration (30.5% against the 30.0% of the RCRH).

A general remark that applies to all the investigated cycle layouts is the fact that, according to Figure 2.c, more than 55% of the total exergy losses is lost as sensible heat because, for the assumptions that have been made, flue gas is exhausted at 500°C to prevent crossing temperature profiles at the heater. Nevertheless, from a WHR perspective, this sensible heat loss has a crucial relevance and must be recovered either through a secondary ORC system or using more complex $sCO_2$ architectures. Apart from the sensible exergy loss, Figure 2.c shows that RC and RCRH have the lowest irreversibility due to plant equipment. The reason can be addressed to the splitting of the regeneration stage, which allows to achieve a better thermal matching between the hot and the cold $sCO_2$ flows in the regenerative heat exchangers. In fact, in the RC and the RCRH layouts the exergy destruction in the high and low temperature regenerators is equal respectively to 2.3% and 2.8% of the total exergy flow entering in the system, while in the RH and SR layouts is equal to 7.9% and 9.8%. The exergy destructions in the recuperators however, are secondary compared with the ones realizing in the heaters (in red and purple color), which, depending on the layout considered, range from 13% to 16% of the entire inlet exergy flow and so represent the most relevant exergy losses between the all plant components.

Regarding the cost analysis, even if the RH and the RCRH layouts achieve higher performances, they also present, due to the increased plant complexity, a higher CAPEX per unit of electric power (Figure 2.f) compared to the Recompression (RC) and the Simple Regeneration (SR) layouts, which have an investment cost respectively of 1175 \$/kWe and 862.5 \$/kWe against the 1675 \$/kWe and the 1425 \$/kWe for the RHRC and the RH configurations.

Focusing instead on the single component cost analysis (Figure 2.e), it is immediate to notice that the heat exchangers constitute the most relevant cost of the $sCO_2$ power generation units and this should be adequately taken in account for the decision of the best cycle layout to adopt in the early design stage.

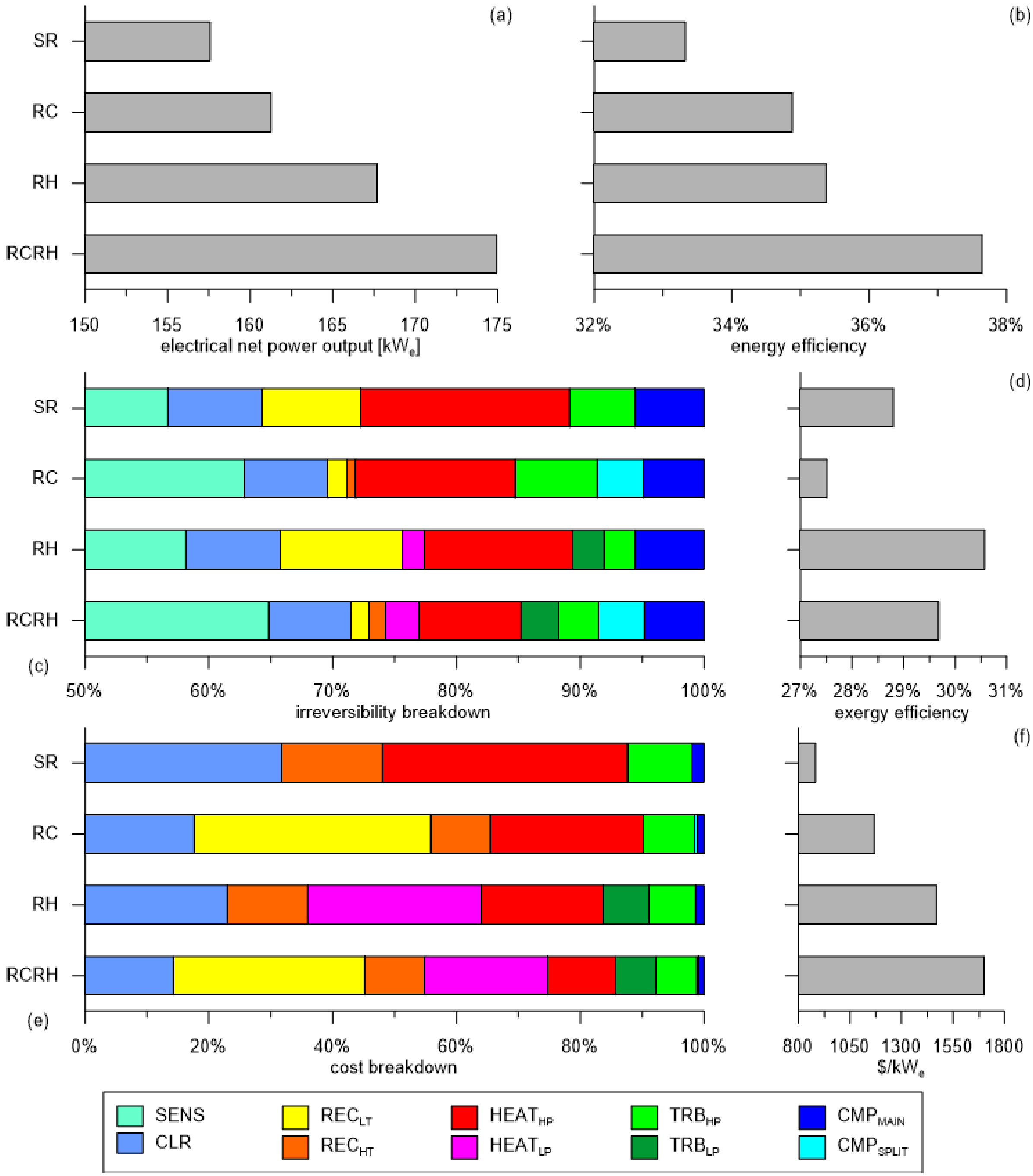


Fig. 2. Comparison of different $sCO_2$ cycle architectures: net power output (a), 1st law efficiency (b), irreversibility breakdown (c), 2nd law efficiency (d), cost breakdown (e) and unitary cost (f)

After this techno-economic analysis, a parametric study has been carried out in order to evaluate how the cycle pressure ratio affects the different layouts performance. The other cycle parameters have been fixed accordingly to the data reported in Table 2 and 3. The results are reported instead in Figure 3.a, which shows that almost all the cycle configurations generate a higher net electric power when the cycle pressure ratio is increased. Only the RC layout presents a maximum power output of 167.4 kWe for a pressure ratio equal to 2.6, overcame this value, the power produced decreases when the pressure ratio is further increased. Also the back work ratio of each layout is reported as function of the cycle pressure ratio in Figure 3.b, which shows that the relation between these two parameters is practically linear. Moreover, in the SR and in the RH schemes, for the maximum cycle pressure ratio of 3.3 only the 35% of the turbine power produced is used to drive the compressor.

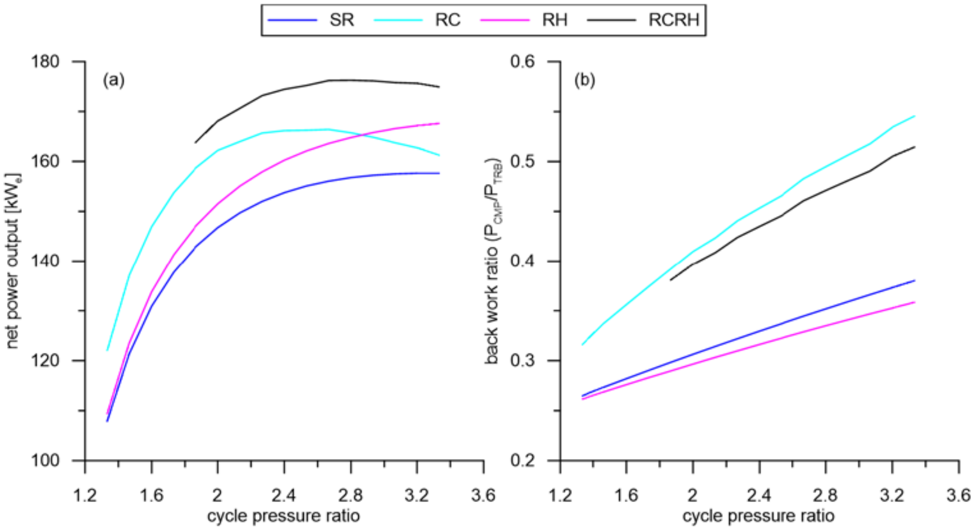


Fig. 3. Parametric analysis and comparison between different cycle layouts: net power output (a) and back work ratio (b) expressed as a function of the cycle pressure ratio.

## 4. Conclusions

In this work the authors presented a techno-economic comparison between several $sCO_2$ Joule-Brayton power cycle schemes addressed for WHR applications. For each of this cycle configuration the net electrical power output, the CAPEX, and the thermal and exergy efficiency have been calculated, from an overall system and a single component perspective. The results showed that the most performant layouts were also the ones characterized by the higher cycle complexity. Furthermore, the economic analysis pointed out that a relevant increase of cycle complexity, and thus of the power plant investment cost, was related to only a small increase of the system performance, both in terms of net power output, thermal efficiency and irreversibility reduction. A prove of that can be found recalling the cost data calculated and referred to the simpler cycle scheme, the Simple Regenerated (SR) layout. In fact, its unitary investment cost per electric power unit is equal to 862.5 $/kWe, which represents respectively the 73.4 %, 60.5 % and 51.5 % of the one related to the Recompression (RC), Reheating (RH) and Recompression Reheating (RCRH) configuration, against a 2.33%, 6.42% and 11% of net power output increase. Then what emerges is that the complex cycle layouts analysed in these work, studied at the beginning for CSP and nuclear power generation plants, are not suitable for WHR applications. For this destinations in fact, the thermal and exergy efficiency of the power generation plants are secondary aspects compared to the net power output produced and thus the amount of heat recovered. The energy supplied to the plant is indeed "free", and will be rejected in the environment if not properly exploited. Moreover, in these kind of systems, of main relevance is the investment payback time, since the power conversion unit is an earn-as-you-save investment, which in turn for the most complex and performant layout (RCRH) is almost the double of the SR configuration.

In conclusion, between the layouts analysed, the conventional simple regenerated $sCO_2$ power cycle appears to be the most suitable alternative for medium to high WHR applications, since its lower initial investment and acceptable performances. Nevertheless, since in all the schemes the sensible exergy rejected is a significant part of the inlet exergy flow, further efforts should be addressed to found alternative and innovative cycle arrangements which will allow to convert more waste heat into electric power and thus a "better use" of the wasted hot source.

## Acknowledgements

The research presented in this paper has received funding from the European Union's Horizon 2020 research and innovation programme under grant agreement No. 680599.

Aspects of the work are also funded by the Centre for Sustainable Energy Use in Food Chains (CSEF). CSEF is an End Use Energy Demand Centre funded by the Research Councils UK, Grant No: EP/K011820/1.

## References

[1] T. Hung, "Waste heat recovery of organic Rankine cycle using dry fluids," Energy Convers. Manag., 2001.
[2] B. F. Tchanche, G. Lambrinos, A. Frangoudakis, and G. Papadakis, "2011 - Tchanche - Low-grade heat conversion into power using organic Rankine cycles A review of various applications," Renew. Sustain. Energy Rev., vol. 15, pp. 3963–3979, 2011.
[3] C. Li and H. Wang, "Power cycles for waste heat recovery from medium to high temperature flue gas sources – from a view of thermodynamic optimization," Appl. Energy, vol. 180, pp. 707–721, Oct. 2016.
[4] G. Sulzer, "Verfahren zur Erzeugung von Arbeit aus Warme," Swiss Pat., 1950.
[5] V. Dostal, M. Driscoll, and P. Hejzlar, "A supercritical carbon dioxide cycle for next generation nuclear reactors," 2004.
[6] S. Wright, P. Pickard, and R. Fuller, "Supercritical $CO_2$ Brayton Cycle Power Generation Development Program and Initial Test Results," ASME 2009, 2009.
[7] P. Freund, S. Bachu, D. Simbeck, and K. Thambimuthu, "Annex I: Properties of CO 2 and carbon-based fuels," IPCC Spec. Rep., 2005.
[8] M. Persichilli, A. Kacludis, and E. Zdankiewicz, "Supercritical $CO_2$ Power Cycle Developments and Commercialization: Why $sCO_2$ can Displace Steam Ste," Power-Gen India &, 2012.
[9] C. Parks, "Corrosion of candidate high temperature alloys in supercritical carbon dioxide," 2013.
[10] J. Sarkar, "Second law analysis of supercritical CO 2 recompression Brayton cycle," Energy, 2009.
[11] C. Turchi, Z. Ma, and J. Dyreby, "Supercritical carbon dioxide power cycle configurations for use in concentrating solar power systems," ASME Turbo, 2012.
[12] Sabau, H. Yin, A. Qualls, and J. McFarlane, "Investigations of supercritical $CO_2$ Rankine cycles for geothermal power plants," 2011.
[13] G. Johnson and M. McDowell, "Issues Associated With Coupling Supercritical $CO_2$ Power Cycles to Nuclear, Solar and Fossil Fuel Heat Sources," Proc. Supercrit. $CO_2$ Power Cycle, 2009.
[14] M. Lehar and V. Michelassi, "System and method for recovery of waste heat from dual heat sources," US Pat. 9,038,391, 2015.
[15] Y. Ahn, S. Bae, M. Kim, S. Cho, S. Baik, and J. Lee, "Cycle layout studies of S-$CO_2$ cycle for the next generation nuclear system application," Trans., 2014.
[16] D. Vitale Di Maio, A. Boccitto, and G. Caruso, "Supercritical carbon dioxide applications for energy conversion systems," Energy Procedia, vol. 82, pp. 819–824, 2015.
[17] Q. Deng, D. Wang, H. Zhao, W. Huang, and S. Shao, "Study on performances of supercritical CO 2 recompression Brayton cycles with multi-objective optimization," Appl. Therm., 2016.
[18] Moisseytsev and J. Sienicki, "Recent Developments in s-$CO_2$ Cycle Dynamic Modeling and Analysis at ANL," Supercrit. $CO_2$ Power Cycles, Pittsburgh, …, 2014.
[19] X. R. Zhang, H. Yamaguchi, D. Uneno, K. Fujima, M. Enomoto, and N. Sawada, "Analysis of a novel solar energy-powered Rankine cycle for combined power and heat generation using supercritical carbon dioxide," Renew. Energy, vol. 31, no. 12, pp. 1839–1854, 2006.
[20] S. N. Laboratories, "Operation of a Supercritical Fluid Compression Loop Using $CO_2$-Based Mixtures," no. 3, pp. 2–3.
[21] Moisseytsev and J. J. Sienicki, "Investigation of alternative layouts for the supercritical carbon dioxide Brayton cycle for a sodium-cooled fast reactor," Nucl. Eng. Des., vol. 239, no. 7, pp. 1362–1371, 2009.
[22] M. Kulhánek and V. Dostál, "Supercritical Carbon Dioxide Cycles Thermodynamic Analysis and Comparison," Proc. $SCCO_2$ Power Cycle Symp., pp. 1–12, 2011.
[23] Y. Ahn, S. J. Bae, M. Kim, S. K. Cho, S. Baik, J. I. Lee, and J. E. Cha, "Review of supercritical $CO_2$ power cycle technology and current status of research and development," Nucl. Eng. Technol., vol. 47, no. 6, pp. 647–661, 2015.
[24] G. Kimzey, "Development of a Brayton Bottoming Cycle using Supercritical Carbon Dioxide as the Working Fluid," EPRI Rep., pp. 1–31, 2012.
[25] X. Wang and Y. Dai, "Exergoeconomic analysis of utilizing the transcritical $CO_2$ cycle and the ORC for a recompression supercritical $CO_2$ cycle waste heat recovery: A comparative study," Appl. Energy, vol. 170, pp. 193–207, 2016.
[26] Arsalis and M. von Spakovsky, "Thermoeconomic Modeling and Parametric Study of Hybrid Solid Oxide Fuel Cell-Gas Turbine-Steam Turbine Power Plants Ranging From 1.5 MWeto10MWe," J. fuel, 2009.
[27] Shen and P. Zhang, "An overview of heat transfer near the liquid–gas critical point under the influence of the piston effect: Phenomena and theory," Int. J. Therm. Sci., 2013.
[28] K. Sridharan, M. Program, G. E. N. I. V Collaborators, G. Iv, F. S. Nf, and S. Co, "Corrosion in Supercritical Carbon Dioxide : Materials , Environmental Purity , Surface Treatments , and Flow Issues," no. 10, pp. 1–2.