David Tomás Sánchez Martínez
Lourdes García Rodríguez
(coordinadores)

Proceedings of the 7th International Seminar on ORC Power Systems

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c/ Porvenir, 27 - 41013 Sevilla.
Tlfs.: 954 487 447; 954 487 451; Fax: 954 487 443
Correo electrónico: info-eus@us.es
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EVALUATION OF THE PERFORMANCE OF MULTIPLE SUPERCRITICAL CO2 POWER CYCLES IN WASTE HEAT RECOVERY APPLICATIONS

Hicham A. Chibli*, Matthew Read, Abdulnaser Sayma

Thermo-Fluids Research Centre, School of Science and Technology, City, University of London, Northampton Square, London, EC1V 0HB, United Kingdom

*Corresponding Author: hicham.chibli@city.ac.uk

ABSTRACT

This paper aims to systematically evaluate the common supercritical carbon dioxide power cycles that are suitable for low to high grade waste heat recovery applications, with source temperatures ranging from 450 to 850 °C. A number of cycles varying from the simple to the fully cascaded split cycles are investigated against a predefined waste heat source for multiple temperatures and flow rates with a target power net yield of the order of 1 MWe. The conditions at the inlet to the main compressor in the sCO_2 loop are also varied to quantify the role of the environmental heat sink in effective heat recovery. The sCO_2 cycle parameters including the mass flowrate and split ratio as well as the effectiveness of the individual heat exchangers are optimized for multiple pressure ratios. Additionally, a detailed exergy analysis is conducted to identify the sources of entropy generation in the various components and enable a detailed comparison of the various cycle configurations reviewed. Improving the quality of the heat source from 450 to 850 °C yielded ~30% increase in cycle performance, while degrading the heat sink quality from 15 to 35 °C reduced the net output power by ~30%. The cascaded and split cycles were determined to achieve the highest performance across all conditions considered.

1 INTRODUCTION

The ever-rising cost of fossil fuels and increasing demand for power generation, have incentivized the implementation of bottoming cycles to extract an improved overall system performance. The elevated density and heat capacity of sCO_2 compared to air allow for an increased efficiency at a reduced footprint, without the limitations of a condensing steam cycle. Consequently, interest in designing and building power cycles that use sCO_2 as a working fluid surged in the last few decades. Feher (1968) pioneered the research in this field, when he proposed a pseudo-supercritical cycle that alleviates the shortcomings of the readily popular Rankine and Brayton cycles. While the former features heat addition and removal at constant temperatures, leading to an improved cycle efficiency, it suffered from a limited temperature range and excessive wetness formation in the low pressure turbine stages. The Brayton cycle allows a wider range of working temperatures, and better use of heat recuperation technologies, at the cost of notably exaggerated compression work intrinsic to the gaseous nature of air. Feher recognized that the abundance, chemical stability and inert nature of CO_2, in addition to its low critical pressure and temperature and high density all contribute to its suitability for use in modern power cycles. Angelino (1968) independently researched the use of CO_2 in recuperated transcritical power cycles to achieve the low compression work of condensing steam cycles, while taking advantage of the elevated turbine inlet temperatures of closed Brayton cycles. Additionally, he identified the recovered heat from the low to high pressure side of the cycle as the largest source of internal irreversibility and devised a series of modifications involving flow splitting to reduce losses and improve performance.

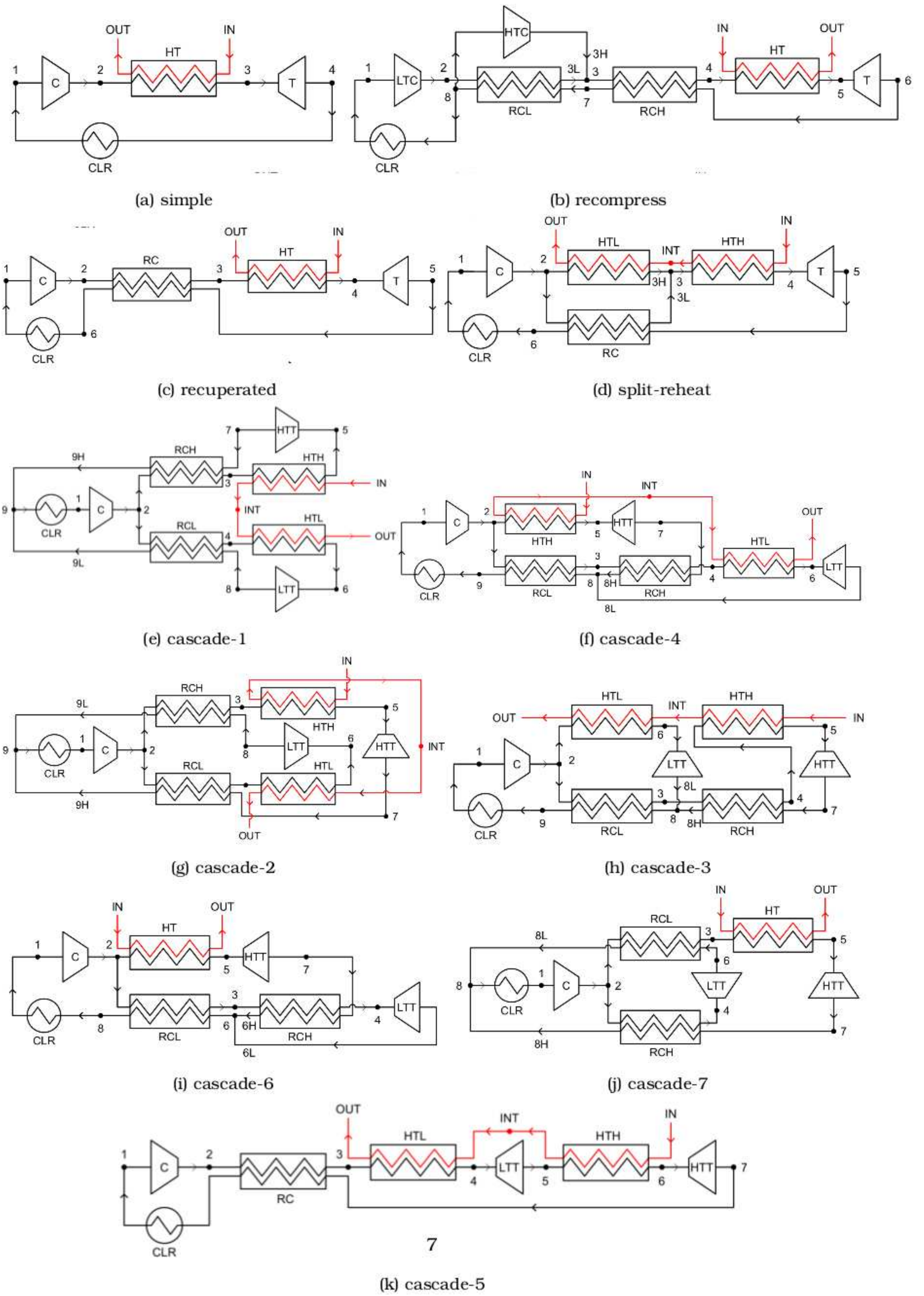

Figure 1: Schematics of the various waste heat recovery cycles considered in this study

Numerous studies have since been conducted, especially in the last decade to evaluate the implementation of sCO_2 cycles in applications ranging from nuclear and concentrated solar power to waste heat recovery. Ahn et al. (2015) reviewed the current status of research and development in Gen IV nuclear reactors, where the application of sCO_2 cycles offers several advantages over conventional water-cooled technology. Increasing the turbine inlet temperatures from the current $300\ °C$ to $500 - 900\ °C$, while maintaining low compression work, at reduced size turbine stages owing to the low specific volume of sCO_2, leads to significant efficiency improvements. The authors also reviewed 12 sCO_2 cycle configurations for efficiency and heat transfer area requirements and concluded that the recompression cycle constitutes the best candidate for nuclear power applications. Additionally, they reviewed the experimental sCO_2 facilities in 5 key locations in USA, Japan and S. Korea, focusing mostly on the mechanical design aspects including bearing stability, thrust mitigation and vibrations. Crespi et al. (2017) reviewed all the published research in the field of sCO_2 power cycles up to date. The authors categorized the different investigative approaches including concepts, applications, and layouts, into a modular classification system using standard building blocks of standalone cycle components. The latter are intuitively utilized to construct more complex cycles and qualitatively evaluate and compare them. An effective quantitative comparison, however, was not provided. Since these cycles were evaluated under widely varying conditions and component parameters, as declared in their respective sources, the resulting heterogenous performance data was not suitable for systemic review.

Table 1: Summary of cases analyzed in this study with their respective conditions

Heat Source Conditions		**Cycle Base Temperature [°C]**	**P_{min} / P_{max} [bar]**	**Reference Figure**
Temperature [°C]	**Flowrate [kg/sec]**			
450	9.00	25	74/122-250	2a
650	5.75	25	74/122-250	2b
850	4.06	25	74/122-250	2c/3a
850	4.06	35	74/122-250	3b
850	4.06	45	74/122-250	3c

Moroz et al. (2015) investigated the application of sCO_2 in a bottoming cycle for a gas turbine, using the recuperated, recompression and precompression cycle configurations. The recuperated cycle was deemed the most suitable. Kim et al (2017) evaluated the performance of the recuperated, split-reheat and cascaded transcritical cycles in heat recovery from gas turbine. For the conditions considered in this study, the split-reheat cycle achieved the best overall performance at reduced cost and complexity. Khadse et al. (2018) completed a thermodynamic optimization of the recuperated and recompression configurations for exhaust heat recovery from a next generation heavy duty simple cycle gas turbine and favored the recompression arrangement for the chosen exhaust gas mass flow rate and temperature. Wang et al. (2021) completed a thermo-economic optimization study of multiple waste heat recovery sCO_2 cycles comprised of the recuperated, split-reheat and cascaded configurations and selected the split-reheat arrangement as the best candidate due to its balanced overall performance and reduced design and operational complexity.

2 SETUP AND ANALYSIS

The supercritical CO2 cycles considered in this study include the simple cycle (base case), the recuperated cycle, the split-reheat cycle, the recompression cycle (2 stage compression) in addition to multiple variations of the cascaded cycle (2 stage expansion). The schematics of these cycles are shown in Figure 1, with high and low temperature components and streams annotated accordingly. *MATLAB* models were prepared that allow simulating the performance of these cycles using the thermodynamic database *CoolProp* that employs the Span-Wagner Polar EOS formulation (Bell and Jäger, 2016 and Span and Wagner, 1996). The turbomachinery stages were modeled with a specified isentropic efficiency, η_{is} = 75% for the compressor and η_{is} = 85% for the turbine. The counter flow type heat exchangers were discretized to maintain a set pinch temperature ΔT_{min} = 10 °C between the cold and warm streams. To simulate additional mechanical losses in the respective cycles, a 1% drop of total pressure was imposed at every heat exchanger. No heat losses to the environment were considered. The minimum pressure in the cycle was maintained right at the critical point to avoid a phase change during the heat rejection process, whereas the maximum pressure was varied to accommodate both the design and performance of turbomachinery stages, not to exceed 250 bar as a reasonable limit of material properties and manufacturability. The minimum and maximum temperatures of the working fluid are dictated by the availability of a suitable heat sink and source. The former is a function of the cooler pinch temperature and cooling fluid temperature, while the latter is the function of the waste heat source temperature and the heat exchange process. To understand the effect of the cooling requirements, which vary geographically, multiple cycle base (minimum) temperatures were considered, that correspond to heat sink temperatures of 15, 25 and 35 °C. Three qualities of waste heat source were evaluated in this study: high (850 °C), medium (650 °C) and low (450 °C) with air as working fluid. In order to maintain equivalent source conditions, the mass flow of hot air was adjusted to yield a constant source rate of exergy at a nominal net cycle output power of 1 MWe. A summary of the cases analyzed in this work is provided in Table 1.

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2.1 Energy and Exergy Balance

The conditions within these cycles are optimized to achieve a maximum net power output. This is reflected through the system efficiency which is the product of the cycle efficiency and the heat recovery efficiency of the cycle, as detailed in Wang et al. (2021):

$$\eta_{sys} = \eta_{cycle} \times \eta_{heat} \tag{1}$$

Where the system and cycle efficiencies are expressed as follows:

$$\eta_{cycle} = \frac{\dot{W}_{net}}{\dot{Q}_{cycle}} \text{ and } \eta_{heat} = \frac{\dot{Q}_{cycle}}{\dot{Q}_{max}} \tag{2}$$

The maximum rate of heat available from the source is defined relevant to the ambient conditions as:

$$\dot{Q}_{max} = \dot{m}_{air}\,(h_{in} - h_0)_{air} \tag{3}$$

The power transfer and exergy loss rate are calculated across the turbine and compressor stages using:

$$\dot{W}_{turbo} = \dot{m}_{CO2}\,(h_{in} - h_{out}) \text{ and } \dot{L}_{turbo} = -T_0\,\dot{m}_{CO2}\,(s_{in} - s_{out}) \tag{4}$$

Similarly, the heat transfer and exergy loss rates through the heater and recuperator can be determined as follows:

$$\dot{Q}_{heater} = \dot{m}_{air}\,(h_{in} - h_{out})_{air} = \dot{m}_{CO2}\,(h_{out} - h_{in})_{CO2} \tag{5}$$

$$\dot{L}_{heater} = -T_0(\,\dot{m}_{air}\,(s_{in} - s_{out})_{air} + \dot{m}_{CO2}\,(s_{in} - s_{out})_{CO2}) \tag{6}$$

$$\dot{Q}_{recuperator} = \dot{m}_{CO2}\,(h_{in} - h_{out})_{warm} = \dot{m}_{CO2}\,(h_{out} - h_{in})_{cold} \quad (7)$$

$$\dot{L}_{recuperator} = -T_0(\,\dot{m}_{CO2}\,(s_{in} - s_{out})_{warm} + \dot{m}_{CO2}\,(s_{in} - s_{out})_{cold}) \quad (8)$$

Since the energy transferred to the cooling fluid is assumed lost to the environment, all the exergy removed out of the sCO_2 stream is treated as destroyed in the cooler for this analysis:

$$\dot{L}_{cooler} = \dot{m}_{CO2}\,((h_{in} - h_{out}) - T_0\,\,(s_{in} - s_{out}))_{CO2} \quad (9)$$

Some of the cycles involve splitting the working fluid flow to achieve better matching of the heat capacities of the warm and cold streams in the heat exchangers. This leads to additional losses when the split streams of dissimilar temperatures are merged back together at the constant pressure. This is captured as shown below:

$$\dot{L}_{mixing} = -T_0\,(\dot{m}_{in,1}s_{in,1} + \dot{m}_{in,2}s_{in,2} - \dot{m}_{out}s_{out})_{CO2} \quad (10)$$

2.2 Cycle Optimization Procedure

The *MATLAB* Global Optimization package using the parallelized *MultiStart* scheme is employed to maximize the net power output of the waste heat recovery power cycles. The heat source temperature and mass flowrate are fixed for each run, as well as the cycle base temperature and pressure and pressure ratio, while the mass flowrate of sCO_2 and the flow split ratio when applicable, are adjusted to identify the optimum performance parameters. The heat transfer effectiveness in the heaters and recuperators are also varied with the stipulation that the specified respective pinch temperature never exceeds the minimum temperature difference between the cold and warm streams along the length of the heat exchanger. The size of the pool of start points of problem variables was increased successively for each run to establish the convergence of the solver to a valid global optimum that satisfies the specified constraints.

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2.3 Comparative Cycle Analysis

The various waste heat recovery cycles are simulated over a range of pressure ratios for different heat source temperatures and shown in Figure 2. As detailed earlier, the waste heat source availability is fixed for all cases by adjusting the hot air mass flowrate. All the cycles benefit from the higher heat source *quality* as evident in the significant performance improvements. However, the more complex cycles that allow flow splitting are more capable of taking advantage of the higher available source temperatures, with the simple cycle achieving the most modest gains. Cascade-3 configuration which allows the high temperature turbine stream double recuperation before being energized from the heat source achieves the best performance over the range of source temperatures. The performance advantages of this cycle in comparison to the other cascaded cycles seem to diminish at low pressure ratios, with the trend being more evident for a lower heat source temperature. It is notable that the solver systemically converges to eliminate the low temperature heater in cascade-4 cycle by assigning a zero effectiveness to the same, rendering the latter identical to cascade-6 cycle. The same results are noted for cascade-2 and cascade-7 cycles. It is to be noted that despite no effective heat transfer in the low temperature heater for both cycles, the pressure drop in the same is still calculated, leading to a measurable drop in the availability of the sCO_2 stream, and a slight downshift of the performance curves.

The split-reheat cycle performs significantly better than the more complex cascaded cycles for a low heat source temperature, with the exception of the cascade-3 configuration. At higher heat source temperatures, the performance of cascade-4/cascade-6 matches the split-reheat cycle, however the latter continues to deliver a higher net power output than the other cascaded setups. These results are quite inconsistent with those from the classical topping cycle performance analysis, that studies the performance relative to the turbine inlet temperature variations, without correcting the results to allow quantifying the heat recovery from the waste heat source. A similar observation is made for the recuperated cycle that matches the performance of the non-split cascade-5 cycle. The solver converges to eliminate the high temperature heater, and the cascaded turbine stages effectively perform as a single expansion stage. It should be clarified that the pressure drop in the high temperature heater, as well as the specification of isentropic efficiency in the distinct turbine stages, result is a slight variation in performance compared to the recuperated cycle. Once more, these observations are unintuitive when contrasted with the isolated cycle performance analysis of sCO_2 power cycles.

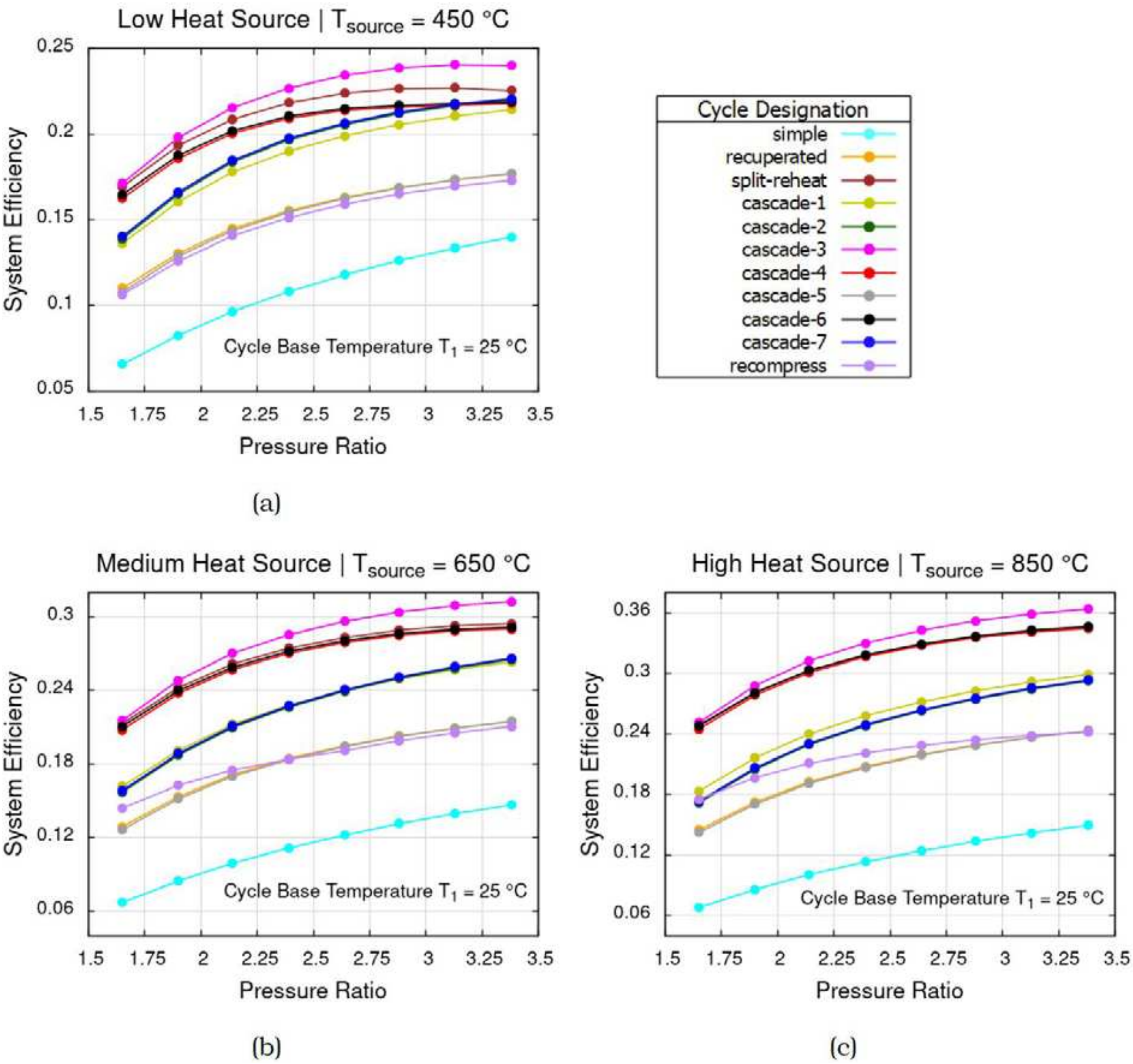

Figure 2: System performance as a function of pressure ratio for the various waste heat recovery cycles evaluated at heat source temperatures of a) 450, b) 650 and c) 850 °C with a cycle base temperature of 25 °C and pressure of 74 bar

To further qualify these findings, the exergy destruction rate in the various cycle components is computed and presented in Table 2 for each cycle at different heat source temperatures. The loss sources are categorized according to the nature of energy transfer to fall into turbomachinery (work transfer), heat exchangers (heat transfer) and stream mixing junctions (isobaric mixing, no energy transfer) groups. Since the energy/exergy transferred through the cooler is removed from the sCO_2 stream, it is grouped separately. The net work output rate of the cycle and the residual exergy rate in the heat source discharge stream (heater outlet) are also presented, all normalized by the available reversible work rate in the heat source (heater inlet).

Table 2: Exergy transfer rate breakdown as percentage of the reversible available work of the heat source stream evaluated at cycle base temperature of 25 °C and pressure of 74 bar and pressure ratio of 3.38 for heat source temperatures of 450, 650 and 850 °C

Cycle Component	Exergy Destruction Rate [%]				Work Out Rate [%]	Exergy Out Rate [%]
	Turbomachinery	Heat Exchangers	Mixing	Heat Sink		
Heat Source temperature = 450 °C						
simple cycle	4.6	7.5	0.0	32.1	24.9	30.9
recuperated cycle	7.1	13.7	0.0	5.7	31.4	42.0
split-reheat cycle	9.8	11.3	0.0	8.2	40.1	30.6
cascade-1 cycle	9.0	12.8	0.0	7.4	38.1	32.7
cascade-2 cycle	8.3	11.0	0.0	6.7	39.1	34.9
cascade-3 cycle	9.4	9.1	0.0	8.2	42.7	30.6
cascade-4 cycle	8.9	12.4	0.0	9.4	38.7	30.6
cascade-5 cycle	6.8	14.3	0.0	5.7	31.5	41.7
cascade-6 cycle	8.9	11.9	0.0	9.4	38.9	30.9
cascade-7 cycle	8.3	10.5	0.0	6.7	39.3	35.2
recompress cycle	7.1	14.0	0.1	5.7	30.8	42.3
Heat Source temperature = 650 °C						
simple cycle	3.4	9.5	0.0	42.4	24.9	19.7
recuperated cycle	6.9	19.6	0.0	5.5	36.3	31.7
split-reheat cycle	9.1	14.1	0.0	7.3	49.9	19.5
cascade-1 cycle	7.7	16.9	0.0	6.2	44.6	24.6
cascade-2 cycle	6.9	14.5	0.0	5.5	45.0	28.1
cascade-3 cycle	8.8	11.5	0.0	7.2	53.0	19.5
cascade-4 cycle	9.0	14.8	0.0	7.5	49.2	19.5
cascade-5 cycle	6.6	20.1	0.0	5.5	36.4	31.4
cascade-6 cycle	9.0	14.4	0.0	7.4	49.4	19.7
cascade-7 cycle	6.9	14.2	0.0	5.5	45.1	28.3
recompress cycle	6.9	19.9	0.1	5.5	35.7	32.0
Heat Source temperature = 850 °C						
simple cycle	3.3	17.7	0.0	41.0	24.0	13.9
recuperated cycle	6.4	22.9	0.0	5.1	39.0	26.6
split-reheat cycle	8.3	16.0	0.0	6.6	55.3	13.8
cascade-1 cycle	7.2	20.3	0.0	5.7	48.0	18.9
cascade-2 cycle	6.5	18.6	0.0	5.1	47.0	22.8
cascade-3 cycle	8.0	13.4	0.0	6.4	58.5	13.8
cascade-4 cycle	8.2	16.0	0.0	6.6	55.4	13.8
cascade-5 cycle	6.2	23.3	0.0	5.1	39.0	26.4
cascade-6 cycle	8.2	15.6	0.0	6.6	55.6	14.0
cascade-7 cycle	6.5	18.4	0.0	5.1	47.1	23.0
recompress cycle	9.5	14.4	0.0	4.3	38.9	32.9

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As predicted the simple cycle yields the highest losses in the cooler section, as the energy from the warm turbine exhaust stream is not recovered; the recuperated cycle in contrast, exhibits the lowest losses in the cooling process due to utilizing the residual energy in the turbine exhaust stream to improve the cycle efficiency. Nonetheless the sharp variations in the heat capacity of the recuperator cold and warm streams, mainly due to the thermodynamic properties of sCO_2 in the vicinity of the critical point, lead to the most elevated heat exchanger losses in the recuperated cycle. By splitting the flow in some of the more complex cycles, a better match of the heat capacity is achieved by the virtue of different mass flowrates of the cold and warm streams in the recuperators. As such, heat transfer occurs between fluids of less dissimilar temperatures, leading to a reduction of exergy destruction in the process. Cascade-3 cycle registers the lowest share of irreversibility creation in the heat exchanger group, consistent with the highest performance discussed earlier. All the cycles show an increase in the heat exchanger losses as the heat source temperature rises, since the temperature gradient between the cold stream, governed by fixed compressor discharge conditions and the hot air stream becomes more prevalent.

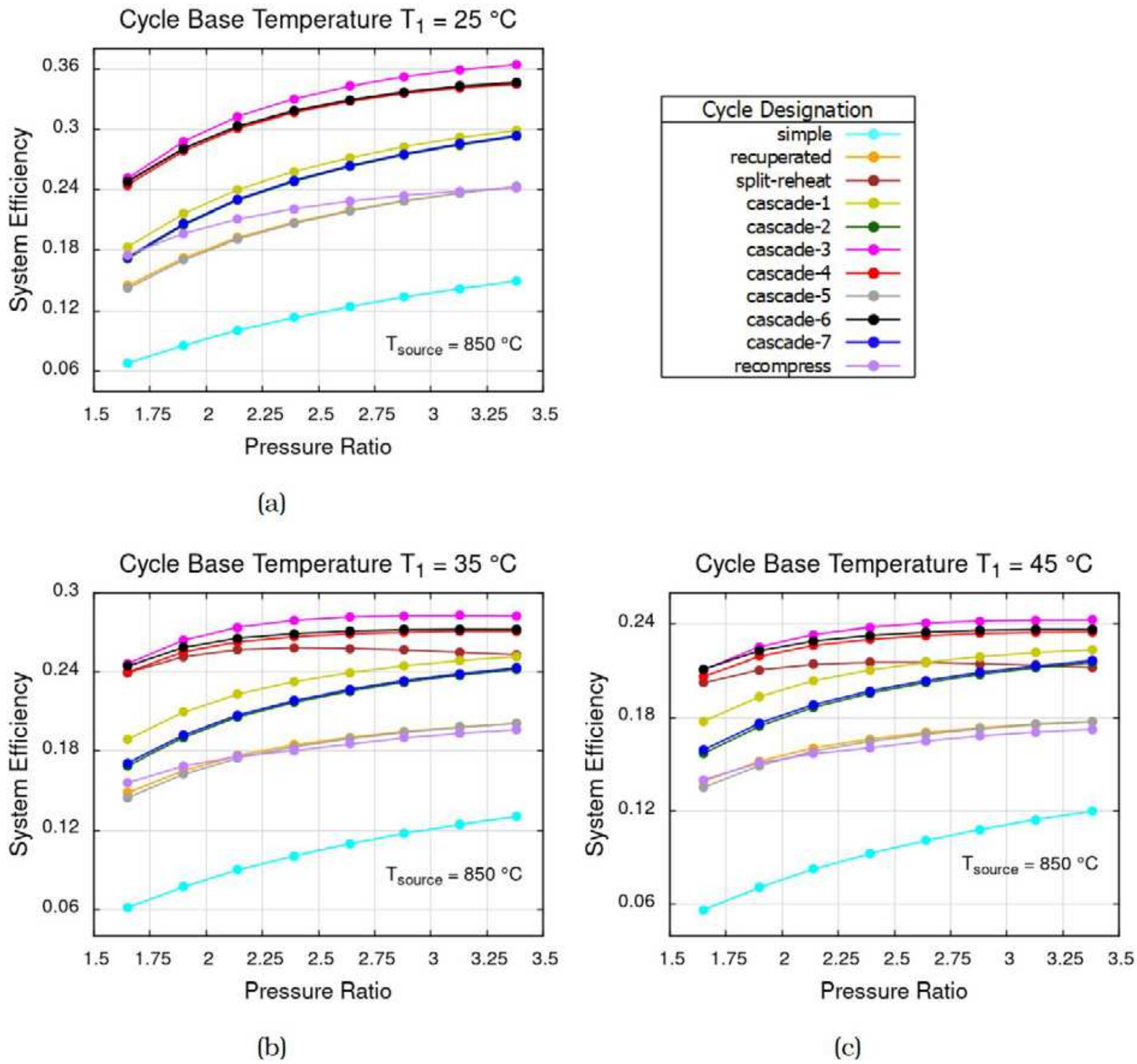

Figure 3: System performance as a function of pressure ratio for the various waste heat recovery cycles evaluated at cycle base temperatures of a) 25, b) 35 and c) 45 °C and base pressure of 74 bar with a heat source temperature of 850 °C

The results displayed in Figure 3 showcase the role of the available heat sink on the effectiveness of waste heat recovery in sCO_2 cycles. The cooling fluid temperature dictates the cycle base temperature through heat transfer across a temperature gradient, modelled in this study as a fixed pinch temperature $\Delta T_{min} = 10$ °C. A significant drop in cycle performance is noted as the base temperature increases from 25 to 45 °C, notably for the cascaded cycles with ~30% reduction in net power output. As the compressor inlet temperature increases, the fluid heat recovery capacity is reduced, together with the ability of flow splitting to reduce the exergy losses in the recuperator sections by achieving better stream matching. This is particularly evident with the split-reheat cycle that no longer outperforms the more complex cascaded cycles that can take advantage of double expansion stages to extract energy from low and high quality streams. The lack of recuperation of the simple cycle configuration causes a less detrimental effect on the performance as the base temperature is elevated.

3 CONCLUSIONS

A number of sCO_2 power cycles were investigated for waste heat recovery applications. These include the simple, recuperated, split-reheat, recompression in addition to several variations of cascaded cycles with distinct flow splitting arrangements. The roles of the heat source and sink availability in the effective cycle energy recovery were reviewed. Increasing the heat source temperature from 450 to 850 °C led to ~30% improvement in performance, while increasing the cycle base temperature from 25 to 45 °C reduced the net output power by ~30%. Detailed analysis of the energy and exergy transfer was conducted to identify the individual cycle characteristics. The cascade-3 cycle was determined to achieve the highest performance across all conditions, while the less complex split-reheat configuration mostly outperformed the rest. The recompression and non-split cascaded cycles offered no performance advantage over the basic recuperated arrangement. However, merely optimizing the cycle for thermodynamic performance is not adequate to determine the suitability of a specific cycle configuration for a given set of requirements. Extending the effort to include an economical component of such optimization is quintessential especially in the current energy market. The authors plan to address this topic in future studies. 262

NOMENCLATURE

h	specific enthalpy	(kJ/kg)
$\dot{L}$	rate of exergy loss	(kW)
$\dot{m}$	mass flowrate	(kg/sec)
P	pressure	(bar)
$\dot{Q}$	heat transfer rate	(kW)
s	specific entropy	(kJ/kg/K)
sCO_2	supercritical CO_2	
T	temperature	(°C)
$\dot{W}$	power	(kW)

Greek

η	efficiency

Schematics Abbreviations

C	compressor
CLR	cooler
HT	heater
HTC	high temperature compressor
HTH	heater high
HTL	heater low
HTT	high temperature turbine
IN	waste heat source inlet

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INT waste heat source internal
LTC low temperature compressor
LTT low temperature turbine
OUT waste heat source outlet
RC recuperator
RCH recuperator high
RCL recuperator low
T turbine
*x*H high temperature stream
*x*L low temperature stream

Subscript
0 ambient conditions
cycle closed cycle attribute
heat heat recovery attribute
is isentropic
source waste heat source
sys system

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This book contains the compilation of works contributed to the *7th International Seminar on Organic Rankine Cycle Power Systems* (ORC 2023), held in Seville between the 4th and 6th of September 2023. The event was hosted by Universidad de Sevilla on behalf of the Knowledge Centre on Organic Rankine Cycle Technology (KCORC), incorporated in The Netherlands.

The ORC conference, organized biennially, stems as the only conference that is specific to ORC technology, therefore gathering a diverse community whose affiliation spans across all the interested stakeholders, not only in this particular technology but also and in a broader context, in the energy transition. Original equipment manufacturers, professional associations, end-users, investors, policy makers, academics, scientists feel at home at ORC 2023.

The almost 100 proceedings in this book cover a wide variety of topics, from fundamentals to system integration through component design, accounting for thermodynamic performance as well as component design. In addition to this, and as a new track in 2023, works on heat pump technology were also accepted in order to raise awareness of the strong ties between both technologies, specifically in energy storage applications.

This book provides an excellent overview of the current maturity of power systems based on Organic Rankine Cycle technology for applications as diverse as geothermal and waste heat recovery in industry or downstream of other prime movers (e.g., marine applications). It is also an excellent source of information to understand the current challenges faced by the technology, stemming from a very competitive market and increasingly stringent environmental regulations.

The organizers of ORC 2023 hope that the reader finds this work as exciting as the attendees to the conference and, maybe, make the decision to join the 8th edition to the conference in 2025.

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