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# Investigation of $sCO_2$ Cycle Layouts for the Recovery of Low Temperature Heat Sources

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## ABSTRACT

Since the development of supercritical carbon dioxide ($sCO_2$) power cycles, several different power cycle architectures have evolved. Additional components like reheaters, recuperators and intercoolers were added and split flow configurations were introduced. These specific configurations are typically tailor-made for an explicit application, mostly in the medium-temperature field ranging between 240-480 °C.

In the waste heat recovery sector, low grade waste heat (< 240 °C) holds the biggest share of the waste heat worldwide. This work focuses on ultra low temperature heat sources as they face big challenges for cycle efficiency because of the low temperature difference of heat source and heat sink. Consequently, the power generation is on low efficiency and subject of improvement. This study therefore investigates different power cycle configurations for a given low temperature air as heat source and ambient air as a heat sink (20 °C). The main objective is to evaluate different cycles with standardized boundary conditions in order to have an equal base for their comparison. Heat source temperature ranges from 60 to 100 °C are considered.

Firstly, $sCO_2$ power cycles from literature are evaluated using the commercial solver EBSILON. This step is meant to validate the numerical set up and results by using a documented cycle configuration from literature with its original boundary conditions. In a second step, the specified low temperature heat source is applied. The configurations are evaluated in terms of mass flow, pressure and thermal performance. Finally, the cycles are classified according to their efficiencies in the low temperature regime.

From the results, it is observed that a recuperator step is not feasible in very low temperatures because of the minor superheating in the supercritical region. For turbine inlet temperatures higher than 80 °C, a recuperator starts to be beneficial. Intercooled compression is not suitable for ultra-low waste heat temperatures. A basic 4-component configuration and split flow before heating perform best in the low temperature region of 60 °C. With increasing turbine inlet temperature, more complex cycle configurations such as reheated expansion and intercooled compression might be considered in order to enhance the system efficiency.

This study provides a dataset of thermal efficiencies of $sCO_2$ power cycle configurations in the low grade waste heat recovery until 100 °C.


## KEYWORDS

Waste heat recovery, power generation, low temperature, ambient air, supercritical $CO_2$, Rankine cycle, cycle configuration


## 1 INTRODUCTION

In times of rising energy demand and limited fossil fuel resources, energy efficiency is an important topic worldwide. The most convenient forms of secondary energy are electricity and heat, which are produced by engines, for example in a cogeneration plant. A global estimation reveals that 72% of primary energy is lost to the surrounding environment in the form of heat during the conversion process to a secondary energy [1]. This means, a large fraction of primary energy remains unused. Waste heat sources can be classified by its temperature into three categories: a) high grade (>650 °C): discharged by e.g. furnaces, power plants; b) medium grade (240-650°C): discharged by e.g. industrial processes, combustion; and c) low grade (<240 °C): discharged by e.g. compressors, cogeneration plant, buildings, biogas plant [1]. The larger portion of waste heat corresponds to low temperature waste heat with approximately 63% [1].

Among current energy recovery technologies, the Organic Rankine Cycle system (ORC) is best suited to generate energy from low-grade waste heat [2]. The ORC relies on organic substances as working fluids, like R11, R12, R113, R123, R134a [2] which evaporate at lower temperatures than water. These organics fluids are harmful to the environment due to their toxicity and high global warming potential [3]. During the last years, supercritical carbon dioxide ($sCO_2$) was found to be a more promising working fluid since it is less toxic and chemically inert [3].

Numerous studies about sCO2 power cycles (PC) for waste heat recovery have been published. To enhance the cycle efficiency, the basic four-component PC consisting of pump, evaporator, expander and condenser is modified. Additional components like reheaters, recuperators and split flow architectures help to maximize the power output of the thermodynamic cycle. Section

1 presents a reviewed of existing cycle configurations. Section 2 shows the methodology of the cycle comparison: First, boundary conditions are defined, followed by a description of the modelling process of the cycle architectures. Afterwards in section 3, their results are compared and influencing factors of the cycle efficiency are defined.

## 1.1 Literature Review

Many different cycle configurations have been proposed, especially for medium and high temperature waste heat. Although significant progress has been made in the design of $sCO_2$ power cycles, their field of application is mainly in temperatures starting from 200 °C [4]. Because very low temperatures have not been investigated extensively, it is not clear which configurations are the most effective in recovering low temperature heat.

Generally, power cycles can be distinguished in single flow cycles and split flow cycles: Single flow means that the working fluid is not divided at any point in the system, while in split flow configuration, the working fluid can be divided into sub streams.

### 1.1.1 Nomenclature of cycle configurations

The names of proposed cycle configurations are as various as configurations itself. Therefore, the uniform configuration labelling of Cespri et al. [4] has been adapted:

The letter R classifies the numbers of recuperators within a PC. R0 stands for zero recuperators until R3 for three recuperators. This term is then combined with other modifications, like reheated expansion (RH) or intercooled compression (IC). When the system is in split flow configuration, the working fluid can be divided at different positions:

- split flow before cooling / compression (SFC)
- split flow before heating (SFH)
- split flow before expansion (SFE),
- split flow before heating and expansion (SFHE)

As an example, if a PC employs one recuperator and has a split configuration for compression and reheated expansion, the naming of the cycle architecture would be R1-SFC-RH.

Resulting from the naming convention, the cycle configurations can be classified from a simple system design (R0) up to complex (R3). Furthermore, the longer the expression (e.g. R2-SFC-SFH-IC), the more system components are utilized.

### 1.1.2 Reviewed cycle configurations

Having identified a general labelling system, the findings from literature can be categorized. It turns out that solely in the waste heat recovery sector, nearly 50 cycle configurations exist, without even distinguishing between the number of reheat stages or intercoolers (e.g. three reheated expansion steps and one reheated expansion both lead to the expression “RH”).

Table 1 shows the single-flow cycle architectures on the left column with their literature source on the right column. It seems that several publications investigated the basic four-component PC with a pump, evaporator, turbine and condenser (labelled R0). Also, several researchers suggested a simple cycle configuration with one recuperator (R1). The most complex configurations for single flow configurations are found to be R2, R2-IC and R2-RH.

Table 1: Single flow PC architectures with their sources from literature

| PC architecture in single flow | Source from literature | PC architecture in single flow | Source from literature |
|---|---|---|---|
| R0 | [5]-[15] | R1-RH | [16] [17] |
| R0-IC | [4] | R1-IC-RH | [18] |
| R0-RH | [16] [19] | R2 | [4] [20] |
| R0-IC-RH | [21] | R2-IC | [22] [23] |
| R1 | [8] [22] [24]-[39] | R2-RH | [37] |
| R1-IC | [38] [40]-[42] | | |

Available split-flow configurations are shown in Table 2. Split-flow architectures offer greater variety of configurations in literature.

Table 2: Split flow PC architectures with their sources from literature

| PC architecture in split flow | Source from literature | PC architecture in split flow | Source from literature |
|---|---|---|---|
| R0-SFC | [43] | R2-SFC-SFE-IC-RH | [44] |
| R0-SFC-SFH | [4] | R2-SFC-SFH | [4] |
| R1-SFC | [22] [42] | R2-SFE | [20] |
| R1-SFC-IC | [45] | R2-SFE-RH | [44] [4] |
| R1-SFC-IC-RH | [46] | R2-SFH | [4] |
| R1-SFC-RH | [47] | R2-SFHE | [22] [24] [27] |
| R1-SFC-SFH | [22] | R2-SFHE-IC | [22] |
| R1-SFE | [48] | R2-SFH-SFHE-IC | [22] |
| R1-SFH | [20] [23] | R3-SFC | [4] |

| | | | |
|---|---|---|---|
| R1-SFHE | [13] | R3-SFCHE | [39] |
| R2-SFC | [20] [23] | R3-SFC-IC | [4] |
| R2-SFC-IC | [30] [49] | R3-SFC-SFE-IC-RH | [4] |
| R2-SFC-IC-RH | [17] [30] [46] | R3-SFC-SFH | [4] |
| R2-SFC-RH | [14] [17] [29]-[31] [34] [36] [38]-[41] [50]-[56] | R3-SFC-SFHE | [4] |
| R2-SFC-SFE | [4] | R3-SFC-SFHE-IC-RH | [4] |
| R2-SFC-SFE-IC | [4] | R3-SFC-SFH-RH | [29] |
| R2-SFC-SFE-RH | [44][48][57] | R3-SFHE | [27] [58] |

From the literature review, it seems from Table 2 that the configuration of R2-SFC-RH is the most frequent investigated. The complex cycle configurations R3 are found in the higher temperature waste heat recovery from 400 °C (673.15 K) such as coal fired power plants [29] and internal combustion engines [58]. While most of the remaining references deal with medium grade waste heat, only a few deal with low grade waste heat: [4-7],[9],[10],[26][43].

It is observed that all findings mentioned above are used for different applications with different boundary conditions. Thus, if one wants to know the best performing cycle configuration for a specific temperature region (here ultra-low temperatures between 60-100 °C), the stated efficiencies in literature cannot be compared as they all base on different setups.  In order to bridge this gap, this work addresses the comparability of thermal efficiencies of $sCO_2$ power cycle configurations in the low-grade waste heat recovery until 100 °C. The objective is to compare different cycle configurations regarding their thermodynamic first law efficiency. Economic viewpoints such as equipment cost and payback time are not considered at the present.

First, boundary conditions are defined, followed by a description of the modelling process of the cycle architectures. Afterwards, their results are compared and influencing factors of the cycle efficiency are defined.

## 2 METHODOLOGY

As a base for the cycle comparison, the Rankine Cycle will serve for the cycle comparisons. This means that the power cycle includes a full condensation of the working fluid: it alternates between gaseous and liquid state.

In order to compare all cycle architectures, several boundary conditions were applied as shown in Table 3.

Table 3: Simulation parameters for the comparison of the sCO2 cycles [59]

| **Parameter** | **Value** |
|---|---|
| Waste Heat Source Temperature | 60-100 °C |
| Waste Heat Source Pressure | 1.013 bar |
| Waste Heat Source Mass Flow | 1000 kg/s |
| Heat Sink Temperature | 20 °C |
| $CO_2$ Condensation Temperature | 25.43 °C |
| $CO_2$ Condensation Pressure | 65 bar |
| Turbine Isentropic Efficiency | 80% |
| Pump Isentropic Efficiency | 80% |
| Heat Exchanger Effectiveness | 95% |

Ambient air serves as a heat sink for condensing the $CO_2$. The mass flow of the air in the condenser is adjusted according to the cooling load. Thereby, a condensation pressure and temperature is fixed in order to reassure a full condensation of the working fluid.

Furthermore, the cycle should operate in steady state condition. No friction losses in pipes and heat exchangers occur as well as no kinetic and potential energy losses. Saturated liquid is supposed to exit the condenser.

In a first step, the findings from the literature research are proven for applicability for the scope of this study's boundary conditions. Wu et al. stated that R3 configurations are suitable for waste heat sources starting from 412 °C [27]. Thus, all R3 cycle architectures are sorted out and are not considered for the comparison for a low grade waste heat recovery.

The remaining 37 configurations for R0, R1 and R2 are then modeled in EBSILON®Professional (EBSILON) [60], a commercial solver for energy and mass balances within cyclic processes of power plants. Thermyphysical properties are calculated via REFPROP [61], which uses the Span-Wagner equation of state [62] for $CO_2$.

The different power cycle configurations are implemented in EBSILON with the previously specified low temperature heat source. The PCs are compared regarding their thermodynamic first law efficiency $\eta_{th}$, which is calculated via the following equation:

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \qquad (1)$$

In equation (1), $\dot{Q}_{in}$ represents the heat flow from the heat source into the system. $\dot{W}_{net}$ denotes the net power output of the cyclic system, namely:

$$\dot{W}_{net} = \dot{W}_{Turbine(s)} - \dot{W}_{Pump(s)} \qquad (2)$$

The Carnot efficiency $\eta_{Carnot}$ is determined through the general equation:

$$\eta_{Carnot} = 1 - \frac{T_{cold}}{T_{hot}} \qquad (3)$$

In equation (3), $T_{cold}$ denotes the temperature of the heat sink and $T_{hot}$ the temperature of the heat source in Kelvin. When working with ultra-low temperatures as little as 60 °C and a heat sink of 25 °C, the temperature difference between heat source and heat sink is rather small. Calculating the Carnot efficiency (eq. 3), a theoretical maximum efficiency of 10.5% is possible.

The cycle architectures were modeled in the order from simple to complex configurations. For each cycle architecture, the thermal efficiency is maximized by optimization of pressure ratio and mass flow. The cycle pressure ratio is the quotient of cycle pressure after pump and cycle pressure before pump.

To validate each studied power cycle, the original setup with its specific adjustments was implemented EBSILON. Only if the resulting efficiency agreed with the stated efficiency from the publication, the parameters were unified to meet the boundary conditions of this comparative study. The validation process showed deviations less than 4.7 percentage points. Thus, adaptions of boundary conditions could be made with confidence. Their outcome is presented in the next section.

## 3 RESULTS AND DISCUSSION

Figure 1 shows the most representative cycle configurations for the previously defined boundary conditions. The influence on the cycle efficiency can be grouped by adding extra steps to the basic configuration (R0):

- recuperator
- intercooled compression
- reheated expansion
- intercooled compression and reheated compression

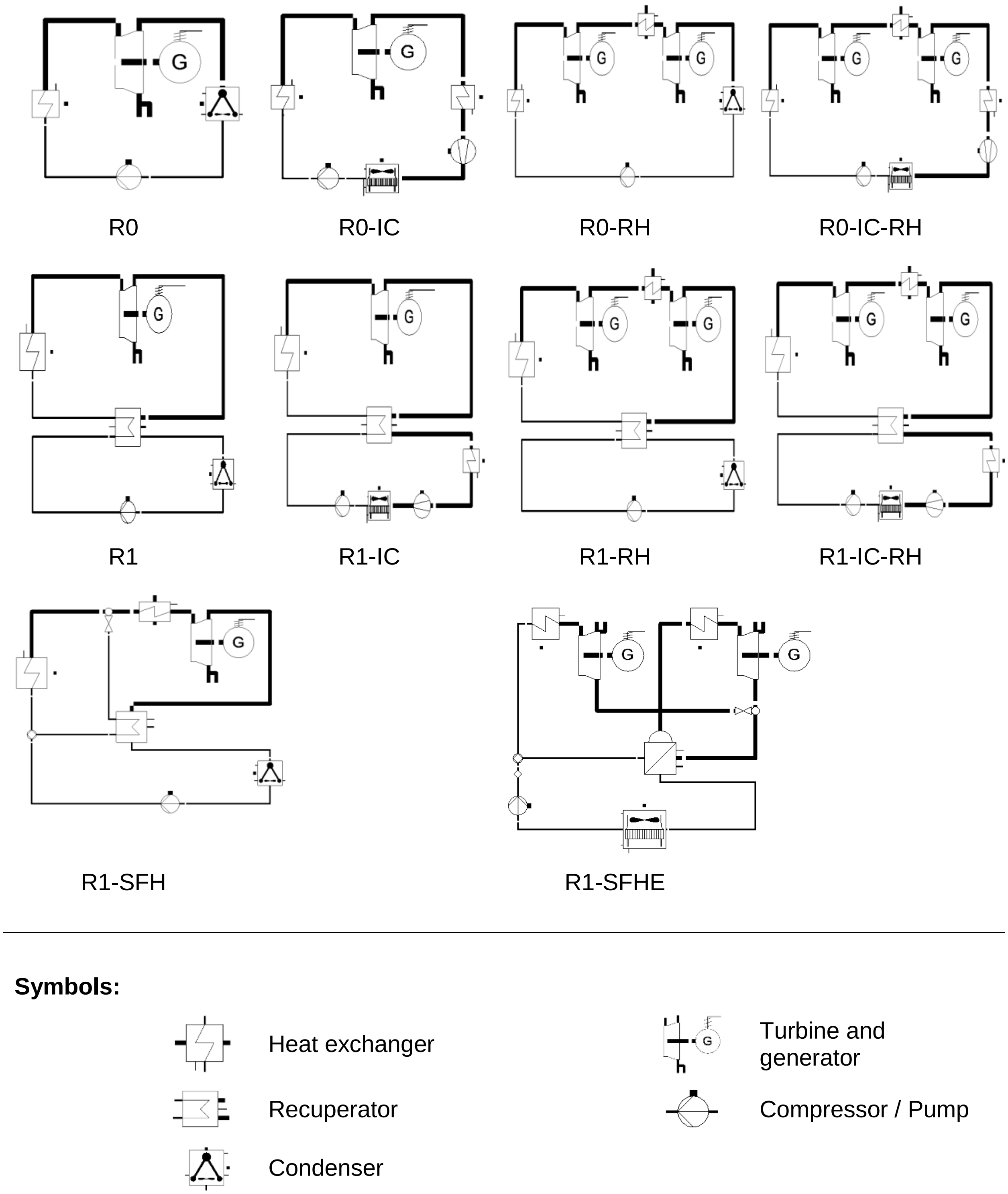


Figure 1: PC configurations modeled in EBSILON

### 3.1 Influence of mass flow and pressure ratio

When optimizing a cyclic process in terms of pressure, there is one peak, at which the first law efficiency is the highest. Figure 2 shows the thermal efficiency for a R0 configuration at a heat source temperature of 60 °C. One can see that the efficiency is shaped parabolic. It is observed that this shape is not unique for this R0 example, but for all modeled configurations.

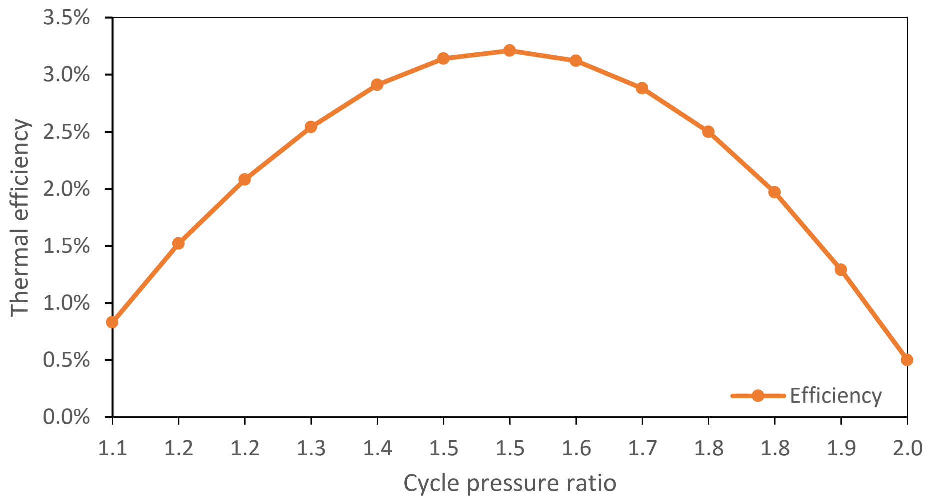


Figure 2: Cycle efficiency in terms of pressure ratio for R0 configuration.

The thermal efficiency of the PC is affected also by its mass flow. More precisely, the net power output (equation 2) changes with a changing mass flow. Figure 3 indicates the net power output and the thermal efficiency of the same R0 configuration with 60 °C heat source. It is observed that the net power output rises until it reaches a maximum. From this point on, the required pumping power rises at a faster rate than the turbine power output, resulting in a decreasing net power output (eq. 2) and consequently a decreasing cycle efficiency (eq. 1). This phenomenon is observed for all investigated cycle configurations.

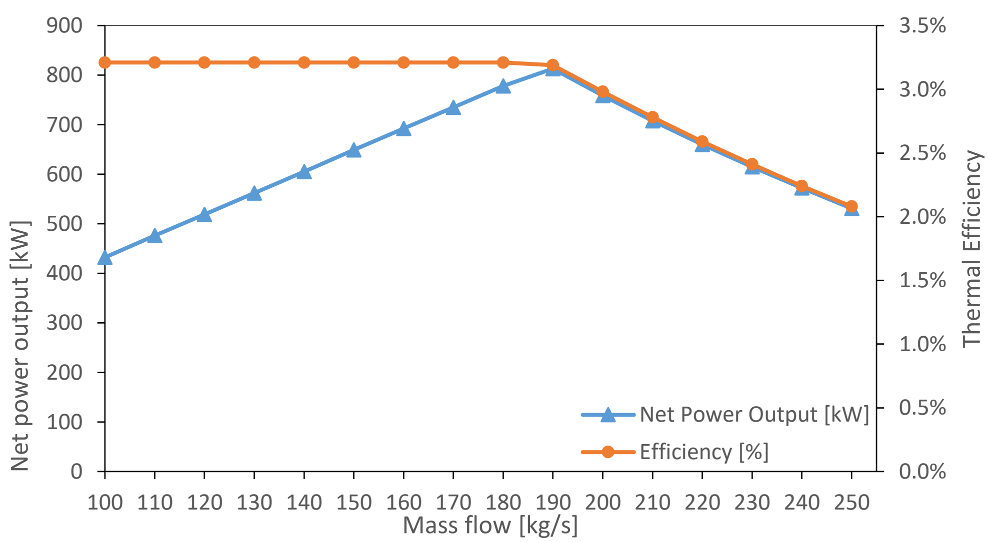


Figure 3: Net power output and cycle efficiency in terms of mass flow for R0 configuration.

### 3.2 Influence of ultra low source temperature

Beginning with the simplest configuration R0, a maximum thermal efficiency of 3.21 percentage points is reached for a waste heat temperature of 60 °C (figures 1 and 2). To understand the reason one needs to look at the system setup shown in Figure 4: In this cycle configuration the pumping process (state 4 → 1) requires a high amount of energy in relation to the energy release by the expansion process (state 2 → 3). Furthermore, it can be seen that the temperature of the $CO_2$ after the expansion (27.7 °C) is close to the condensation temperature of state 4 (25.4 °C). This small temperature difference makes it difficult to fit in further steps like intercooled compression. Also a recuperator, which should transfer heat from state 3 to state 1 is not possible, as the temperature after the pump (33.2 °C) is already higher than the turbine outlet temperature (27.4 °C).

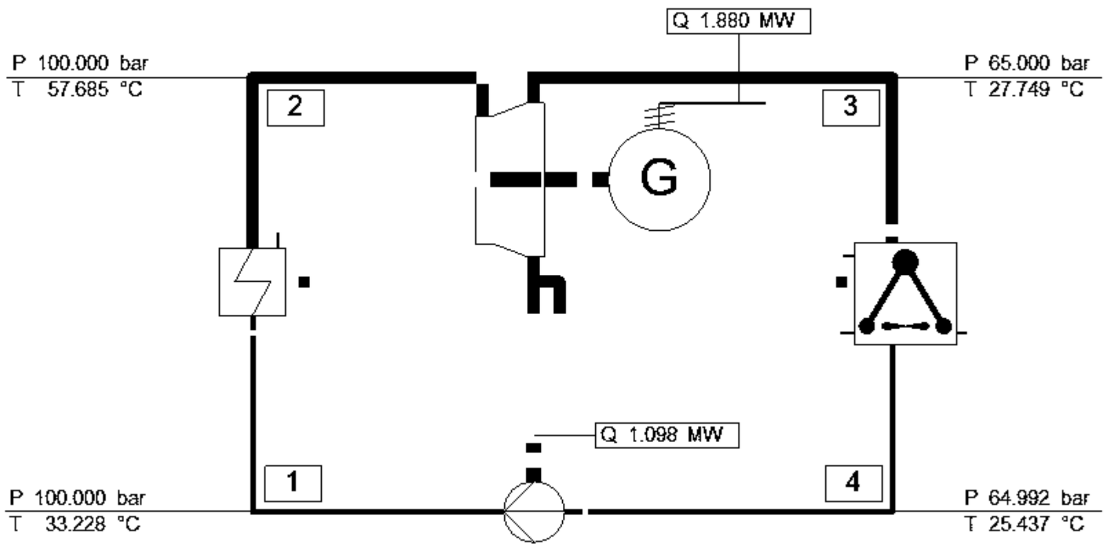


Figure 4: R0 with a source temperature of 60 °C.

These circumstances lead to the fact that the R0 cycle architecture performs better when compared to other investigated cycle configurations in the temperature regime of 60 °C.

Figure 5 shows the resulting efficiencies from the comparison of all configurations with zero and one recuperator with respect to the heat source temperature. As can be seen, for the source temperatures from 60-100 °C, only 10 out of the remaining 37 configurations lead to a result due to the mentioned temperature restrictions.

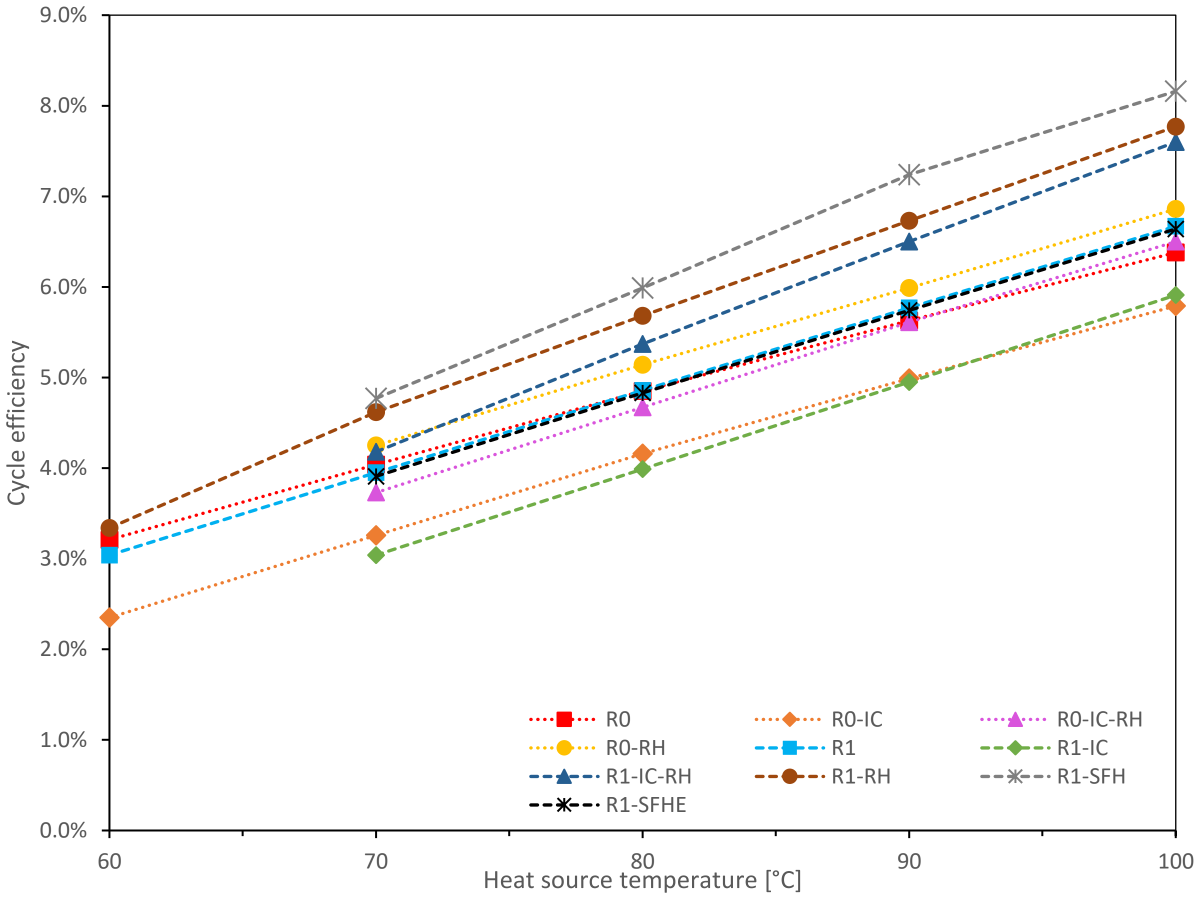


Figure 5: Cycle efficiencies of PC configurations with respect to the heat source temperature

Figure 5 does not contain plots for cycle architectures with two recuperators. The reason therefore is that the small temperature difference between heat source and heat sink the temperatures after the expansion are too restrictive so that employing two recuperators is not possible. It was found that two recuperators could be used for heat source temperatures greater than 225 °C. Also, several split flow configurations mentioned in Table 1 did not function in an ultra-low temperature regime.

### 3.3 Influence of a recuperator step

When comparing the configurations R0 and R1 in Figure 5, one can conclude that a recuperator is beneficial from a heat source greater than 80 °C. Furthermore, the plots can be categorized in mainly two different slopes: configurations with a recuperator have a steeper rise in efficiency with increasing temperature. This means that in a temperature range between 77°C and 93 °C, most of the complex cycle architectures start to outplay the simple cycle architectures. Thus, from lower to higher temperature, the difference in efficiency between the cycles increases, this is observed from the diverging plots from left to right in Figure 5.

### 3.4 Influence of an additional heat exchanger for energy source extraction

Figure 5 further shows that the configuration R1-SFH provides the highest efficiency among the investigated configurations. The reason behind this finding is that this configuration utilizes two heat exchangers in a row to extract energy from the heat source (similar to increasing the heat exchanger surface). It was found that employing an extra heat exchanger in the heat addition step leads to an efficiency increase of approximately 1.4 percentage points. Considering the computed efficiencies ranging from 1.68 to 7.77 percentage points, an augmentation of the efficiency by 1.4 percentage points leads to a relevant improvement of the cycle performance.

### 3.5 Influence of reheated expansion

Cycle configurations with reheated expansion showed to be beneficial from source temperatures of 70 °C. At this point, a reheated expansion improves the efficiency by 0.3 percentage points. The surplus of efficiency rises with increasing the source temperature: for example at a source temperature of 100 °C, adding reheated expansion already contributed to 0.7 percentage points higher efficiency. Adding a reheated expansion step to the cycle leads to a rather small efficiency improvement but becomes more significant with rising waste heat temperatures.

### 3.6 Influence of intercooled compression

For a low temperature heat source, adding intercooled compression leads to an efficiency decrease of 1 percentage point due to small temperature difference of the fluid after expansion and the condensation. It was detected that for low grade temperatures, an intercooled compression step does not lead to a better performance. Only for heat source temperatures higher than 190 °C, intercooled compression contributes towards a higher thermal efficiency.

## CONCLUSIONS

In this study, a comparison in terms of thermal efficiency for several $sCO_2$ power cycle configurations with standardized boundary conditions has been presented. Efficiencies range between 1.68 and 7.77 percentage points for a waste heat temperature regime up to 100 °C. The following conclusions can be drawn:

- the basic R0 configuration performs best in heat source temperatures from 60 °C – 80°C
- a recuperator (R1) starts to be beneficial from source temperatures higher than 80 °C
- a second heat exchanger for heat source extraction results in 1.4 percentage points higher efficiency
- reheated expansion augments the efficiency from a heat source temperature of 70 °C and increases the benefit on the efficiency with rising source temperature

- split flow before cooling (SFC) was not feasible for an ultra-low temperature regime of less than 100 °C
- intercooled compression starts to be beneficial for heat source temperatures greater than 190 °C
- for source temperatures higher than 225 °C two recuperator steps (R2) show reasonable results
- the required pumping power presents one of the major reasons for low cycle efficiencies
- within ultra-low temperature regime, more complex system configurations produce only minor efficiency improvements while rising higher heat source temperatures

Comparing the cycle architectures one can say that adding extra components to the basic PC leads to a change in efficiency. In the case of additional heat exchanger for heat source extraction and recuperation, the cycle efficiency is influenced in a positive way while intercooled compression downgrades the performance for low temperatures. Nevertheless, to fully decide whether additional components are reasonable or not also depends on an economical viewpoint, which is not in the scope of this study.

This work provides a comparison of cycle architectures at ultra-low temperatures that has not been carried out yet. It demonstrates the effect of modifying the basic PC and its resulting thermodynamic efficiency. The findings of this study contribute to the state of the art about recovering low temperature waste heat in providing a road map of cycle efficiencies in different temperature conditions.

## NOMENCLATURE

| | |
|---|---|
| EBSILON | = EBSILON®Professional by Steag Energy Services |
| R0 - R3 | = zero to three Recuperators |
| RH | = Reheated expansion |
| IC | = intercooled compression |
| SFC | = Split Flow before Cooling / Compression |
| SFH | = Split Flow before Heating |
| SFE | = Split Flow before Expansion |
| SFHE | = Split Flow before Heating and Expansion |
| SFCHE | = Split Flow before Compression, Heating and Expansion |

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