Off-design analysis of ORC and CO_2 power production cycles for low-temperature surplus heat recovery

Harald Taxt Walnum*, **Daniel Rohde and Yves Ladam**
SINTEF Energy Research, 7465 Trondheim, Norway

Abstract

Electricity production with an organic Rankine cycle and a transcritical Rankine cycle is investigated in this paper with R-123 and CO_2 as working fluids, respectively. The analysis focuses on the off-design behavior with different control strategies to show some of the occurring difficulties. It was found that both cycles need an advanced control strategy to avoid non-feasible operation (R-123) or significant losses in work output (CO_2). A challenge for the advanced control is the required large change in expander speed, which can lead to compatibility problems with the grid.

Keywords: ORC; CO_2; power production; low temperature; off-design

*Corresponding author:
harald.taxt.walnum@sintef.no

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1 INTRODUCTION

In process industry, large amounts of energy are rejected to the ambient. Recovery of this surplus energy is a wide topic. Among the strategies for energy recovery, production of electricity is very interesting, due to the versatility of this form of energy.

Power production from surplus heat sources is largely dominated by the steam process. It can be found in nuclear and oil or gas-fired power plants as well as large biomass-fired plants or even solar power plants. However, the steam process suffers from high capital cost and poor efficiency for medium-to-low temperature heat sources (the borderline being around 400°C) [1].

The organic Rankine cycle (ORC, Figure 1) is a well-established technology for power production from low-temperature heat sources. It combines improved efficiency with lower capital and operating costs. The working fluids used are organic compounds of the halocarbon or hydrocarbon families, fluids commonly used in the refrigeration industry.

Common applications for the technology are electricity production from geothermal fields [2, 3], biomass plants [4] or bottoming cycles for gas turbines [5, 6]. More scarce applications are solar application [7, 8] or energy recovery from industrial waste heat [1, 9]. A commonly accepted limit for a profitable energy recovery plant is 200°C for a gas heat source and 90°C for a liquid heat source (S. Koren, 2008, private communication with Ormat sales manager). However, lower temperatures might become economical with further R&D work. Research in ORC technology is very active, focusing both on component development [10] and on working fluid selection [11–14].

Despite substantial improvements, power production from low-to-medium temperature heat sources is still handicapped by large investment costs and relatively poor efficiency. In addition, working fluids used are either toxic (ammonia), flammable (hydrocarbons) or very potent greenhouse gases, contributing to global warming (HFC refrigerants).

The transcritical Rankine Cycle recently received special attention [15–18] due to its performances for energy recovery from low-temperature sources. The transcritical process differs from the others, in that it absorbs heat at a supercritical pressure. Due to the temperature glide during heating of a single-phase fluid (compared with the constant temperature of an evaporating single component fluid), it is possible to achieve a much better temperature approach with the heat source in the main heat exchanger. To achieve low temperature differences in a heat exchanger is important, as the exergy losses are directly coupled with the temperature difference between the fluids.

CO_2 is a natural candidate as working fluid for this technology. It combines high performance, low cost, low toxicity, is non-flammable and has no environmental impact. A transcritical CO_2 power cycle operates at relatively high pressures,

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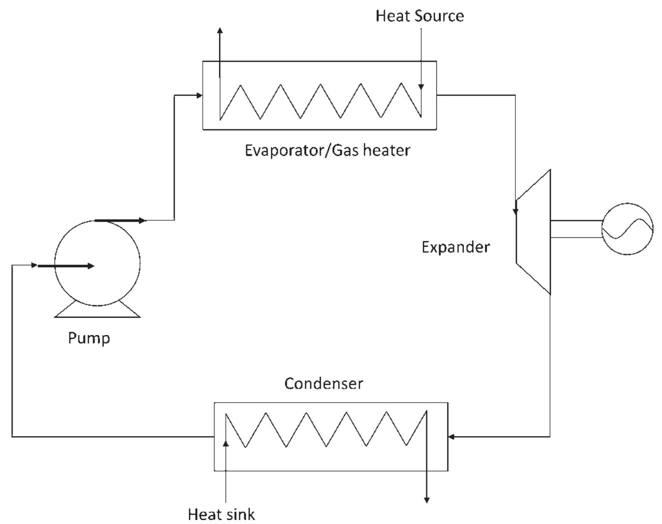

Figure 1. *Principle layout of the Rankine cycle.*

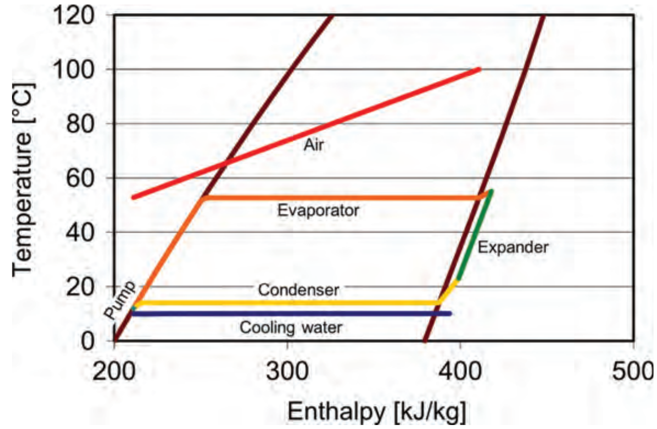

Figure 2. T–h *chart, R-123.*

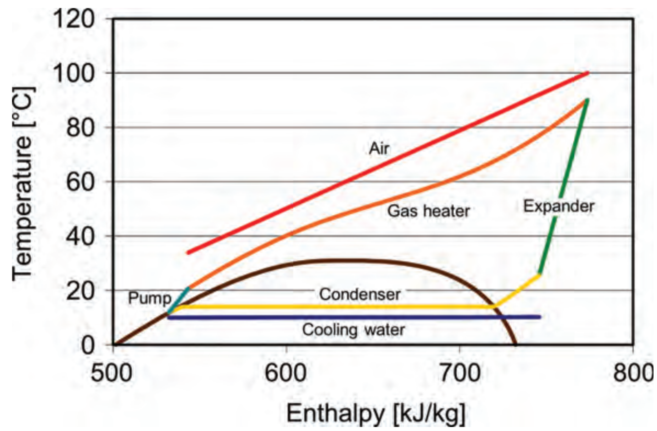

Figure 3. T–h *chart, CO_2.*

typically 10 MPa at heat absorption. This gives a potential for component size reduction which leads to investment cost reduction. In addition, heat absorption without phase change can possibly ease source integration. It has also been shown that a CO_2 power cycle is suitable to take advantage of LNG regasification if available on the site [15].

Earlier studies have discussed and compared cycles running at their design point [16, 17]. However, there exists very little literature on how sensitive the cycles are to changes in the condition of the heat source. The present article is the second part of a study on the performance of transcritical and subcritical Rankine Cycles outside the design point. The first article [19] focused on the performance with a constant expander speed control strategy. The present article compares the performance of that strategy with the performance of a constant high-pressure control strategy and the optimum performance operation points. The aim is to show the differences between the working fluids and to point out difficulties at off-design operation.

2 SIMULATION MODEL

2.1 The simulation model principles

A spreadsheet simulation model was built in Excel, based on a refrigerant property library developed by SINTEF Energy Research and NTNU. The Span–Wagner equation of state [20] is used for CO_2 (R-744) properties, while the Chan–Haselden equation of state [21] with fluid coefficients from AlliedSignal is used for R-123. T–h charts for the two cycles at the design conditions are shown in Figures 2 and 3.

The model's solver calculates the heat transfer in the heat exchangers (gas heater/evaporator and condenser), based on specified (constant) heat transfer coefficients. Since the heat transfer coefficients highly depend on the phase of the fluid (liquid, two-phase or gas), a factor of 0.65 is used to reduce the heat transfer for the gas phase. This assumption is validated in [19].

If the correct values are set in the GUI, the model will output the missing ones. This allows the user to make different kinds of calculations (Table 1).

2.2 Model constraints and parameters

An installation of the cycles in an aluminum production plant in Norway was assumed. This produces hot air (100°C), which was used as the heat source. The mass flow was set to 1 kg/s, since it has no effect on the results (it will just scale up the whole cycle proportionally). It was assumed that unlimited amounts of water at 10°C were available as the heat sink.

The efficiency of the expander is set to 80% for both CO_2 and R-123. The CO_2 pump's isentropic efficiency is assumed to be 70%, while R-123 is assumed to be incompressible at this stage. The pump work is therefore calculated as the product of volume flow and pressure difference, which is acceptable because the value is insignificant. A subcooling of 2 K in the condenser was also defined, to avoid cavitation in the pump.

Table 1. *Calculation overview.*

	Step 1 Finding heat exchanger area	Step 2 Finding design point	Step 3 Expander speed const.	Step 4 High pressure const.	Step 5 Optimum control	Step 6 Optimum control
Heat source mass flow (kg/s)	1.0	1.0	0.7 - 1.6	0.7 - 1.6	1.0	0.7 - 1.6
Heat source temperature (°C)	100	100	90 - 120	90 - 120	90 - 120	100
Working fluid mass flow			Design Point	Design Point		
Working fluid high pressure				Design Point		
Heat exchanger area		Area	Area	Area	Area	Area
Volume flow at expander inlet			Design Point			
Heat exchanger minimum ΔT	10 K					

Legend Constant input: Optimized for max. work: Variable:

Irreversibilities such as pressure drop and heat loss to the ambient were neglected.

2.3 Optimization and simulation cases

2.3.1 *Case setup*

The design point for each cycle was found in two steps. First, a minimum temperature difference of 10 K in the main heat exchanger was assumed. Based on this value, the optimum combination of working fluid mass flow, high pressure and heat exchanger area was found (step 1 in Table 1). During this optimization, the mass flow and high pressure were varied to find the maximum work output, while the heat exchanger area was continuously changed to fulfill the constraint on temperature approach. The object function for optimization was based on absolute work output and not thermal efficiency, since it has earlier been shown that this is not a good evaluation parameter for Rankine cycles when utilizing a constrained heat source with gliding temperature in a case where there is no other use for surplus heat [17]. The heat exchanger area found in step 1 was then used for all future calculations.

In step 2, the minimum temperature difference constraint was removed and instead the area found in step 1 was fixed. The cycle's final design point was then found by optimizing the mass flow and high pressure of the working fluid again. A 2% increase in work output compared with step 1 was found for both cycles. This was achieved with a decreased mass flow, which led to lower temperature differences in the heat exchangers (5.0 K for CO_2 and 8.6 K for R-123). This shows that the temperature difference alone is not a sufficient design parameter.

2.3.2 *Off-design investigation*

To analyze the off-design behavior, case studies were performed for each cycle and different control strategies, with air temperatures from 90 to 120°C (step size of 2.5 K) and air mass flows from 0.7 to 1.6 kg/s (step size of 0.05 kg/s).

As the first control strategy, the rotational speed of the expander was kept constant (step 3) by locking the volumetric flow through the expander inlet. The mass flow rate of the system was kept constant, since it was assumed that the density at the pump inlet was constant (small changes in the pressure and temperature in a liquid fluid) and that the pump was running on constant speed.

The second control strategy (step 4) was to vary the expander speed to keep the high pressure constant (the pump speed was still constant).

The third control strategy was to vary both expander speed (high-pressure control) and pump speed (mass flow control). One case with varying heat source temperature was run with a constant heat source mass flow of 1 kg/s (step 5) and one case with varying heat source mass flow was run with a constant heat source temperature of 100°C (step 6). The optimum mass flow and high pressure were found in each point.

3 RESULTS

Simulation results for the two processes are illustrated in Figures 4 and 5. The work output has been normalized with the work output in the design point (heat source: 1 kg/s at 100 °C), which was calculated to be 4.14 and 4.97 kW for

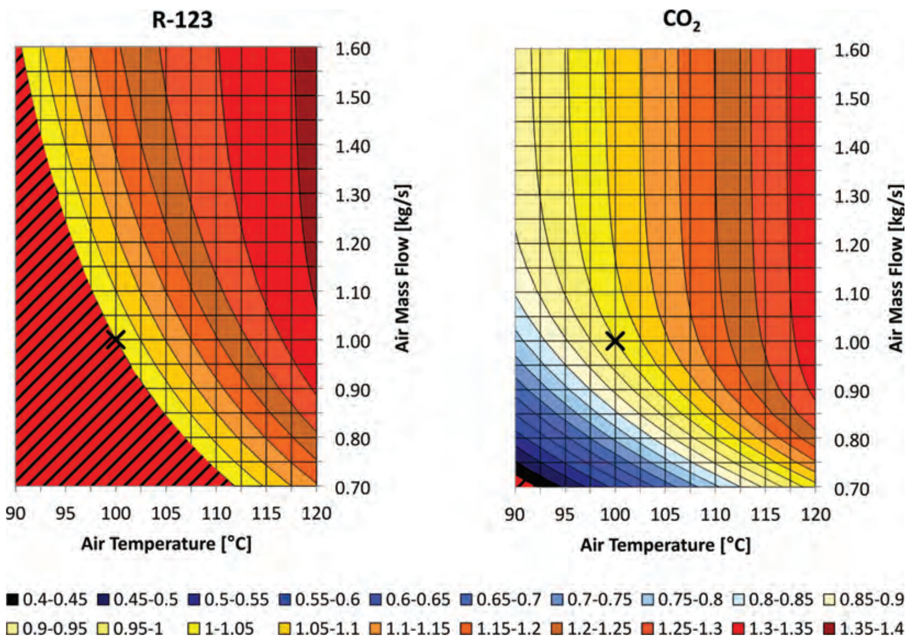

Figure 4. *Simulation results for constant expander speed control (black cross, design point).*

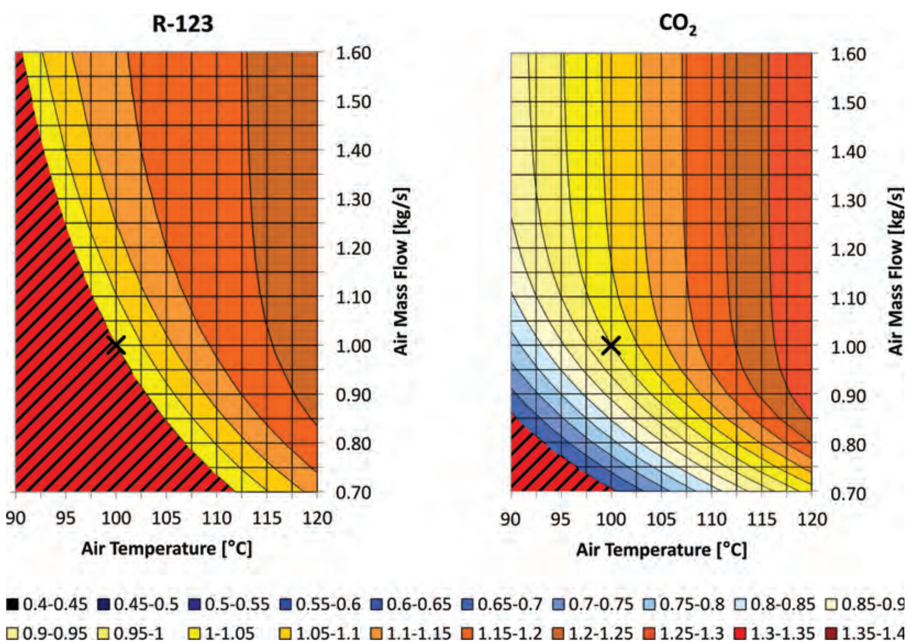

Figure 5. *Simulation results for constant high-pressure control (black cross, design point).*

the R-123 and CO_2 cycle, respectively. The black and red shaded area shows where the simulations indicate either wet inlet or outlet of the expander. This is considered a non-feasible area of operation, as erosion in the expander should be avoided. One could discuss how damaging it is for an expander to run with some liquid droplets; however, it is not desirable to operate at this condition as it would certainly reduce the expander's efficiency.

4 DISCUSSION

4.1 General observations

As described in [19], the R-123 cycle is much more vulnerable to reduction in available heat (either temperature or mass flow) compared with the CO_2 cycle. The work output from the CO_2 cycle is of course reduced when the amount of available heat is reduced, but unlike the R-123 cycle, it will not move into the non-feasible area. For R-123, the effect of superheating the fluid at the design point (to reduce the risk of droplets in the expander) was also investigated in [19]. However, it was found that the amount of superheat needed to significantly reduce the vulnerability would yield a substantial decrease in performance.

When the amount of available heat is increased, both cycles will operate at feasible conditions, but the increase in work output depends on the control strategy.

A more detailed analysis of how the cycles perform during off-design operation is given below.

4.2 Change in heat source mass flow

Figure 6 shows how the work output is influenced by a change of the heat source mass flow. Both cycles were plotted with all control strategies so that they can easily be compared. The areas of non-feasible operation were not plotted, which applies only to the R-123 cycle.

For small changes in the heat source mass flow, there is almost no difference between the two cycles: if the mass flow is increased by 5%, the work output from both cycles will increase by ~2.5%, independent from the control strategy.

For larger variations in the heat source mass flow, the performance of the CO_2 cycle rapidly declines for the simple control strategies. This shows that even though it will not move into the non-feasible area, the CO_2 cycle strongly depends on advanced control strategies. From heat pumping systems, it is well known that high-pressure control is very important for transcritical CO_2 systems. With constant expander speed, the amount of heat transferred from the heat source directly controls the high pressure and also the efficiency.

With optimum control, the CO_2 cycle has the potential to outperform the R-123 cycle when the mass flow of air increases. For a decrease in heat source mass flow, the R-123 cycle performs slightly better.

The optimum control for a change in heat source mass flow is further analyzed below. The required relative changes of high pressure, working fluid mass flow and expander speed for both cycles are shown in Figures 7–9, respectively.

As can be seen, the optimum control strategy for both cycles is basically a constant pressure strategy.

The optimum mass flow of the working fluid highly depends on the flow of the heat source and linear adjustment

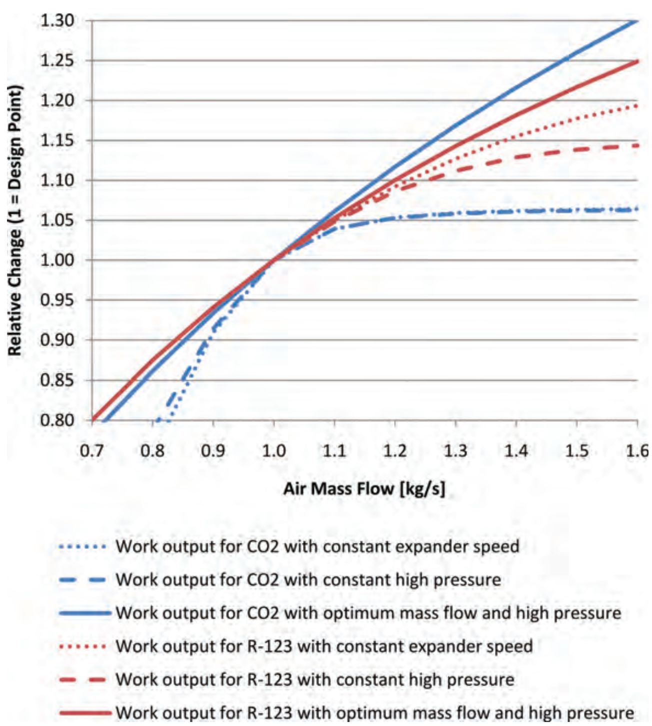

Figure 6. *Influence of heat source mass flow on work output for different control strategies.*

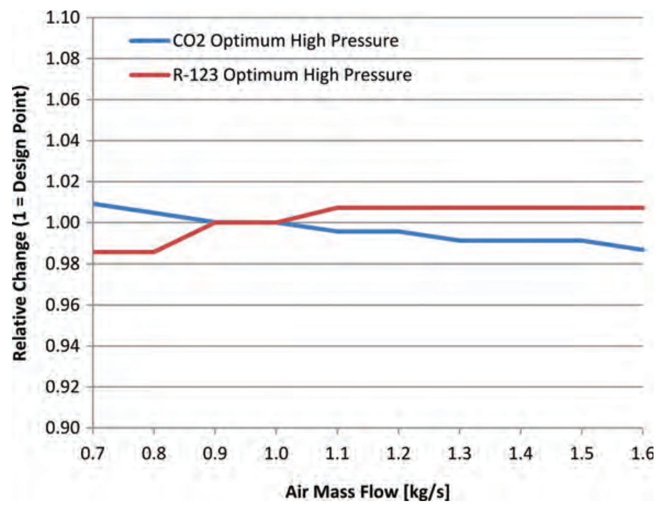

Figure 7. *Optimum high-pressure control for changes in heat source mass flow.*

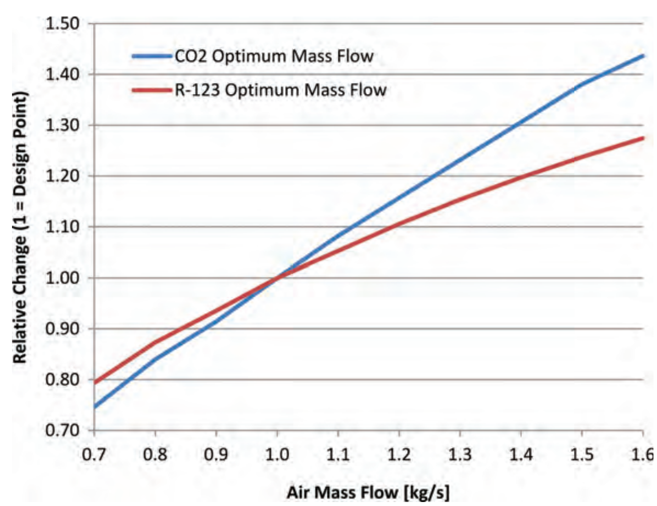

Figure 8. *Optimum mass flow control for changes in heat source mass flow.*

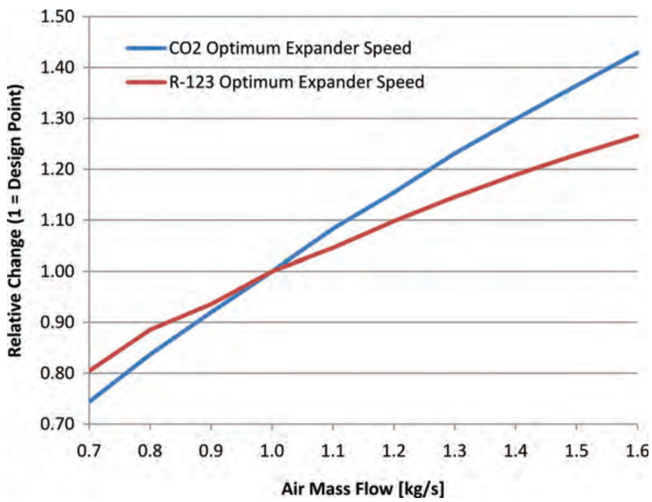

Figure 9. *Optimum expander speed control for changes in heat source mass flow.*

seems reasonable, as one would expect that the profile in the main heat exchanger should be maintained. The relative change for CO_2 is higher than for R-123.

Since the high pressure should be kept constant, the required change in expander speed is very similar to the change in mass flow. Continuous control of the expander speed in order to maintain optimum pressure is feasible, but additional power electronics are required to make the produced electricity compatible to the grid. Modern permanent magnetic generators could offer efficient and simple control for the turbine speed [22]. These components are not further described here, as it is not within the scope of this article. Their performance and costs have to be considered in an economic evaluation though. Since a 40% increase of volumetric flow is difficult for most turbines, the constant efficiency assumption is clearly no longer valid. Because of the higher relative change, this effect might be a bigger challenge for the CO_2 cycle. However, the aim of this study is not to find the optimum off-design operation curve, but to illustrate the challenges at off-design operation.

4.3 Change in heat source temperature

Figure 10 shows how the work output is influenced by a change of the heat source temperature. Again, both cycles are shown with all control strategies and only the feasible conditions are plotted.

The differences at small changes are marginal again: If the temperature is increased by 5%, the work output from both cycles will increase by ~10% independent from the control strategy.

For a large decrease in temperature, the feasible results are very similar. This means that the control strategy is not important for the CO_2 cycle, whereas the R-123 needs optimum control.

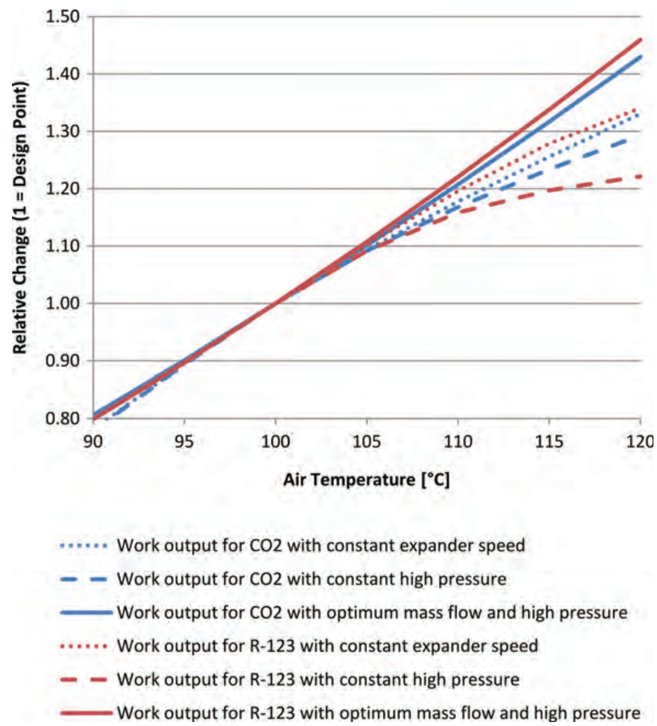

Figure 10. *Influence of heat source temperature on work output for different control strategies.*

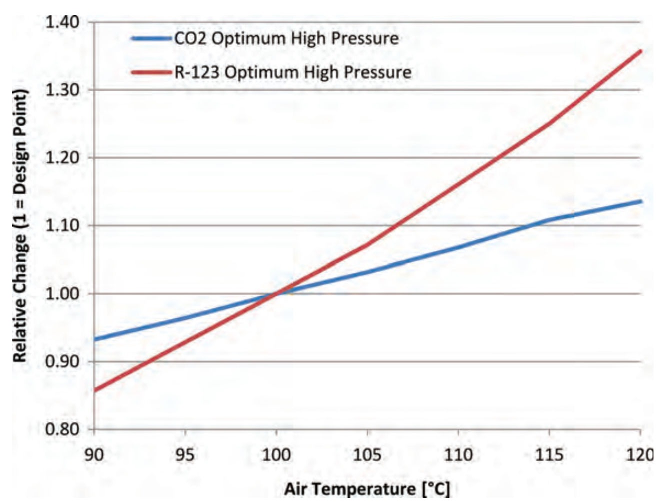

Figure 11. *Optimum high-pressure control for changes in the heat source temperature.*

For a large increase in air temperature, the control makes a much bigger difference. As expected, an optimum control leads to higher work output for both cycles; the difference between the cycles at optimum control is not significant though.

The required relative changes of high pressure, working fluid mass flow and expander speed for both cycles are shown in Figures 11–13, respectively.

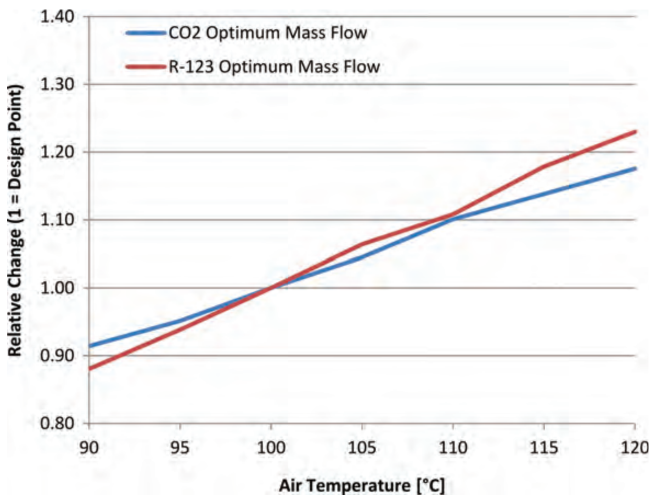

Figure 12. *Optimum mass flow control for changes in the heat source temperature.*

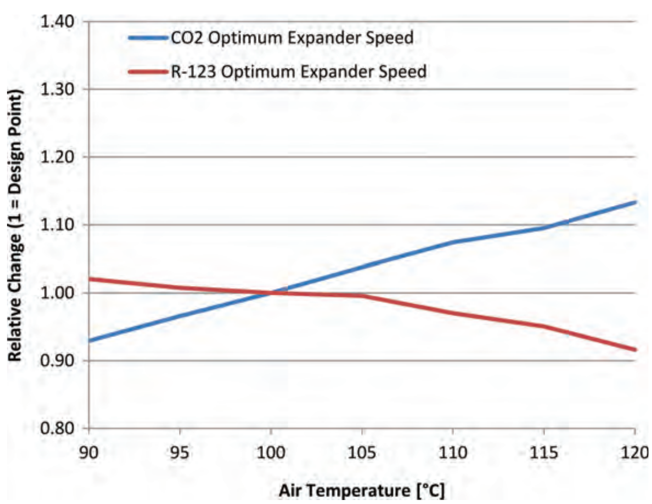

Figure 13. *Optimum expander speed control for changes in the heat source temperature.*

The mass flow has to be adjusted similarly for both cycles again and linearly follows the amount of available energy in the heat source.

The pressure change on the other hand is quite different: although both increase linearly, the relative change rate for R-123 is much higher. The absolute change is higher for CO_2 though, due to the much higher absolute level.

This leads to an interesting behavior for the change rate of the expander speed: the slope is positive for CO_2 and negative for R-123. For both cycles, the volumetric flow through the expander (and thus the expander speed) is increased by a mass flow increase and decreased by a pressure increase. It can be seen that for CO_2, the mass flow effect is bigger while for R-123, the pressure change has a bigger influence. This is important for the off-design control and shows that each cycle can have very different requirements.

As mentioned before, the compatibility to the grid has to be assured for both cycles, which will add costs and complexity to the systems.

5 CONCLUSION

The main goal of this work was to compare how an ORC and a transcritical CO_2 Rankine cycle respond to operation outside the design point with different control strategies and to illustrate challenges at off-design operation.

It was shown that the R-123 cycle needs a detailed control strategy when the amount of available heat is reduced to ensure the required superheat condition in the expander. For an increase in available heat, a constant expander speed yields better results than a constant high pressure, but the highest work output can be achieved with optimum control.

The CO_2 cycle is more robust and could be operated without advanced control. However, the performance decreases significantly for a change in heat source mass flow. If the heat source does not supply a constant mass flow, an advanced control is therefore highly desirable. Changes in heat source temperature are not that critical, although the optimum control leads to higher work output, especially for increasing heat source temperatures.

In order to obtain maximum work output, rather large adjustments in mass flow and expander speed are required for both cycles, especially for changes in heat source mass flow. This can lead to problems with the expander performance and grid compatibility.

The simulations showed only small differences in performance between the two cycles at optimum control. For a better direct comparison, a more thorough investigation on an absolute level is required. An advanced simulation tool including detailed models for the components is needed for this and should therefore be built. Other possible working fluids for the ORC should also be considered.

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