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

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ABSTRACT: 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. For the relatively low temperature heat, which is most commonly encountered for industrial surplus heat, the Organic Rankine Cycle is a well established technology. The transcritical Rankine Cycle recently received special attention due to its performances for energy recovery from low temperature sources. 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. This paper focuses on the off-design operation of Rankine cycles and compares the behaviour of transcritical CO_2 cycles and an ORC cycle with R-123 as working fluid. The main observation is that the ORC is very sensitive to reduction in available heat, and will with only small changes get droplets in the inlet of the expander. This shows that it is reasonable to operate the ORC with some degrees superheat, to have a buffer. Superheating the outlet 5 to 10 K has only a small effect on the cycle performance. However the gained robustness is also relatively small. With small increments in the available heat source, the CO_2 cycle also seem to have a marginally better response without control of the process, which indicates that it is more robust and less in need of detailed control.

Keywords: ORC, Power Production, Surplus heat

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 400C) [1].

The Organic Rankine Cycle (ORC) is now a well established technology for power production from low temperature heat sources. It combines improved efficiency, and lower capital and operating costs. The working fluids used are organic compounds 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 [4] plants or as used as bottoming cycle for gas turbines [5-6]. More scarce applications are to be found in solar application [7-8] or energy recovery from waste heat in industry [1, 9]. A commonly accepted limit for a profitable energy recovery plant is 200ºC for a gas heat source and 90ºC for a liquid heat source [10].

Research in ORC technology is very active, focusing both on component development [11] and working fluid selection [12-15].

Despite substantial improvements, power production from low to medium temperature heat sources is still handicapped by large investment cost 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 power cycle recently received special attention [16-19] 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 to the constant temperature of a 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 as low temperature differences as possible 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, typically 100bars at heat absorption. This gives a potential for component size reduction and then, 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 [16].

Earlier papers has discussed and compared cycles running at the design point [17-18]. However, there exists very little literature on how sensitive the cycles are to changes in the condition of the heat source. The current paper will focus on operation 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 Figure 1 and Figure 2

The high-side pressure and mass flow rate of the working fluid is set and the model solves the heat exchangers (gas heater/evaporator and condenser), based on specified heat transfer coefficients and area and heat sink and source specifications. For off-design simulations, the model can be set to keep the volume flow through the expander constant and instead change the high pressure side pressure. The gas heater/evaporator and condenser are divided into 30 and 20 segments each for the ORC and the CO_2 cycle respectively, with equal heat transfer area. The solver function in excel is used to find the outlet pressure from the expander that matches the heat transfer capabilities of the heat exchangers. This constrains the whole process to one possible solution.

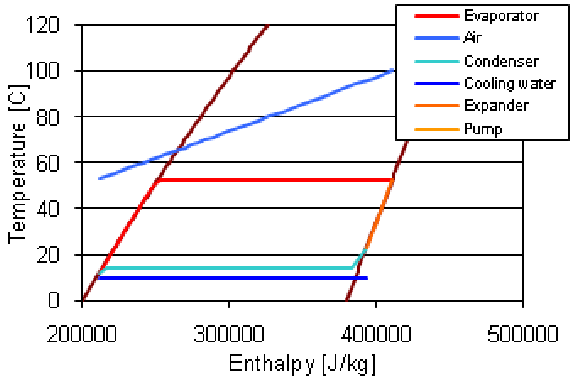

Figure 1: T-h chart, R-123

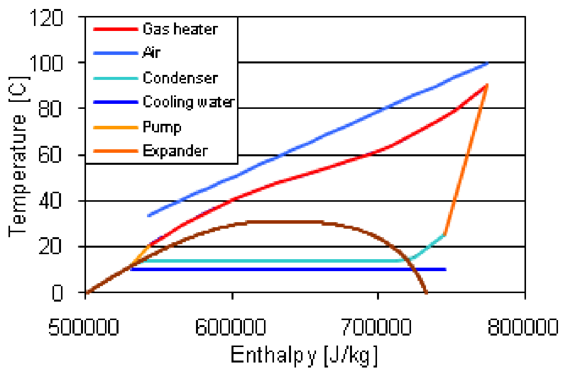

Figure 2: T-h chart, CO_2 cycle

Since the models uses the total UA value for calculation of the heat transfer in the heat exchangers, the actual heat transfer coefficients of each fluid is not important. However, since there are large differences in the heat transfer properties of a fluid during phase change (evaporation or condensation) and when superheated, the assumed heat transfer coefficient for a condensing or evaporating fluid is multiplied by 0,65. This assumption was validated by using an in-house heat exchanger simulation tool. The resulting change in heat transfer coefficient through a typical evaporator is shown in Figure 3 (the Gnielinski correlation [22] is used for single-phase and the Moser/Webb for two-phase [23] heat transfer, with polynomial transition between the correlations). It may be observed that the heat transfer coefficient of the liquid reduces the average HTC for the liquid/two-phase region. For supercritical CO_2, a constant heat transfer coefficient is assumed.

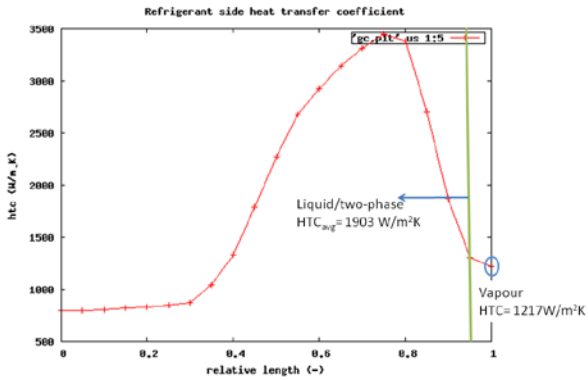

Figure 3: Heat transfer coefficient through an evaporator.

Since the heat exchangers are divided into elements with uniform area, one may risk that there is superheated gas at the inlet of an element and two-phase fluid at the outlet, or vice versa. In these cases the heat transfer coefficient is calculated with a linear interpolation between the heat transfer coefficient for superheated gas and two-phase flow, based on the inlet and outlet state of the element. An example of this is shown in Figure 1, where it seems as the temperature of the working fluid (R123) changes inside the two-phase area. This result in an error in the arithmetic mean temperature difference (AMTD) for the element, and thereby an error in the heat transferred in the element. However, this error is small and therefore neglected.

2.2 Model constraints and parameters

Air at 100°C and a flow of 1 kg/s was used as heat source. It was assumed that unlimited amounts of water at 10°C were available for heat rejection in the base case simulations. These assumptions are representative for an aluminium production plant in Norway (air mass flow rate is chosen for simplicity and has no effect on the results).

The efficiency of the expander is set to 80% for both CO_2 and R-123. The CO_2 pump isentropic efficiency is assumed to be

70 %, while R-123 is assumed to be incompressible at this stage. Even though R-123 might not be incompressible and a pump efficiency should be added, the pump work is so small that this minor error is neglectible.

A minimum temperature difference of 10 K in the main heat exchanger and 2 K in the condenser were design criteria used for the two systems. A subcooling of 2 K in the condenser was also defined, to avoid cavitation in the pump.

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

2.3 Optimization and simulation cases

Based on these assumptions, the working fluid high pressure and mass flow were optimized to obtain maximum work output at a design condition. The area of the heat exchangers was continuously changed during optimization, to fulfil the constraints. 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 the surplus heat [18].

The main goal for the simulations was to see how the cycles responded to operation outside the design point. When the heat exchanger areas for optimum pressure and mass flow were found, it was locked, and the conditions were changed. A case study was performed for each cycle with air temperatures from 90 to 120 °C (step size of 2.5 K) and air mass flow from 0.70 to 1.6 kg/s (step size of 0.05 kg/s). The simulations were performed as if the rotational speed of the expander was kept constant, 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) and that the pump was running on constant speed.

It appeared from the simulations that the R-123 cycle was very sensitive to variations in the heat source temperature and mass flow before entering into a non-feasible operating condition with a liquid fraction at the outlet of the expander. The effect of superheating when operating at the design point was therefore investigated in addition.

3. RESULTS

Simulation results for the two processes are illustrated in Figure 4. The output work has been normalized with the work output in the design point (Tair source = 100 °C and mass flow air = 1 kg/s), which was calculated to be 4.1 kW and 4.9 kW for the R-123 and CO_2 cycles 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 one want to avoid erosion in the expander. One could discuss how damaging it is for an expander with some liquid droplets, however it is not desirable to operate with this condition and it would certainly reduce the isentropic efficiency of the expander.

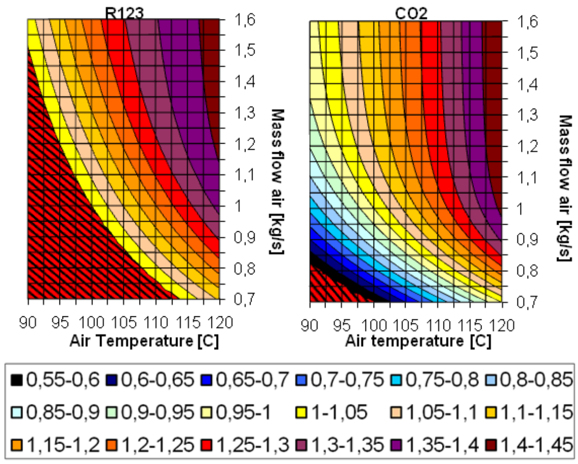

Figure 4: Simulation results from the two cycles when the volume flow through the expander is kept constant. The black cross indicates the design point.

Figure 5 shows the results from simulations of the R-123 cycle with different degrees of superheating at the evaporator outlet. Figure 5a) shows at what air mass flow and temperature the outlet of the R-123 evaporator will reach the dew point line, i.e. if the available heat is reduced further there will be liquid droplets in the expander inlet. Figure 5b) shows the relative reduction in work with increased amount of superheating.

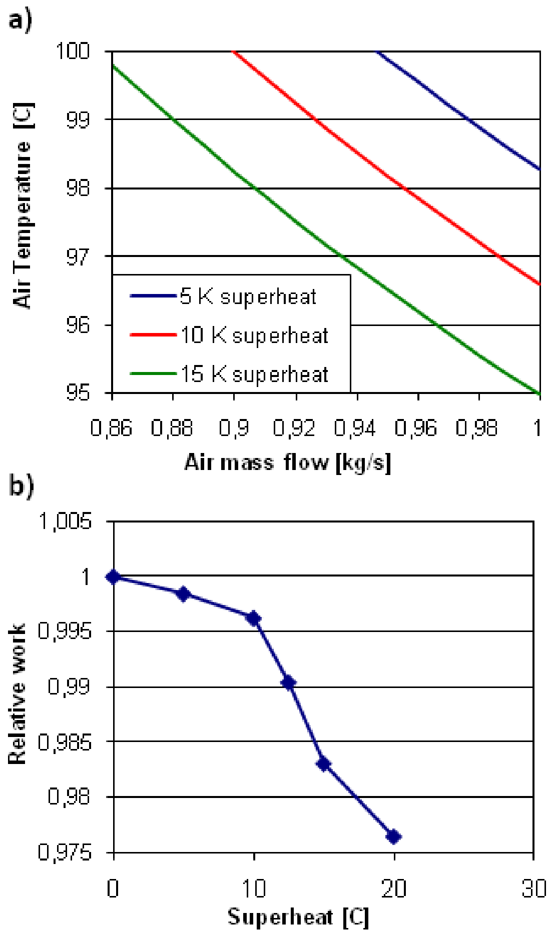

Figure 5: Superheating, a) critical operation line, b) reduction in work at design point.

4. DISCUSSION

The simulations at the design point show a 20 % higher work output for the CO_2 cycle than the R-123 cycle. The results are not necessarily directly comparable. ΔT_{min} is used as design criterion. It results in a much higher heat exchanger area for the CO_2 cycle (the CO_2 cycle can reach a much smaller mean temperature difference than the R123 cycle with the same ΔT_{min}). However, an equal mean temperature difference would not be correct either, as R-123 would be able to efficiently utilise a much smaller heat transfer area and one could expect better heat transfer coefficients from CO_2. (chen06) showed a difference of about 2 % in favour of the CO_2 cycle with air temperature of 150 °C, mass flow rate of 0,4 kg/s and equal thermodynamic mean temperatures for the two fluids in each heat exchanger. It is however expected that the advantage of CO_2 increases with decreasing heat source temperature as the good temperature match with the source will be more important.

The most obvious observation when studying the simulation results, is how much more vulnerable the R-123 cycle is to reduction in available heat (either reduction in heat source temperature or mass flow), compared to the CO_2 cycle. The work output from the CO_2 cycle is of course reduced when the amount of heat added is reduced, but the R-123 cycle will move straight into the non-feasible area. This is due to the fact that the optimal operation point (design point) for the R-123 cycle will always be with the inlet of the expander at the saturation line. With the operating conditions set in these simulations the outlet of the evaporator will always be wet when the amount of heat transferred is reduced.

Because of the ORCs sensitivity to reduction in the heat source temperature and mass flow, it would be reasonable to operate with some superheat at the inlet for the expander, as a buffer. Superheating of the gas, however, results in lower work output. Results from simulations illustrated in Figure 5b), show that with 5 or 10 K superheating the reduction in work output is small.

Figure 5a) shows at what air mass flow and temperature the outlet of the R123 evaporator will reach the dew point line, i.e. if the available heat is reduced further there will be liquid droplets in the expander inlet. In most literature it is normal to assume 5 K superheating in the evaporator (holdman07a for example). As one can see from Figure 5a), a cycle would then be able to handle 6 % reduction in air mass flow or 2 % reduction in air temperature with constant turbine speed.

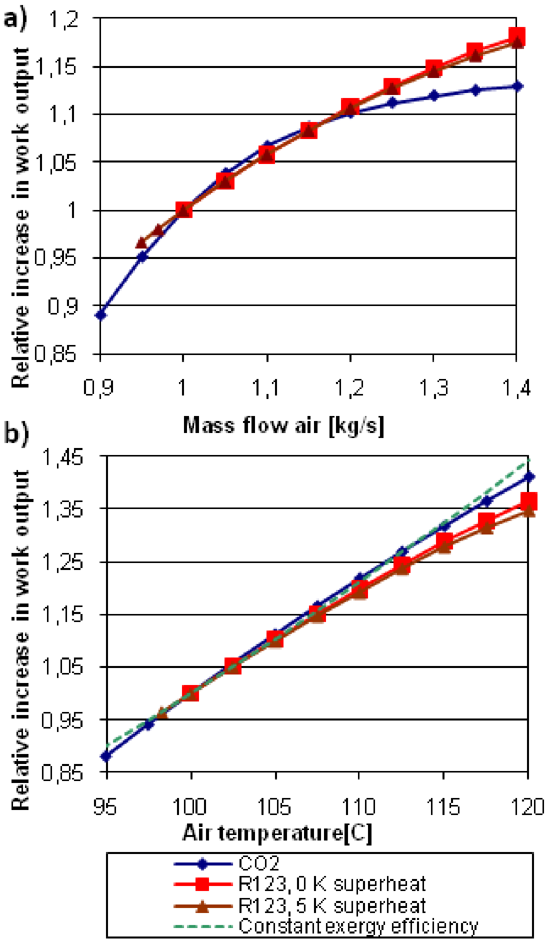

Figure 6: Relative reduction in work output with: a) constant heat source temperature (100 °C), b) constant a mass flow (1 kg/s).

The CO_2 cycle can accept much higher reduction in available heat, but the work output is rapidly reduced. From Figure 6 one can see that the R-123 cycle (with 5 °C superheat) has a lower relative reduction in work output than the CO_2 cycle when the amount of available heat is reduced within the defined operating limits. This is because the amount of superheat is reduced and the cycle actually moves closer to a thermodynamically optimal operating point. However, the superheating aggravates the response when the heat source is increased.

When the amount of available heat is increased, both cycles will operate at feasible conditions, but the increase in work output is different between the two cycles and results in different control strategies. If the mass flow of air is increased with 5 % while the rotational speed of the expander is kept constant, the work output from the CO_2 cycle increases with 3.8 %, while the work output of the R-123 cycle increases with 3 %. For larger changers in the heat source, a good control strategy would be important.

Figure 6a) shows how the work output increases with increasing mass flow of air. This indicates that the CO_2 cycle adapts better to small changes in the air mass flow, but the gradient of the increment will rapidly be reduced, compared to the R-123 cycle. However, with a variation of 20-40 % in the air flow rate, the system should be controlled to meet the optimum conditions.

From heat pumping systems, it is well known that high pressure control is very important for transcritical CO_2 systems. With constant turbine speed the amount of heat transferred from the heat source will directly control the high pressure and the efficiency will rapidly decline. Continuous control of the expander speed is no problem, since a high speed turbine with generation of direct current will be used in a real plant.

The main reason for the R-123 cycle's being superior to the CO_2 cycle at large increments in the heat source mass flow, is that the constant speed control partially counteracts superheating of the R-123 in the evaporator, by increasing the pressure.

When the air temperature is increased (with constant air mass flow), the CO_2 cycle is again better with small increments, but now the CO_2 cycle is also better with larger changes, see Figure 6b). If the air temperature is increased with 5 % while the rotational speed of the expander is kept constant, the work output from the CO_2

cycle increases with 11 %, while the work output of the R-123 cycle increases with 10 %.

For the same reason as with reduction in the heat source mass flow, the R-123 cycle is marginally better than the CO_2 cycle when the heat source temperature is reduced.

In Figure 6b), a green dotted line has been added that show what the relative increase in work output would be if the exergy efficiency of the cycle was kept constant. It is interesting to see that the exergy efficiency of the CO_2 cycle actually increases with small increments in the air temperature. It should however be possible to reach higher exergy efficiencies with higher air temperatures, as the relative exergy loss in the evaporator/gas heater becomes smaller (with the same temperature differences).

5. CONCLUSION

Main goal in this work was to compare how an ORC and a transcritical CO_2 Rankine cycle responds to operation outside the design point without active control of the system. The main observation is that the ORC is very sensitive to reduction in available heat, and will with only small changes get droplets in the inlet of the expander. This shows that it is reasonable to operate the ORC with some degrees superheat, to have a buffer. Superheating the outlet 5 to 10 K has only a small effect on the cycle performance. However the gained robustness is also relatively small.

With small increments in the available heat source, the CO_2 cycle also seem to have a marginally better response without control of the process, which indicates that it is more robust and less in need of detailed control.

It is clear that the R123 and CO_2 cycle demands quite different control strategies. While the R123 cycle should be controlled to give minimum superheat, the optimisation of the CO_2 cycle is more complex. The optimum operation point based on the process and conditions is much more difficult to define, and detailed simulations and optimization for each case will be necessary. However, experience from laboratory plants indicates that the CO_2 Rankine cycle is easier to control than CO_2 heat pumps, since the mass flow is almost only dependent on the pump speed and not on the heat rejection pressure.

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