COMBINING AN ORGANIC RANKINE CYCLE AND A HEAT PUMP CYCLE IN A TEST PLANT FOR REVERSIBLE STORAGE OF ENERGY

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ABSTRACT

The heat pump cycle is a well-known concept to lift the temperature of a medium by using electrical energy and waste heat. The ORC-process on the other side offers the possibility to obtain electrical energy from thermal energy. Combining these two processes and adding a thermal storage enables an innovative and reversible energy storage concept called *Combined Heat Pump - Organic Rankine Cycle (HP-ORC)*. This paper describes preliminary considerations, design methods and the development of a HP-ORC-pilot-plant. The upper and lower storage temperature are critical factors, as they determine the storage capacity and influence the overall efficiency (power-to-power-efficiency). Methods of multi-criteria decision-making, such as Pareto-Optimization, identify favorable combinations. The selection of the working fluid has also a strong impact on the combined process, as apparatus properties alter, in some cases effecting significant increase in costs. The realization of the compressor of the heat pump and the expander of the ORC in one single machine is another complex task to be considered. Especially the lubrication system has to fulfill the demands of both modes, thus innovative designs of the oil supply and separation are shown.

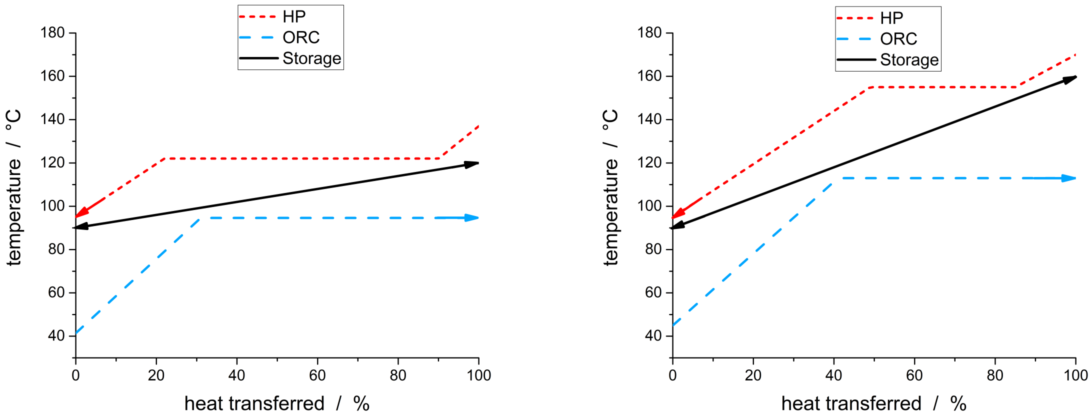
1. INTRODUCTION

A rising ratio of renewable energies results in the need for buffering the fluctuating energy input via storage systems (Henning and Palzer, 2013). The *Combined Heat Pump - Organic Rankine Cycle (HP-ORC)* offers an alternative to common concepts. As Staub et al. (2018) described the approach is based on thermal processes and a hot water storage tank: A heat pump process is used to store electrical energy as thermal energy in case of excess in the electrical grid. The thermal storage on the other hand fuels an organic Rankine cycle to provide electrical energy in case of need. The combination of a heat pump and an (organic) Rankine cycle to store and use electrical energy is a topic in several projects. Steinmann (2014) describes large scale applications at high temperatures, Lenk et al. (2015) present the idea within an sector integration and show the results of a lab scale high temperature heat pump. Within the *Energie Campus Nürnberg* there is an ongoing project to investigate the thermal upgrading of wast heat and the storage of electrical energy. With an experimental HP-ORC plant setup in pilot scale the whole concept will be evaluated, supporting the research on this topic, which is mainly theoretical so far. The design of this plant requires several considerations concerning temperature levels, fluid selection and lubrication, which are discussed in this paper.

2. OPTIMAL UPPER AND LOWER STORAGE TEMPERATURE

Depending on the operating mode, the storage medium acts as cooling liquid in the heat pump process and as heating liquid in the organic Rankine cycle. Since the storage medium is water, it can be circulated and fed directly into the heat exchanger. Figure 1 shows the loading and unloading processes in t-Q diagrams. It illustrates the amount of heat transferred in the heat exchanger at a specific temperature level (heat losses not included). The figure presents two different scenarios. The first (a) has a temperature lift from 90 °C lower water storage temperature to 120 °C upper water storage temperature, the second (b) from 90 °C to 160 °C storage temperature.

To optimize the exergy loss, the area between the graphs of the working fluid and the storage medium should be minimized. To achieve this goal the heat transfer surface has to be increased. To balance the costs for heat exchangers the minimal temperature difference at the PINCH point is set to 5 K. With this fixed minimal temperature difference, several combinations of upper and lower storage temperature were calculated via a MATLAB-script (steady process calculation, no losses, isenthalpic efficiencies of compressor and expander were assumed to be 0.7) regarding the HP and ORC, with an efficiency optimization for each combination and for several fluids. Hereafter the results for the fluid R1233zd(E) are exemplary shown. To evaluate the best combination significant criteria such as the effective electrical storage output of the heat storage and the overall efficiency of the whole process must be defined.



(a) storage temperature rise 90 °C to 120 °C (b) storage temperature rise 90 °C to 160 °C

Figure 1: t-Q diagrams for the energy transfer between sub-processes for different scenarios

2.1 ELECTRICAL STORAGE CAPACITY

The electrical storage capacity quantifies the amount of electrical energy, which can be released from the thermal storage tank via the organic Rankine cycle. Hence the definition is shown in (1)

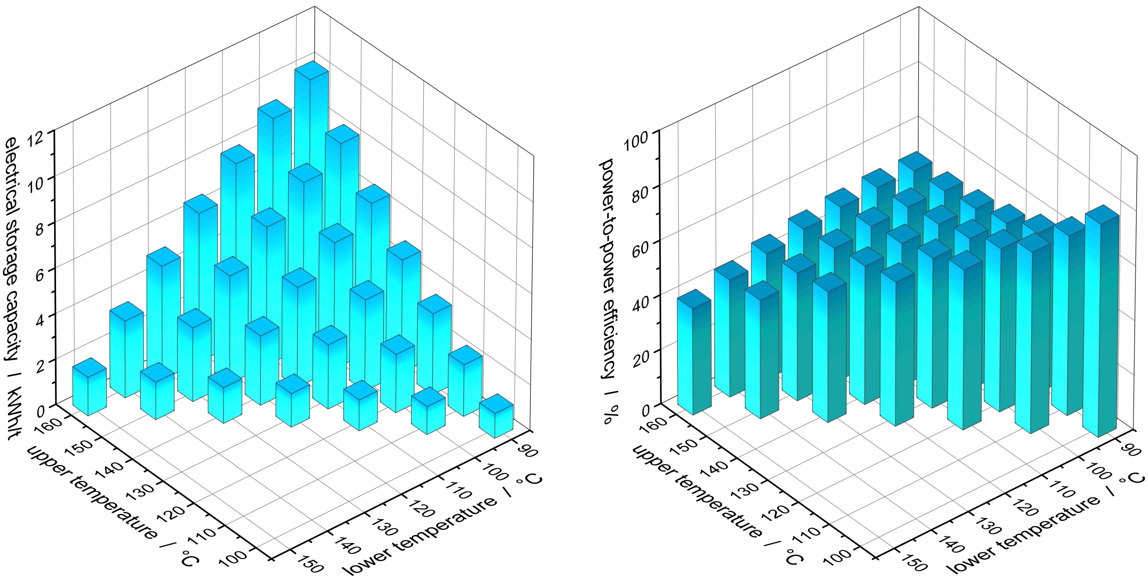
$$E_{\text{out}} = Q_{\text{stored}} \cdot \eta_{\text{ORC}} \tag{1}$$

where Q_{stored} is the thermal capacity of the storage and η_{ORC} is the efficiency of the ORC process.

It is

$$Q_{\text{stored}} = m \cdot \bar{c}_{\text{p}} \cdot \Delta T = m \cdot \frac{c_{\text{p,upper}} + c_{\text{p,lower}}}{2} \cdot (T_{\text{upper}} - T_{\text{lower}}) \tag{2}$$

with m being the mass of the water and $\bar{c}_{\text{p}}$ being the arithmetic mean of the heat capacities at upper ($c_{\text{p,upper}}$ at T_{upper}) and lower ($c_{\text{p,lower}}$ at T_{lower}) storage temperature. Figure 2 (a) shows the electrical storage capacity for several combinations of upper and lower storage temperatures. Obviously the storage capacity increases proportionally to the temperature difference, the variation of the heat capacity is marginal in that case. At constant temperature differences (diagonal groups of the bars) the ORC-efficiency rises for higher temperature levels, which is plausible regarding the definition of Carnot-efficiency.



(a) electrical storage capacity

(b) power-to-power-efficiency

Figure 2: Comparison of two relevant criteria to find the best the storage temperatures

2.2 POWER-TO-POWER-EFFICIENCY

Another important evaluation criteria of an energy-storage system is the overall or power-to-power-efficiency *PTP*. It describes the ratio between the amount of electrical energy provided by the storage E_{out} to the amount of the electrical energy which was stored E_{in}. An optimal storage system without any losses has a power-to-power-efficiency of $PTP = 1$ (the special case of reaching a theoretical $PTP > 1$ as Staub (2018) described is excluded here). Concerning the current case of a *Combined Heat Pump - Organic Rankine Cycle* and assuming no losses in the storage tank the power-to-power-efficiency is defined as:

$$PTP = \frac{E_{\text{out}}}{E_{\text{in}}} = \frac{Q_{\text{stored}}}{E_{\text{in}}} \cdot \frac{E_{\text{out}}}{Q_{\text{stored}}} = COP_{\text{HP}} \cdot \eta_{\text{ORC}} \tag{3}$$

with COP_{HP} being the coefficient of performance of the HP process.

Figure 2 (b) gives the power-to-power-efficiency of the complete process for several combinations of upper and lower storage temperatures. As the value consists of two factors and for the current case of using waste heat, the discussion of the influence of the temperatures must be divided: On the one hand the ORC-efficiency rises with higher temperature levels as seen before. On the other hand the decrease of the COP is much more dominant achieving the highest COP at lowest storage temperature respectively lowest temperature difference (see definition of the Carnot-COP). Therefore the overall efficiency shows a temperature dependency having the lowest value for highest temperature level and the highest value for lowest temperature difference and lowest temperature level.

2.3 PARETO OPTIMIZATION

As figure 2 shows, the two presented criteria do not point out an explicit optimal operation point. While the electrical storage capacity shows the best value at high temperature differences, the power-to-power-efficiency behaves contrary. Such optimization problems can be solved by methods of multi-criteria decision making algorithms. Classically a multi-criterion is calculated as a sum or a product of several criteria, which in some cases are weighted via a specific factor or exponent. The drawback of these methods is the determination of such weights since they crucially influence the solution. In the current case it is pointless to grade one of the two criteria higher than the other without knowing the exact impact these changes would have elsewhere.

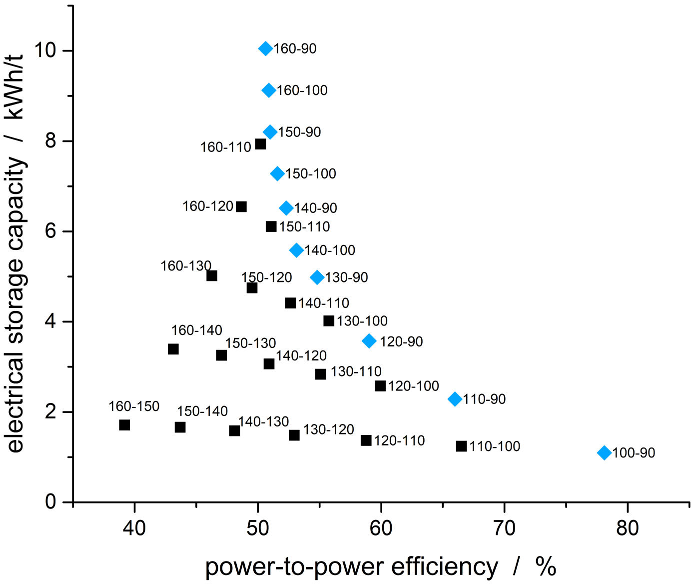


Figure 3: Pareto Optimization diagram (labels: upper storage temperature - lower storage temperature; Pareto frontier in blue)

An unbiased way to rearrange the considered cases is the *Pareto optimization* (Miettinen, 1999). Figure 3 shows the corresponding diagram, where the power-to-power efficiency is displayed on the x-axis and the electrical storage capacity on the y-axis. Each dot references a combination of upper and lower storage temperature indicated with two numbers. Highlighted with blue diamonds is the so called Pareto frontier, a set of Pareto optimal solutions. For each point beneath the Pareto frontier a better solution having higher values in PTP or E_{out} can be found. Within the Pareto optimal solutions it is hard to rate the cases, as long as it is unclear whether or how much one criterion is preferred. Although the *Pareto optimization* does not point out one best opportunity it can still exclude some less appropriate alternatives. To not overweight one of the two criteria a solution close to the symmetry line $y = x$ can be chosen. In the current case it is the combination of 120 °C upper and 90 °C lower storage temperature.

3. INFLUENCE OF THE WORKING FLUID ON APPARATUS COSTS

So far no influence of the working fluid has been considered. Previous simulations with capable working fluids showed, that for similar temperature levels there is just a marginal difference in the parameters *PTP* or E_{out}. Hence it is reasonable to take further criteria into account. This should be fluid properties like the pressure level at phase change, handling (flammability, toxicity, GWP, ODP) or availability on the market as they can have significant influence on the investment and operational costs. Another interesting factor is the specific enthalpy of vaporization Δh_{vap}.

3.1 ENTHALPY OF VAPORIZATION AS SELECTION CRITERION

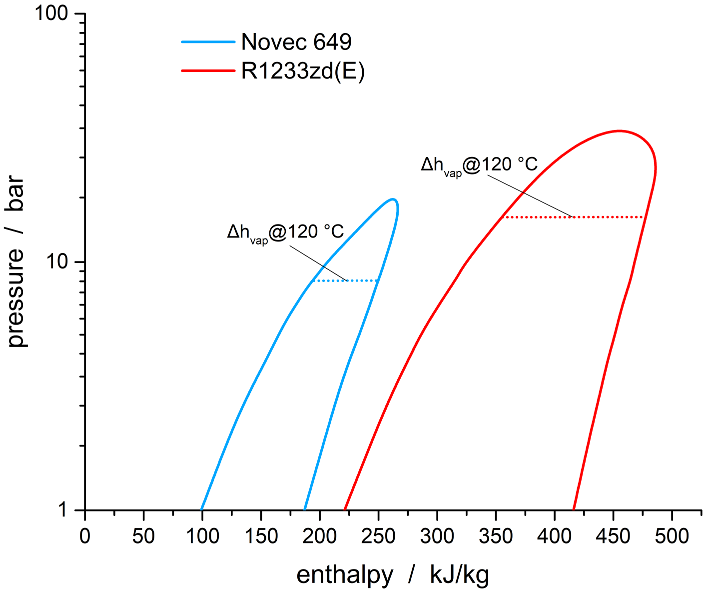


Figure 4: log(p)-h-diagram of 3M™Novec™649 and R-1233zd(E)

Henceforth two fluids shall be compared, both being suitable for the *Combined Heat Pump - Organic Rankine Cycle (HP-ORC)*, namely 3M™Novec™649 and R-1233zd(E). Figure 4 shows a log(p)-h-diagram of both fluids. The dashed lines indicate the enthalpy of vaporization at 120 °C. The curve of 3M™Novec™649 has a more narrow shape than that of R-1233zd(E), which can also be seen comparing the enthalpy of vaporization at 120 °C:

$$\Delta h_{\text{vap,Novec649}} = 61,31\ kJ/kg \approx \frac{1}{2} \cdot \Delta h_{\text{vap,R-1233zd(E)}} = \frac{1}{2} \cdot 123,44\ kJ/kg \tag{4}$$

Since the transferred heat ΔH_{vap} shall be the same for both cases, with the corresponding mass flows $\dot{m}_{\text{Novec649}}$ and $\dot{m}_{\text{R-1233zd(E)}}$

$$\Delta H_{\text{vap}} \overset{!}{=} const. = \dot{m}_{\text{Novec649}} \cdot \Delta h_{\text{vap,Novec649}} = \dot{m}_{\text{R-1233zd(E)}} \cdot \Delta h_{\text{vap,R-1233zd(E)}} \tag{5}$$

it follows equation (6):

$$\dot{m}_{\text{Novec649}} \approx 2 \cdot \dot{m}_{\text{R-1233zd(E)}} \tag{6}$$

Hence, if the transferred heat is supposed to be equal for the fluids, the mass flow in the 3M™Novec™649-scenario must be about the double of the mass flow using R-1233zd(E) as working fluid. Because of different density, the volume flows differ by the factor of 2.5.

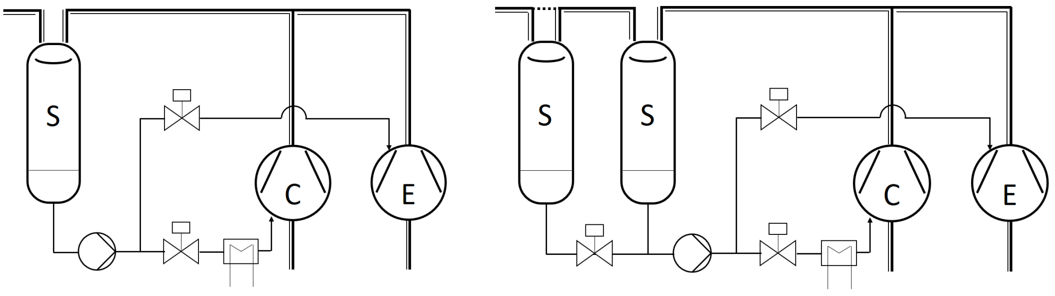
3.2 EFFECT ON APPARATUS DESIGN

The mass flow of course has an effect on the apparatus design. Higher mass flow (and therefore higher volume flow) often induces bigger cross section diameters to stay within proposed velocity borders. Having a huge impact in total costs already, particularly the heat exchangers can be a decision making part in the plant. In the current case heat exchangers for both fluids were designed and requested. The three heat exchangers (condenser, evaporater and internal heat exchanger) are in sum about 60 % cheaper for R-1233zd(E) compared to 3M™Novec™649. Although the mass flow is not the only factor for the cost of a heat exchanger, it can be considered to be the most crucial one. Similarly, the costs of the working machines are expected to increase with the suction volume flow. Besides this, the maximum pressure and the pressure ratio have to be considered.

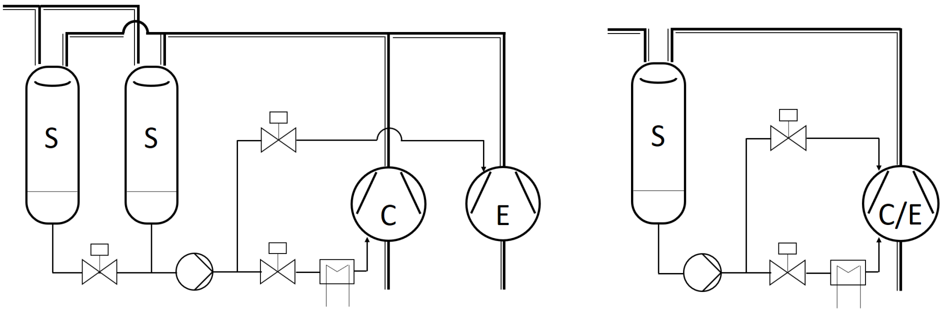
4. LUBRICATION CONCEPT OF A COMBINED COMPRESSOR-EXPANDER-MACHINE

To reduce investment costs to a minimum, the combination of apparatuses in both processes HP-cycle and ORC is pursued. Beside the reversible design of condenser, evaporator and IHX, the second main challenge is to realize the compressor (HP) and the expander (ORC) in one single machine. Previous works of Dumont (2017) show, the combination works well with rotating volumetric machines. Their common feature is the lubrication system ensuring the reduction of friction, sealing and heat transport. The combination of a compressor and expander complicates the design of the lubrication system, of course.

4.1 FLOW CHARTS OF LUBRICATION CONCEPTS



(a) Two machines (b) Two machines and adjustable serial oil separation



(c) Two machines and adjustable parallel oil separation (d) One combined machine

Figure 5: Several flow charts for the lubrication system (S: separator; C: compressor; E: expander)

The lubrication of a compressor and an expander differ. While the heat transport (cooling) is an important task of the oil supply in a compressor, the expander does not need further cooling, as the refrigerant itself lowers the temperature during expansion and the inlet temperature is already lower. Thus the mass flow of the oil can be reduced in the latter case. The oil separation is another topic. In the compressor it classically takes place directly after the compression on the high pressure side. Since the flow direction is reversed in the expander-case, one could assume the separator to be on the low pressure side. Yet often the opposite is the case: Using the lower volume flow on the high pressure side the separator can be significant smaller. Consequently the lubrication oil has to pass through the whole process until it is separated right before the expander, injected at the correct point and thereby mixed with the refrigerant again.

Figure 5 shows several possibilities for the lubrication system. Combining the both processes it starts with a parallel setting of expander and compressor sharing one oil separator (a). The separation for both cases takes place on the high pressure side for reasons described above. The oil is injected on each inlet side of the machines. Regarding the diverse oil flow rates, it can be necessary to adjust different separation rates with a serial or parallel setup, which leads to the configurations (b) and (c). According to demand the first separator can be bypassed to adapt to the right separation rate. A combined compressor-expander-machine needs two injection inlets as the flow direction varies (d). It is also conceivable to expand concept (d) by adding a second oil separator as described in (b) or (c). The pump is necessary to inject the oil at the high pressure side of the expander.

4.2 FURTHER CHALLENGES IN THE LUBRICATION SYSTEM

Apart from the challenging set up of the lubrication system, the oil separation in a reversible vessel needs further consideration. As there is no distinguished flow direction, the inlet and outlet and all inner devices of the separator must be symmetrical. For that reason gravity separators are more suitable than others (e.g. vortex separators). The high temperature at the compressor outlet is another issue to pay attention to. Besides the impact on the machine itself, the oil behavior (e.g. viscosity) changes with the temperature rise within the machine. The viscosity must not fall below the mandatory value to fulfill the lubrication task. Additionally the refrigerant has an influence on the viscosity of the lubricant. Further research on the given points is in progress and soon to be published.

According to the described basic engineering a HP-ORC pilot plant is designed and built. Having a nominal electrical power input of 15 kW the concept will be studied in short term and long term runs. The lubrication is one interesting focus of the experimental campaigns.

5. CONCLUSION

Within the paper the following new insights for the planning of reversible HP-ORC-plants which are of general relevance were presented:

- Crucial factors of the storage system are the upper and lower storage temperature, as they determine the power-to-power efficiency and the effective electrical power output. To find a favorable compromise between these two characteristic numbers, the *Pareto optimization* is a useful tool.
- Besides frequently discussed properties (GWP, ODP, flammability, etc.) the enthalpy of vaporization of the working fluid is very relevant in process design, as it can significantly influence the investment costs of the apparatuses.
- The lubrication system of machines in reversible processes is a diverse task. Several ways of arranging the components were shown, as well as further considerations like maximum temperature and viscosity.
- A HP-ORC pilot plant is currently under construction to get experimental experience on the storage concept and its partial aspects.

NOMENCLATURE

ORC	Organic Rankine Cycle
HP	Heat Pump
t	temperature
Q	thermal energy
E	electrical energy
m	mass
c_p	heat capacity at constant pressure
T	absolute temperature
PTP	power-to-power efficiency
GWP	Global Warming Potential
ODP	Ozone Depletion Potential

Subscript

out	released
stored	concerning the storage
upper	at upper temperature
lower	at lower temperature
in	brought in
vap	vaporization

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