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Experimental Investigations Of Waste Heat Utilization Of High-Temperature Heat Pump Compressors

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Experimental Investigations of Waste Heat Utilization of High-Temperature Heat Pump Compressors

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ABSTRACT

Renewable energy sources are playing an increasingly important role in the supply of electricity. However, in the European context, renewable energy sources still play a minor role in the heating sector, with about 21 % in 2018, although this sector accounts for more than 50 % of final energy consumption (World Energy Council, 2020). In order to decarbonize the heating sector, the integration of high-temperature heat pumps (HTHPs) into renewable energy systems is a promising approach. Potential areas of application are geothermal systems or the use of waste heat from industrial processes. The goal is to utilize HTHPs to guarantee coverage during peak loads, enhance the thermal output of renewable systems, or enable waste heat utilization. Such system integration requires flexibility and robust part-load characteristics to offset significant fluctuations in demand. This study aims to examine the part-load performance of an HTHP at a laboratory scale experimentally.

The test system represents a HTHP with a thermal output of 35 kW and supply temperatures of up to 130 °C. The refrigerant trans-1-Chlor-3,3,3-Trifluoropropen (R1233zd(E)), with low global warming potential (GWP) and ozone depletion potential (ODP), is used as working medium. An internal heat exchanger (IHX) as well as water-cooled cylinder heads (CHC) are implemented to investigate their potential of optimizing the performance of the test rig.

The system's part load behavior is examined in a defined base scenario at a heat source temperature of 50 °C and a supply temperature of 100 °C. Additionally, increasing supply temperatures up to 130 °C are realized in conjunction with (and without) the CHC. The analysis is focused on the influence of the installed cylinder head cooling. The results show that the cylinder head cooling reduces the discharge gas temperature of the reciprocating compressor and thus ensures material-friendly operation, while the dissipated heat can be recovered and improves the efficiency of the system by up to 8 %. In addition, major influences on heat transfer can be identified, like a decrease of the pinch point in the condenser. Further recommendations for action can then be derived from this in the context of economic and technical optimization.

1. INTRODUCTION

Due to a very low share of renewable energies in the heating and cooling sector in Europe (World Energy Council, 2020), high-temperature heat pumps (HTHPs) becoming more and more important. The number of publications on HTHPs is increasing (Web of science, 2023) as are the review articles. Bamigbetan et al. (2017) identified the current challenges for research in refrigerant selection, innovative refrigerant mixtures and cascaded systems. In particular, the authors point out challenges regarding higher pressure ratios and compressor cooling. Arpagaus et al. (2018) emphasized the need to minimize heat losses, oil lubrication and to investigate scale up for industrial applications. The great potential of transcritical systems is identified by Adamson et al. (2022). Additionally, the authors formulated six challenges with the corresponding solution proposals. Sun et al. (2023) and Jiang et al. (2022) indicated an increase in efficiency at high-temperature lifts and the development of large-scale HTHPs above 1 MW thermal capacity as important steps in development. Khalid et al. (2023) also focused on improving the compressors by oil-free operation and optimizing heat transfer through lower mean temperatures. The mentioned review papers ((Arpagaus et al., 2018), (Bamigbetan et al., 2017), (Adamson et al., 2022), (Sun et al., 2023), (Jiang et al., 2022), (H. Khalid et al., 2023)) showed that piston compressors are often used in HTHPs, and the state-of-the-art heat pump cycle designs include internal heat exchangers. Increasing the lifetime and efficiency of the entire system, and of the compressor in particular, can therefore be defined as crucial aspects for the decarbonization of the energy market. One way of achieving these goals is to decrease the discharge gas temperature downstream of the compressor and thus reduce the thermal stress of the compressor itself and the working fluids and oils. The resulting waste heat can be utilized further for heating purpose. A possible plant design for such a use case of compressor waste heat utilization is presented in this study, in form of water-cooled cylinder heads (cylinder head cooling (CHC). In this context, the return temperature of a district heating network is used to cool down the discharge temperature of the compressor and the corresponding flow is reinjected into the heat sink inlet flow stream, entering the condenser. Thus, the heating capacity of the entire system is increased, and the thermal stress of the compressor is decreased. Such water-cooled cylinder heads in HTHP applications have been not examined extensively in literature yet. Exemplarily, Brendel et al. (2023) investigated different concepts of the waste heat utilization theoretically. The study shows well promising results. In particular, the COP increased of up to 18.7 % by using the heat sink as cooling medium for the CHC.

In this study, the results of experimental investigations of water-cooled cylinder heads in reciprocating compressors for HTHPs are presented. The test rig is a HTHP with a thermal capacity of 35 kW and the hydrochlorofluoroolefine (HCFO) R1233zd(E) is used as refrigerant. In the following chapters, the methodology is presented, followed by the results and the conclusions.

2. Methodology

In this chapter, the heat pump test rig is described, the measurement procedure and evaluation methodology are presented.

2.1 Experimental setup

In order to investigate the influence of the CHC in different reliable part load and off-design operating points, a test rig of a HTHP with a thermal capacity of 35 kW is analyzed. The experimental setup was initially introduced in 2022 (Jeßberger et al., 2022a) and is analogous to the experimental setup presented in the work of Jeßberger et al. (2022b). As working fluid trans-1-Chloro-3,3,3-trifluoropropene (R1233zd(E)) is used. This hydrochlorofluoroolefine (HCFO) has a GWP of 1 CO_2e and an ODP of 0.00034 R-11e (IPCC, 2014). The flow chart of the test rig is presented in Figure 1. All pipes and heat exchangers are insulated to minimize the system's heat losses. In the following section, the heat pump circuit is described based on the flow chart. At state point 1, the suction gas enters the reciprocating piston compressor and is compressed and superheated at state point 2. The compressor speed is varied using a software-controlled frequency converter, ranging from 758 rpm to 2100 rpm, to investigate the part load behavior of the HTHP. The refrigerant then is led through the oil separator, which separates the entrained oil and ensures optimal heat transfer in the condenser. At state point 4, the working fluid is subcooled as it releases thermal energy during condensation.

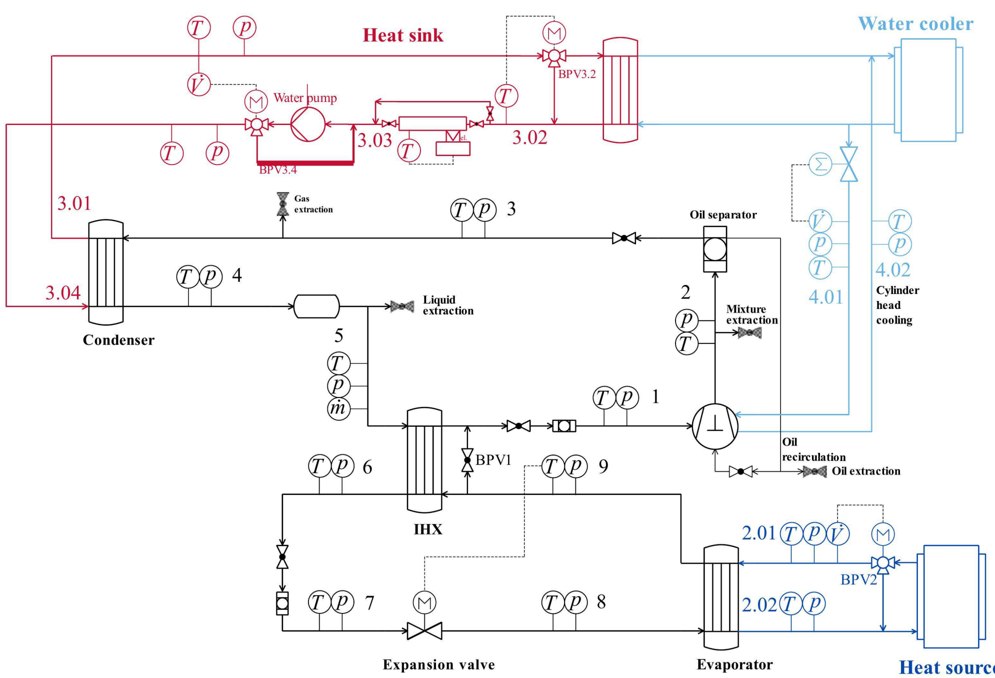

Figure 1: Flow chart of the HTHP test rig.

After the liquid tank, the subcooled working medium enters the internal heat exchanger (IHX) and is coupled to the outlet stream of the evaporator (state point 9 to 1) to ensure sufficient superheating of the suction gas. On the high-pressure side, the expansion valve leads to an isenthalpic expansion from state point 8 to state point 9. In the following, the refrigerant is evaporated and superheated by 5 K. The heat source is simulated by a tempering device that uses water as a medium. The temperatures and volume flow rate can be controlled through software.
In addition, the test rig enables the following analysis and plant configuration:

- Sampling points for liquid, gaseous and oil-refrigerant mixtures.
- Variable internal heat exchanger surface by using a bypass valve.
- Water-cooled cylinder heads for compressor waste heat utilization (see also (Jeßberger et al., 2024)).

A primary water circuit is used as a heat sink. The water pump and bypass valve (BPV3.4) control the heat sink outlet temperature ($T_{3.01}$), and the bypass valve after state point 3.1 (BPV3.2) controls, in combination with a heating device, the inlet temperature of the condenser ($T_{3.04}$). The house cooling water is used to cool the cylinder heads, with a mean temperature of 16 °C, illustrated in Figure 1. This must be considered, as the temperature difference is not so great that the outlet temperature can be used for heating purpose as in real applications. A simulation of the real scenario, where the heat sink inlet temperature is used (compare Figure 2), is carried out in order to compare the test results and confirm the statements, but will not be focussed in this study.

2.2 Measurement procedure and error analysis

To investigate the influence of the CHC on the system performance as well as on the performance of the single components, the HTHP system is evaluated at different part load operation points. A base scenario is defined where the CHC is used to decrease the discharge gas temperature downstream of the compressor and thus reduce the thermal stress concerning the compressor itself, the working fluids and lubrication oil. The resulting waste heat can be utilized further for heating purpose. Figure 2 presents the flow chart for a possible design of the compressor waste heat utilization by using CHC. Here, the return temperature of a district heating network is used to cool down the discharge temperature of the compressor and the corresponding flow is reinjected into the heat sink inlet flow stream, entering the condenser. Thus, the heating capacity is increased, and the thermal load is decreased. So, the new coefficient of performance (COP) leads to:

$$COP_{\mathrm{CHC}} = \frac{\dot{Q}_{\mathrm{sink}} + \dot{Q}_{\mathrm{CHC}}}{P_{\mathrm{el}}} \tag{1}$$

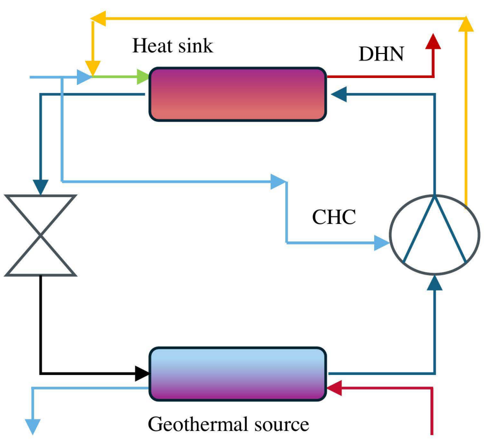

Figure 2: Flow chart for compressor waste heat utilization with CHC.

Where $\dot{Q}_{\mathrm{sink}}$ is the thermal power at the condenser, $\dot{Q}_{\mathrm{CHC}}$ the absorbed thermal power by the CHC, and P_{el} the consumed electrical power by the compressor. Regarding this base scenario, the temperatures are fixed in the heat source and heat sink. The heat source inlet temperature is set to 50 °C with a return temperature of 40 °C. The heat sink simulates a district heating network (DHN) with a supply temperature of 100 °C and a return temperature of 90 °C. Using these fixed temperatures, the compressor speed is used to serve the required flexibility and to vary the thermal power. Additionally, the volume flow rate in the cylinder heads is also varied from 5 L·min^{-1} to 10 L·min^{-1}. Each operating point is measured in a steady state over a period of 10 minutes. Additionally, the temperature lift (ΔT_{lift}) is varied by increasing the supply temperature from 100 °C to 130 °C in 10 K steps, at heat source temperatures of 60 °C and 70 °C.

For the conducted experiments, the steady-state conditions are defined according to a change in the discharge temperature (T_2), and the heat sink temperatures ($T_{3.01}$ and $T_{3.04}$) differ in a range below 1.5 K over a period of 10 minutes (see measuring points in Figure 1). Figure 3 illustrates this steady state and shows the behavior of the software based PID controllers. The measurement uncertainties are caused by the uncertainty of sensors and of the measurement board. It is essential to distinguish between a full-scale uncertainty (FS) and the operating range uncertainty (OR). FS refers to the maximum measurement deviation as a percentage of the total measurement range, while OR indicates the maximum deviation within the actual range used during a measurement.

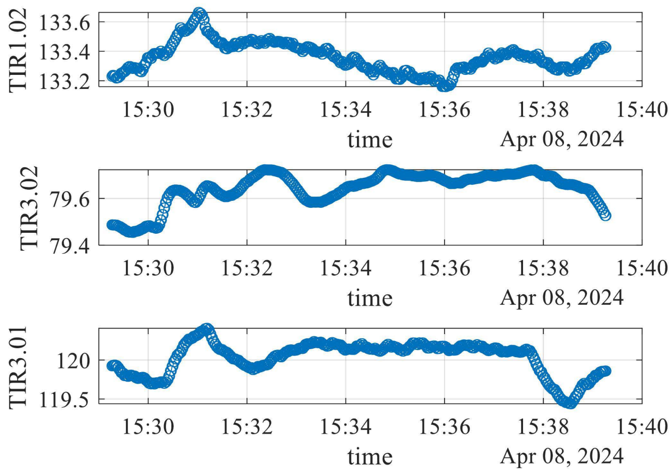

Figure 3: Steady-state conditions for an exemplary operating point.

The reciprocal influence of the individual errors is calculated with the help of the Gaussian error propagation, according to DIN 1913-4 (Deutsches Institut für Normung e.V., 1999). By calculating a parameter y as a function of different parameters x_i, the combined uncertainty Δy is defined as:

$$\Delta y = \sqrt{\left(\frac{\partial y}{\partial x_1}\cdot\Delta x_1\right)^2 + \left(\frac{\partial y}{\partial x_2}\cdot\Delta x_2\right)^2 + \cdots + \left(\frac{\partial y}{\partial x_n}\cdot\Delta x_\mathrm{n}\right)^2} \quad (2)$$

where Δx_i is the independent variable, generally provided by the manufacturer and shown in Table 1. Δx_i compromises two components: the measurement uncertainty of the sensor itself and the error introduced by the measurement board.

Table 1: Sensors and uncertainties

Sensor	Type	Range	Sensor uncertainty	Board uncertainty
Temperature	Omega, PR-22-3-100-1/3-M3-100-M12	-30 °C to 350 °C	dT = ± (1/3 · (0.30 °C + 0.005 · T))	± 0.15 °C
Pressure	Omega, PAA23SY-C-5-M12, 5 bar abs.	-1 bar to 5 bar	± 0.7 % FS	± 0.76 %
Pressure	Omega, PAA23SY-C-20-M12, 20 bar abs.	-1 bar to 20 bar	± 0.7 % FS	± 0.76 %
Mass flow refrigerant	Endress+Hauser, Proline Promass 40E	0 kg/h to 18,000 kg/h	± 0.5 % OR	negligible
Volume flow water circuits	Siemens, SITRANS FM MAG 3100 P/5100 W	-	± 0.4 % OR ± 1 mm/s	negligible
Electrical Power	-	-	± 0.7 % OR	negligible

Following this methodology, the base scenario depending on the boundary conditions of the DHN is investigated. To show additionally the impact on higher heat sink temperatures and different temperature glides, the temperature lift is varied, with varying the heat source inlet temperature at the same time. The temperature glide is defined as the difference between the inlet and outlet of the water circuit on the condenser ($T_{3.01}$ - $T_{3.04}$) and evaporator ($T_{2.01} - T_{2.02}$). The glide at the condenser and evaporator are kept constant, with 40 K at the condenser and 10 K at the evaporator. This corresponds to a decreased return temperature of the DHN, and its effect on the COP of the heat pump could be quantified. The operating points are presented in Figure 4.

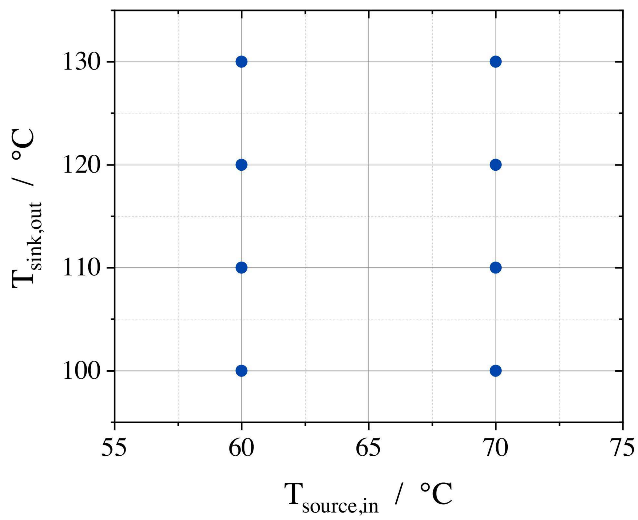

Figure 4: Operating points concerning heat sink temperatures for the measurement campaign .

3. Results

The results are evaluated on the compressor level, regarding the isentropic and volumetric efficiency, as well as the discharge temperature and superheating. On system level, COP and the heating capacity are analyzed. Additionally, the heat transfer characteristics in the condenser are examined. Figure 5 shows the discharge temperature as function of the compressor speed on the left hand side and the discharge superheating on the right hand side. The CHC5 campaign leads to a temperature drop of up to 23 K and CHC10 of up to 28.7 K at a compressor speed of 1058 rpm. At compressor speeds below 1000 rpm, superheating is too low. Therefore, the corresponding results are not analyzed further here. With a higher compressor speed the temperature drop due to the CHC decreases to 14 K and to 17 K, caused by the increasing mass flow of the refrigerant by increasing the compressor speed.

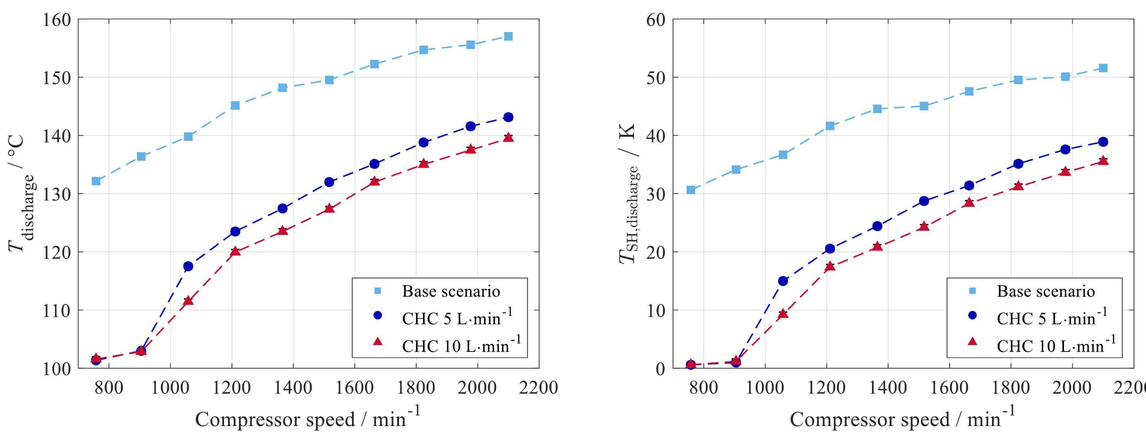

Figure 5: Discharge temperature and discharge super heating as function of the compressor speed.

Due to constant condensation temperatures the superheating on the right-hand side behaves similarly. Concerning a critical temperature of the refrigerant of 165.5 °C, the base scenario is on the temperature limit for the fluid with 8 K below the critical temperature. By using the cylinder head cooling, the pressure can stay constant, and the discharge temperature can be decreased. At compressor speed under 1000 rpm, Figure 5 shows that care must be taken to ensure that the superheat after the compressor is still sufficiently high, and the refrigerant is not compressed with a certain liquid fraction. Figure 6 illustrates the isentropic and volumetric efficiencies as function of the compressor speed. In both diagrams it can be seen that the efficiencies decrease with increasing compressor speed, for compressor speed higher than 1000 rpm. The CHC shows a positive effect on both efficiencies with a maximum increase in the isentropic efficiency of 2.6 % and a maximum increase of the volumetric efficiency of 1.2 %. At compressor speeds below 1000 rpm the volumetric efficiency shows a significant decrease, caused by the decrease of suction gas temperature and corresponding insufficient superheating at the compressor discharge side. The isentropic efficiency of the compressor is also affected by the insufficient superheating of the discharge gas.

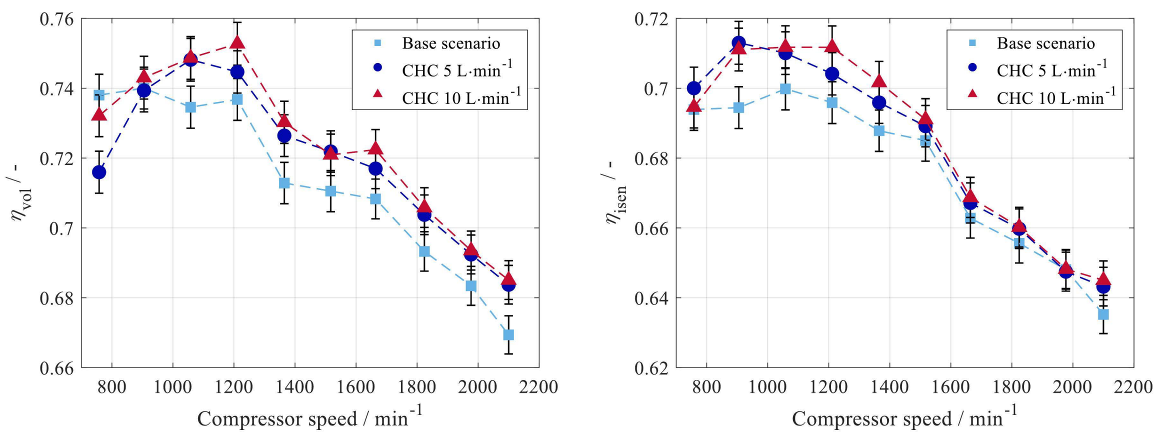

Figure 6: Isentropic and volumetric efficiency as function of the compressor speed.

Different parameters have an influence on the heat transfer efficiency of a plate heat exchanger. Figure 7 presents the minimum temperature difference ($\Delta T_{\mathrm{MITA,Cond}}$) and the logarithmic mean temperature ($\Delta T_{\mathrm{m,log,Cond}}$) as function of compressor speed.

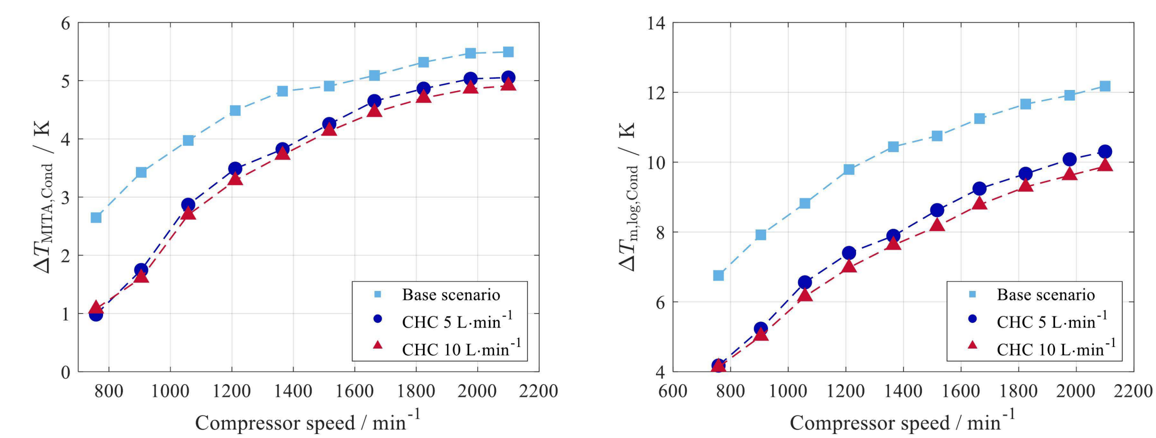

Figure 7: Minimum temperature approach and logarithmic mean temperature as function of the compressor speed.

The CHC leads to a decrease of the minimum temperature difference (pinch point). This is caused by the reduction of the condensation temperature to a minimum compared to the required heat sink outlet temperature. The logarithmic mean temperature's decrease is caused by the low discharge temperature as well as the decreased condensation temperature. Following the logarithmic mean temperature the overall heat transfer coefficient at the condenser is presented in Figure 8. Due to the very low logarithmic mean temperature the overall heat transfer coefficient for CHC10 measurement campaign is in average 18 % higher than the base scenario.

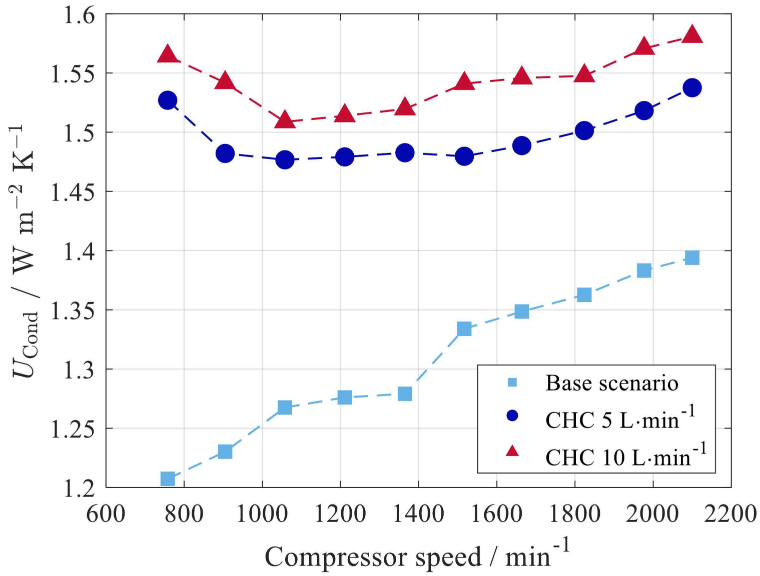

Figure 8: Overall heat transfer coefficient as function of the compressor speed.

The increase at compressor speed below 1000 rpm is due to the decreasing heat sink capacity caused by the insufficient superheating and a possible partial condensation on the way from the compressor to the condenser. On system level, the CHC shows a positive impact on the heat capacity as well as the COP. On the left-hand side of Figure 9 the heat capacity shows an increasing trend with increasing compressor speed, as well as the increase of the CHC water volume flow. At round about 2000 rpm the capacity lift shows its maximum with 6.9 % in CHC5 and 8.9 % in CHC10. This leads to an increase of the COP, presented on the right-hand side. The maximum difference to the base scenario in CHC5 of 6.4 % at 2100 rpm and of 8 % at 2000 rpm in CHC10.

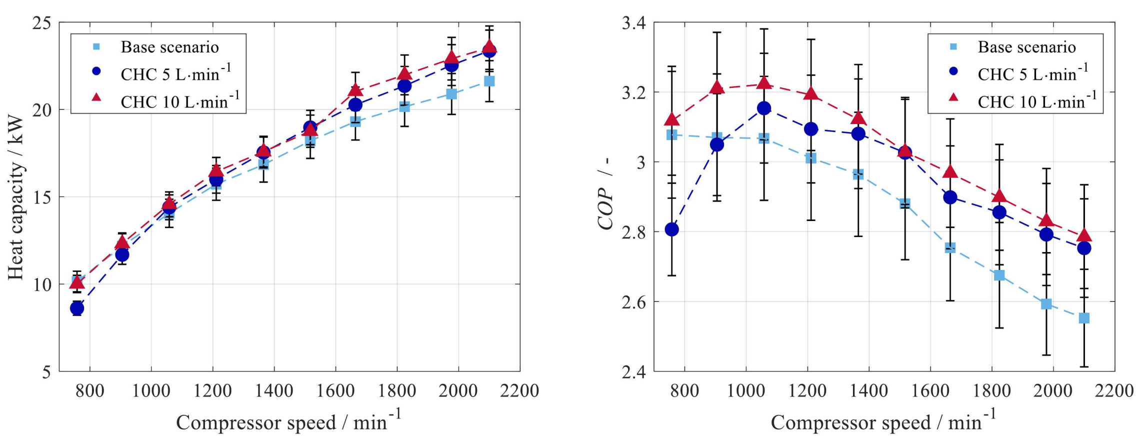

Figure 9: Heat capacity and COP as function of the compressor speed.

The decrease at low compressor speeds is related to previous discussed effects. The presented results show the possibility of decreasing the discharge temperature of a HTHP to extend the lifetime and reduce the load on working fluids with an increase of the system efficiency, at moderate supply temperatures.

In a last step the supply temperature is increased up to 130 °C, at source temperatures of 60 °C and 70 °C. The glide at the evaporator is kept constant with 10 K, while the glide at the condenser is increased to 40 K to simulate a more application-oriented scenario. Also, the superheating of the suction gas is kept constant at 25 K, to minimise the thermal stress on the compressor suction side, what leads to a not so high impact in system efficiency compared to higher suction gas superheating. Figure 10 shows the discharge temperature and the COP as function of the temperature lift.

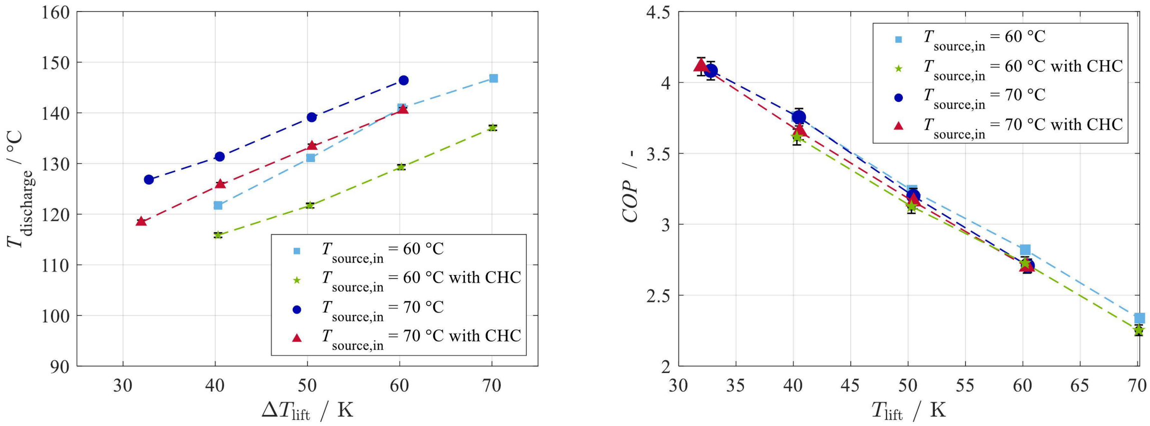

Figure 10: Discharge temperature at high supply temperature and COP as function of the temperature lift.

With a source temperature of 60 °C the discharge temperature can be reduced by up to 11.7 K. In case of a source temperature of 70 °C the difference is up to 8.4 K. The COP decreases with increasing temperature lift and the compressor waste heat utilization by using the CHC can only keep the COP approximately constant. Further studies will investigate whether increasing the suction gas superheating will increase the COP using the CHC, at high supply temperatures. The results show, that also for high temperatures the CHC can reduce the discharge temperature without crucial efficiency losses and can be one approach to use the full potential of a refrigerant.

4. CONCLUSIONS

- The part load behavior of a high temperature heat pump was investigated in a base scenario.
- The COP decreases with increasing compressor speed due to the increasing relative heat exchanger surface, while the heat capacity can be increased at higher compressor speeds.
- Based on these measurements the influence of water-cooled cylinder heads was investigated.
- The cylinder head cooling can increase the COP by up to 8 % and reduce the discharge temperature by up to 29 K.
- The pinch point in the condenser can be reduced from 2.7 K to 1 K in maximum, what leads to an increasing overall heat transfer coefficient, using the CHC.
- The CHC leads to a minimal increase of isentropic and volumetric efficiency.
- The CHC can lead to insufficient superheating of the discharge gas at low compressor speeds.
- The suction gas superheating is the most important factor to decide how much the CHC can influence the system performance.
- A discharge temperature of 137 °C was obtained for a water supply temperature of 130 °C, thus higher supply temperatures can be realized in future without reaching the critical temperature of the refrigerant.
- In further work an experimental multi-objective optimization will be carried out to identify the optimal interaction between IHX-surface (change in suction gas superheating) and CHC in every operation point.
- Furthermore, the results will be transferred to natural hydrocarbons and subsequently implemented in techno-economic models.

NOMENCLATURE

COP	coefficient of performance	(–)
CHC	cylinder head cooling	(–)
DHN	district heating network	(–)
FS	full scale uncertainty	(%)
GWP	global warming potential	(CO_2e)
HCFO	hydrochlorofluoroolefine	(–)
HTHP	high-temperature heat pump	(–)
IHX	internal heat exchanger	(–)
ODP	ozone depletion potential	(R-11e)
OR	operating range uncertainty	(%)
P	electrical power	(kW)
$\dot{Q}$	heat capacity	(kW)
T	temperature	(°C)
U	overall heat transfer coefficient	(W/(m²K))
ΔT_{lift}	temperature lift	(K)
Δx	sensor uncertainty	(%)
Δy	combined uncertainty	(%)
η	efficiency	(%)

Subscript

CHC	cylinder head cooling
MITA	minimum temperature approach
SH	superheating
cond	Condenser
discharge	compressor discharge
el	electrical
in	inlet
isen	isentropic
log	logarithmic
m	mean
out	outlet
sink	heat sink
source	heat source
vol	volumetric

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