



Research Paper

Experimental optimisation of the performance of a high-temperature heat pump under part load conditions[☆]

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ABSTRACT

Integrating high-temperature heat pumps (HTHPs) into industrial energy systems and district heating networks offers a promising pathway towards CO₂-neutral heat supply. To ensure reliable performance under fluctuating demand profiles, relevant research tasks include suitable compressor design, working fluid selection, and detailed understanding of variable operating behaviour. The present study investigates the off-design characteristics and part load optimisation of a laboratory-scale HTHP test rig equipped with an internal heat exchanger (IHX), reciprocating compressor with water-cooled cylinder heads (CHC) and a maximum thermal capacity of 37 kW. Off-design behaviour is analysed by varying the supply temperature from 75 °C to 140 °C at nominal compressor speed. The results show a linear decrease in the coefficient of performance (COP) with increasing temperature lift, ranging from a maximum COP of 4.83 at minimal lift to 1.93 at the highest supply temperature. Part load behaviour is investigated across compressor speeds from 758 rpm to 2100 rpm for multiple supply temperatures, revealing speed-dependent COP optima shifting towards higher speeds with increasing supply temperature. Optimisation measures include variation of superheat at evaporator outlet through an electronic expansion valve and the control of suction gas superheat by thermal recuperation using the IHX. At a supply temperature of 120 °C, these strategies result in COP improvements of up to 22% at minimum speed. The results demonstrate that a variable IHX heat-transfer area has the greatest influence on HTHP efficiency. Additionally, CHC further enhances performance at higher speeds by reducing discharge temperatures and enabling increased heat recuperation without exceeding thermal limits.

1. Introduction

Mitigating global warming through the decarbonisation of the energy sector represents one of the most pressing challenges of our time. Although the heating sector accounts for more than 50% of final energy consumption in Europe, the share of renewable energy sources in this sector remained at approximately 26.2% in 2023 [1]. The integration of large-scale or high-temperature heat pumps (HTHPs) into the energy system offers a promising pathway for providing thermal energy in a sustainable and resource-efficient manner. However, HTHPs should not be deployed solely for base load supply. Their use in modular and cascaded configurations, combined with coordinated integration into the electrical energy system, is particularly relevant. Furthermore, applications involving peak load coverage and simultaneous heating and cooling are of considerable interest.

Jeßberger et al. [2] show that district heating networks and geothermal heat sources exhibit significant hourly variations in heat demand and supply temperatures, requiring flexible part load operation of integrated heat pumps. Similarly, Foslie et al. [3] report that industrial facilities, such as dairies, have also highly dynamic hourly heating, cooling and electricity loads, which highlights the need for adaptive, load-dependent heat pump control. Furthermore, Persson and Bernström [4] demonstrate that seasonal and operational fluctuations can significantly affect heat demand profiles and the potential for energy savings. Consequently, the importance of designing heat pump systems that can maintain efficient performance even under unstable conditions is emphasised. This underscores the substantial need for efficient and flexible system operation in many different applications.

The corresponding HTHP market is experiencing significant growth. In this context, Arpagaus et al. [5] identified 26 commercially available

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HTHPs with maximum supply temperatures above 90 °C as early as 2018, with investment costs ranging from 100 €kW⁻¹ to 1000 €kW⁻¹. Overall, commercially mature technologies in 2018 achieved heating capacities between 20 kW and 20 MW.

Project 68 of the International Energy Agency (IEA) provides a detailed overview of the current state and development of industrial HTHPs with supply temperatures above 100 °C in its 2025 annual report [6]. The report is based on data from 55 manufacturers and 40 demonstration projects and clearly shows that the market is growing rapidly. While technologies operating in the 120 °C to 160 °C range are leaving the pilot stage and becoming commercially available, solutions for temperatures above 160 °C and large-scale systems exceeding 10 MW are also making substantial progress.

The market is segmented into TRLs (Technology Readiness Levels) according to the temperature range [6]:

- <120 °C (**Established**): Broad availability (TRL 9) based on industrial refrigeration technology employing reciprocating and screw compressors. Refrigerants predominantly include ammonia (R717), hydrochlorofluoroolefins (HCFOs), and, in some cases, hydrocarbons.
- 120 °C – 160 °C (**Commercialisation**): This range represents the transition from demonstration to standardised commercial products. Screw compressors and, increasingly, turbo compressors are used, operating with hydrofluorolefine (HFOs), hydrocarbons, or water/steam.
- >160 °C (**Demonstration & Pilot**): The focus here lies on steam compression and specialised cascade systems (e.g., hydrocarbon cycle coupled with steam compression). Turbo compressors, oil-free reciprocating compressors, and rotary compressors are applied.

For large-scale systems (> 10 MW), turbo compressors using refrigerants such as water or CO₂ dominate, as economies of scale significantly reduce costs. Investment costs vary widely between 300 €kW⁻¹ and 1500 €kW⁻¹, with standardised modules (0.5 MW to 2 MW) often reaching a cost optimum around 500 €kW⁻¹.

Due to the lack of detailed performance data, experimental studies are required to understand and investigate system performance beyond the design point. The number of publications on HTHPs has been increasing annually [7], as has the number of review articles. Bamigbetan et al. [8] identify current research challenges related to refrigerant selection, innovative refrigerant blends, and cascade systems. In particular, they highlight the need to investigate higher pressure ratios and compressor cooling. Arpagaus et al. [5] emphasise the importance of minimising heat losses and oil lubrication, as well as studying upscaling for industrial applications. Adamson et al. [9] identify the significant potential of transcritical systems and formulate six challenges together with corresponding solution approaches. Sun et al. [10] and Jiang et al. [11] complement this by addressing efficiency improvements at high temperature lifts and the development of large-scale HTHPs exceeding 1 MW of thermal capacity. Khalid et al. [12] also focus on improving compressors and optimising heat transfer by reducing temperature differences and enabling oil-free operation. Ma et al. [13] and Klute et al. [14] provide a comprehensive overview of steam-generating HTHPs, presenting different technologies such as direct and indirect steam generation, as well as various refrigerant cycle configurations, including single-stage, multi-stage, and cascade systems. El Samad et al. [15] give a detailed overview of compressor technologies for HTHPs and their associated challenges at high temperatures. They conclude that reciprocating and screw compressors are the most commonly used technologies for HTHPs, while turbo compressors may gain importance in smaller capacity ranges in the future due to their good temperature resistance and efficiency. Tijani et al. [16] provide an overview of refrigerant–oil compatibility for various cycle concepts and highlight the importance of oil selection for the reliable operation of HTHPs, considering both synthetic and natural refrigerants.

The previously mentioned review studies indicate that reciprocating compressors are frequently employed in HTHPs and that state-of-the-art heat pump cycle designs include internal heat exchangers (IHxs). The open research questions primarily concern efficiency improvements, refrigerant–oil interactions, optimisation of compressor technologies, and the scaling-up of HTHPs for industrial applications. The following section provides a detailed overview of the current state of research on HTHPs, with a particular focus on experimental work. Table 1 and Table 2 summarise the most relevant experimental HTHP studies of recent years, listing the study focus, thermal capacity of the heat pump, and publication year. The works differ in their research emphasis: while early studies such as that by Pan et al. [17] investigate the technical implementation and initial characterisation of HTHP, later studies focus on the use of different refrigerants. Arpagaus and Bertsch [18,19] compare various HFO/HCFO refrigerants in terms of their performance at high supply temperatures of up to 150 °C. Xu et al. [20] implement an innovative HTHP concept by providing 100 °C hot water from ambient air. Bamigbetan et al. [21] develop the first HTHP based on hydrocarbons (butane) and investigate compressor performance at high temperatures. Navarro-Esbriet al. [22] use experimental results from two single-stage HTHPs to simulate a cascade configuration. Jiang et al. [23] present an innovative heat pump design featuring a radial compressor and achieve a COP of 3.67 at supply temperatures of 100 °C. Ramirez et al. [24] introduce a novel HTHP design incorporating a subcooler and an IHX. Studies by Ma et al. [25] and Wu et al. [26] focus on cascade and two-stage HTHPs to achieve high supply temperatures while maintaining acceptable pressure ratios. Overall, the experimental studies demonstrate the continuous evolution of HTHP technology, with a particular emphasis on improving efficiency, expanding the application range, and applying new refrigerants.

The presented studies emphasise the broad field of experimental investigations; however, only limited datasets for part load conditions are available and analysed [33,34,37]. These include the performance maps of Hassan et al. [34] who evaluated an HTHP under varying compressor speeds and heat source temperatures between 40 °C and 100 °C, but with a fixed heat sink temperature above 130 °C, and without an IHX, thereby excluding any influence of recuperation or controllable suction gas superheat. Jeßberger et al. [33,37] present different measurement campaigns focusing on off-design performance at varying heat sink temperatures for a fixed compressor speed, as well as part load behaviour across the entire compressor speed range at a fixed heat sink temperature of 120 °C. Additionally, the influence of suction gas superheating was investigated by adjusting the recuperated heat in the IHX to optimise part load behaviour. However, these previous part load behaviour studies focused only on a single supply temperature level without assessing how part load characteristics change across multiple supply temperature levels. Due to fluctuating heat demands and heat source conditions, the part load behaviour of HTHPs is a critical aspect to consider when designing and operating a heat pump for various applications. The present study aims to address this research gap by characterising the part load and off-design performance of a lab-scale HTHP at several supply temperature levels. Subsequently, the part load performance at a source temperature of 60 °C and a supply temperature of 120 °C is optimised by controlling the recuperated heat in the internal heat exchanger, utilising the compressor's waste heat and varying the superheating after the evaporator. The results of this study provide valuable insights for the integration of HTHPs into industrial energy systems and district heating networks.

2. Methodology

The present study extends the work of Jeßberger et al. [37], which introduced initial investigations into the optimisation of the part load behaviour of an HTHP. The current work builds upon this foundation, expanding the scope by incorporating the characterisation of part load performance across multiple supply temperature levels, a detailed

Table 1
Overview of experimental studies on high-temperature heat pumps (part 1).

Author	Focus	$\dot{Q}_{\text{sink}}$	Year
Pan et al. [17]	• Comparison of different natural and synthetic refrigerants • Evaporation temperature from 30 °C to 55 °C • Condensation temperature from 60 °C to 100 °C	5 kW	2011
Zhijiang et al. [27]	• Characterisation of an HTHP using R1234ze(E) • Supply temperatures up to 85 °C • Discharge temperature limited to 110 °C	15 kW	2015
Lee et al. [28]	• Solar-assisted HTHP with R1233zd(E) • Comparison with R134a • Variation of refrigerant charge	3 kW	2018
Arpagaus and Bertsch [18]	• Comparison of HFO/HCFO refrigerants • Temperature lift from 30 K to 70 K • Supply temperature up to 150 °C	11 kW	2019
Arpagaus and Bertsch [19]	• Comparison of R1233zd(E) and R1336mzz(Z) • Temperature lift from 30 K to 70 K • Supply temperature up to 150 °C	11 kW	2019
Shah et al. [29]	• Comparison of R1233zd(E) and R245fa • R245fa simulated • Supply temperature up to 135 °C	13.8 kW	2019
Mateu-Royo et al. [30]	• Characterisation of an HTHP with R245fa using a scroll compressor • Supply temperatures up to 140 °C • Heat source temperature from 60 °C to 80 °C	17 kW	2019
Bamigbetan et al. [21]	• Experimental analysis of a hydrocarbon compressor for high temperatures • Suction gas temperatures up to 80 °C • Discharge gas temperatures up to 140 °C	20 kW	2019
Arpagaus and Bertsch [31]	• Comparison of R1233zd(E) and R1224yd(Z) • Temperature lift from 30 K to 70 K • Supply temperature up to 150 °C	11 kW	2020
Arpagaus and Bertsch [32]	• Comparison of HFO/HCFO refrigerants • Temperature lift from 30 K to 70 K • Supply temperature up to 150 °C	11 kW	2021

evaluation of various thermodynamic cycle parameters, and systematic optimisation strategies. Consequently, the fundamental experimental setup and control concepts established in the preceding study are adopted from [37] without modification. A comprehensive dataset of all experimental measurements is publicly available, see Jeßberger et al. [38], ensuring full comprehensibility and transparency of the presented analyses. The following sections describe the system setup, control and safety concept, as well as the experimental procedures applied in this study.

2.1. Experimental setup

Previous publications have provided comprehensive descriptions of the experimental setup (see [2,37,39]). However, for completeness, the concept of the HTHP test rig is briefly reiterated here, following the description provided in [37]. As shown in Fig. 1, the flowchart illustrates the key components and layout of the HTHP test rig. The working fluid utilised in this study is trans-1-Chloro-3,3,3-trifluoropropene (R1233zd(E)), an HCFO with a global warming potential (GWP) of 1 CO₂e and an ozone depletion potential (ODP) of 0.00034 R-11e [40]. Due to its low GWP and non-flammable nature, this refrigerant represents a safe and efficient choice for laboratory-based experimental investigations. The heat pump circuit, featuring insulated pipes and heat exchangers, is described below with reference to the flow chart.

Referring to Fig. 1, the compressor is located between state points 1 and 2. From there, the compressed and superheated working fluid proceeds to the oil separator manufactured by ESK Schultze (model: BOS2-R-35/28F), where liquid oil droplets are filtered out of the high-pressure gas to prevent impairment of heat transfer. Earlier substance data measurements show that the oil separator separates over 95% of the oil out of the refrigerant and is therefore negligible in the

heat exchanger calculations. The oil is returned via an electronic oil level regulator located at the compressor sump when the oil level drops. From the oil separator, the high-pressure gas at state point 3 enters the condenser to release thermal energy during condensation and subsequent subcooling of approximately 3 K to the heat sink. The condenser is a plate heat exchanger manufactured by SWEP (model: SWEP B15T). All heat exchangers in the system operate in a counterflow configuration to ensure the highest possible heat transfer efficiency. The subcooled working fluid then enters a liquid receiver, which compensates for load changes and serves as a reserve, before transferring thermal energy to the low-pressure side via the internal heat exchanger (IHX). Following the IHX, the working fluid undergoes an isenthalpic expansion in the expansion valve prior to entering the evaporator. The evaporator is also a plate heat exchanger from SWEP (model: SWEP V80 x 40HT). It is equipped with a distribution device that ensures the working fluid, which is in wet vapour condition after expansion, is evenly distributed over the plates. This is important to guarantee uniform heat transfer and to prevent the occurrence of dry vapour in individual channels. The distribution device requires a pressure difference of approximately 1 bar to ensure optimal heat transfer; consequently, the expansion valve does not expand the fluid to the evaporator pressure directly, but part of the expansion takes place within the distribution device itself. From the evaporator, the superheated working fluid proceeds to the IHX, which can be bypassed. The IHX is a plate heat exchanger manufactured by Alfa Laval (model: CB30-34H) and serves to recuperate thermal energy not utilised in the condenser back to the low-pressure side. The bypass enables electronic control of the amount of recuperated heat and thus the regulation of suction gas superheat. From the IHX, the working fluid returns to the compressor, and the cycle repeats. The test facility is also equipped with four sampling ports that allow for the extraction of both refrigerant

Table 2
Overview of experimental studies on high-temperature heat pumps (part 2).

Author	Focus	$\dot{Q}_{\text{sink}}$	Year
Xu et al. [20]	- Air-to-water HTHP with R134a - Supply temperatures up to 100 °C - Heat source at 25 °C	2.8 kW	2021
Jeßberger et al. [33]	- Part load behaviour of an HTHP - Temperature lift from 25 K to 45 K - Supply temperature up to 104 °C	35 kW	2022
Hassan et al. [34]	- Part load behaviour of an HTHP - Development of characteristic maps for thermal storage - Condensation temperature of 138 °C	22 kW	2022
Chen and Kyung Kwon [35]	- Comparison of R1233zd(E) and R245fa - Condensation temperature from 66 °C to 74 °C - Evaporation temperature from 32 °C to 38 °C	9 kW	2022
Navarro-Esbríet al. [22]	- Two single-stage HTHPs using R1234ze(E) and R1336mzz(Z) - Adaptation for cascade operation - Supply temperatures up to 150 °C	14 kW	2022
Jiang et al. [23]	- Innovative heat pump design with radial compressor - Temperature lift from 30 K to 50 K - Supply temperature up to 100 °C	381 kW	2023
Navarro-Esbríet al. [36]	- HTHP with R1336mzz(Z) using a scroll compressor - Focus on temperature and compressor performance - Supply temperatures up to 160 °C	12 kW	2023
Ramirez et al. [24]	- Innovative HTHP design with subcooler and IHX - Provision of sensible and latent heat - Supply temperatures up to 148 °C	66 kW	2024
Jeßberger et al. [37]	- Characterisation and optimisation of part load behaviour of an HTHP - Variation of rotational speed - Compressor waste heat utilisation	35 kW	2025
Ma et al. [25]	- Characterisation of the operating behaviour of a cascaded HTHP - Investigation of the effect of refrigerant charge - Supply temperatures up to 142 °C	20 kW	2025
Wu et al. [26]	- Two-stage HTHP with R1233zd(E) - Maximum temperature lift of 100 K - Supply temperatures up to 145 °C	100 kW	2025

Table 3
Components of the test rig.

Component	Description
Compressor	Four-cylinder reciprocating compressor, Bitzer 4JE-22Y Swept volume: 0.73 l
Frequency inverter	25 Hz–70 Hz, Bitzer FOY+ 39-4
Oil separator	ESK-Schultze BOS2-R-35/28F
Condenser	SWEP B15T, plate heat exchanger, 1.29 m ²
Liquid receiver	Bitzer F202H
Expansion valve	Danfoss, ICMTS
Internal heat exchanger	Alfa Laval CB30-34H, plate heat exchanger, 1.2 m ²
Evaporator	SWEP V80 × 40HT, plate heat exchanger, 2.28 m ²
Refrigerant charge	28 kg
Oil	POE oil Bitzer BSE 85K, viscosity of 80 mm ² /s at 40 °C [41]
Refrigerant	Honeywell Solstice® R1233zd(E)
Heating capacity	≈ 35 kW

and oil samples. These sampling ports are equipped with commercially available ball valves and Swagelok valves, enabling easy and rapid sampling without impacting the system. The sampling locations, depicted in Fig. 1, include the high-pressure gas line, the oil return line, the section between the oil separator and the condenser, and after the liquid receiver between state points 4 and 5. These ports allow for fluid analysis to investigate the influence of high-temperature operation without the need to drain the system. For a better overview, Table 3 shows the components of the test rig.

2.2. Secondary circuits

The secondary circuits are responsible for heat transfer between the working fluid and the heat source and heat sink, both operated with water. The heat source is simulated by a temperature control unit from HB-Therm (model: HB-Therm Thermo-5). A three-way valve regulates the volumetric flow of the heat transfer medium and enables control of the return temperature $T_{2,02}$. The supply temperature is automatically controlled by the temperature control unit to the setpoint. The heat sink requires a more complex control strategy to maintain constant supply and return temperatures. For this purpose, a secondary circuit between the building cooling network (light blue) and the HTHP is used, shown in red in Fig. 1. This circuit is driven by a frequency-controlled circulation pump from Grundfos (model: Magna3). The heat sink circuit is connected to the building cooling network via a plate heat exchanger from SWEP (model: B15Hx60). The hot water exits the condenser at state point 3.01 and is cooled by the plate heat exchanger to the set return temperature $T_{3,02}$. An electronic heater is connected downstream of the plate heat exchanger to compensate for control-related fluctuations. To protect the circulation pump from high temperatures, it is located on the cold side downstream of the electronic heater. In addition to frequency control, the volumetric flow is regulated by a bypass around the pump to enable particularly low flow rates and thus large temperature differences.

2.3. Control concept

The test facility is controlled via a LabVIEW system by National Instruments, enabling easy and fast adjustment of operating parameters

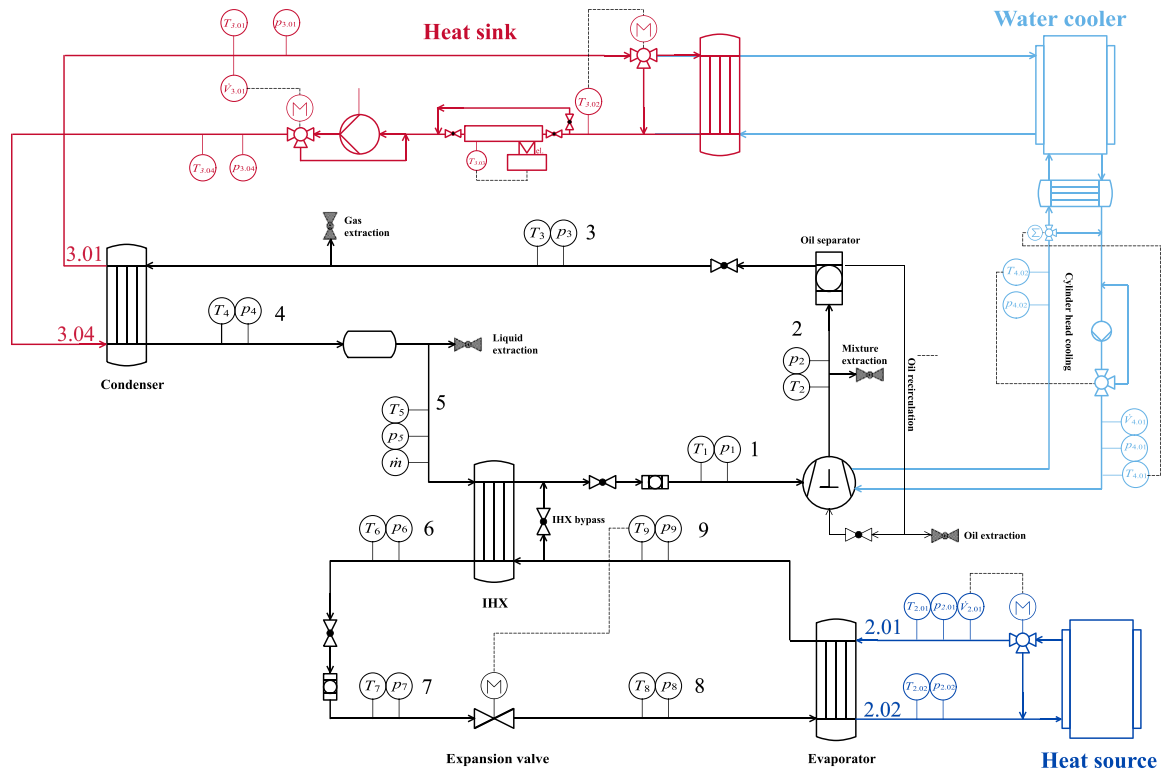


Fig. 1. System layout of the HTHP test rig.

and automatic data acquisition. The control is based on a PID controller concept that regulates variable process parameters. The implemented control loops are indicated with dashed lines in Fig. 1. In the refrigeration cycle, the opening degree of the expansion valve controls the superheat after the evaporator (state point 9), and the bypass position before the IHX regulates the suction gas superheat (state point 1). While the primary function of the expansion valve is superheat control, it also indirectly governs the refrigerant mass flow rate and the degree of evaporator filling dynamics. However, over the 8 K superheat heat range considered, the mass flow rate remained within a 2% difference. On the heat source side, a bypass controls the return temperature $T_{2,02}$, while the supply temperature is set directly at the temperature control unit. On the heat sink side, the supply temperature $T_{3,01}$ is regulated by the position of the three-way valve, while the return temperature $T_{3,02}$ is controlled by the bypass position before the cooling water heat exchanger and the heating element. In the cylinder head cooling (CHC) circuit, a manually operated three-way valve regulates the flow rate, and a PID controller controls the flow through the cooling water heat exchanger to regulate the injected temperature $T_{4,01}$.

2.4. Safety concept

The test facility is equipped with various safety devices to ensure safe operation. These include high-pressure and low-pressure switches, expansion tanks in the water circuits, and an emergency stop switch that shuts down the entire system in an emergency. Additionally, all system components are designed to withstand the high pressures and temperatures. The compressor includes internal safety shutoffs; however, only the oil differential pressure shutoff is activated because the process temperatures exceed the manufacturer's approved range.

2.5. Experimental procedure

The measurement uncertainties, as well as the error analysis using Gaussian error propagation, are reported in Jeßberger et al. [39].

Independently of these sensor uncertainties, steady-state operating conditions were defined a priori to ensure efficient and reproducible measurement campaigns. For experimental data acquisition, the heat sink inlet and outlet temperatures ($T_{3,04}$ and $T_{3,01}$), as well as the discharge temperature (T_2), were required to remain within a band of ± 1.5 K over a period of 10 min. Detailed uncertainty budgets for each sensor are published in Jeßberger et al. [39] and are therefore not reiterated in this work. The uncertainties of all reported quantities are determined using Gaussian error propagation. The variable contributions to the uncertainty are sufficiently small to be fully covered by the symbol size in all figures, while the constant components do not influence the interpretation of the results and are thus not considered further.

Table 4 summarises the conducted measurement campaigns and illustrates the ranges of the experimentally varied parameters, including compressor speed n , suction gas superheat, and temperatures. The base scenario (BS) conditions are defined as follows:

- Heat source: inlet temperature 60 °C, temperature difference 10 K
- Heat sink: outlet temperature 120 °C, temperature difference 60 K

The first measurement campaign investigates the off-design behaviour of the HTHP by varying the heat sink outlet temperature from 75 °C to 140 °C at a compressor speed of 1517 rpm corresponding to a net frequency of 50 Hz. The evaporator superheat is set to 5 K for all operating points. The IHX bypass valve is adjusted to achieve a suction gas superheat of 20 K at the BS and is subsequently kept constant across all operating points. This approach simulates a fixed IHX heat transfer surface without active superheat control, as would be the case in a real application. Consequently, the suction gas superheat may deviate slightly from 20 K under different operating conditions. In addition to the variation of temperature lift, the influence of temperature differences between the inlet and outlet flow of the heat sink and heat source will be investigated. The heat sink outlet temperature is therefore fixed at 100 °C, while the return temperature is adjusted between 30 °C and 90 °C. Similarly, to investigate the influence of the temperature

Table 4
Ranges of experimentally varied parameters.

Campaign	$T_{\text{sink,in}}$ (°C)	$T_{\text{sink,out}}$ (°C)	$T_{\text{source,in}}$ (°C)	$T_{\text{source,out}}$ (°C)	n (rpm)	$SH_{1,01}$ (K)	$SH_{1,09}$ (K)
BS	60	120	60	50	1517	20	5
Off-design	60	75–140	60	50	1517	20 at BS	5
Part load	60	90–120	60	50	750–2100	20 at BS	5
Influence $SH_{1,01}$	60	120	60	50	750–2100	variation	5
Influence $SH_{1,09}$	60	120	60	50	750–2100	20 at BS	3–11
Variation ΔT_{cond}	30–90	100	50	40	1517	33	5
Variation ΔT_{evap}	50	100	50	35–43	1517	33	5
Optimisation part load	60	120	60	50	750–2100	optimisation	optimisation

difference at the heat source, the heat source inlet temperature is fixed at 50 °C and the return temperature is varied between 35 °C and 43 °C. The second measurement campaign focuses on the part load behaviour of the HTHP by varying the compressor speed between 750 rpm and 2100 rpm for different supply temperatures. The superheat settings are consistent with those applied in the off-design investigations, with a constant evaporator superheat of 5 K for all measurement points and a fixed IHX bypass valve position corresponding to a suction gas superheat of 20 K at the BS conditions.

Based on these investigations, the part load behaviour of the HTHP is optimised by adjusting the superheat at the evaporator outlet and the recuperated thermal energy in the IHX. The optimisation is conducted for each compressor speed, varying the frequency in increments of 5 Hz from 25 Hz to 70 Hz, to identify the optimal combination of evaporator superheat and IHX split flow ratio. For safety reasons, the suction gas temperature is hereby limited to a temperature of 90 °C and the discharge gas temperature to 145 °C. Additionally, the influence of the CHC on the optimisation of part load behaviour is also considered. The optimised scenario is finally defined as the operating point with the highest COP at each compressor speed.

2.6. Evaluation of measurement results

After identifying the steady-state operating points, a detailed thermodynamic analysis is conducted at the system and component level for the heat exchangers and the compressor. The compressor is evaluated with respect to its isentropic and volumetric efficiencies. The isentropic efficiency η_{isen} is defined as:

$$\eta_{\text{isen}} = \frac{\dot{m}_{\text{WF}} \cdot (h_{2s} - h_1)}{P_{\text{el}}} \quad (1)$$

where h_1 denotes the specific enthalpy at state point 1 and h_{2s} the isentropic enthalpy on the discharge side of the compressor. P_{el} is the electrical input power, and $\dot{m}_{\text{WF}}$ is the refrigerant mass flow rate.

The volumetric efficiency of the compressor represents the ratio between the actual refrigerant volume flow rate and the theoretically possible volume flow rate. The theoretical volume flow rate can be calculated from manufacturer specifications as:

$$\dot{V}_{\text{ideal}} = z \cdot \pi \cdot r^2 \cdot s \cdot n \quad (2)$$

where z is the number of cylinders, r the cylinder inner radius, s the piston stroke, and n the rotational speed. Accordingly, the volumetric efficiency is given by:

$$\eta_{\text{vol}} = \frac{\dot{V}_{\text{WF}}}{\dot{V}_{\text{ideal}}} \quad (3)$$

To focus on the system performance, the heat exchangers are analysed with respect to the heat losses. Here, the heat losses of the condenser are in focus relative to the thermal energy supplied to the heat sink circuit. On system level the COP leads to:

$$COP = \frac{\dot{Q}_{\text{sink}}}{P_{\text{el}}} \quad (4)$$

where $\dot{Q}_{\text{sink}}$ is the thermal capacity of the heat pump and P_{el} the electrical power consumption. Taking the CHC into account the COP_{CHC} leads to:

$$COP_{\text{CHC}} = \frac{\dot{Q}_{\text{sink}} + \dot{Q}_{\text{CHC}}}{P_{\text{el}}} \quad (5)$$

where $\dot{Q}_{\text{CHC}}$ is the utilised waste heat of the compressor. The temperature lift of the HTHP is defined as:

$$\Delta T_{\text{Lift}} = T_{\text{sink,out}} - T_{\text{source,in}} \quad (6)$$

3. Results

This section presents the results of the conducted measurements described in Chapter 2.5 on the off-design and part load behaviour of the HTHP. First, the operating behaviour of the HTHP in off-design operation is discussed. Subsequently, the measurement results for part load operation at variable compressor speed are presented and analysed. Finally, the performance in part load operation is optimised in order to improve the efficiency of the HTHP under flexible operating conditions.

3.1. Characterisation of operating behaviour in off-design

The off-design operation is of great importance for the flexibility and adaptability of the HTHP to varying operating conditions. Therefore, the influence of the temperature lift and the effect of variable temperature differences in the heat source and sink circuits are described and analysed below.

The influence of the temperature lift is investigated at a compressor speed of 1517 rpm with a constant evaporator superheat of 5 K and a suction gas superheat of 20 K at BS conditions, with the IHX bypass valve position kept constant across all operating points. The heat source circuit inlet temperature is maintained at 60 °C, with a temperature difference of 10 K. The supply temperature in the heat sink circuit is varied from 75 °C to 140 °C, corresponding to a temperature lift of 15 K to 80 K. Figs. 2–5 present the influence of the temperature lift on the COP , the required electrical compressor power (P_{el}), pressure ratio (Π) and heat sink capacity ($\dot{Q}_{\text{sink}}$). The COP of the HTHP shows a linearly decreasing behaviour with increasing supply temperature and a corresponding increase in temperature lift. A maximum COP of 4.83 is achieved at a minimum temperature lift of 15 K, while the COP drops to 1.93 at a maximum temperature lift of 80 K. This behaviour can be explained primarily by the significant increase of the pressure ratio as the temperature lift rises (see Fig. 4). At a temperature lift of 15 K, the pressure ratio is 3.5, while at 80 K it rises to 7. The increased pressure ratio leads to significantly higher compressor work, which consequently increases the required electrical compressor power (see Fig. 3) and thus negatively affects the COP of the heat pump. Simultaneously, the heat sink capacity decreases with increasing temperature lift, as the thermal capacity of the condenser is reduced at higher supply temperatures because of the increasing pressure ratio. At a temperature lift of 15 K, the heat sink capacity is 32.4 kW, while at 80 K it drops to 18 kW. While the required temperature lift is often predetermined when integrating HTHPs into processes or district heating networks, the variation of

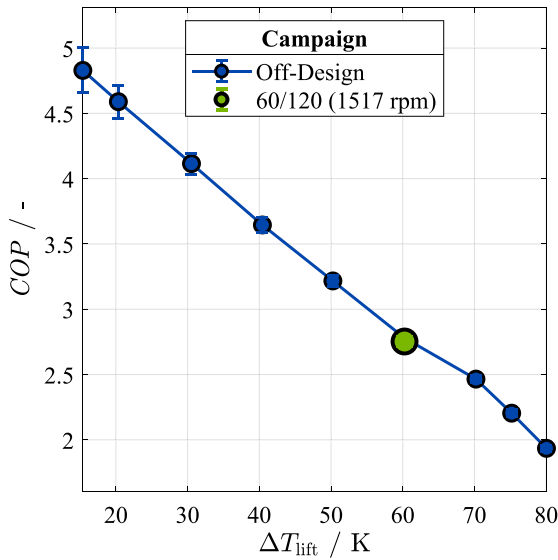


Fig. 2. Off-design operation: COP as a function of temperature lift.

temperature differences in the heat source and heat sink circuits can be used to optimise the operating behaviour of the HTHP. In the following, the effects of varying the temperature difference at the evaporator and condenser are therefore investigated, with a constant heat source inlet temperature of 50 °C for both measurement campaigns. First, the temperature difference at the evaporator is varied from 7 K to 15 K. The supply temperature of the heat sink circuit is fixed at 100 °C to prevent the HTHP cycle from overheating and to ensure safe operation when varying temperature differences. Figs. 6–9 show the results of the off-design analysis with variation of the temperature difference at the evaporator. The *COP* of the HTHP shows a linear decrease with increasing temperature difference at the evaporator. At a temperature difference of 7 K, a maximum *COP* of 3.46 is reached, while at a temperature difference of 15 K, the *COP* drops to 2.95. This behaviour can be explained by the decrease in evaporation pressure from 2.02 bar to 1.51 bar as the temperature difference increases. The reduction in evaporation pressure results in an increased pressure ratio in the compressor, which rises from 5 to 6.3 (see Fig. 8). As described above, the increased pressure ratio leads to a significant increase in electrical power consumption, which negatively impacts the *COP* of the heat pump. The heat sink capacity also shows a decreasing trend with rising temperature difference, which is caused by the increased pressure ratio and the associated reduction in thermal capacity at the condenser at low evaporation pressures. At a temperature difference of 7 K, the heat sink capacity amounts to 22.65 kW, while at 15 K it drops to 16.04 kW.

The results demonstrate the significant impact of the heat source conditions on the efficiency and performance of the HTHP. It is therefore recommended that the temperature difference at the evaporator should be as low as possible in order to optimise the operation of the heat pump.

The influence of the temperature difference at the condenser is also of great importance for system integration. To investigate its impact, the heat sink supply temperature is maintained at 100 °C, while the return temperature is varied between 90 °C and 30 °C. Consequently, the heat sink temperature difference is varied between 10 K and 70 K. The inlet temperature of the heat source is set to 50 °C, with a temperature difference of 10 K. Figs. 10–13 illustrate the influence of varying the temperature difference at the condenser on the operating behaviour of the HTHP. Fig. 10 shows the *COP* as a function of the temperature difference at the condenser. With increasing temperature difference, the *COP* of the HTHP increases. At a temperature difference of 10 K, a minimal *COP* of 2.88 is reached, while at a temperature difference

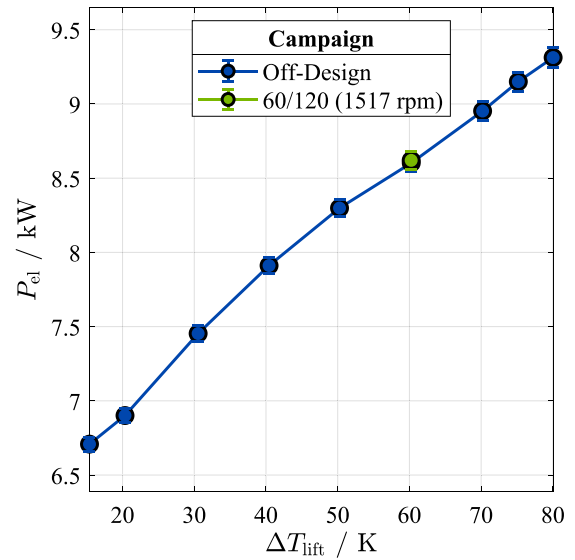


Fig. 3. Off-design operation: Electrical power as a function of temperature lift.

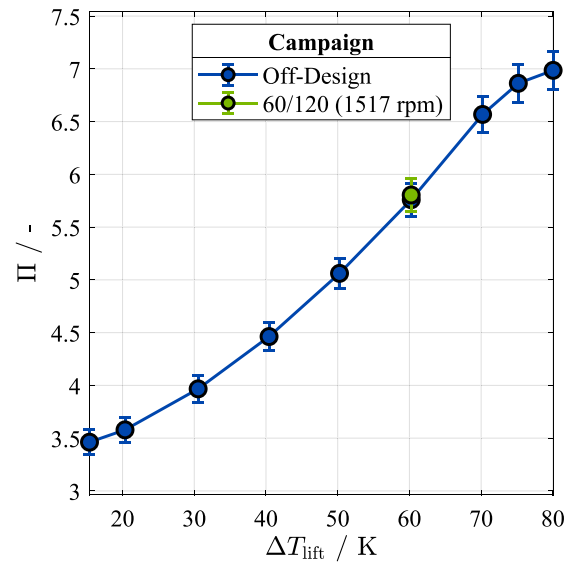


Fig. 4. Off-design operation: Pressure ratio as a function of temperature lift.

of 70 K, the *COP* rises to 3.39. This behaviour can be explained by the strong decrease in condensing pressure (see Fig. 12) which accompanies increasing temperature difference. The condensing pressure decreases from 11.1 bar to 6 bar, resulting in a reduced pressure ratio in the compressor. This, in turn, leads to a reduction in the required compressor work, which has a positive impact on the *COP* of the heat pump. Additionally, the heat sink capacity increases with higher temperature differences at the condenser, as the thermal capacity of the condenser is increased at lower condensing pressures. In this section, the off-design behaviour of the HTHP was analysed in detail, whereby the influence of the temperature lift as well as the temperature differences at the evaporator and condenser were investigated. The results highlight the importance of these parameters for the efficiency and performance of the HTHP.

3.2. Characterisation of the part load performance

As described in Section 2.5, the part load behaviour of the HTHP is examined by varying the compressor speed under constant boundary

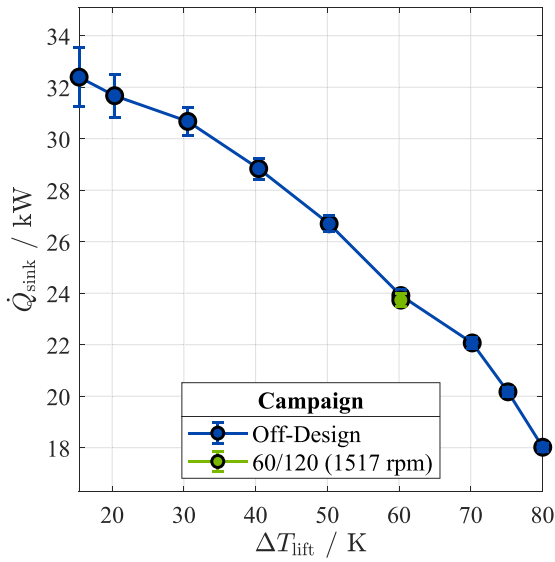


Fig. 5. Off-design operation: Heat sink capacity as a function of temperature lift.

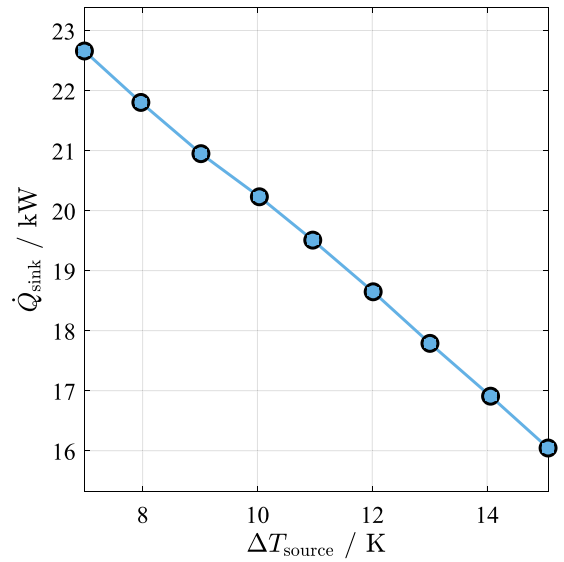


Fig. 7. Off-design operation: Heat sink capacity as a function of temperature difference at the evaporator.

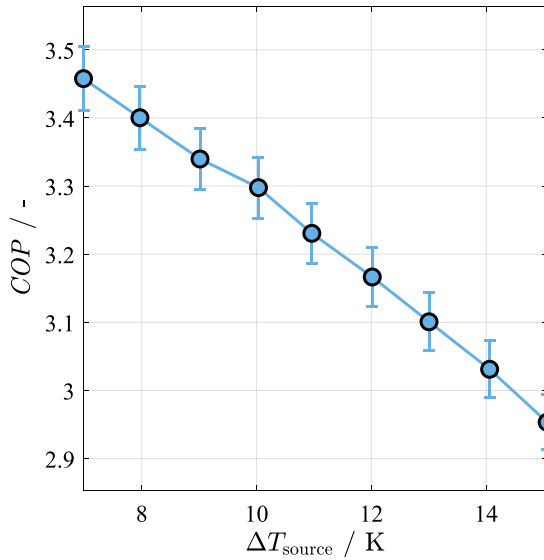


Fig. 6. Off-design operation: COP as a function of temperature difference at the evaporator.

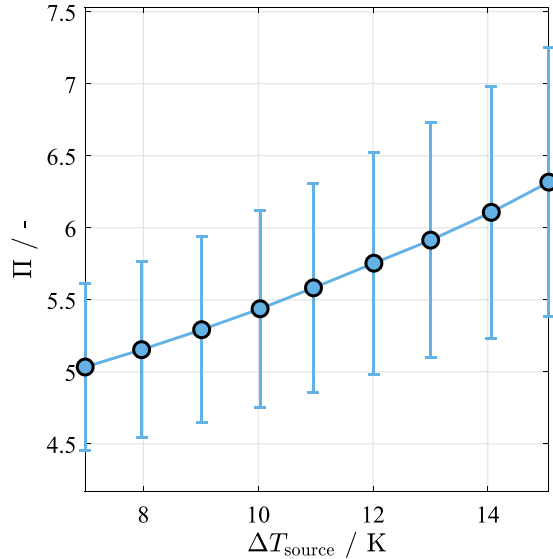


Fig. 8. Off-design operation: Pressure ratio as a function of temperature difference at the evaporator.

conditions. The supply temperature in the heat sink circuit is set to four different levels: 90 °C, 100 °C, 110 °C, and 120 °C, with a constant return temperature of 60 °C. The inlet temperature of the heat source is set to 60 °C for all measurement series, with a temperature difference of 10 K. The compressor speed is varied for each supply temperature, ranging from 750 rpm to 2100 rpm, corresponding to a frequency range of 25 Hz to 70 Hz.

Fig. 14 shows the $\log(p)$ - h diagram of the refrigerant R123zd(E) at a nominal speed of 1517 rpm for the different supply temperatures. The overhanging wet-vapour region underscores the need for sufficient superheat at the compressor inlet to prevent droplet erosion. For optical clarity, the state point between expansion and evaporation is recalculated to account for the use of a distribution device. Consequently, pressure losses within the evaporator are not shown (see Chapter 2). Figs. 15 and 16 show the COP as well as the electrical compressor power P_{el} as a function of the compressor speed n . The compressor power increases linearly with increasing compressor speed

and increasing supply temperature. In contrast, the COP does not show consistent behaviour for the different supply temperatures. At a low supply temperature of 90 °C, the heat pump reaches a maximum COP of 4.67 at 750 rpm, which decreases with increasing speed. As the supply temperature increases, the COP deterioration at lower speeds intensifies, while at higher speeds it is less pronounced, resulting in a speed- and temperature-dependent optimum. This optimum shifts towards higher speeds as the supply temperature increases. At 100 °C, this is at 905 rpm with a COP of 3.92; at 110 °C, at 1058 rpm with a COP of 3.33; and at 120 °C, at 1211 rpm with a COP of 2.78.

To explain the COP curve, the following diagrams show various process variables as a function of compressor speed. In Figs. 17 and 18, the thermal capacities of the heat source $\dot{Q}_{\text{source}}$ and the heat sink $\dot{Q}_{\text{sink}}$ are presented as a function of the compressor speed n . Both capacities increase with increasing speed, with the heat sink capacity increasing significantly more than the heat source capacity. At a supply temperature of 90 °C and a compressor speed of 2100 rpm, the heat

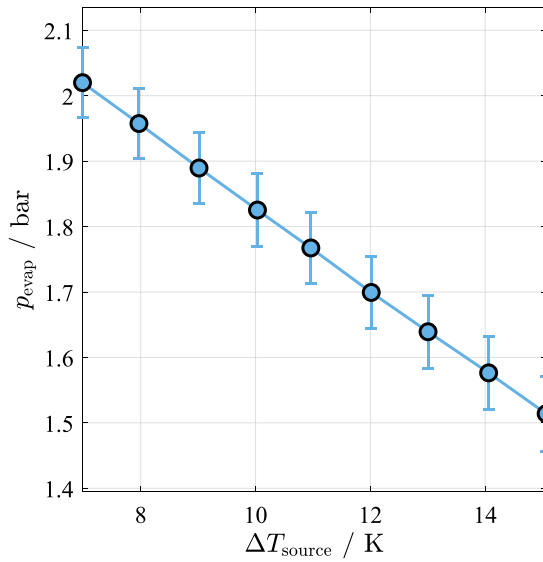


Fig. 9. Off-design operation: Evaporating pressure as a function of temperature difference at the evaporator.

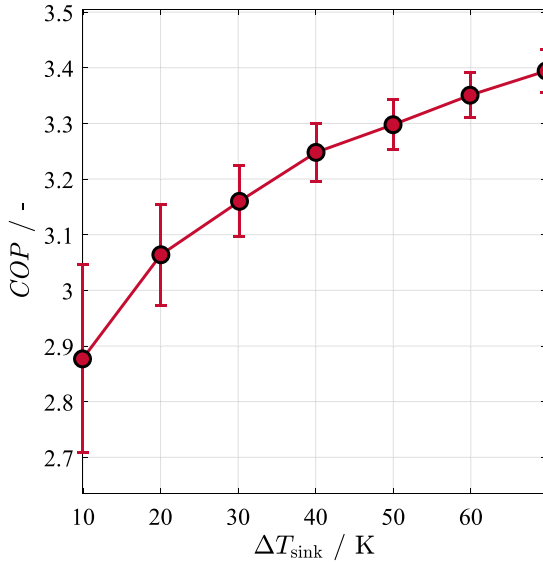


Fig. 10. Off-design operation: COP with variation of the temperature difference at the condenser.

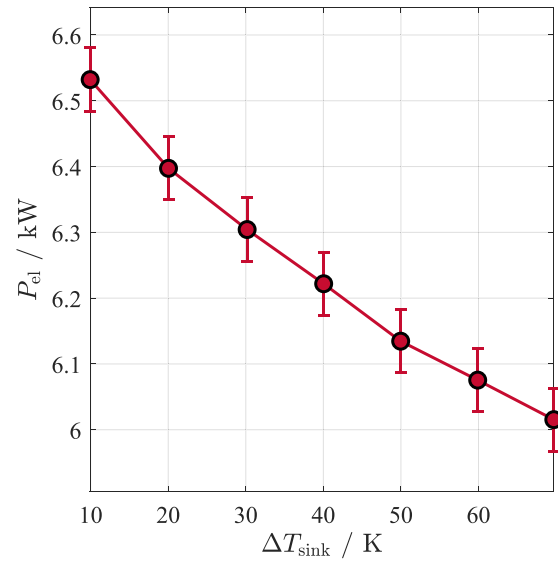


Fig. 11. Off-design operation: Electrical power with variation of the temperature difference at the condenser.

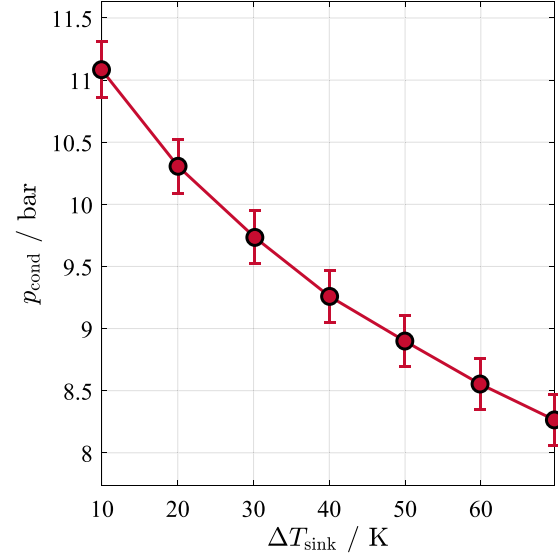


Fig. 12. Off-design operation: Condensing pressure as a function of temperature difference at the condenser.

sink capacity reaches its maximum of 37.0 kW, while the heat source capacity is at 27.7 kW. At the minimum speed of 750 rpm, the heat sink capacity is 15.65 kW and the heat source capacity is 13.4 kW. This corresponds to a reduction in thermal capacity of approximately 42.3% for the heat sink capacity and 48.3% for the heat source capacity compared to operation at maximum speed. At a supply temperature of 120 °C, a reduction in heat sink capacity to 35% at minimum speed can be achieved. Figs. 19 and 20 show the isentropic and volumetric efficiencies of the compressor as a function of compressor speed. The isentropic efficiency shows a similar trend for all supply temperatures. A maximum is reached at a speed between 1000 rpm and 1250 rpm, while it subsequently drops by up to 8% absolute with increasing speed. The volumetric efficiency shows a similar trend, although the decline after the optimum is less drastic. The decreasing efficiencies at low speeds below 1000 rpm can be explained by increased relative heat losses, generated by the longer residence times in the compressor. This behaviour can be explained by the dominant loss mechanisms of a

reciprocating compressor. At low speeds, the compression duration is relatively long, which results in increased heat losses to the cylinder wall. The compression thus proceeds closer to an isothermal change of state, which leads to increased irreversible entropy production and thus reduced isentropic efficiency. Concurrently, the longer residence time in the clearance volume favours re-expansion processes and backflows, thereby reducing the volumetric efficiency. With increasing speed, these time-dependent losses decrease, and both efficiencies initially increase significantly. Above the optimum speed, the trend reverses. Higher piston speeds lead to greater flow and pressure losses at the valves and in the inlet and outlet channels, which typically increase with the square of the flow velocity. Additionally, there is an increase in mechanical friction losses in bearings and piston rings. These effects require greater real compression work and cause the working gas to heat up above the isentropic level. This results in a significant drop in isentropic efficiency as the speed increases. The volumetric efficiency reacts less strongly to these effects, as it is primarily characterised by

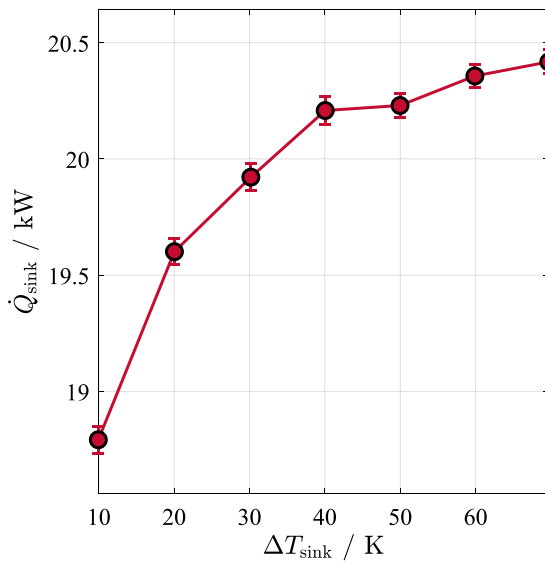


Fig. 13. Off-design operation: Heat sink capacity as a function of temperature difference at the condenser.

the filling, clearance volume, and valve losses. Although valve inertia and backflows also increase, they do not increase to the same extent as the thermal and mechanical losses that dominate the isentropic efficiency. Overall, this results in a shallower decline in volumetric efficiency at high speeds.

The significant reduction in volumetric efficiency at elevated supply temperatures results from the corresponding increase in condensation temperature and pressure (see Fig. 23), which leads to an increase in the pressure ratio of the compressor at an almost constant evaporation pressure. Consequently, the volumetric losses that are characteristic of reciprocating compressors increase significantly. A higher discharge pressure leads to stronger re-expansion of the residual gas in the clearance volume. At higher condensation pressures, the residual gas has a higher density and temperature, resulting in greater expansion during the return stroke and a larger displacement of the theoretically available intake volume. This reduces the effective cylinder filling and, accordingly, the volumetric efficiency. Furthermore, valve losses also increase with increasing pressure ratio. Consequently, the inlet valves must open against increased residual pressure, thus increasing the effective pressure drop across the valve plate and throttling the suction process. At the same time, increased backflows occur at the exhaust valves, as the pressure differences become larger during the switching processes. Both effects reduce the mass of the refrigerant that is actually sucked in per working cycle and thus also worsen the volumetric efficiency. Figs. 21–24 show both suction gas and discharge gas pressures as well as condensing and evaporating pressures as a function of compressor speed at the different supply temperatures. While the suction gas pressure (p_{suc}) and the evaporating pressure (p_{evap}) behave identically and decrease linearly with increasing compressor speed at all supply temperatures, the discharge pressure (p_{dis}) and the condensing pressure (p_{cond}) show a temperature-dependent behaviour.

The linear decrease of suction gas and evaporating pressure with increasing compressor speed can be explained by two coupled effects. Firstly, the compressor delivers a higher mass flow at higher speeds, leading to increased flow losses in the suction line and the evaporator. These pressure losses typically grow quadratically with the flow velocity and thus approximately linearly with the speed. Secondly, the thermal capacity provided by the heat source and the heat transfer surface area of the evaporator are limited. The increased refrigerant mass flow increases the evaporator heat demand, thus necessitating an increase in the temperature difference to transfer the required heat.

Consequently, the evaporation temperature and thus the evaporation pressure are reduced. Both effects combined lead to the observed linear drop in suction gas and evaporating pressure with increasing compressor speed. The discharge pressure, measured at state point 2 (see Fig. 1), shows a clearly different behaviour depending on the supply temperature. For the lower supply temperatures of 90 °C and 100 °C, the increase is approximately linear with increasing speed. This is due to compression resulting in higher outlet pressures at higher mass flows. However, at higher supply temperatures of 110 °C and 120 °C, this behaviour changes fundamentally. In 110 °C operation, the discharge gas pressure increases up to a speed of about 1200 rpm and subsequently remains constant, while in 120 °C operation, it remains almost constant over the entire speed range. This behaviour indicates that at higher condensing temperatures, heat dissipation in the condenser becomes limiting and the pressure cannot rise further because the heat sink does not dissipate the increased heat output effectively enough.

The condensation pressure behaves inversely to the discharge pressure. At 90 °C p_{cond} remains almost constant while p_{dis} increases with speed, whereas at 120 °C, p_{cond} decreases while p_{dis} remains constant. This can be explained by the pressure losses in the oil separator, which is located between the compressor outlet (state point 2) and the condenser inlet. The pressure difference between discharge gas and condensing pressure ($\Delta p = p_{\text{dis}} - p_{\text{cond}}$) corresponds to the pressure loss in the oil separator, which increases with the square of the flow velocity and thus approximately linearly with the speed. In 90 °C operation, the increase in discharge gas pressure exactly compensates for the increasing pressure losses in the oil separator, so that the condensing pressure remains almost constant. Conversely, in the 120 °C operation, the discharge gas pressure remains constant across the entire speed range, while the pressure losses in the oil separator exhibit an increase with increasing speed. This results in a linear decrease in the condensing pressure. The high pressure losses can be explained by the strong solubility of the refrigerant in the oil.

Figs. 25 and 26 show the relative heat losses in the condenser as well as the pressure loss in the oil separator as a function of compressor speed. The heat losses in the evaporator are significantly lower, so they can be neglected in the analysis. The highest heat losses in the condenser are observed at low compressor speeds for all supply temperatures. In 90 °C operation, the losses at 750 rpm amount to about 2% of the heat sink capacity and decrease with increasing speed to about 0% at 1058 rpm. At higher supply temperatures, the losses are significantly higher. In 120 °C operation, the losses at 750 rpm amount to about 8% of the heat sink capacity and decrease with increasing speed to about 2% at 2100 rpm. This behaviour can be explained by the significantly larger temperature difference between refrigerant and ambient temperature at higher supply temperatures. Furthermore, the longer residence times in the condenser at low speeds result in increased heat losses through the condenser shell. In combination with the presented curves of the compressor efficiencies, the behaviour of the COP in part load operation can be explained. At low speed, the heat losses in the condenser, as well as the reduced compressor efficiencies, are dominant and lead to a poor COP. With increasing speed, the losses decrease and the efficiencies improve, which leads to an increased COP. However, above the optimal speed, the decreasing compressor efficiencies become the dominant factor, resulting in a decline in the COP.

3.3. Optimisation of part load performance

This section presents various measures for optimising the part load operating behaviour of the HTHP at a supply temperature of 120 °C. First, the influence of varying the evaporator superheat is analysed, followed by an investigation of the influence of suction gas superheat on system performance. Finally, the optimal setting for maximising the COP is determined for each operating point by considering combinations of IHX usage, evaporator superheat, and CHC.

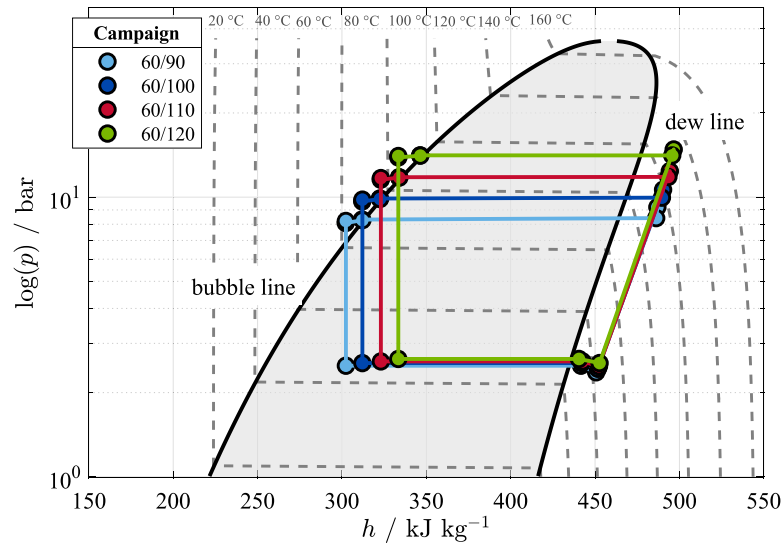
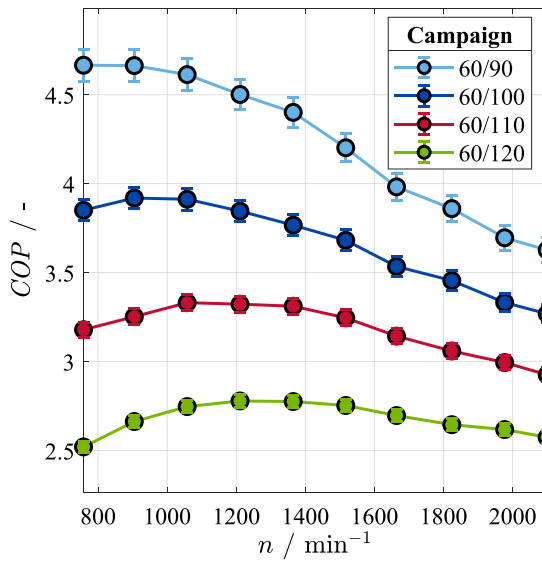
Fig. 14. $\log(p)$ - h diagram at nominal speed (1517 rpm)

Fig. 15. COP as a function of compressor speed.

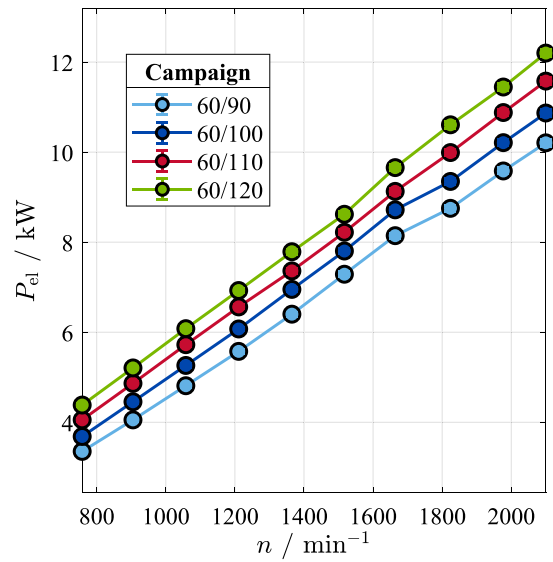


Fig. 16. Electrical power consumption as a function of compressor speed.

Variation of Evaporator Superheat

The superheat after the evaporator is controlled by the opening degree of the expansion valve. If this opens further, the mass flow through the evaporator increases, which leads to a reduction in superheat. Figs. 27 and 28 show the influence of superheat after the evaporator on the COP and the heat source capacity at different compressor speeds. Three speeds were selected for the measurements. 905 rpm represents the lower part load range (frequency = 30 Hz), 1517 rpm the medium part load range (frequency = 50 Hz), and 2100 rpm the upper part load range (frequency = 70 Hz). At the nominal speed of 50 Hz, the COP increases linearly with increasing superheat up to about 8 K, whereafter it continues to rise slightly to a maximum of 2.89 at 10 K. In the lower part load range, a certain fluctuation of the measured values can be seen, which can be attributed to suboptimal control in this operating range. However, this observed increase in the COP with increasing superheat, is influenced by competing effects including reduced mass flow rate and altered evaporator conditions and therefore should not be generalised beyond the conditions studied. In the upper part load range, a comparatively strong increase in the COP at low superheat

can be seen, remaining at an almost constant level from about 5 K. This phenomenon can be explained by the actual flow pattern after the evaporator. At low nominal superheats below 5 K, no completely superheated vapour is present at the evaporator outlet. Instead, a high-quality wet vapour is observed. The remaining liquid fraction is entrained by the high flow velocities in the three-way valve and in the pipes.

The apparent superheat indicated by the temperature sensor, despite the two-phase flow, results from the inhomogeneous flow pattern. At high vapour qualities above 95%, an annular flow typically forms, in which a superheated vapour core is surrounded by a thin liquid film on the pipe wall. The temperature sensor, which is typically positioned within the vapour core, measures the local temperature of this vapour portion, which may already exceed the saturation temperature. Simultaneously, the turbulent flow continues to carry liquid droplets and wall films. This results in a thermal non-equilibrium, with the vapour portion being locally superheated while liquid is still present in the system. At minimum superheat, the measured sensible heat transfer on the gaseous side is approximately 34% lower than that on the

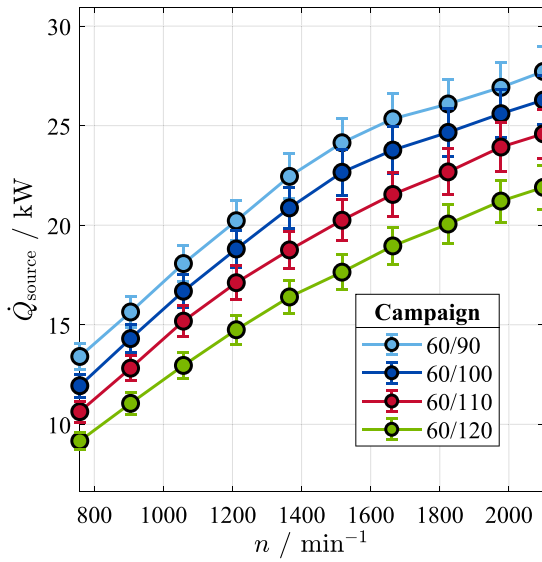


Fig. 17. Heat source capacity as a function of compressor speed.

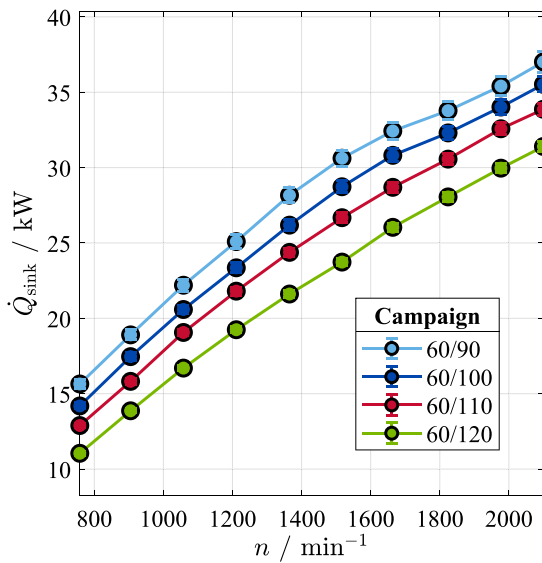


Fig. 18. Heat sink capacity as a function of compressor speed.

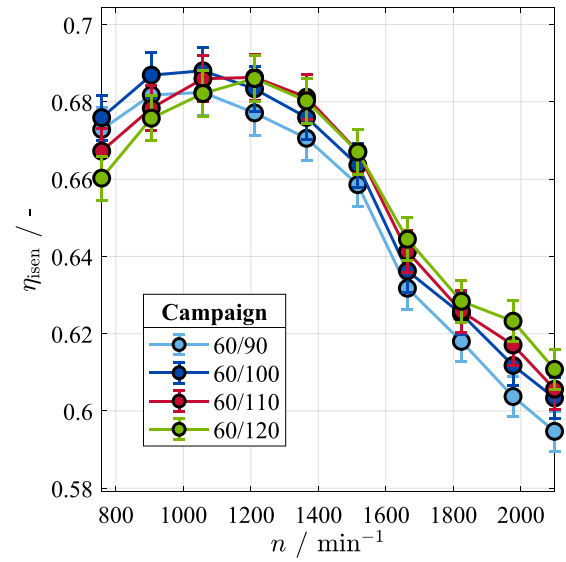


Fig. 19. Isentropic efficiency as a function of compressor speed.

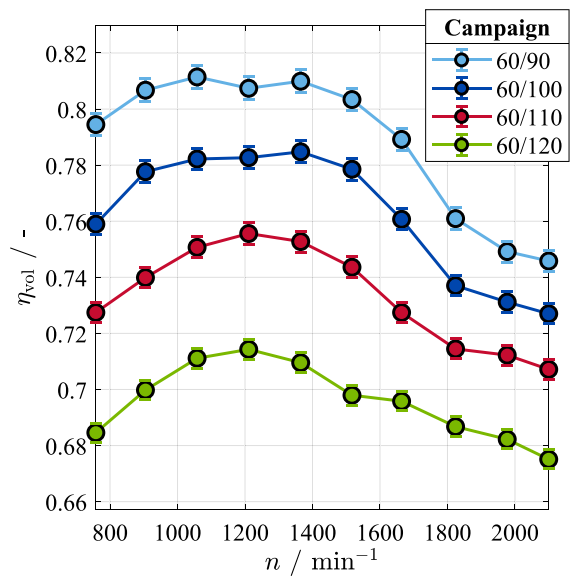


Fig. 20. Volumetric efficiency as a function of compressor speed.

liquid side of the IHX. With increasing superheat, this discrepancy decreases and approaches a plateau, reaching a deviation of about 14% from a superheat of 5 K onward. The measured superheat of 2 K to 5 K, therefore, does not reflect the complete evaporation of the entire mass flow, but merely reflects the thermodynamic state of the vapour core portion. A measured superheat of approximately 5 K to 8 K is required to ensure that no liquid remains in the near-wall regions and throughout the flow, and that a homogeneous single-phase vapour flow is present. At high compressor speeds and thus high mass flows, this effect is intensified, as the turbulent flow favours the entrainment of liquid droplets. This two-phase flow at the compressor inlet leads to several negative effects. Firstly, the volumetric efficiency of the compressor is reduced, as the incompressible liquid phase occupies volume but does not contribute to the delivery capacity. Secondly, there is a risk of liquid slugging, which can cause mechanical damage to the compressor. Additionally, heat absorption in the evaporator is inhibited, as the incompletely evaporated liquid blocks part of the evaporator surface and thus worsens heat transfer.

The lower COP observed at higher compressor speeds is consistent with the part-load results discussed earlier. As compressor speed increases, a reduction in isentropic and volumetric efficiencies can be expected, accompanied by increased pressure losses in the system. In addition, higher discharge temperatures at elevated speeds may impose operational constraints or necessitate protective control strategies, further limiting performance. These combined effects contribute to the observed decrease in COP at high compressor speeds. With increasing nominal superheat at the evaporator outlet, evaporation becomes progressively complete. From around 5 K, the evaporation process is practically complete and superheated vapour is present. This explains the notable increase in heat source capacity from 2 K to 5 K by about 7%, as the entire evaporator surface is now being utilised effectively and liquid blockage no longer occurs. Above 5 K, the heat source capacity remains almost constant, as further superheating only causes a slight increase in heat absorption. Consequently, preventing two-phase flow at the IHX inlet through adequate superheat settings results in the observed significant COP improvement.

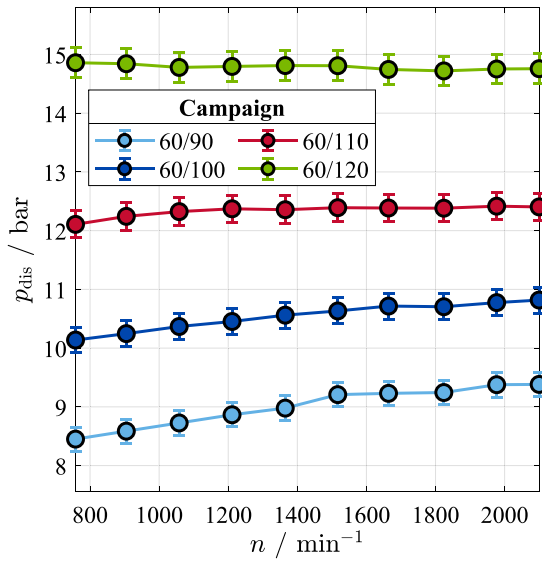


Fig. 21. Discharge gas pressure as a function of compressor speed.

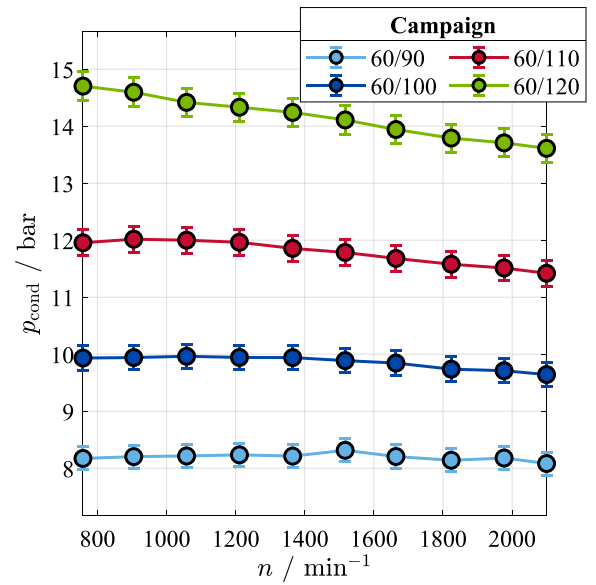


Fig. 23. Condensing pressure as a function of compressor speed.

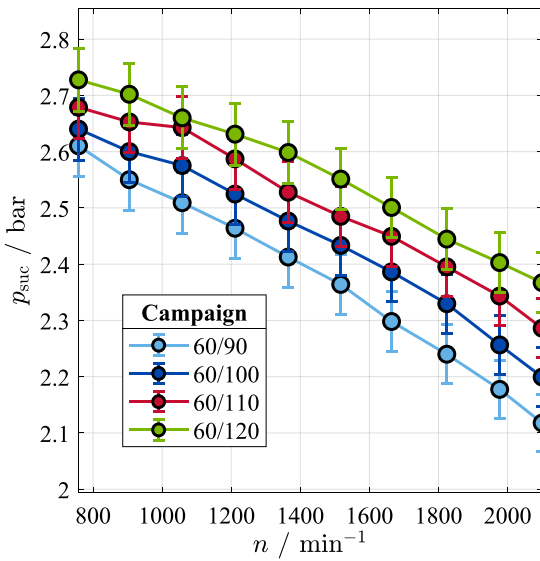


Fig. 22. Suction gas pressure as a function of compressor speed.

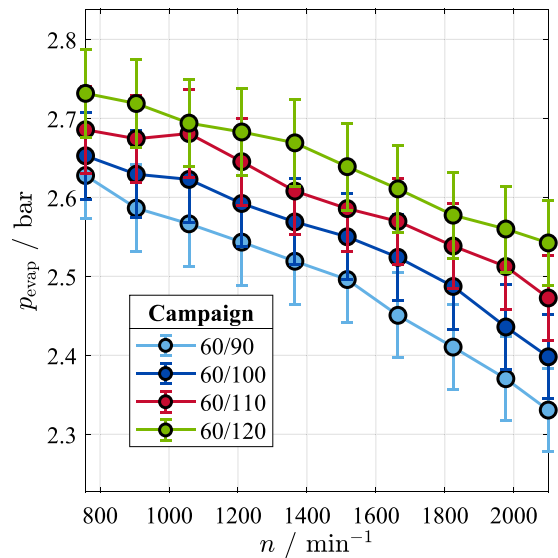


Fig. 24. Evaporating pressure as a function of compressor speed.

The energy balance of the IHX is consistent with the incomplete evaporation described above in the case of superheating below 5 K. The heat released by the liquid refrigerant on the high-pressure side is considerably greater than the sensible heat absorbed on the low-pressure side from the supposedly superheated vapour. This discrepancy can be explained by the evaporation of residual liquid droplets in the IHX, whereby the evaporation enthalpy absorbed significantly exceeds the sensible heat absorption. The IHX thus functions as an essential safety component, ensuring complete evaporation and protecting the compressor from liquid slugging, even if the main evaporator does not achieve complete evaporation and superheating.

From the presented results, it can be deduced that a minimum superheat of about 8 K should be set at the evaporator outlet to ensure safe and efficient operation of the HTHP in all operating states. However, the influence is not so significant that a new value needs to be optimised for every operating point.

Variation of Suction Gas Superheat

By adjusting the bypass position to vary the heat exchanger surface area of the IHX, the superheating of the suction gas at the compressor

inlet can be controlled, as described in Section 2.5. Figs. 29 and 30 show the influence of suction gas superheating on the COP and the heat sink capacity at a compressor speed of 1517 rpm. The green dot indicates the base case scenario with a suction gas superheat of 20 K. In order to avoid overheating of the compressor motor and not to exceed the permissible compressor outlet temperature, the operating limits are 90 °C for the suction gas temperature and 145 °C for the discharge gas temperature. With increasing suction gas superheat, the COP increases approximately linearly. The same behaviour can also be observed for the heat sink capacity, whereby there is no further increase in thermal capacity above 30 K. The linear increase of the COP with increasing suction gas superheat can be explained by several interrelated thermodynamic effects. The opening of the bypass results in a greater proportion of the refrigerant mass flow passing through the IHX, thereby increasing internal heat recovery. The liquid refrigerant on the high-pressure side releases heat after the condenser and becomes subcooled, while the vapour-phase refrigerant on the low-pressure side continues to superheat after the evaporator. The subcooling of the

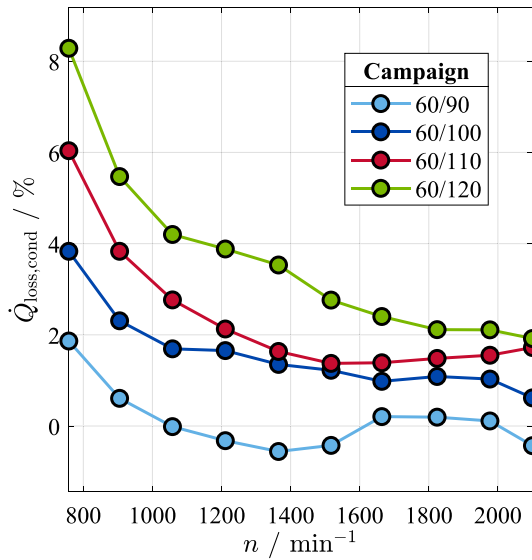


Fig. 25. Condenser losses as a function of compressor speed.

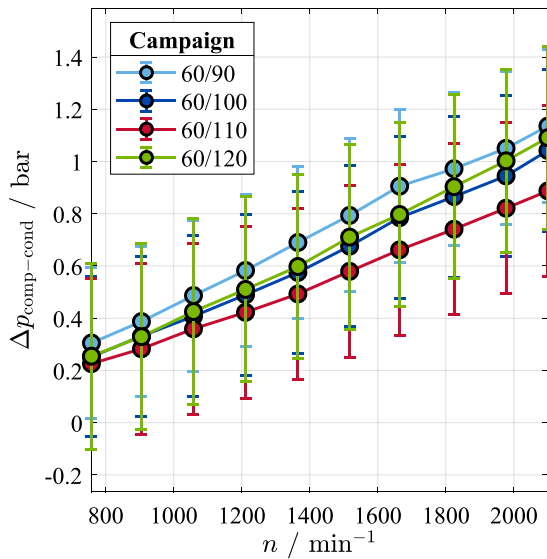


Fig. 26. Pressure losses as a function of compressor speed.

condensate on the high-pressure side positively impacts the entire cycle. A low refrigerant temperature upstream of the expansion valve significantly reduces flash gas formation during the expansion process. Consequently, less refrigerant has already evaporated when it enters the evaporator, leaving more evaporator surface area available for the actual heat absorption from the heat source. As the temperature difference at the heat source remains constant, more heat can be absorbed due to the improved surface utilisation in the evaporator, which leads to an increase in heat source capacity and thus heat sink capacity. Figs. 31 and 32 illustrate this correlation. As the suction gas superheat increases, the heat source capacity increases while the condensation pressure decreases linearly. This seemingly counterintuitive pressure reduction can be explained by the interaction of enhanced condensate subcooling, increased evaporator capacity, and elevated discharge gas temperature, which leads to optimised heat transfer in the condenser, thus enabling condensation at a lower pressure level. The reduction in condensation pressure has a direct positive effect on compressor efficiencies, as demonstrated in Figs. 33 and 34. With increasing

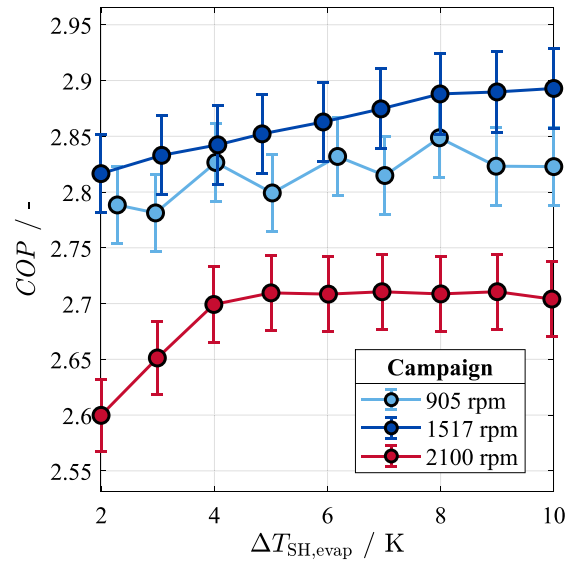


Fig. 27. Influence of superheat after evaporator on COP as a function of compressor speed.

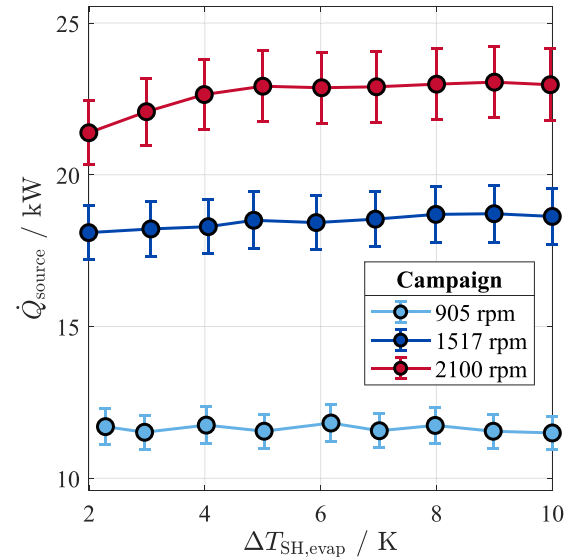


Fig. 28. Influence of superheat after evaporator on heat source capacity as a function of compressor speed.

suction gas superheat, both isentropic and volumetric efficiencies increase approximately linearly and the electrical power consumption of compressor decreases (see Fig. 35). This is primarily attributed to the reduced pressure ratio in the compressor, resulting from the lower condensation pressure at higher suction gas superheat. The results highlight the significant influence of suction gas superheating on the efficiency and performance of the HTHP. In order to optimise the operation of the heat pump, a maximum suction gas superheating should be achieved. The combination of increased suction gas superheat and reliable evaporator superheat thus represents an effective measure for improving the part load behaviour of the HTHP. The potential of this approach is illustrated in the following section by combining both measures.

Optimisation of the Part Load Behaviour of an HTHP

Figs. 36 and 37 illustrate the results of the optimisation measures in part load operation across the entire speed range. The colour of the measurement points indicates the suction gas superheat setting,

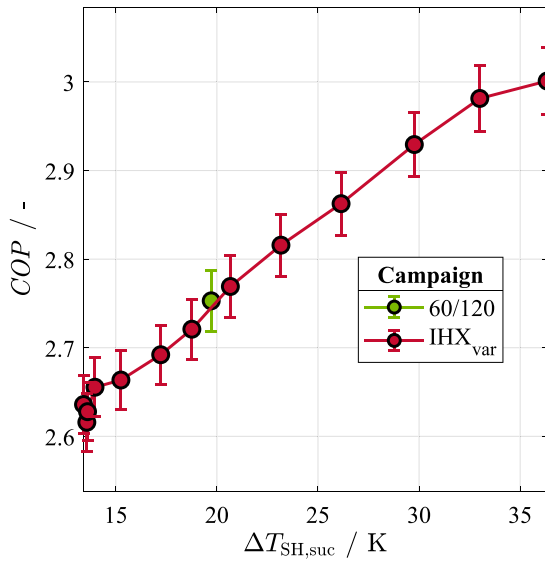


Fig. 29. COP as a function of suction gas superheat.

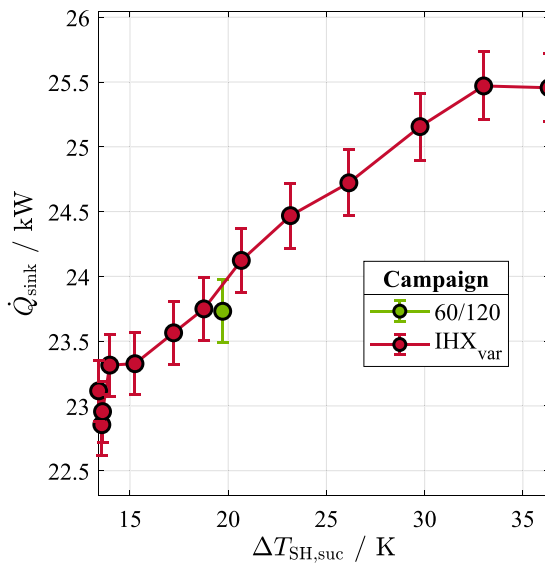


Fig. 30. Heat sink capacity as a function of suction gas superheat.

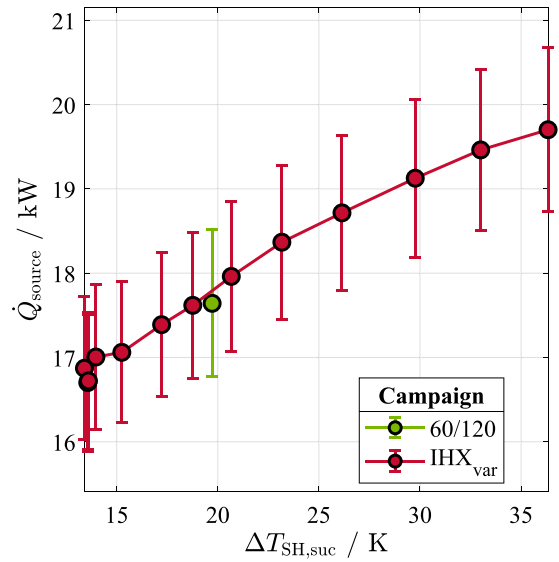


Fig. 31. Heat source capacity as a function of suction gas superheat.

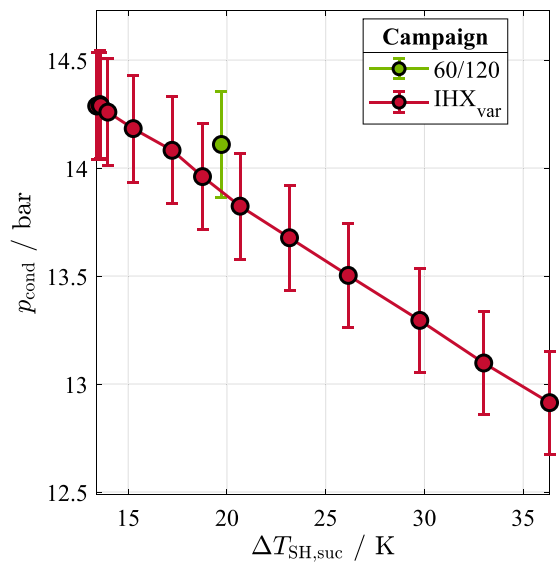


Fig. 32. Condensing pressure as a function of suction gas superheat.

while the marker shape distinguishes the measurement series. The base scenario is illustrated as a circle with a constant suction gas superheat of 20 K, whereas squares represent the optimised scenario combining evaporator superheat with variable suction gas superheat. Based on the previous analyses, the evaporator superheat is set to 8 K for all measurements in the optimised scenario, ensuring complete evaporation and efficient IHX utilisation. As illustrated in Fig. 36, the suction gas superheat decreases as the compressor speed increases. This can be explained by the decreasing isentropic efficiency with increasing speed, which leads to increased discharge gas temperatures. Consequently, the amount of heat recuperated in the IHX must be reduced in order to comply with the operating limits of the compressor. Consequently, a reduction in suction gas superheat is required at higher speeds. Nevertheless, it can be seen that an adjustment to the IHX usage and evaporator superheat results in a significant improvement in the part load behaviour of the HTHP. The optimal operating point at 750 rpm shows an increase in the COP of 22% compared to the base scenario and a maximum COP of 3.21 at a speed of 1058 rpm is reached. This corresponds to a COP_{Carnot} of 49% and a COP_{Lorenz} of 31%.

Fig. 37 presents the optimised measurement series with additional cylinder head cooling (CHC) at higher compressor speeds, represented by diamond symbols. Jeßberger et al. [42] demonstrated that the use of CHC has only a minimal direct impact on compressor efficiencies. Furthermore, they show that the isolated use of the CHC without suction gas superheat optimisation does not positively influence system efficiency of an HTHP. The discharge gas temperature can be lowered by up to 12 K, but the reduction in heat sink capacity caused by this cannot be compensated for by waste heat utilisation in the compressor. However, in combination with increased suction gas superheat, CHC contributes positively to efficiency improvement by enabling higher suction gas superheat at elevated compressor speeds without exceeding operational limits.

As can be seen in Fig. 37, the additional use of the CHC in combination with the optimised settings for the evaporator and suction gas superheating allows the part load behaviour in the higher speed range to be further optimised. Thus, at maximum speed of 2100 rpm, a COP of 2.8 can be achieved. This corresponds to an improvement of 9% compared to the base scenario and an improvement of 2.6%

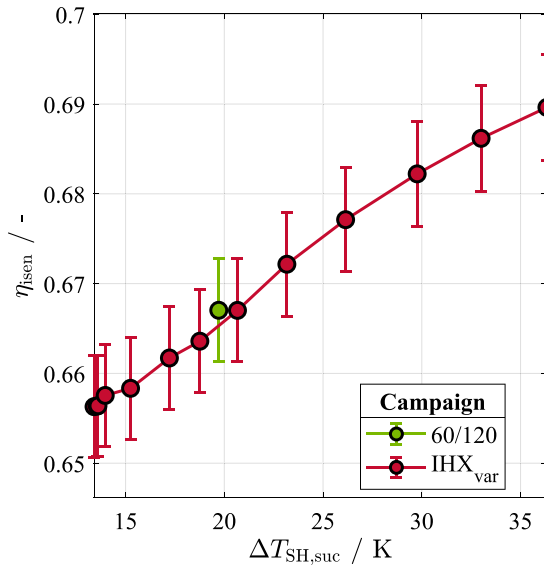


Fig. 33. Isentropic efficiency as a function of suction gas superheat.

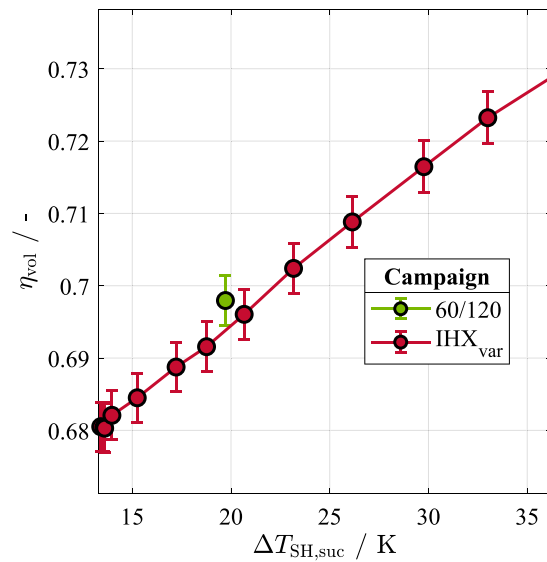


Fig. 34. Volumetric efficiency as a function of suction gas superheat.

compared to IHX-optimisation. Through the CHC, suction gas superheat can be further increased without exceeding the operating limits of the compressor. At lower compressor speeds, the suction gas temperature limit is the limiting factor for further optimisation, so the usage of the CHC would not be beneficial. Overall, it has been demonstrated that the combination of increased suction gas superheat, reliable evaporator superheat, and the use of the CHC represents an effective strategy for optimising the part load behaviour of the HTHP. The efficiency increases achieved highlight the potential of these measures for the practical use of HTHPs in variable operating conditions. Fig. 38 shows the $\log(p)$ - h diagram of the optimised part load operation with IHX and CHC at a compressor speed of 2100 rpm. The increased suction gas superheat at the compressor inlet and the reduced condensing temperature are clearly visible through the use of the CHC.

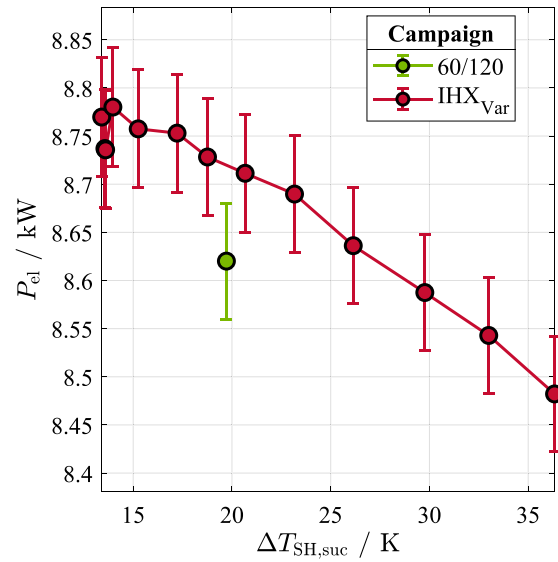
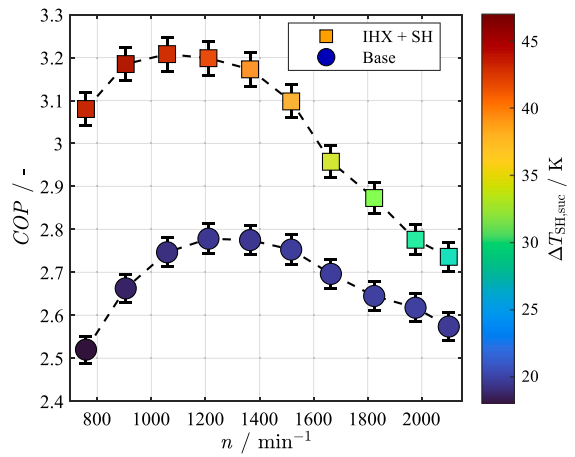
Fig. 35. P_{el} as a function of suction gas superheat.

Fig. 36. Performance optimisation in part load operation: Influence of IHX and suction gas superheating (IHX+SH).

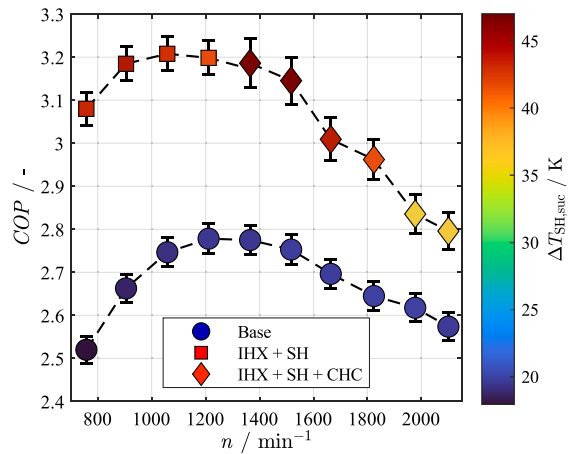


Fig. 37. Performance optimisation in part load operation: Influence of IHX, suction gas superheating and cylinder head cooling (IHX+SH+CHC).

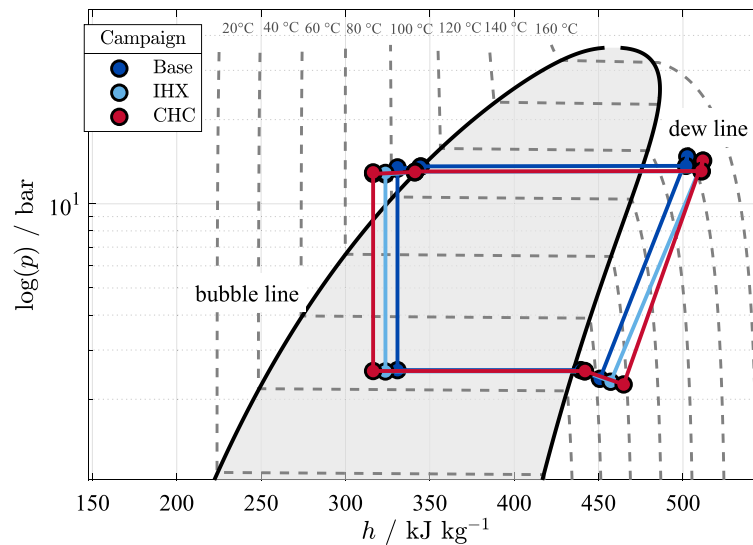


Fig. 38. $\log(p)$ - h diagram base scenario, optimised with IHX and CHC at 2100 rpm.

4. Conclusion

This study highlights the critical role of the part load performance of HTHPs in achieving decarbonisation goals, particularly in industrial energy systems and district heating networks. Comprehensive experimental measurement campaigns were conducted for a wide range of part load and off-design conditions. The test rig has a maximum thermal capacity of 37 kW and enables supply temperatures of up to 140 °C. In addition to an IHX, the system is equipped with water-cooled cylinder heads, enabling the recovery of compressor waste heat. The IHX includes a bypass line, which enables variable control of the recuperated heat and thus precise regulation of the suction-gas superheat. The refrigerant used is the hydrochlorofluoroolefin R1233zd(E), a synthetic refrigerant with low global warming potential (GWP = 1) and favourable thermodynamic properties for use in HTHPs.

The off-design analysis revealed a linear decrease in COP from 4.83 to 1.93 with increasing temperature lift up to 80 K, while secondary side variations showed that a larger temperature difference at the condenser raised the COP from 2.88 to 3.39, whereas a larger temperature difference at the evaporator reduced it from 3.46 to 2.95. In part load operation, the efficiency characteristic depends heavily on the supply temperature; at 90 °C, the COP dropped continuously from 4.67 to 3.63, while at 120 °C, increased relative heat losses shift the optimum to 1211 rpm with a peak COP of 2.78. Optimisation of the system parameters demonstrated that the suction gas superheat has the most significant impact, allowing for a COP increase of up to 22% at 120 °C supply temperature and net frequency. Conversely, the evaporator superheat had only a minor influence on the system efficiency and was fixed at 8 K. It was further shown that utilising cylinder head waste heat is beneficial at high speeds, as it reduces discharge temperatures and increases the potential use of recuperated energy. However, it proves detrimental at low speeds, where the suction gas temperature becomes the limiting factor. Nevertheless, the study is subject to certain limitations. The experimental setup utilised R1233zd(E), a synthetic refrigerant. In addition, the effective variation of suction gas superheat requires an oversized IHX, which may impact system cost and volume in practical applications. Furthermore, the results are specific to the tested piston compressor with a frequency converter and may differ for other compression and part load control technologies.

Overall, the results highlight the necessity of adapted control strategies for suction gas superheat and active cooling to maximise efficiency across the entire operating envelope.

The analysis of the operational behaviour of the laboratory system presented in this study clearly reveals significant optimisation potential

under part load and off-design conditions. The following key findings at the system integration level can be highlighted:

- The temperature lift has the strongest influence on the efficiency of the HTHP. The system design should therefore aim to minimise the temperature lift within the intended operating range.
- A maximum mass flow rate on the heat source side leads to small temperature differences at the evaporator and consequently increases the evaporation pressure. This reduces the pressure ratio and improves the heat pump efficiency.
- The temperature difference on the heat sink side should be selected as large as possible (i.e. minimum return temperature of the district heating network). This reduces the condensation pressure and increases the heat pump efficiency.
- The superheat after the evaporator is strongly dependent on the refrigerant. Optimising the suction gas superheat can enhance heat pump efficiency, particularly under part load conditions.
- The use of an oversized internal heat exchanger allows flexible adjustment of the suction gas superheat and can improve part load efficiency. The superheat should be maximised within the compressor's operational limits.
- The utilisation of the compressor waste heat can further increase the HTHP efficiency and enable more reliable compressor operation.
- The part load behaviour of the HTHP can be optimised through the implementation of advanced control strategies, enabling the system to operate more efficiently at low compressor speeds than at its nominal design point.

These findings provide a foundation for further advancements in HTHP technology, paving the way towards more efficient and sustainable energy systems. Future studies can focus on validating these optimisation approaches with natural low-GWP refrigerants and different compressor technologies to enhance the generalisability of the findings. In addition, a techno-economic analysis is recommended to evaluate whether the efficiency gains associated to the oversized internal heat exchanger are sufficient to justify the increased system cost and volume. Finally, the operational limits of steam generation and its control under off-design conditions should be investigated to further enhance the operating range of HTHPs.

BS	Base scenario (–)
COP	Coefficient of Performance (–)
cond	Condenser (–)
dis	Discharge (–)
evap	Evaporator (–)
$\dot{Q}_{\text{sink}}$	Heat sink capacity (kW)
$\dot{V}_{\text{WF}}$	Refrigerant Volume flowrate ($\text{m}^3 \text{s}^{-1}$)
Δ	Difference (–)
P_{el}	Electrical power consumption (kW)
Π	Pressure ratio (–)
η_{vol}	Volumetric efficiency (–)
η_{isen}	Isentropic efficiency (–)
$\dot{m}_{\text{WF}}$	Refrigerant mass flowrate (kg s^{-1})
h	Enthalpy (kJ kg^{-1})
n	Compressor speed (min^{-1})
p	Pressure (bar)
SH	Superheat (–)
suc	Suction (–)
T	Temperature ($^{\circ}\text{C}$)
IHX	Internal Heat Exchanger (–)
r	Cylinder inner radius (m)
rpm	Rounds per minute (min^{-1})
GWP	Global Warming Potential (CO_2e)
ODP	Ozone Depletion Potential (R-11e)
HCFO	Hydrochlorfluorolefine (–)
HFO	Hydrofluorolefine (–)
HTHP	High Temperature Heat Pump (–)
CHC	Cylinder head cooling (–)
s	Piston stroke (–)
sink	Heat sink (–)
source	Heat source (–)
WF	Working fluid (–)
z	Number of cylinder (–)

Nomenclature

CRedit authorship contribution statement

Jaromir Jeßberger: Conceptualization, Methodology, Investigation, Writing – original draft. **Lynn Briese:** Investigation, Writing – review & editing. **Echezona Obika:** Investigation, Writing – review & editing. **Florian Heberle:** Supervision, Project administration, Review. **Dieter Brüggemann:** Supervision, Funding acquisition.

Declaration of generative AI and AI-assisted technologies in the manuscript preparation process

AI-based tools (such as DeepL [43], Grammarly [44] and Gemini 3 Pro [45]) were used to reformulate and improve the linguistic quality of sections of text. After using this tools, the authors reviewed and edited the content as needed and take full responsibility for the content of the published article.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Appendix. Supplementary data

The experimental dataset is made publicly available (see Jeßberger et al. [38]) and provides comprehensive measurements of the high-temperature heat pump under part load and off-design conditions. It can be directly used for modelling, simulation validation, and the development of control strategies. Furthermore, the data support research on system-level optimisation, working fluid assessment, and industrial integration of high-temperature heat pumps.

Data availability

Data available at: <https://data.mendeley.com/datasets/56m3dd55zf/1> Jeßberger, Jaromir; Briese, Lynn; Obika, Echezona; Heberle, Florian; Brüggemann, Dieter (2026), “Dataset of Experimental Optimisation of the Performance of a High Temperature Heat Pump Under Part Load Conditions”, Mendeley Data, V1, doi: [10.17632/56m3dd55zf.1](https://doi.org/10.17632/56m3dd55zf.1).

References

- [1] Eurostat, EU renewable energy for heating and cooling reaches 26 %, 2025, URL <https://ec.europa.eu/eurostat/de/web/products-eurostat-news/w/ddn-20250305-1>.
- [2] J. Jeßberger, F. Heberle, D. Brüggemann, Maximising the potential of deep geothermal energy: Thermal output increase by large-scale heat pumps, Appl. Therm. Eng. (257) (2024) [http://dx.doi.org/10.1016/j.applthermaleng.2024.124240](https://doi.org/10.1016/j.applthermaleng.2024.124240).
- [3] S.S. Fosløe, B.R. Knudsen, M. Korpås, Integrated design and operational optimization of energy systems in dairies, Energy 281 (2023) 128242, [http://dx.doi.org/10.1016/j.energy.2023.128242](https://doi.org/10.1016/j.energy.2023.128242).
- [4] J. Persson, T. Berntsson, Influence of seasonal variations on energy-saving opportunities in a pulp mill, Energy 34 (10) (2009) 1705–1714, [http://dx.doi.org/10.1016/j.energy.2009.07.023](https://doi.org/10.1016/j.energy.2009.07.023).
- [5] C. Arpagaus, F. Bless, M. Uhlmann, J. Schiffmann, S.S. Bertsch, High temperature heat pumps: Market overview, state of the art, research status, refrigerants, and application potentials, Energy 152 (2018) 985–1010, [http://dx.doi.org/10.1016/j.energy.2018.03.166](https://doi.org/10.1016/j.energy.2018.03.166).
- [6] International Energy Agency (IEA), Project 68 - Industrial High-Temperature Heat Pumps, Task 1 - Technologies, Yearly report 2025 [http://dx.doi.org/10.23697/9b7b-sz34](https://doi.org/10.23697/9b7b-sz34).
- [7] Web of science, HTHP Veröffentlichungen pro jahr: 2010–2025, 2026, URL <https://www.webofscience.com/wos/woscc/summary/b37c05aa-01b0-478e-8820-54a23f8e6381-019647e5ef/relevance/1>.
- [8] O. Bamigbetan, T.M. Eikevik, P. Neksa, M. Bantle, Review of vapour compression heat pumps for high temperature heating using natural working fluids, Int. J. Refrig. 80 (2017) 197–211, [http://dx.doi.org/10.1016/j.ijrefrig.2017.04.021](https://doi.org/10.1016/j.ijrefrig.2017.04.021).
- [9] K.-M. Adamson, T.G. Walmsley, J.K. Carson, Q. Chen, F. Schlosser, L. Kong, D.J. Cleland, High-temperature and transcritical heat pump cycles and advancements: A review, Renew. Sustain. Energy Rev. 167 (2022) 112798, [http://dx.doi.org/10.1016/j.rser.2022.112798](https://doi.org/10.1016/j.rser.2022.112798).
- [10] J. Sun, Y. Wang, Y. Qin, G. Wang, R. Liu, Y. Yang, A review of super-high-temperature heat pumps over 100 °C, Energies 16 (12) (2023) 4591, [http://dx.doi.org/10.3390/en16124591](https://doi.org/10.3390/en16124591).
- [11] J. Jiang, B. Hu, R.Z. Wang, N. Deng, F. Cao, C.-C. Wang, A review and perspective on industry high-temperature heat pumps, Renew. Sustain. Energy Rev. 161 (2022) 112106, [http://dx.doi.org/10.1016/j.rser.2022.112106](https://doi.org/10.1016/j.rser.2022.112106).
- [12] H. Khalid, S. Uzair, M.U. Ahrens, S. Ren, P. Ganesan, I. Tolstorebrov, A. Arshad, Z. Said, A. Hafner, C.-C. Wang, R. Wang, T.M. Eikevik, Potential evaluation of integrated high temperature heat pumps: A review of recent advances, Appl. Therm. Eng. 230 (2023) 120720, [http://dx.doi.org/10.1016/j.applthermaleng.2023.120720](https://doi.org/10.1016/j.applthermaleng.2023.120720).
- [13] X. Ma, Y. Du, T. Zhao, T. Zhu, B. Lei, Y. Wu, A comprehensive review of compression high-temperature heat pump steam system: Status and trend, Int. J. Refrig. 164 (2024) 218–242, [http://dx.doi.org/10.1016/j.ijrefrig.2024.04.024](https://doi.org/10.1016/j.ijrefrig.2024.04.024).
- [14] S. Klute, M. Budt, M. van Beek, C. Doetsch, Steam generating heat pumps – overview, classification, economics, and basic modeling principles, Energy Convers. Manage. 299 (2024) 117882, [http://dx.doi.org/10.1016/j.enconman.2023.117882](https://doi.org/10.1016/j.enconman.2023.117882).
- [15] T. El Samad, A. Zabnienska-Gora, H. Jouhara, A.I. Sayma, A review of compressors for high temperature heat pumps, Therm. Sci. Eng. Prog. 51 (2024) [http://dx.doi.org/10.1016/j.tsep.2024.102603](https://doi.org/10.1016/j.tsep.2024.102603).

- [16] M.E.H. Tijani, G.J.O. Rodriguez, M. Ramirez, J.L. Poulsen, S. Spoelstra, Selection of lubricant oils for high temperature heat pumps: A review and selection methodology guidelines, *Appl. Therm. Eng.* 273 (2025) [http://dx.doi.org/10.1016/j.applthermaleng.2025.126483](https://doi.org/10.1016/j.applthermaleng.2025.126483).
- [17] L. Pan, H. Wang, Q. Chen, C. Chen, Theoretical and experimental study on several refrigerants of moderately high temperature heat pump, *Appl. Therm. Eng.* 31 (11–12) (2011) 1886–1893, [http://dx.doi.org/10.1016/j.applthermaleng.2011.02.035](https://doi.org/10.1016/j.applthermaleng.2011.02.035).
- [18] C. Arpagaus, S. Bertsch, Experimental results of HFO/HCFO refrigerants in a laboratory scale HTHP with up to 150 °C supply temperature, in: 2nd Conference on High Temperature Heat Pumps 2019.
- [19] C. Arpagaus, R. Kuster, M. Prinzing, M. Uhlmann, E. Büchel, S. Frei, J. Schiffmann, S. Bertsch, High temperature heat pump using HFO and HCFO refrigerants - system design and experimental results, in: 25th IIR International Congress of Refrigeration (ICR 2019), 2019.
- [20] Y. Xu, Z. Zhu, C. Fu, W. Xia, Experimental investigation on an ultra-high temperature air source heat pump water heater, *J. Therm. Sci. Eng. Appl.* 13 (6) (2021) [http://dx.doi.org/10.1115/1.4050234](https://doi.org/10.1115/1.4050234).
- [21] O. Bamigbetan, T.M. Eikevik, P. Neksa, M. Bantle, C. Schlemminger, Experimental investigation of a prototype R-600 compressor for high temperature heat pump, *Energy* 169 (2019) 730–738, [http://dx.doi.org/10.1016/j.energy.2018.12.020](https://doi.org/10.1016/j.energy.2018.12.020).
- [22] J. Navarro-Esbrí, A. Fernández-Moreno, A. Mota-Babiloni, Modelling and evaluation of a high-temperature heat pump two-stage cascade with refrigerant mixtures as a fossil fuel boiler alternative for industry decarbonization, *Energy* 254 (2022) 124308, [http://dx.doi.org/10.1016/j.energy.2022.124308](https://doi.org/10.1016/j.energy.2022.124308).
- [23] J. Jiang, B. Hu, R. Wang, T. Ge, H. Liu, Z. Zhang, Y. Zhou, Experiments of advanced centrifugal heat pump with supply temperature up to 100 °C using low-GWP refrigerant R1233zd(e), *Energy* 263 (2023) 126033, [http://dx.doi.org/10.1016/j.energy.2022.126033](https://doi.org/10.1016/j.energy.2022.126033).
- [24] M. Ramirez, F. Trebilcock-Kelly, J.L. Corrales-Ciganda, J. Payá, A.H. Hassan, Experimental and numerical investigation of a novel high-temperature heat pump for sensible and latent heat delivery, *Appl. Therm. Eng.* 247 (2024) 122961, [http://dx.doi.org/10.1016/j.applthermaleng.2024.122961](https://doi.org/10.1016/j.applthermaleng.2024.122961).
- [25] X. Ma, Y. Fang, Q. Dai, Z. He, J. Yang, J. Liu, H. Zou, Experimental evaluation of a high temperature cascade heat pump based on refrigerant charge, *Case Stud. Therm. Eng.* 73 (2025) [http://dx.doi.org/10.1016/j.csite.2025.106573](https://doi.org/10.1016/j.csite.2025.106573).
- [26] D. Wu, J. Wei, B. Hu, Theoretical analysis, experimental research and industrial verification of high temperature heat pump based on R1233zd(e), *Energy* 319 (2025) [http://dx.doi.org/10.1016/j.energy.2025.135175](https://doi.org/10.1016/j.energy.2025.135175).
- [27] Z. Wu, J. Liu, X. Zhang, Experimental investigation on high-temperature heat pump water heater of R1234ze, in: *Proceeding of the 5th International Conference on Advanced Design and Manufacturing Engineering*, vol. 39, Atlantic Press, France, 2015, pp. 1403–1406.
- [28] S.J. Lee, B.H. Shon, C.W. Jung, Y.T. Kang, A novel type solar assisted heat pump using a low GWP refrigerant (r-1233zd(e)) with the flexible solar collector, *Energy* 149 (2018) 386–396, [http://dx.doi.org/10.1016/j.energy.2018.02.018](https://doi.org/10.1016/j.energy.2018.02.018).
- [29] N.N. Shah, D. Cotter, N. Hewitt, Overview on HCFO-r1233zd(e) use for high temperature heat pump application, in: 25th IIR International Congress of Refrigeration (ICR 2019), vol. 286, 2019, [http://dx.doi.org/10.18462/iir.icr.2019.0286](https://doi.org/10.18462/iir.icr.2019.0286).
- [30] C. Mateu-Royo, J. Navarro-Esbrí, A. Mota-Babiloni, F. Molés, M. Amat-Albuixech, Experimental exergy and energy analysis of a novel high-temperature heat pump with scroll compressor for waste heat recovery, *Appl. Energy* 253 (2019) 113504, [http://dx.doi.org/10.1016/j.apenergy.2019.113504](https://doi.org/10.1016/j.apenergy.2019.113504).
- [31] C. Arpagaus, S. Bertsch, Experimenteller vergleich von R1224yd(z) und R1233zd(e) in einer hochtemperatur-wärmepumpe, *KI Kälte* (03) (2020) 46–55.
- [32] C. Arpagaus, S. Bertsch, Experimental comparison of HCFO and HFO R1224yd(z), R1233zd(e), R1336mzz(z), and HFC R245fa in a high temperature heat pump up to 150 °C supply temperature temperature heat pump up to 150 °C supply temperature, in: *International Refrigeration and Air Conditioning Conference 2021*, vol. 2611.
- [33] J. Jeßberger, F. Heberle, D. Brüggemann, Integration von hochtemperatur-wärmepumpen in erneuerbare energiesysteme, in: *DKV Tagungsband*, 2022, AA IV 19. URL <https://dkv.org/index.php?id=160>.
- [34] A.H. Hassan, J.M. Corberán, M. Ramirez, F. Trebilcock-Kelly, J. Payá, A high-temperature heat pump for compressed heat energy storage applications: Design, modeling, and performance, *Energy Rep.* 8 (2022) 10833–10848, [http://dx.doi.org/10.1016/j.egy.2022.08.201](https://doi.org/10.1016/j.egy.2022.08.201).
- [35] T. Chen, O. Kyung Kwon, Experimental analyses of moderately high-temperature heat pump systems with R245fa and R1233zd(E), *Energy Eng.* 119 (6) (2022) 2231–2242, [http://dx.doi.org/10.32604/ee.2022.021289](https://doi.org/10.32604/ee.2022.021289).
- [36] J. Navarro-Esbrí, A. Mota-Babiloni, Experimental analysis of a high temperature heat pump prototype with low global warming potential refrigerant R-1336mzz(z) for heating production above 155 °C, *Int. J. Thermofluids* 17 (2023) 100304, [http://dx.doi.org/10.1016/j.ijft.2023.100304](https://doi.org/10.1016/j.ijft.2023.100304).
- [37] J. Jeßberger, F. Heberle, D. Brüggemann, Experimental optimization of part-load behaviour in a high-temperature heat pump, in: *8th International Seminar on ORC Power Systems 2025*.
- [38] J. Jeßberger, L. Briese, E. Obika, F. Heberle, D. Brüggemann, Dataset of experimental optimisation of the performance of a high-temperature heat pump under part-load conditions, 2026, [http://dx.doi.org/10.17632/56m3dd55zf.1](https://doi.org/10.17632/56m3dd55zf.1).
- [39] J. Jeßberger, C. Arpagaus, F. Heberle, L. Brendel, S. Bertsch, D. Brüggemann, Experimental investigations of upscaling effects of high-temperature heat pumps with R1233zd(E) études expérimentales des effets d'échelle des pompes à chaleur haute température au R1233zd(E), *Int. J. Refrig.* (2024) [http://dx.doi.org/10.1016/j.ijrefrig.2024.04.023](https://doi.org/10.1016/j.ijrefrig.2024.04.023).
- [40] Intergovernmental Panel on Climate Change (IPCC), Climate change 2022 - mitigation of climate change, 2023, [http://dx.doi.org/10.1017/9781009157926](https://doi.org/10.1017/9781009157926).
- [41] FUCHS, Safety data sheet, BSE85k, 2020.
- [42] J. Jeßberger, F. Heberle, D. Brüggemann, Experimental investigations of waste heat utilization of high-temperature heat pump compressors, in: *International Refrigeration and Air Conditioning Conference At Purdue*, 2024, URL <https://docs.lib.purdue.edu/iracc/2608>.
- [43] D. S.E., DeepL translator, 2026, KI-basiertes Übersetzungstool. URL <https://www.deepl.com>.
- [44] Grammarly, Inc., Grammarly: AI writing assistance, 2024, <https://www.grammarly.com>.
- [45] Google DeepMind, Gemini 3 pro (preview), 2026, Large Language Model. URL <https://deepmind.google/technologies/gemini/>.