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Micro heat pump system with thermal storage for domestic appliances

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Abstract

The application of heat pump systems in domestic appliances offers significant energy savings compared to the use of electric heating elements. Required cooling or dehumidification can provide the heat source for the heat pump. Peculiarities of small-scale domestic applications are the strong variability in both operating conditions and load, combined with a large temperature lift between cooling and heating demand.
To tackle these challenges, a refrigerant cycle consisting of a low-temperature reservoir (e.g. ice storage or cold water reservoir), a high-temperature reservoir (e.g. hot water storage) and an air/refrigerant heat exchanger with a thermal storage at an intermediate temperature level is investigated. Heat can be shifted between these three storage tanks at different temperature levels by means of a heat pump system. In addition, heat can be extracted from or transferred to the ambient air using the air/refrigerant heat exchanger. The heat pump system utilizes the natural refrigerant R600a and a hermetic reciprocating compressor.
In the scope of this paper, a possible refrigerant circuit is presented. Experimental investigations of a small-scale water/water heat pump with a hermetic reciprocating compressor form the basis for the further investigation of the proposed system supported by simulation models.

Keywords: hermetic compressor, R600a, simulation, experimental investigation

1. Introduction

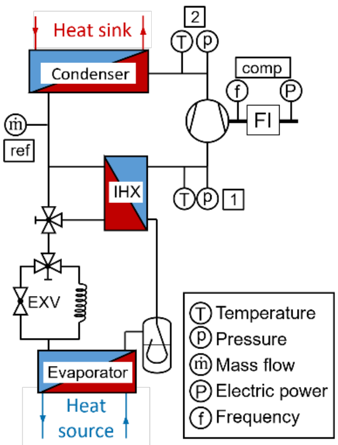
In recent years, heat pump technology has been increasingly integrated into domestic appliances to improve their energy efficiency and reduce their carbon footprint. Market available products cover dishwashers, tumble dryers and washer dryers. Recent work covers a heat pump water heater of small size [1], using R600a (Isobutane) as refrigerant. R290 (Propane) is applied in market available tumble dryers, with one example reported by [2]. Other applications of natural refrigerants are currently under research such as R600a – which is currently the standard refrigerant for household refrigerators – in a dishwasher [3, 4, 5]. R744 (CO_2) was investigated for the use in tumble dryers by [6]. [3] investigated a heat pump using a capillary tube, constant speed reciprocating compressor and ice storage as heat source in a dishwasher, reaching COPs between 1.2 and 1.5 in experiments. [4] further investigated utilizing the ice storage for condensing humid air during the drying process, where the overall electricity consumption decreased by 39 Wh compared to a dynamic open drying method using a fan only, but the system required a well-balanced amount of ice in the thermal storage and put rather strict limitations on the temperatures at the start of the drying process and drying time to achieve good performance. [7] simulatively compared the application of R134a, R600a and R290 in dishwashers and found a 25 % reduction in electric energy consumption for the heat pump variant compared to direct electric heating. [6] investigated tumble dryers with R744 (CO_2) by means of simulation, where the focus was on cycle modifications such as ejectors and internal heat exchangers. Though, the experimental and simulation results were promising, the use of R744 requires very high pressure resistances of the components (more than 100 bar) as well as a rather good control over the heat pump cycle due to trans-critical operation (additional components like gas coolers are needed possibly too). [1] developed an air to water heat pump for heating up 4 liters of water up to 50 °C using the refrigerant R600a and a capillary tube as expansion device. Experimental results showed that the evaporating temperature was very low (-5 °C to 5 °C) resulting in high energy demand of the compressor. Compared to electrical resistance heaters the energy demand was reduced by 26 % at a high increase of operation time (+65 min). The achieved time averaged COPs showed good results (3 to 7.7), yet the corresponding ambient air temperatures (25 °C to 33 °C) do not resemble typical room temperatures.

Key challenges associated with the application in domestic appliances are high condensing temperatures, large temperature lift in case of ambient air as heat source, transient operation and limited available space. In the course of the project "ECHODA" (Energy Efficient Cooling and Heating of Domestic Appliances) [8], innovative concepts for household appliances with cooling and heating requirements are being developed to improve efficiency and sustainability. The present work covers a micro heat pump system designed to replace direct electric heating elements in household applications that enables pumping heat between three temperature levels, a cold reservoir (CR), a hot reservoir (HR) and a thermal energy storage (TES) at intermediate temperature level which also acts as an air/refrigerant heat exchanger by switching on a fan. The temperature level of the hot reservoir reaches up to 100 °C, covering the challenge of high condensing temperatures, while both the cold reservoir temperature (below 10 °C) as well as using ambient air as the heat source demands a high temperature lift. The introduced TES enables to split the required temperature lift by acting as a heat sink when cooling the CR and as a heat source for heating the HR at an intermediate temperature level, especially during transient operation. Due to the spatial limitations, reflected by the mass of the TES of 4 kg, it is possible to use ambient air as a heat sink or heat source. The simulation is based on experimental results of a water/water heat pump which confirmed the compressor performance in the considered operating envelope. Furthermore, polynomial approximations of the evaluated compressor efficiencies are used to model the compressor in the simulation model.

2. Small scale water/water heat pump

2.1. Refrigerant cycle and auxiliary hydraulics

The small-scale water/water heat pump used to gain experimental data for the compressor efficiencies is based on the work of Illmer [9], with additional modifications. The one-stage cycle utilizes a variable speed hermetic reciprocating compressor with a displacement of 7ccm and a maximum speed of 4500 rpm, an optional internal heat exchanger (IHX), a low-pressure accumulator (LP-accu) and different expansion devices such as an electric expansion valve (EXV) and a capillary tube. Modifications to the refrigerant cycle concern the addition of electric expansion valves (EXVs) of different size and a different LP-accu. Evaporator and condenser are plate heat exchangers, the IHX is of tube-in-tube type, all of them operate in counter-flow. Fig. 1 (left) depict the initial configuration of the refrigerant cycle. For better visibility, the picture in Fig. 1 (right) shows the refrigerant cycle before applying thermal insulation and the mentioned modifications. Prior to the experiments, the components of the refrigerant cycle with exception of the compressor and inverter were insulated.



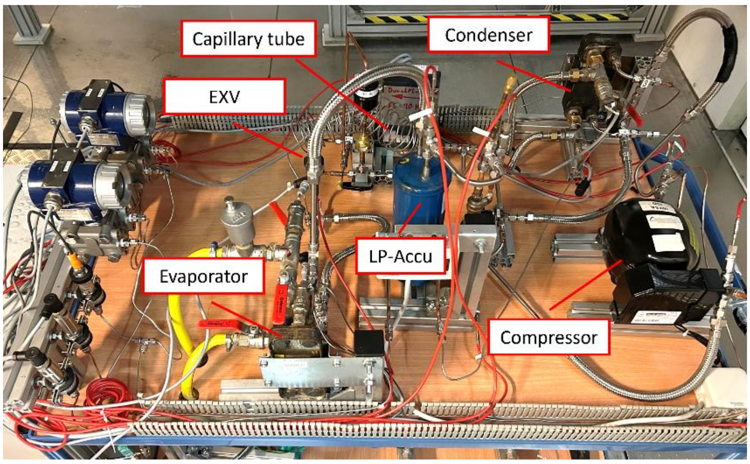


Figure 1. Schematic with measurement equipment used for the evaluations within this work (left) and picture (right) of the refrigerant cycle before applying thermal insulation, based on Illmer [9]

Hydraulic circuits supply water at a set temperature and flow rate to the evaporator and condenser. In the heat source cycle, an electric heater is used to heat water, covering the cooling capacity of the test rig. While a PID-controller controls the electric heater via pulse-width-modulation to reach the set inlet temperature, the flow rate is adjusted manually using valves. The heat sink hydraulic circuit uses a process thermostat with cooling function to cover the heating capacity of the test rig and to set the inlet temperature to the condenser. The water flow rate is adjusted manually using a valve.

2.2. Measurement equipment

The measurement equipment used to evaluate the compressor efficiencies is listed in Table 1 including the expected uncertainties of reading at the measured operating points of the prototype. The accuracy of the temperature sensors was determined through calibration. For the other measured quantities manufacturers' specifications regarding the accuracy of the sensor as well as the acquisition system were taken into consideration. The uncertainty of measurement of evaluated quantities are calculated using the Gaussian Error Propagation implemented in the Engineering Equation Solver (EES) [10]. Property data of R600a and water are evaluated with EES.

Table 1. Measurement equipment used for evaluating the compressor efficiencies with associated uncertainties of reading

Measured quantity	Location	Sensor	Uncertainty of reading
Temperature	1, 2	Pt100	0.17 K
Pressure	1, 2	Pressure transducer (absolute)	0.023 bar (p_1), 0.083 bar (p_2)
Mass flow	ref	Coriolis-type mass flowmeter	0.04 g/s
Electric Power	comp	Power meter	1% of the measured power
Frequency	comp	Custom built inductive sensor	1% of the measured frequency

2.3. Measured compressor efficiencies

The volumetric efficiency, inner isentropic efficiency and "mechanical" efficiency are evaluated according to Eq. (2), (3) and (4), respectively. Polynomials depending the pressure ratio of the compressor, see Eq. (1), are used as approximations.

$$\Pi = \frac{p_2}{p_1} \tag{1}$$

$$\lambda_{vol} = \frac{\dot{m}_{ref}}{\rho_{ref,1} \cdot \dot{V}_{swept}} = 0.97 - 0.0345 \cdot \Pi \tag{2}$$

$$\eta_{is,i} = \frac{h_{2s} - h_1}{h_2 - h_1} = 0.3777 + 0.0802 \cdot \Pi - 0.0064 \cdot \Pi^2 \tag{3}$$

$$\eta_m = \frac{\dot{m}_{ref} \cdot (h_2 - h_1)}{P_{el}} = 0.321 + 0.175 \cdot \Pi - 0.0245 \cdot \Pi^2 + 0.009 \cdot \Pi^3 \tag{4}$$

Multiplying the inner isentropic efficiency with the "mechanical" efficiency yields the overall isentropic efficiency, as shown with equation (5).

$$\eta_{is,i} \cdot \eta_m = \eta_{is,ov} = \frac{\dot{m}_{ref} \cdot (h_{2s} - h_1)}{P_{el}} \tag{5}$$

Fig. 2 depicts the operating points in terms of evaporating and condensing temperatures for which the compressor efficiencies were evaluated.

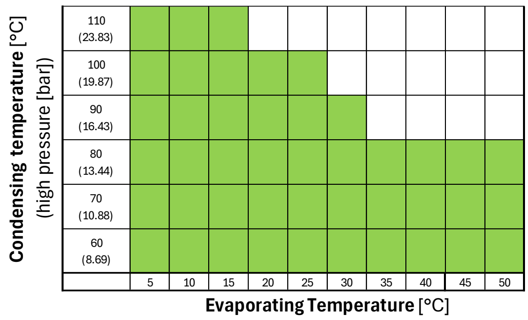


Figure 2. Test envelope (evaporating and condensing temperatures) when evaluating compressor efficiencies

Fig. 3 depicts the evaluated compressor efficiencies - the volumetric efficiency (top left), "mechanical" efficiency (top right), inner isentropic efficiency (bottom left) and overall isentropic efficiency (bottom right) - with the associated uncertainties and polynomial approximations used in the simulation model, plotted over the pressure ratio according to Eq. (1). The volumetric efficiency shows an almost linear decrease with increasing pressure ratio. Different values for the volumetric efficiency at the same pressure ratio are caused by different evaporation temperatures. The approximation as a linear function of the pressure ratio cannot depict such effects and also leads to a volumetric efficiency different to one at a pressure ration of one but for the sake of a better fit across the tested operating range. Further detailed investigations might use a more sophisticated compressor model but for the scope of this work, simple polynomial correlations are chosen. Similar considerations apply as well to the approximations of the mechanical efficiency and the inner isentropic efficiency.

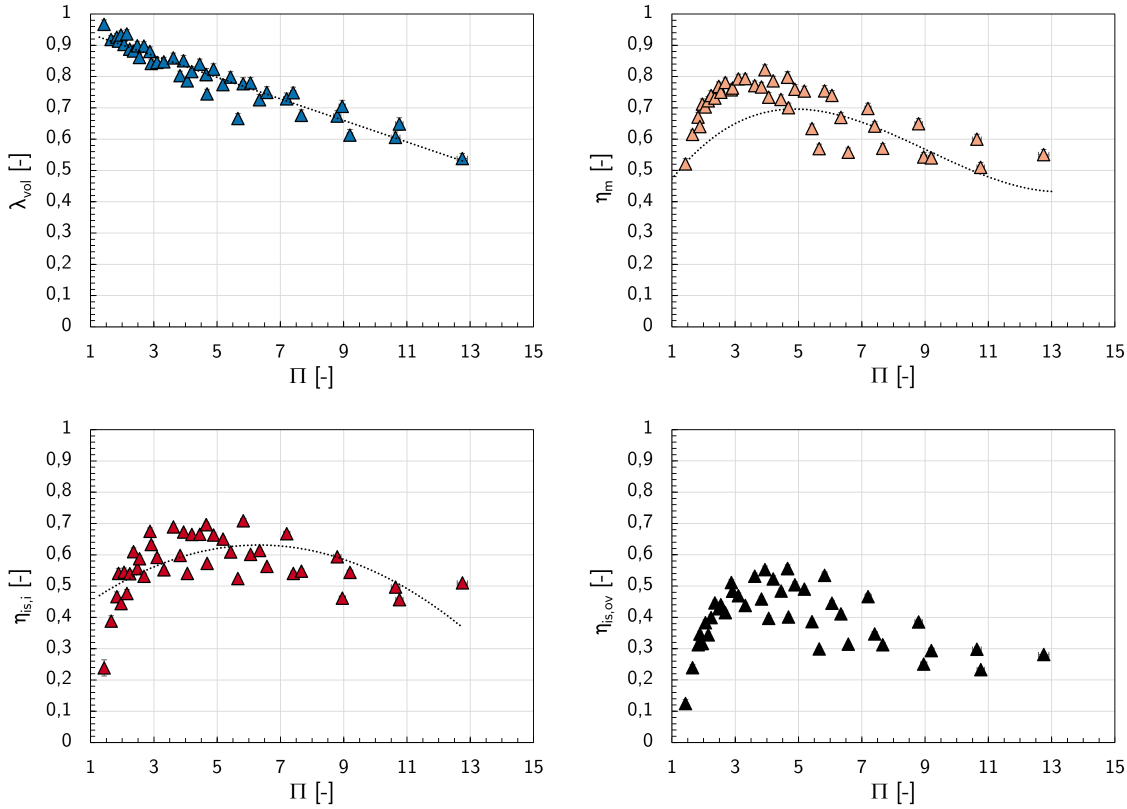


Figure 3. Compressor efficiencies evaluated from experimental data and polynomial approximations used in the simulation model

3. Cascade concept with TES

3.1. Refrigerant cycle

The investigated refrigerant cycle allows to pump heat between the three temperature levels cold reservoir (CR) consisting of 3 l water, hot reservoir (HR) with 5 l of water and thermal energy storage (TES) of 4 kg AL1100 with a specific thermal capacity of 914 J/(kg·K) [11] at intermediate temperature. The TES can also be used as a refrigerant/air heat exchanger. 3-way valves in the refrigerant cycle allow for three operating modes. Cooling the CR and heating up the TES (mode 1), using the TES as evaporator when heating up the HR (mode 2) in which the fan can be activated to improve heat transfer from the ambient air, or cooling the CR while heating the HR (mode 3). Fig. 4 depicts the described refrigerant cycle schematically.

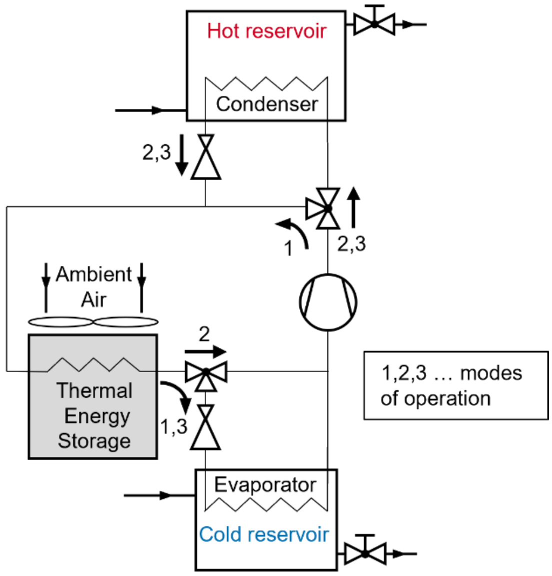


Figure 4. Concept of the refrigerant cycle to pump heat between three temperature levels (mode 1: CR-TES, mode 2: TES-HR, mode 3:CR-HR) using one compressor

Fig. 5 shows t/h-Diagrams of the three modes of operation, the assumptions of uniform temperature in the CR and TES, and temperature stratification in the HR are visible.

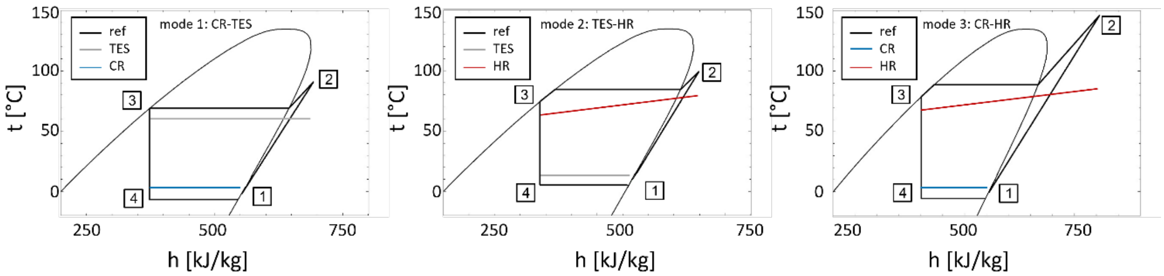


Figure 5. t/h-Diagrams of the three modes of operation: CR-TES (mode 1), TES-HR (mode 2), CR-HR (mode 3)

3.2. Simulation model

The simulation model set up in EES [11] combines a steady-state model of the refrigerant cycle with transient models of the cold reservoir, the TES and the hot reservoir. At the hot reservoir, a linear temperature stratification is considered which enables subcooling of the refrigerant. The cooling and heating capacity of the refrigerant cycle are extracted from or supplied to the reservoirs or the TES according to the mode of operation.

The refrigerant cycle considers 5 K suction gas superheat in the heat exchanger used as evaporator (either cold reservoir or TES, depending on the mode of operation), no subcooling in case of the TES being used as condenser and if the hot reservoir is used as condenser, the subcooling at the outlet of the condenser depends on the temperature stratification. Table 2 lists the coupling of reservoirs and TES with the refrigerant cycle depending on the mode of operation. The heat exchangers are modelled using temperature differences, where at the TES $\Delta T_{ref,TES} = 10$ K and at the reservoirs $\Delta T_{ref,w} = 10$ K is assumed. Further work is needed regarding the actual design of the heat exchangers, however possible options include coil tubes immersed in the water reservoirs. Considering the TES with the air/refrigerant heat exchanger, on of the main challenges will be to both minimize the heat losses of the TES if no heat exchange with ambient air is desired while enabling good heat transfer and thus a small temperature difference between ambient air and refrigerant if ambient air is used as heat source or heat sink.

Table 2. Coupling of the refrigerant circuit with the reservoirs according to the mode of operation

Mode	t_0	t_C	t_3, x_3	$\dot{Q}_{CR}$	$\dot{Q}_{TES}$	$\dot{Q}_{HR}$
1 (CR-TES)	$t_{CR} - \Delta T_{ref,w}$	$t_{TES} + \Delta T_{ref,TES}$	t_C (x_3=0)	$-\dot{Q}_0$	$\dot{Q}_h$	
2 (TES-HR)	$t_{TES} - \Delta T_{ref,TES}$	$t_{HR,high} + \Delta T_{ref,w}$	$t_{HR,low} + \Delta T_{ref,w}$		$-\dot{Q}_0$	$\dot{Q}_h$
3 (CR-TES)	$t_{CR} - \Delta T_{ref,w}$	$t_{HR,high} + \Delta T_{ref,w}$	$t_{HR,low} + \Delta T_{ref,w}$	$-\dot{Q}_0$		$\dot{Q}_h$

The compressor is modelled using polynomial approximations for the compressor efficiencies listed in Eq. (1) to (3). Expansion is assumed isenthalpic. Cooling and heating capacity are modelled according to Eq. (6) and Eq. (7), respectively with the state points according to Fig. 5.

$$\dot{Q}_0 = \dot{m}_{ref} \cdot (h_1 - h_4) \tag{6}$$

$$\dot{Q}_h = \dot{m}_{ref} \cdot (h_2 - h_3) \tag{7}$$

The temperature of the cold reservoir and TES are modelled by integrating Eq. (8) and Eq. (9), respectively. Heat transfer to and from the refrigerant cycle is considered with $\dot{Q}_{\mathrm{CR}}$ and $\dot{Q}_{\mathrm{TES}}$ according to Table 2. Heat transfer to and from the ambient ($t_{amb} = 20\ °C$) is considered with $UA_{CR} = 2\ W/K$ for the CR and a variable UA-value ($UA_{TES} = 2 \ldots 70\ W/K$) for the TES, representing the option to turn on a fan and increase the heat transfer with the ambient air.

$$V_{CR} \cdot \rho_w \cdot c_w \cdot \frac{\mathrm{d}t_{\mathrm{CR}}}{\mathrm{d}\tau} = \dot{\mathrm{Q}}_{CR} + UA_{CR} \cdot (t_{amb} - t_{CR}) \tag{8}$$

$$m_{TES} \cdot c_{TES} \cdot \frac{\mathrm{d}t_{\mathrm{TES}}}{\mathrm{d}\tau} = \dot{\mathrm{Q}}_{TES} + UA_{TES} \cdot (t_{amb} - t_{TES}) \tag{9}$$

The mean temperature of the hot reservoir is modelled by integrating Eq. (10) which considers heat supply from the refrigerant cycle via $\dot{Q}_{\mathrm{HR}}$ and heat losses to the ambient ($UA_{HR} = 3\ W/K$). The assumed linear temperature stratification between $t_{\mathrm{HR,min}}$ and $t_{\mathrm{HR,max}}$ of 10 K is considered by evaluating a mean enthalpy according to the high and low temperature in the reservoir.

$$V_{HR} \cdot \rho_w(t_{HR,mean}) \cdot c_w(t_{HR,mean}) \cdot \frac{\mathrm{d}t_{HR,mean}}{\mathrm{d}\tau} = \dot{\mathrm{Q}}_{HR} + UA_{HR} \cdot (t_{amb} - t_{HR,mean}) \tag{10}$$

$$h(t_{HR,mean}) = \frac{h(t_{HR,high}) + h(t_{HR,low})}{2} \tag{11}$$

4. Results

Fig. 6 shows the results of the transient simulation model. Temperatures of CR, TES and HR, condensing and evaporating temperature on top, the operating modes and overall heat transfer coefficient from the TES to the ambient – indicating whether the fan is on or not – in the middle and heating capacity, cooling capacity, electric power consumption of the compressor and COP for heating and cooling in the bottom. Within the first 400 seconds, the CR is cooled down from 15 °C to 4 °C, while the TES is heated up (mode 1) to 67 °C. This is followed by further heating the HR, using the TES as heat source (mode 2 without fan). Within further 200 seconds, the temperature of the TES drops to 8 °C and thus significantly below ambient temperature. So the fan is activated (considered in the simulation model by increasing the overall heat transfer coefficient from 2 to 70 W/K) to use ambient air as heat source for heating up the 5 l HR (mode 2 with fan). After in total 1750 seconds of operation, the HR is heated to a mean temperature of ca. 60 °C. Heat influx from the ambient (t_{amb} =20 °C) leads to an increase of the CR temperature to 7 °C, necessitating another cooling cycle for the CR. This is accomplished in mode 3 (cooling the CR and heating the HR) for 250 seconds, leading to a CR temperature of 3 °C. This is followed by the final heating of the HR in mode 2 with fan up to a water temperature of 100 °C, which is eventually reached after 3864 seconds. In order to enable temperatures exceeding 100 °C, the hot water reservoir is pressurized.

During most of the operation, the heating capacity is in a range between 500 and 1000 W, using a compressor speed of 4500 rpm. The electric power consumption is between 170 and 370 W. Capacity and power peaks occur during the shift to mode 2 between 400 and 500 seconds when the TES at a temperature of approximately 60 °C causes high evaporating temperatures while the low temperature in the HR does not require high condensing temperatures.

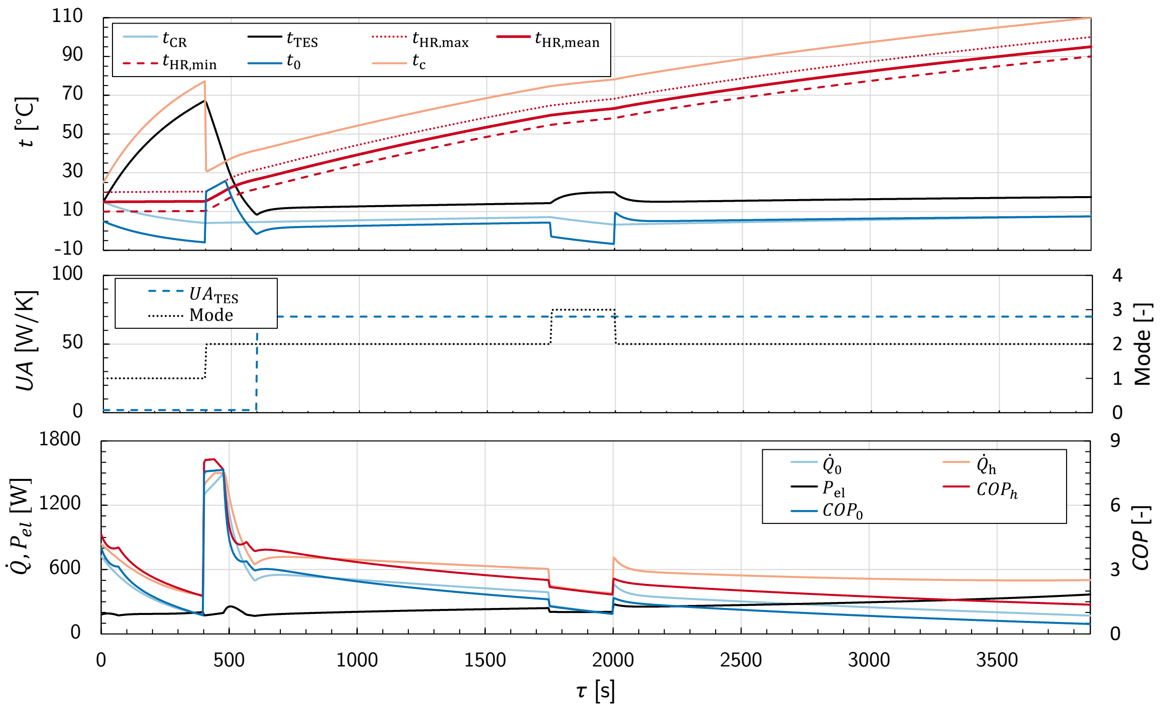


Figure 6. Results of the transient simulation model, temperatures (top), operating mode and fan overall heat transfer coefficient of the TES (middle) and COPs, heating and cooling capacity, and electric power consumption (bottom)

During cooling down of the CR in the first 400 seconds and between 1750 and 2000 seconds, 57 Wh of heat were extracted from the CR, requiring 35 Wh of electric energy which corresponds to a cooling performance factor, see Eq. (12), of 57/35 = 1.63. Heating up the HR between 400 and 3864 seconds supplied 581 Wh to the HR at the cost of 248 Wh electric energy, leading to a heating performance factor, see Eq. (13), of 581/248 = 2.34.

$$PF_0 = \int \frac{\dot{Q}_o}{P_{el}} \mathrm{d}\tau \tag{12}$$

$$PF_h = \int \frac{\dot{Q}_h}{P_{el}} \mathrm{d}\tau \tag{13}$$

5. Conclusions and outlook

The investigated small scale heat pump system allows to pump heat between three temperature levels, a cold reservoir, a hot reservoir and a thermal energy storage at intermediate temperature level which can also be used as an air/refrigerant heat exchanger.

The presented simulation model combines a steady-state model for the refrigerant cycle with transient models for the reservoirs and TES, modeled as lumped thermal masses, considering temperature stratification in the hot reservoir. The used compressor was previously investigated experimentally and the evaluated compressor efficiencies used in the simulation model. This was carried out using a small-scale water/water heat pump which enables to test different components due to its modular design. By means of the simulation model, the transient operation of the proposed system can be studied. During 3864 seconds, the 3 l cold reservoir was cooled from 15 °C to 4 °C, and later again from 7 °C to 3 °C, while the hot reservoir was heated from a mean temperature of 15 °C to a mean temperature of 95 °C, leading to a maximum temperature of 100 °C considering the 10 K stratification. The corresponding performance factors for cooling and heating were 1.63 and 2.34, respectively.

Possible topics for future work include transient simulations with increased level of detail. Considering the refrigerant cycle, refrigerant migration might be of concern, an alternative approach would be to use a compact refrigerant cycle with e.g. plate heat exchangers for evaporator and condenser and connect the different reservoirs and the TES via water cycles.

Nomenclature

Abbreviations and Indices

0	Cooling	h	Heating
COP	Coefficient of Performance	HR	Hot reservoir
comp	Compressor	IHX	Internal heat exchanger
CR	Cold reservoir	LP-accu	Low-pressure accumulator
EXV	Electric expansion valve	*s*	Isentropic change of state
ref	Refrigerant	TES	Thermal Energy Storage
amb	Ambient	w	Water

Symbols

c	Specific heat capacity (J/(kg·K))	$\dot{Q}_0$	Cooling capacity (W)
$\eta_{is,ov}$	Overall isentropic efficiency (-)	$\dot{Q}_h$	Heating capacity
$\eta_{is,i}$	Inner isentropic efficiency (-)	$\Delta T_{ref,w}$	Temperature difference between refrigerant and water
η_m	"Mechanical" efficiency (-)	$\Delta T_{ref,TES}$	Temperature difference between refrigerant and TES
h	Specific enthalpy (J/kg)	t	Temperature (°C)
λ_{vol}	Volumetric efficiency	t_0	Evaporating temperature (°C)
m	Mass (kg)	t_c	Condensing temperature (°C)
$\dot{m}_{ref}$	Refrigerant mass flow (kg/h)	UA	Overall heat transfer coefficient (W/K)
p	Pressure (bar)	$\dot{V}$	Volume flow
P_{el}	El. power of compressor and inverter (W)	V	Volume
PF	Performance factor	$\dot{V}_{swept}$	Compressor swept volume flow (m³/h)
Π	Pressure ratio (-)	x	Vapour fraction (-)
ρ	Density (kg·m³)		

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