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Control optimization of a double stage heat pump with desuperheater in multi-family NZEBs

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Abstract

Two multi-family buildings (MFB) with together 26 flats were designed to achieve a net Zero Energy (NZE) balance for heating, domestic hot water (DHW) preparation and all auxiliary energies. The roof of one of the MFB is covered by PV, the other is partly used for PV and partly for solar thermal (ST). A double-stage ground water heat pump (HP) with hot gas desuperheating (DSH) provides the remaining energy for heating and DHW. The system was designed with a low temperature distribution system (floor heating) and separate decentral fresh water preparation (DHW plate heat exchanger) for low thermal losses and optimal efficiency.
Detailed performance maps of the double-stage HP pump with DSH were generated analyzing the monitoring data and by means of refrigerant cycle simulation. Simulation based optimization was performed with the aim to operate the HVAC system with improved performance. Different system configurations (i.e. with or without ST, with different buffer storage sizes, with different control strategies and set points) can be compared and evaluated also considering (seasonal) variations of the share of renewables in the electricity mix.

Keywords: NZEB; Multi-family buildings; double stage heat pump; desuperheater; building and HVAC simulation

1. Introduction

In order to reach the ambitious targets of a sustainable energy system, strong efforts in improving energy efficiency in the building sector are required, see [1]. Only with a combination of passive (i.e. building envelope) and active (i.e. HVAC system) measures as well as integrating renewable energies (RE) these targets can be reached. System solutions, i.e. well designed and matching components with a robust control scheme and the possibility of easy quality control, are required.

While there is a discussion among experts about the most economic building energy standard (e.g. whether it is Passive House standard or less ambitious), it is undiscussed that heat pumps (HP) will play a major role in heating (and cooling) of buildings. HPs exist in a large variety with respect to the source (air, water, ground, waste heat) and the sink (air, water; low temperature heating, DHW etc.) and in almost any size (from some 100 W to several 100 kW). Even though there is experience with HPs since several decades and a strong development of HPs within at least the last 20 years, there is still a lack of knowledge of HP operation and optimal system design for nZEB [2,3]and in particular nZE multi-family buildings [4, 5]. In [6, 7, 8, 9] increasing performance of installed HP systems within the recent years is reported. However, the market and its players lack experience and guidelines for HP system for nZEB are still missing (see also e.g. IEA HPT Annex 49).

In this paper, a case study of two NZE multi-family buildings in Passive House standard is presented with an innovative HVAC system. In this work, it is presented, how by means of simulation with models that were parameterized using detailed monitoring data (refrigerant cycle as well as building and HVAC simulations), failures could be detected and how the performance could be improved. The results can be used to derive guidelines for HP system design and failure detection and quality control.

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Nomenclature

Abbreviation	Description	Abbreviation	Description
BMS	Building management system	HW	Hot water
BS	Buffer storage	HX	Heat exchanger
CW	Cold water	HVAC	Heating ventilation air conditioning
comp	Compressor	LEH	Low energy house
cond	Condenser	MFB	Multi-family building
DHW	Domestic hot water	NZE(B)	Net Zero energy (building)
DSH	Desuperheater	nZEB	Nearly Zero energy building
el	Electric	PH	Passive house
evap	Evaporator	PM	Performance map
FH	Floor heating	PV	Photovoltaic
GW	Ground water	RE	Renewable energy
H	Heating	SFH	Single-family house
HD	Heating demand	SC	Solar collector
HP	Heat pump	ST	Solar thermal

Symbol	Description	Unit	Symbol	Description	Unit
A	area	m²	Q	Heat	kWh
Cp	Specific heat	kJ/(kg K)	SPF	Seasonal Performance Factor	-
COP	Coefficient of Performance	-	T	Absolute Temperature	K
D	Compressor Displacement	m³	U	Heat transfer coefficient	W/(m² K)
f_{DSH}	Share of DSH to DHW preparation	-	η_C	Carnot performance factor	-
h	Specific enthalpy of refrigerant	kJ/kg	η_{vol}	Volumetric efficiency	-
$\dot{m}$	Mass flow	Kg/s	η_{is}	Isentropic efficiency	-
N_{min}	Rotational speed of compressor	min^{-1}	ρ_{suc}	Refrigerant vapour density	kg/m³
P	Pressure	Pa	ϑ	Temperature	°C
P	Power	W	τ	Compression Ratio	-

2. Case Study

In Innsbruck Vögelebichl, two multi-family buildings (MFB) were designed to achieve a net Zero Energy (NZE) balance for heating, domestic hot water (DHW) preparation and auxiliary energies. The roof of one of the MFB is covered by PV, the roof of the other MFB is partly used for PV and partly for solar thermal (ST). A double-stage ground water (GW) heat pump with hot gas de-superheating provides the remaining energy for heating and DHW.

2.1. Buildings

The two buildings with together 26 flats and together 2149 m² of treated area (see Fig. 1) were already described in detail in [10, 11, 12, 13]. The buildings were designed to reach Passive House Standard (Heating Demand HD ≤ 15 kWh/(m² a)). One roof of the MFB is covered by PV (99.8 m²), the other is partly used for PV (52.5 m²) and partly for ST (73.6 m²).

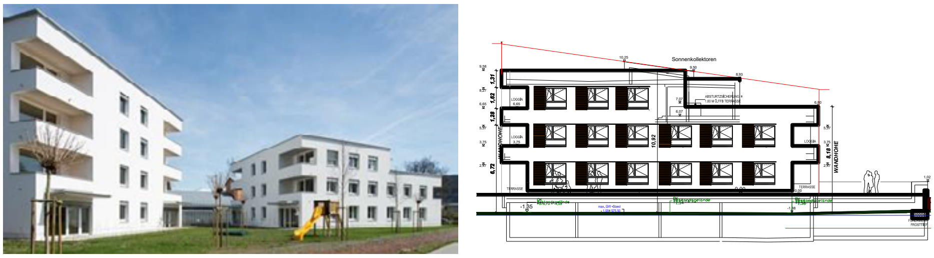


Fig. 1. (left) two multi-family buildings in Passive House standard (NHT, Innsbruck Vögelebichl), (right) west view on south building with 10 flats

2.2. Heat Pump System and Hydraulic Scheme

A ground water (GW) heat pump (HP) (with two stages, i.e. two parallel compressors) and solar thermal collector (SC) field charge the buffer store which provides heat with a low temperature heat distribution and the separate decentral fresh water preparation (DHW plate HX) to the buildings, see Fig. 2. The double-stage HP is equipped with hot gas (HG) de-superheating. Depending on the operation mode (heating or DHW preparation), the flow of the HP enters the buffer store (BS) at the top or at 1/3 of the height from the top. The combined return of the heating and DHW loop enters the large 6 m³ buffer store depending on the temperature level either at the bottom or at about 1/3 of the height of the tank in order to enhance stratification. The electric backup heater (BH) is currently not in use.

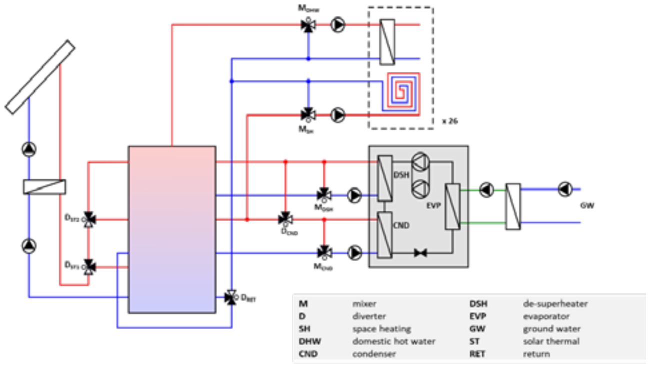


Fig. 2. Simplified hydraulic scheme with Solar Collectors (SC), Buffer Store (BS), 2-stage ground water heat pump (HP) with hot gas HG) desuperheating (DSH) in heating mode with floor heating (FH) and decentral heat exchanger (HX) for domestic hot water (DHW) preparation

2.3. Heat Pump

The heat pump has a nominal capacity of 52 kW (at W55, see Table 1). The performance map (PM) data of the HP according to the manufacturer is available only for double-stage operation and without DSH loop. PM data is available for other source temperatures, but no data is available for single-stage operation and for operation with desuperheating loop. Hence, the required performance maps have to be generated using dynamic monitoring data, which will be described shortly in the next section.

According to the EN 14511 measurements (compare table 1) there is not DSH or the DSH is in series to the condenser.

Table 1. Performance map (PM) of the double-stage HP with R410A according to data sheet (EN 14511)

(data sheet) EN 14511	W35	W45	W55
Heating Power at W10 / [kW]	58	55	52
COP at W10 / [-]	6	4.6	3.7
El. Power at W10 / [kW]	9.7	11.8	13.9
Cooling Power at W10 / [kW]	49	43	38

2.4. Double stage heat pump

Double-stage HPs are on the market already for several years. Double-stage HPs feature higher investment costs but have potentially lower operation costs. A double-stage compressor has a low capacity, which is 50% or about 70% in most models, and high capacity (100%). Variable-speed or modulating compressors vary capacity from about 40% to 100% in increments of less than 1% and are the most efficient. They have the highest advantage if either the source (e.g. air) and or the sink have variable temperature or power level (e.g. for heating).

With a single-stage HP, the compressor runs only at full load capacity, i.e. once the required temperature level is satisfied, the compressor turns off. A double-stage compressor can operate at part load or full load capacity. The two-stage HP compressors turn on and run until stage two of the thermostat is satisfied, then one compressor continues to run at a lower level, stage one. If the HP is connected directly to the heating system, this would allow to avoid swings, i.e. for a more even room temperature.

Potentially, the double-stage HP consumes less energy and does not have to work as hard to adjust to swings in room temperature. This could result in a lower operating cost while keeping occupants at the same comfort level. In [14] it was found that such a coupling process improved energy efficiency ratio by 20 % compared to a purely air source HP.

2.5. Heat Pump with Desuperheater

One possibility to improve the performance of a HP in residential applications is a desuperheater (DSH). The additional heat exchanger between the compressor and the condenser of the HP transfers the heat of the superheated refrigerant to a secondary circuit. It is suited for DHW preparation (directly or indirectly via a storage tank) because of the relative high temperature level that can be reached (e.g. 55 °C to 60 °C). The condensation of the de-superheated refrigerant takes place in the condenser of the HP at lower temperature level, e.g. at 35 °C in case of floor heating system and the heat is transferred to the heating system directly or via a separate buffer storage tank.

An energetic advantage is obtained as a part of the heat for DHW preparation that has to be provided at high temperature with usually relative low performance in "DHW preparation mode" is provided at higher efficiency at low condensing temperature in "heating mode". During this simultaneous operation (heating and DHW preparation), the HP operates at a lower condensing pressure compared to DHW preparation only. The amount of high temperature heat that can be provided and that reduces operation hours at low efficiency depends on the heating load and heating demand of the building (load duration curve). An extensive investigation for a single family home can be found in [15] and in [16]. In [13] the theoretical upper limit of performance increase by use of a DSH was shown as a function of the ratio of heating and DHW demand.

2.6. Buffer Tank

A scheme of the buffer tank is shown in Fig. 3 with inlets and outlets for (left) the heat generation part and (right) for the load. The dots indicate the position of the temperature sensors. The tank was modelled with the storage model of the Carnot Blockset in Matlab/Simulink.

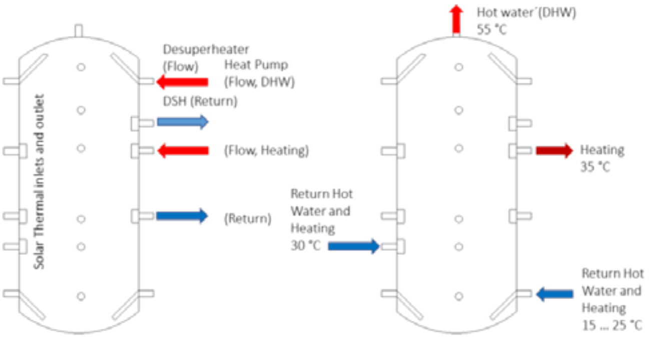


Fig. 3. Hot water tank (6 m³) with (left) heat generation (ST, HP DHW, HP Heating and desuperheating and (right) outlets for hot water preparation and heating and common return inlets

2.7. Hydraulic integration of desuperheater

The sizing and integration of the desuperheater is a nontrivial problem. Generally, the desuperheater can be connected in a parallel loop, as currently implemented in this system (Fig. 4), or it can be connected with a controlled diverter valve and can be operated in mixed parallel/series operation (Fig. 5). In parallel operation, either speed controlled circulation pumps or flow temperature control mixing loops are required to achieve the appropriate flow temperature. The latter is implemented in this system.

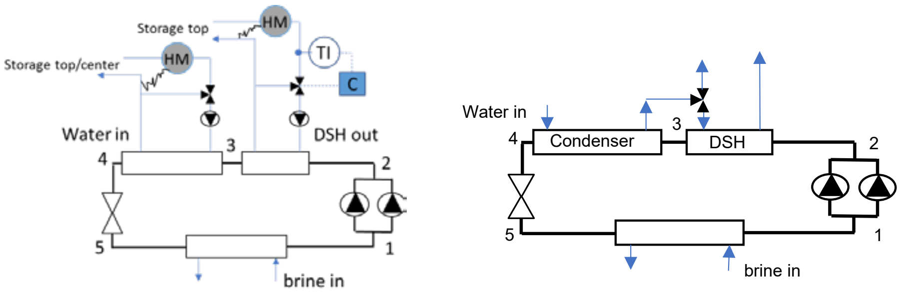


Fig. 4. (a) Parallel operation of condenser and desuperheater with flow temperature control of condenser and DSH with heat meter (HM), temperature indicator (TI) and controller (C) (TI and C of the condenser loop are not shown for better visibility); (b) Flow temperature control of condenser and DSH (mixed parallel / series operation)

2.8. Control

Theoretically, only for the heat pump eight different operation modes exist as shown in the following figure. The current implementation of the system does not allow operating the desuperheater in series, a slight change of the hydraulic scheme would be necessary. Currently, the HP is operated either in DHW mode (if the top third of the storage tank is below 48 °C, set point temperature of storage top temperature = 52 °C) in stage II or in heating mode in stage I or II depending on the outdoor temperature. An improved system concept is shown in the discussion section.

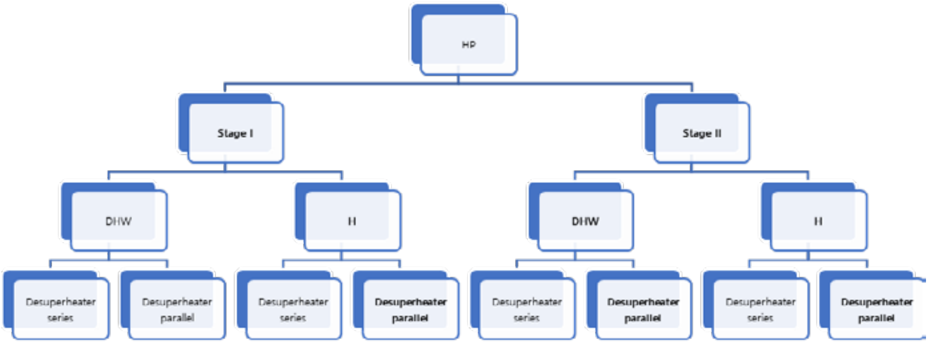


Fig. 5. Theoretical control options of a double stage heat pump with desuperheater and current control implementation in bold (stage I, and stage II heating mode, stage II DHW mode; DSH always in parallel operation)

2.9. Monitoring System

Detailed monitoring of the buildings (all flats in one of the two MFB), the HVAC system including thermal losses and the performance of PV and ST allowed to study the behavior of NZEBs in practice. The simulation-assisted analysis of the monitoring data allowed to identify problems and faults of the installation and of the operation. By means of simulation, influences of climate and of the user behavior (set points, etc.) could be considered and the monitoring results could be normalized to standard conditions.
The measurement system consists of Pt100 (1/3 DIN), heat meters and electricity meters. The measurement system is connected to the BMS and readings are stored with a 1 min. time step. Data is post-processed and average or integrated values are reported.

Detailed monitoring results were collected for a period of three years, analyzed by means of dynamic simulation and the performance of the buildings was evaluated with respect to the aim of achieving the NZE balance. The dynamic building and HVAC simulation was performed with Matlab/Simulink.

3. Research Questions

The following research questions can be defined:

- Does a double-stage outperform a single-stage HP?
- Is there a cost benefit and is there a benefit with respect to service life (i.e. can excessive on/off cycles be avoided)?
- What is the optimum control strategy for a double stage HP?
    - for heating and for DHW preparation
    - in combination with a buffer storage
- What is the theoretical and real benefit of the DSH?
- Does a 2 + 2-pipe system outperform the 3 pipe system (with common return of heating and DHW loop)?
- By how much can the storage stratification and the system performance be improved by variable input heights of the return flow to the tank?

The particular question in this project is whether the performance of a HP can be improved in these MFBs with very high energy standard (PH Standard) with relative low heating demand (15 kWh/(m² a) and relative high DHW demand (ca. 25 kWh/(m² a)) and by how much? And can the goal of NZEB be achieved?

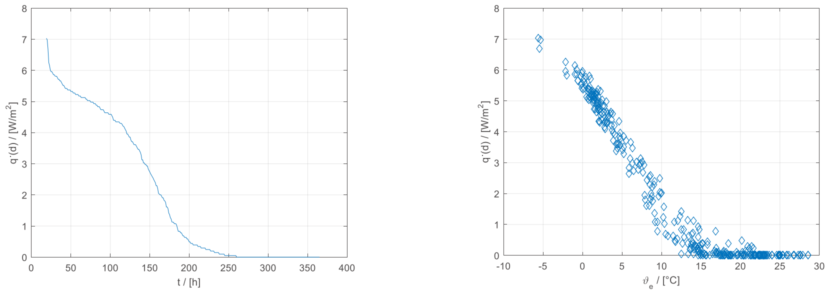
4. Method

Firstly, based on the analysis of monitoring results, conclusions on the actual performance are derived and typical operation conditions of the heat pump are identified. Secondly, refrigerant cycle simulations are performed in order to identify the performance map for the different operation modes. Finally, dynamic simulations are performed in order to detect faults in the current system and to identify possible improvements with respect to the control strategy and set points but also regarding improved system design.

5. Monitoring Results

5.1. Heating Load and DHW demand

The heating load is 24 kW for both buildings according to the design (PHPP, max. daily average). DHW load adds approx. further 12 kW with a buffer tank acc. to the standard recommendations. For the south building detailed data of each flat is available. The maximum daily heating load is 7 W/m² (see Fig. 6). There is a clear trend of heating load vs. ambient temperature, which is not typical for passive houses. The heating capacity of the heat pump in stage I is always sufficient to cover the heating load of the building. However, in stage II higher share of desuperheater can be expected.



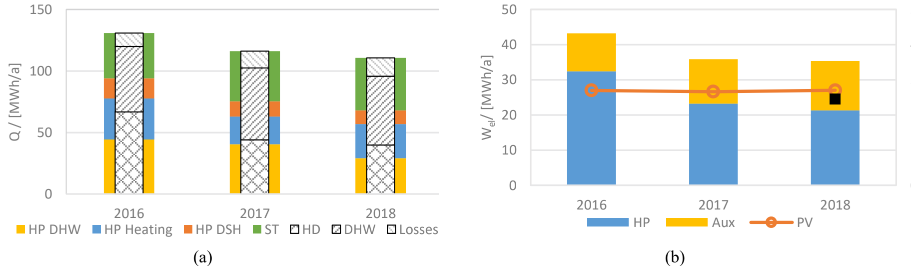
(a) (b)

Fig. 6. (a) Load duration curve of the south building and (b) daily heating load vs. ambient temperature

Domestic hot water demand is not monitored in detail. The total delivered DHW from the storage is measured and includes distribution losses. In 2018 it was 26 kWh/(m² a) and in the range of the predictions. The flow temperature to the decentral fresh water heat exchangers is in the range of 52 °C and always sufficient to provide hot water with sufficient comfort.

5.2. Electricity Energy Demand and Net Zero Energy Balance

In 2018, the electricity consumption of the HP was 21 MWh and auxiliary energies were responsible for 14 MWh. The PV field produced in average around 27 MWh. Fig. 7 reports the electric energy balance for the years 2016 to 2018. In spite of the reduced total energy demand in 2018 (mainly because of the decreased heating demand), the NZE balance was not achieved. (Remark: Actually, the PV production was 24.5 MWh in 2018 because of a technical failure of the inverter of the north building PV field. A yield between 26.6 MWh and 27.2 MWh can be predicted based on data of 2016 and 2017 for the case without the failure).



(a) (b)

Fig. 7. Thermal (a) and electric (b) balance with HP, auxiliary energies (aux) and photovoltaic (PV)

5.3. HP Operation

A series of typical HP operation modes is shown in Fig. 8. The HP operates with full capacity (stage II) in DHW mode. Also the DSH delivers around 11 kW. The condenser power is 28 kW. With the electric power of the compressor of 14 kW, the COP is 2.8. In heating mode (with single compressor operation) the heating power is 25 kW. No useful desuperheating can be measured. The COP under these conditions is 6.25 (4 kW of electric power). There is one short period with heating mode at stage II with a small contribution by the DSH. It has to be mentioned here, that the heat meters measure the heat that is delivered to the storage tank. In order to deliver the energy at the required temperature level, flow temperature control is implemented, as shown in Fig. 4, above. Furthermore, it is important to remark that the presented operation of the HP does not represent the optimum configuration but is the one implemented and is used to identify the HP operation and generate the PMs.

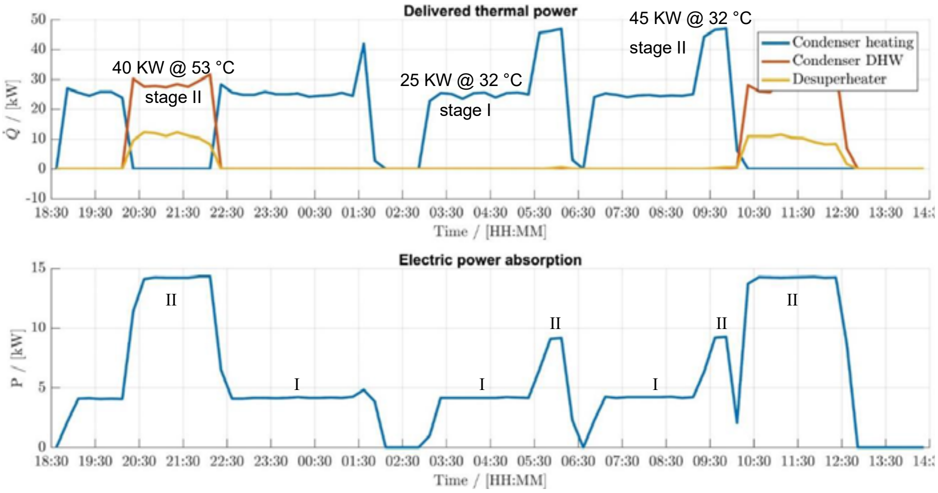


Fig. 8. Heat Pump operation with different operation modes (stage I and stage II, DHW and Heating with and without DSH)

6. Component and System Modelling

6.1. Refrigerant Cycle

A simplified scheme of the refrigerant cycle of a double-stage HP with DSH is shown in Fig. 5. In order to predict the performance of the HP under various conditions, a simplified physical vapour cycle model was developed in Matlab using the CoolProp database for refrigerants [17]. Pressure losses in and between the components are disregarded with the exception of the pressure loss in the condenser, which can be considered by a given pressure difference Δp_{cond}. Subcooling and superheating are considered in a simplified way with given temperature differences ΔT_{sub} and ΔT_{super}, respectively.

The evaporator is divided in three sections (film evaporation, evaporation and superheating). The heat transfer coefficients are assumed to be constant in each section. The following equations are applied.

$$\dot{Q}_{evap} = \dot{m}_b \cdot \Delta h_b = \dot{m}_{ref} \cdot (h_1 - h_5) \tag{6}$$

$$\dot{Q}_{evap} = \sum UA_{evap,i} \cdot \Delta\vartheta_{log,evap,i} \tag{7}$$

Correspondingly, the condenser is divided in desuperheating, condensation and sub-cooling sections with constant (i.e. given) heat transfer coefficients.

$$\dot{Q}_{cond} = \dot{m}_w \cdot c_w \cdot \Delta T_w = \dot{m}_{ref} \cdot (h_3 - h_4) \tag{8}$$

$$\dot{Q}_{cond} = \sum UA_{cond,i} \cdot \Delta\vartheta_{log,cond,i} \tag{9}$$

Then for the DSH it applies that

$$\dot{Q}_{DSH} = \dot{m}_{w,DSH} \cdot c_{p,w} \cdot \Delta T_{w,DSH} = \dot{m}_{ref} \cdot (h_2 - h_3) \tag{10}$$

$$\dot{Q}_{DSH} = UA_{DSH} \cdot \Delta\vartheta_{log,DSH} \tag{11}$$

The mass flow of the refrigerant

$$\dot{m}_{ref} = D \frac{N_{min}}{60\ \mathrm{s/min}} \cdot \rho_{suc} \cdot \eta_{vol} \tag{12}$$

is a function of the displacement and of the volumetric efficiency. The compressor power in the refrigerant cycle is calculated based on the isentropic efficiency η_{is}.

$$P_{comp} = \dot{m}_{ref} \cdot \frac{h_{2s} - h_1}{\eta_{is}} \tag{13}$$

where $h_{2s} = h(p_{dis}, s_1)$ is the enthalpy of superheated gas at isentropic compression. The isentropic efficiency η_{is} depends on the compression ratio, which is a function of the compressor discharge p_{dis} and suction p_{suc} pressure.

$$\tau = \frac{p_{dis}}{p_{suc}} \tag{14}$$

Both, the volumetric and the isentropic efficiency can be approximated based on literature data by a polynomial or a 2D-lookup table.

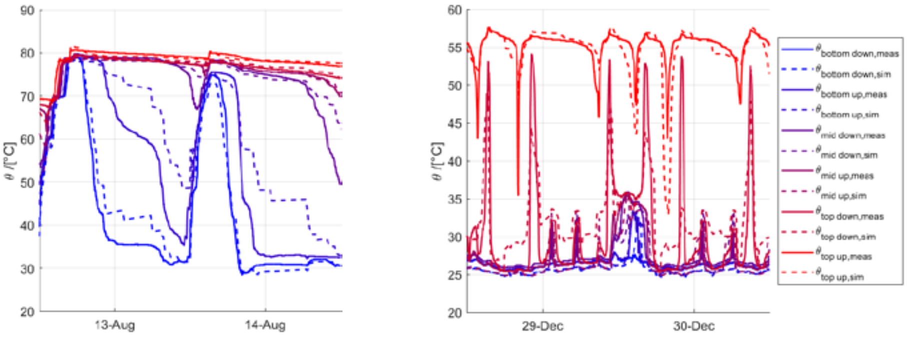
The electrical efficiency of the compressor η_{el} is approximated with a linear function depending on the compressor temperature. Then the performance of the HP can be calculated as follows

$$COP_{HP} = \frac{\dot{Q}_{cond} + \dot{Q}_{DSH}}{P_{el,comp}} \tag{15}$$

7. Results

7.1. Buffer Storage

The measured and simulated storage temperatures for a period in winter and in summer (with ST operation) are shown in Fig. 9. Overall, the agreement is acceptable. In detail, there are some deviations in the dynamic behavior during some days for some sensors, but with respect to the research question discussed in this paper, the deviations are acceptable.



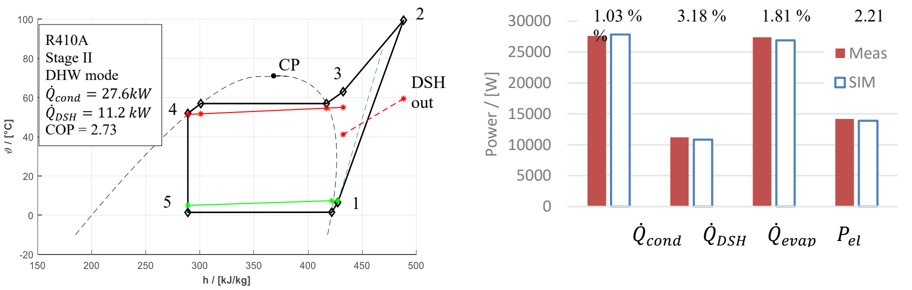
(a) (b)

Fig. 9. Measured (meas) and simulated (sim) storage temperatures for a period in summer (a) and in winter 2018 (b)

The measured thermal losses are within an average factor of about 10 times higher than the predicted ones (for a tank of that size and with that insulation level, corresponding to a ErP class C tank, EU efficiency classes for storage losses). One main reason is that losses from pipes, valves and pumps between the tank and the heat meters are included in the energy balance. Unwanted buoyancy flows could be excluded by analyzing the monitoring data for all hydraulic loops except for the cooling loop (in summer the ground water can be used with a heat exchanger to do some cooling with the floor heating system.) This is still under investigation.

7.2. HP Model Parameterization

The simplified physical HP model, described above, was parameterized using measured data from monitoring. The source temperature showed little variation between around 8 °C and 12 °C over the course of the year. Two typical operation modes were identified: DHW operation with desuperheating at double-stage and heating mode at single-stage without significant desuperheating. The resulting refrigerant cycle is shown on the following Fig. The corresponding heating power (condenser, DSH and evaporator) and electric power (compressor) are shown in Fig. 10. The deviations are highest for the DSH with 3.2 %.



(a) (b)

Fig. 10. (a) Refrigerant Cycle at DHW operation mode with DSH, stage II; (b) Comparison of measured and simulated heating powers and electric powers and relative deviations (remark: the presented operation of the HP does not represent the optimum configuration)

7.3. PM of Heat Pump Model DSH

With the parameterized HP model, performance maps for the different operation modes were generated. These include

- Stage I or stage II
- DSH in series (i.e. without separate DSH loop) or separate DSH loop
- DSH inlet temperature

$$\dot{Q}_{cond}, \dot{Q}_{DSH}, P_{el} = \mathrm{f}(\vartheta_{src}, \vartheta_{snk,}\vartheta_{DSH}, stage, DSH) \qquad (16)$$

Fig. 11 (left) reports the heating power of the HP. The uncertainty of the model is indicated by the error bars and is in the range of 6 kW. Fig. 11 (right) shows the corresponding COP. It is significantly lower than the COP according to the data sheet. One reason for lower performance is the lower volume flows on the source and sink side compared to the measurement acc. to the standards. Furthermore, a slight degradation of the heating capacity of the HP could be identified within the first three years of operation, which could be explained by some refrigerant leakage. Further detailed investigations will be carried out to identify the reason for the deviation of the measured data and the data sheet.

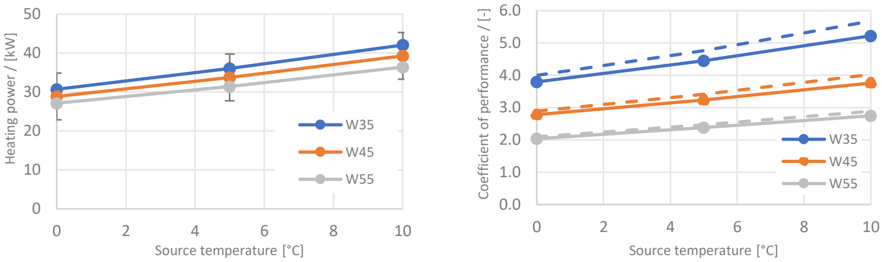


Fig. 11. Heating power for stage 2 (left) and COP for stage I and stage II (right) for the configuration with DSH in series to the condenser as in Fig. 5 (series configuration)

7.4. Dynamic Heat Pump Model DSH

The comparison of measured and simulated thermal power in condenser and desuperheater as well as electric power of the compressor(s) (bottom) is shown in Fig. 12 for a typical day in winter. The agreement is overall acceptable for all the three operation modes (heating stage I and II, DHW preparation stage II).

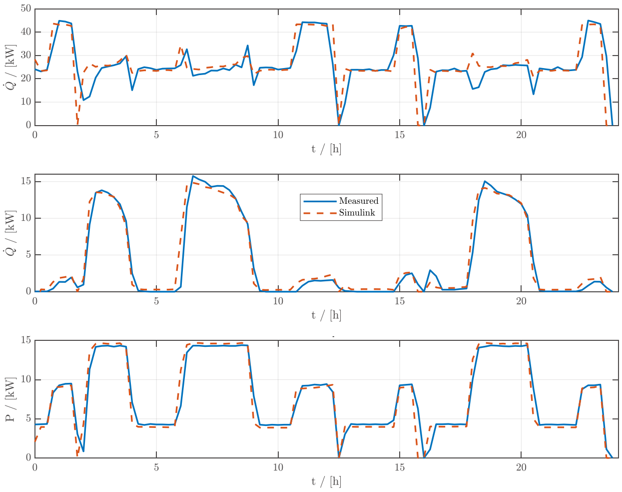


Fig. 12. Comparison of measured and simulated thermal power in condenser (top) and desuperheater (center) as well as electric power of the compressor(s) (bottom) for a typical day in winter

8. Discussion

The simulation assisted analysis of the monitoring results of the two buildings and of the HVAC system (including distribution losses, heat pump performance factors, auxiliary electricity demand, solar thermal and PV yield) allows to identify and solve problems that frequently occur during commissioning of the system. With the help of the simulation study, the control of the system can be optimized in order to reach the anticipated performance.

The net zero energy balance (for heating and DHW and auxiliary energies) could not be achieved during the first three years of monitoring. The implementation of first optimization measures shows a reduction of the demand. The performance of the HP is significantly lower than expected based on data sheet PM. There is a potential improvement of the performance by using the DSH loop, however, due to the high COP in heating mode in stage I that mode should be preferred. the following control scheme is recommended see Fig. 13. Optimal switching between single and double stage operation in heating mode is subject of current investigations.

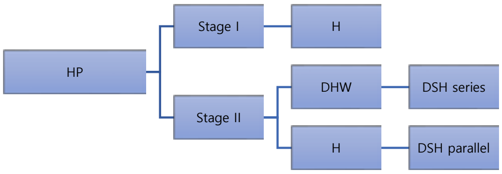


Fig. 13. Recommended operation modes: stage II in winter, stage I in intermediate season; (DSH in series in heating mode stage I)

Predictions based on results after introduction of improvements show that Net Zero could be achieved, see Fig. 14. In future projects more care should be taken in the appropriate dimensioning of the components (in particular heat pump capacity and storage volume) Passive House (PH) standard is key for achieving NZEB level for heating, DHW and aux. energies. If appliances was included in the energy balance, additional PV on the south façade would be required.

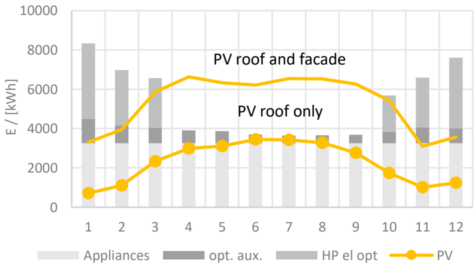


Fig. 14. Predicted monthly electric energy balance after improvements with appliances and with additional PV on the façade

A Simulink model of the HP was developed. The multi-dimensional PM model allows to simulate different HP operation strategies with and without DSH as well as single and double-stage operation. The optimization of both, the control strategy and of the components, is performed with the aim to operate the HVAC system with improved performance. Predictions show that the NZE can be achieved with improved control strategy and when the auxiliary energies are reduced.

9. Conclusions

Two multi-family buildings (MFB) with together 26 flats were designed to achieve a net Zero Energy (NZE) balance for heating, domestic hot water (DHW) preparation and all auxiliary energies. The roof of one of the MFB is covered by PV, the other is partly used for PV and partly for solar thermal (ST). A double-stage ground water heat pump (HP) with hot gas desuperheating (DSH) provides the remaining energy for heating and DHW. The system was designed with a low temperature distribution system (floor heating) and separate decentral fresh water preparation (DHW plate heat exchanger) for low thermal losses and optimal efficiency.
Detailed monitoring of the buildings, the HVAC system including thermal losses and the performance of the

HP, PV and ST allowed to study the behavior of NZEBs in practice. The simulation-assisted analysis of the monitoring data allowed to identify problems and faults of the installation and of the operation (i.e. control). Detailed performance maps of the double-stage HP pump with DSH were generated analyzing the monitoring data and by means of refrigerant cycle simulation. Simulation based optimization was performed with the aim to operate the HVAC system with improved performance. Different system configurations (i.e. with or without ST, with different buffer storage sizes, with different control strategies and set points) can be compared and evaluated also considering (seasonal) variations of the share of renewables (RE) in the electricity mix. Recommendations for the design and for the operation of the HP system (including control optimization) for MFB and blocks of buildings can be derived from the simulation results. It is evident that basic monitoring of the performance of the HVAC system is required to detect faults and to be able to guarantee efficient operation (quality control).

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