Simulation-assisted Optimization of the HVAC system of NZE Multi-family Buildings

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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 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 a hot gas desuperheater (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 HX) for optimal efficiency. By means of monitoring over a period of three years and by simulation assisted analysis of the main components, the buffer tank, the double -stage HP with DSH and the system, the performance is improved to achieve the goal of NZE.

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 BPIE 2015. 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 and in particular nZE multi-family buildings. Miara et al. 2014 and 2017 show that the performance, i.e. the seasonal performance factor (SPF), of installed HP systems improved within the recent years. 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. 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.

Case study: NZEB NHT Vögelebichl

In Innsbruck Vögelebichl (see Figure 1), two multi-family buildings (MFB) with together 26 flats and together 2149 m² of treated area 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.

Figure 1: Two multi-family buildings in Passive House standard (NHT, Innsbruck Vögelebichl).

The NZEB project NHT Vögelebichl was already described in Ochs et al. 2014 and Ochs et al. 2018. Here only the most relevant aspects are given that are required as basis for the optimization that will be discussed in the following sections.

Hydraulic Scheme

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). A GW heat pump (with two stages, i.e. 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 Figure 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.

Figure 2: 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.

Double-stage Heat Pump with Desuperheater

- 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 desuperheated 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 Hengel et al. (2016).

- Doublestage 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. Wang et al. (2005) found that such a coupling process improved energy efficiency ratio by 20 % compared to a purely air source HP.

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

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?

Method

Theoretic improvement of system efficiency with DSH

The theoretic improvement of the system efficiency can be estimated by simple calculation. The theoretic improvement arises from delivering a part of the energy needed for DHW preparation with high performance of e.g. $SPF_H = 5$ in heating mode instead of delivering it at high temperature, i.e. 55 °C and thus with low performance, e.g. with $SPF_{DHW} = 2.8$. The share of DHW that can be provided by the HP with a DSH in heating and mode is illustrated in Fig 3.

The contribution of the DSH to DHW preparation $Q_{DHW,DSH}$ depends on the boundary conditions and is a share of the heating demand Q_H as shown in eq. 1. The realistic range for the values of f_{DSH} is between 5 and 15 %. For specific conclusions dynamic simulations are obviously required.

$$Q_{DHW,DSH} = Q_H \cdot f_{DSH} \quad (1)$$

The theoretical possible savings depend on the ratio of heating demand (HD) and DHW demand, see Figure 4. Furthermore, it depends on the SPF of the HP in heating and in DHW mode. The electric consumption can be estimated using SPF for heating (eq. 2) and for DHW preparation (eq. 3). Electricity savings between the case with the DSH (eq. 5) and the reference case without DSH (eq. 4) are presented in Figure 4.

Figure 3: Schematic sorted load duration curve (base load DHW and peak load heating) with double stage HP with possible share of DHW contribution from DSH in heating mode.

The following applies:

$$SPF_H = \eta_{c,H} \cdot \frac{T_{max,H}}{T_{max,H} - T_{min}} \quad (2)$$

$$SPF_{DHW} = \eta_{c,DHW} \cdot \frac{T_{max,DHW}}{T_{max,DHW} - T_{min}} \quad (3)$$

$$W_{tot,ref} = \frac{Q_{DHW}}{SPF_{DHW}} + \frac{Q_H}{SPF_H} \quad (4)$$

$$W_{tot,DSH} = \frac{Q_{DHW} - Q_{DHW,DSH}}{SPF_{DHW}} + \frac{Q_H + Q_{DHW,DSH}}{SPF_H} \quad (5)$$

Figure 4: Theoretic reduction of electricity consumption by using a DSH depending on the ratio of domestic hot water (DHW) and heating demand (HD), corresponding to low energy house (LE) and Passive house (PH) standard for single-family house (SFH) or multi-family building (MFB) with the DHW share of desuperheater f_{DSH} (eq. 1) as parameter.

Depending on the ratio of DHW over HD and depending on the contribution of DSH to DHW preparation in heating mode, electricity savings between 1 % and 9 % can be obtained. Whether this theoretic reduction of electricity consumption by using a DSH can be achieved for a specific case, it needs to be proven by dynamic building and system simulation and depends on the COP of the HP at different operation conditions and also on the control strategy.

Heating Load and Heat Pump Capacity

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. 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 (i.e. with DSH in series to the condenser, see Figure 5). 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.

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

Refrigerant Cycle

A simplified scheme of the refrigerant cycle of a double-stage HP with DSH is shown in Figure 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 (www.coolprop.org). 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.

Figure 5: Scheme of HP with desuperheater (DSH) and two parallel compressors (double-stage HP; according to the EN 14511 measurements (compare table 1) the DSH is in series to the condenser

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.

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

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

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

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

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

Then for the DSH it applies that

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

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

The mass flow of the refrigerant

$$\dot{m}_{ref} = D \frac{N_{min}}{60 \text{ s/min}} \cdot \rho_{suc} \cdot \eta_{vol} \quad (12)$$

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} .

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

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.

$$\tau = \frac{p_{dis}}{p_{suc}} \quad (14)$$

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

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

With DSH operation, the condenser mass flow $\dot{m}_{water}$ is different (usually higher) than the mass flow through the DSH $\dot{m}_{DSH}$. Without explicit DSH operation, it is assumed that the water outlet is connected to the DSH inlet (series configuration, see Figure 5).

Results

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.

Energy Balance

Operation of the buildings started winter 2015. The HD of the buildings was with about 20.5 kWh/(m² a) in 2017 and 18.5 kWh/(m² a) in 2018 slightly higher than the planned 15 kWh/(m² a) for Passive House standard. The DHW demand was in the range of the expectations with about 25 kWh/(m² a), see Ochs et al. 2018a and Ochs et al. 2018b. A summary of the energy balance is given in Figure 6. The overall demand was decreasing in the first three years of operation. The thermal losses slightly increased in absolute value leading to an increase relative contribution of 14 % and are one of the major reasons why the NZE balance could not yet be fulfilled.

Figure 6: Energy balance with delivered energy for heating, for DHW and thermal losses (tank + pipes to heat meters) as well as contributions from solar thermal (ST) and heat pump (condenser DHW, condenser Heating and DSH).

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. Figure 7 reports the electric energy balance for the years 2016 to 2018.

Figure 7: Electric Balance with HP, auxiliary energies (Aux) and photovoltaic (PV).

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 PV field of the north building. 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).

Buffer Tank

A scheme of the buffer tank is shown in Figure 8 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 tanks was modelled with the storage model of the Carnot Blockset in Matlab/Simulink.

The measured and simulated storage temperatures for a period in winter and in summer (with ST operation) are shown in Figure 9.

Figure 8: 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.

Figure 9: Measured and simulated storage temperatures for a period in winter (a) and in summer (b).

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.

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.

Heat pump operation

A series of typical HP operation modes is shown in Figure 10. In DHW mode the HP operates with full capacity (stage II). 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.

Figure 10: Heat Pump operation with different operation modes (stage I and stage II, DHW and Heating with and without 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 Figure 11.

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.

Figure 11: 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).

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 Figure

Figure 12: Refrigerant Cycle at DHW operation mode with DSH, stage II.

The corresponding heating power (condenser, DSH and evaporator) and electric power (compressor) are shown in Figure 13. The deviations are highest for the DSH with 3.2 %, see Figure 13.

Figure 13: Comparison of measured and simulated heating powers and electric powers and relative deviations.

The heating power of the HP (condenser and DSH in series) is significantly lower than the numbers given in the data sheet. One of the reasons might be insufficient refrigerant fill and furthermore, different, i.e. lower mass flows compared to the measurements acc. to the standard (EN 14511).

Performance Maps (PM) of double-stage heat pump with desuperheater (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

$$\dot{Q}_{cond}, \dot{Q}_{DSH}, P_{el} = f(\vartheta_{src}, \vartheta_{snk}, \vartheta_{DSH}, stage, DSH) \quad (18)$$

Figure 14 (a) 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. Figure 14 (b) 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.

Figure 14: Heating power for stage 2 (a) and COP for stage I and stage II (b) for the configuration with DSH in series to the condenser as in Figure 5.

The theoretical maximum share of the DSH with respect to the total heating power of the HP is shown in Figure 5 for different operation conditions taking stage I as an example. For stage II, the trend is the same with a share

being slightly higher (about 1.2 % higher for W35 and 0.2 to 0.6 % higher for W55). The share is highest with low source temperatures and high sink temperatures. For a realistic operation range of W35, a DSH inlet temperature of 50 °C and source temperatures around 8 °C, the DSH power is about 8 % of the total power.

Figure 15: Influence of sink and source temperature as well as stage on the share of the DSH.

Modelling and Simulation of the HVAC System

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. Figure 14 gives the resulting electric energy balance. Net energy balance for heating, DHW and auxiliary energy is achieved with PV on the roof. PV on the façade is required, if appliances are considered, too. The contribution to the winter load is insignificant.

Figure 16: Predicted monthly electric energy balance after improvements with and without appliances and with additional PV on the façade.

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 Figure 17.

Figure 17: Recommended operation modes: stage II in winter, stage I in intermediate season; DSH off means desuperheater in series to the condenser, on means parallel operation in heating mode.

Predictions based on results after introduction of improvements show that Net Zero could be achieved. 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 electricity consumption of appliances was included in the energy balance, additional PV on the south façade would be required (see Figure 16). NZEB do not significantly reduce winter grid load. The mismatch between (electricity) demand and PV yield has to be considered, e.g. by means of different electricity prices for purchase/sell or by monthly primary energy (PE) conversion factors.

Conclusions

The challenge to make the building sector climate-neutral was already tackled with the project Innsbruck Vögelebichl. The demo project involves two multi-storey multi-family buildings with a total of 26 apartments in Passive House standard. The aim of the project was to achieve the net zero energy balance for heating and hot water preparation and the necessary auxiliary energies. An energy and cost efficient central heat supply based on a low temperature heat distribution and a solar HP system coupled with PV was investigated and optimized by means of building and HVAC simulation.

Recommendations for the design and dimensioning for the HVAC system and for the operation of the system (including control optimization) can be derived from the simulation results. Basic monitoring of the performance of the HVAC system is required to detect faults and to be able to guarantee efficient operation (quality control).

As a next step, by means of simulation, different system configurations (i.e. with or without ST, with different buffer storage sizes, with different control strategies) can

be compared and evaluated considering seasonal variations of the share of RE in the electricity mix. Design recommendations will be elaborated.

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Nomenclature

Abbreviation	Description	
BMS	Building management system	
BS	Buffer Storage	
CW	Cold water	
comp	Compressor	
cond	Condenser	
DHW	Domestic hot water	
DSH	Desuperheater	
el	Electric	
evap	Evaporator	
FH	Floor heating	
GW	Ground water	
H	Heating	
HD	Heating demand	
HP	Heat pump	
HW	Hot water	
HX	Heat exchanger	
HVAC	Heating ventilation air conditioning	
LEH	Low energy house	
MFB	Multi-family building	
NZE(B)	Net Zero energy (building)	
nZEB	Nearly Zero energy building	
PH	Passive house	
PM	Performance map	
PV	Photovoltaic	
RE	Renewable energy	
SFH	Single-family house	
SC	Solar collector	
ST	Solar thermal	
Symbol	Description	Unit
A	area	m ²
cp	Specific heat	kJ/(kg K)
COP	Coefficient of Performance	-
D	Compressor Displacement	m ³
f _{DSH}	Share of DSH to DHW preparation	-
h	Specific enthalpy of refrigerant	kJ/kg
$\dot{m}$	Mass flow	Kg/s
N_{min}	Rotational speed of compressor	min ⁻¹
p	Pressure	Pa
P	Power	W
Q	Heat	kWh
SPF	Seasonal Performance Factor	-
T	Absolute Temperature	K
U	Heat transfer coefficient	W/(m ² K)
η_C	Carnot performance factor	-
η_{vol}	Volumetric efficiency	-
η_{is}	Isentropic efficiency	-
ρ_{suc}	Refrigerant vapour density	kg/m ³
ϑ	Temperature	°C
τ	Compression Ratio	-

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