EXPERIMENTAL AND ANALYTICAL EVALUATION OF EXHAUST AIR HEAT PUMPS IN AIR BASED HEATING SYSTEMS

Arsalan Shirani[1*], Alexander Merzkirch[2], Jennifer Roesler[2], Stephan Leyer[1], Frank Scholzen[1], Stefan Maas[1]

[1] *Université du Luxembourg, 6, rue Richard Coudenhove-Kalergi, L-1359 Luxembourg*
[2] *Bosch Thermotechnology, Junkersstraße 20, 73249 Wernau (Neckar), Germany*

*Corresponding author: *E-mail: Arsalan.Shirani.001@student.uni.lu, Phone: +49 7153 306-2229*

ABSTRACT

Due to the significant energy loss in the building sector, energy efficiency of the buildings and their heating systems is gaining interest especially in the last decades. Application of exhaust air heat pumps as an energy efficient heating technology to cover the heating and venting demand of the building is becoming conventional. The exhaust air heat pumps are attracting considerable interest due to their normally reasonable price and small footprint. This heat pump type is often installed as part of an air based heating system together with electrical heaters. These electrical heaters are installed to cover the residual heating demand and/or to insure thermal comfort. The technology of exhaust air heat pumps has been investigated in several studies. Available studies on exhaust air heat pumps have tended mostly to focus on comparison between different technologies utilized in efficient buildings. Almost all of the existing studies have ignored the actual application of this technology in air based heating systems together with electrical heaters. In this sense, there is still a need for discussion on the influence of different buildings' standards and control strategies on the heating system performance. The present paper aims to call into question the energy efficiency of exhaust air heat pumps under different boundary conditions. In this study, the results of a long time field monitoring are utilized in order to model the dynamic behavior of an exhaust air heat pump in MATLAB/Simulink. The impact of different boundary conditions on the heat pump's efficiency is studied and additionally compared to previous studies. The developed model is used for annual simulations of air based hybrid heating systems in three different building standards. Finally, the influence of control strategy on the system performance is investigated. It is shown that under certain boundary conditions and control strategy, the studied system could provide an efficient heating system with acceptable user comfort. Nevertheless, applying conventional heating control methods or installing the system in buildings with high heating energy demand could lead to high electrical energy consumption and/or undesirable thermal comfort.

ABBREVIATION

Aux.	Auxiliary
Biv.	Bivalent
EHA1	Exhaust Air condition 1 (after air-to-air heat exchanger)
EHA2	Exhaust Air condition 2 (after evaporator)
ETA	Extract Air from building
Evap.	Evaporator
COP	Coefficient Of Performance
HEX	Heat Exchanger
HP	Heat Pump
HRV	Heat Recovery Ventilation
NTR	Nighttime reduction
ODA	Outdoor Air
SCOP	Seasonal Coefficient Of Performance
Surf.	Surface
SUP1	Supply Air condition 1 (after air-to-air heat exchanger)
SUP2	Supply Air condition 2 (after condenser)
Temp.	Temperature

INTRODUCTION

The loss of energy in the building sector accounts for approximately 40% of overall European final energy consumption **[1]**. Moreover, in most developed countries buildings are responsible for more than 30 % of greenhouse gas emissions **[2]**.

Firstly, the improvements of energy efficiency in heating systems is gaining importance. For example, one of the key elements in improvement of efficient heating systems is the use of the heat pump technology. If the heat pump works in the right condition, it could offer a sustainable solution for both heating and cooling systems in the buildings **[3]**. A study on the potential savings in energy consumption using heat pump systems in residential buildings has shown, a reduction of 40% in CO2 equivalent emissions is possible in case 50% of new buildings and 30% of renovated buildings are equipped with heat pumps until 2030 **[4]**. In **[5]** is shown that use of HPs in residential buildings reduces the CO2 emissions to more than 50% compared to conventional boiler systems.

Secondly, efficient and airtight buildings are often equipped with Heat Recovery Ventilation (HRV) systems in order to reduce the ventilation heat losses and to ensure high indoor air quality **[6]**. Due to rising construction costs a possible approach to decrease investment costs is to distribute the heat via air instead of the water-based distribution system such as floor heating or radiators. There are several available publications, related to utilize the air-based heating systems in efficient buildings.

In **[7]** it was shown that a passive house could be heated by means of hygienically required minimum air change rates. Exhaust air heat pumps (EHA-HPs) are well known as one of the main heating concepts of passive houses. Since under winter conditions the enthalpy of the exhaust air leaving the HRV unit is higher than the enthalpy of the outdoor air (higher moisture contents and temperature) this could be used in EHA-HPs as a source for a heat pump **[8]**.

There are several studies dealing with the topic of exhaust air heat pump heating systems. An extensive theoretical and experimental investigation about different EHA-HPs was done in a dissertation by Buehring **[9]**. According to **[9]** such system could guarantee the necessary heat supply with a sufficient efficiency by a wide range of user behaviors in passive houses. In **[1]** four different heating system variants, including an EHA-HP, were compared with each other through dynamic simulations. The EHA-HP is introduced as the "more favorable" system in colder climates for buildings with low heating demand due to its significant potential of heat recovery.

A comprehensive practical comparison of EHA-HPs is done by **[10]** in Switzerland. In this practical study different energy and comfort measurements were taken place in eight apartments in an apartment building with MINERGIE-P[1] standard for over a year. In this study, only in four of eight apartments the measured heating energy demand was below the maximum allowed (based on MINERGIE-P requirements).

The heating power of air-sourced heat pumps reduces with reduction of outdoor air temperature. In other words, usually in very cold outdoor air conditions (when the building's heating demand is at maximum) the heating power of the heat pump is at its lowest. It would be typically avoided to design the heat pump, covering the maximum heating load, mainly due to the cycling loses which would reduce the heat pump performance. For this reason, normally the air-sourced heat pumps are dimensioned to cover only a part of the building heating demand and the rest should be covered via a second heating source **[11]**. Moreover, due to lower temperature of extract air zones in the buildings, which are equipped with an air based heating system, the heating demand of extract air zones (especially bathroom) should be covered with a second heating source. As a result of the above mentioned, the EHA HPs would be normally designed as part of a hybrid heating system, which usually contains electrical heaters for covering the residual heating demand of the building.

Although there are several studies about the technology of EHA-HPs available, two interesting edges of this technology are not well considered:

1. Effect of different boundary conditions on the EHA-HP performance

The number of publications dealing with the impact of exhaust air temperature and humidity on the EHA HP performance is few. Since the source air volume rate of EHA HP is limited, the role of the source air enthalpy is more significant in this type of air sourced heat pumps. Besides, increasing the exhaust air humidity leads to higher number of needed defrost cycles. With this in mind, it is important that the impact

[1] A building standard in Switzerland; heating and hot water specific primary energy demand must be under 30 kWh/m^{2}a

of air humidity on the EHA-HP performance be investigated along with influence of the defrost cycles. This significant correlation has not been dealt with in depth yet and would be one of this work's focus areas.

2. Effect of building standard and control strategy on the performance of a hybrid heating system, containing an EHA-HP and electrical heaters as back up

The EHA HPs are widely used in efficient residential buildings as an efficient system for covering the heating and venting demand of the building. As it mentioned before, this type of heat pump is generally utilized in air based hybrid heating systems together with electrical heaters. The system control strategy plays a significant role in the hybrid heating system efficiency. However, previous works have only focused on the comparison of EHA HPs with other possible technologies for efficient buildings and ignored the common combination of EHA HPs with electrical heaters and the heating system control. This would be the second focus area of the present paper.

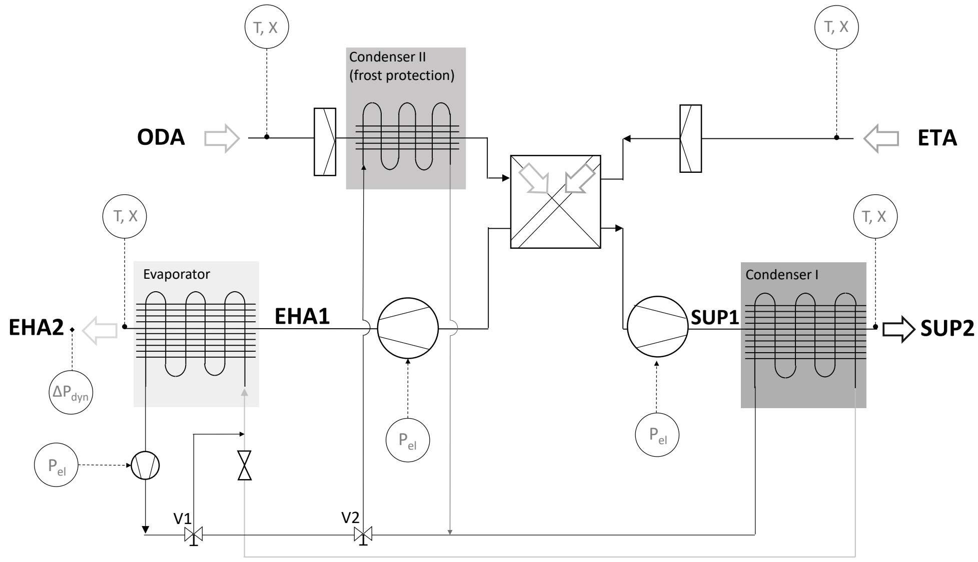


Figure 1: Simplified illustration of the measured EHA HP

FUNCTIONAL PRINCIPLE

Figure 2 shows a sample of a winter case air conditions in the EHA-HP on a psychrometric chart. The air would be conditioned in two processes. The first process would be through the air-to-air heat exchanger: Extract Air from the building (ETA) cools down to Exhaust Air condition one (EHA1), in the meantime, the Outdoor Air (ODA) heats up to Supply Air condition one (SUP1). The transferred power in the air-to-air heat exchanger could be calculated as:

$$\dot{Q}_{\mathrm{Ait-to-air}} = \dot{m}_{\mathrm{air}} \times (h_{\mathrm{SUPI}} - h_{\mathrm{ODA}})$$

Afterwards EHA1 flows through the HP evaporator and evaporates the refrigerant (EHA1 to EHA2). This process uses mostly the latent heat of condensation of water. The transferred power in the evaporator could be described as:

$$\dot{Q}_{\mathrm{Evaporator}} = \dot{m}_{\mathrm{air}} \times (h_{\mathrm{EHA1}} - h_{\mathrm{EHA2}})$$

In the end, heated and pressurized refrigerant, after HP's compressor, heats up SUP1 to SUP2 conditions. The heating power of the condenser could be written as:

$$\dot{Q}_{\mathrm{Condenser}} = \dot{m}_{\mathrm{air}} \times (h_{\mathrm{SUP2}} - h_{\mathrm{SUP1}})$$

The coefficient of performance (COP) of the heat pump would be defined as the useful power of heat pump divided by power input. In this work it is calculated with the following equation:

$$\mathrm{COP} = \frac{\dot{Q}_{\mathrm{Condenser}}}{P_{\mathrm{Compressor}} + P_{\mathrm{Ventilators}}}$$

When the heat pump is not activated, $\dot{Q}_{\mathrm{Condenser}}$ and $P_{\mathrm{Compressor}}$ are zero and the unit works like a normal Heat Recovery Ventilation (HRV) unit. For calculating of EHA1 and SUP1 conditions, a simulation model for the air-to-air heat exchanger based on the measured data was developed, which will be described later.

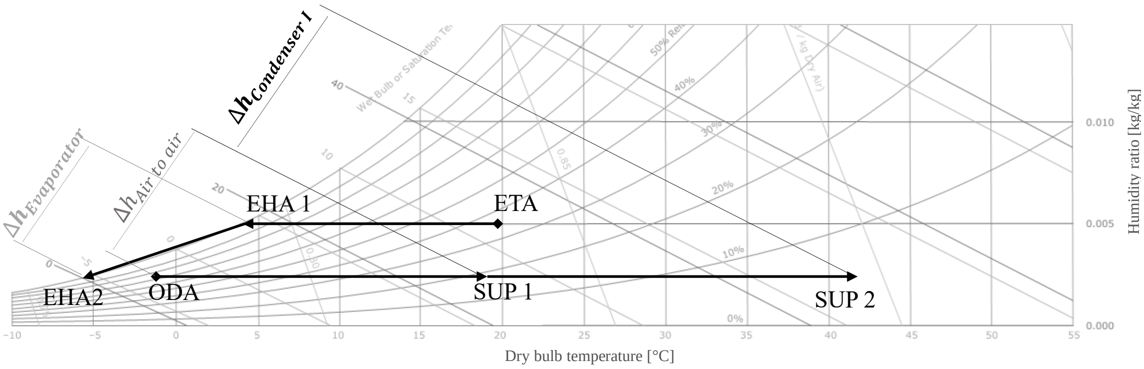


Figure 2: Air conditions in the EHA-HP on the air psychrometric chart (chart from **[12]**)

FIELD MONITORING

Figure 1 shows a hydraulic illustration of the investigated EHA-HP and the position of the installed sensors. Air temperature and humidity on all four sides of the unit (outdoor air, extract air coming from the building, supply air to the building and exhaust air leaving the house) and electrical power consumption of the unit (compressor and ventilators) were measured and logged. The measurement period was 55 winter days with an interval of 1 second.

In addition, the dynamic pressure difference (as an indication of the air volume rate) on the exhaust side was logged during the measurement period. Finally, the air volume rates of the unit at different ventilation levels was measured using constant injection tracer gas method as described in **[13]**. Table 1 summarizes the measured air volume rates and ventilators' powers usage in different levels. Based on the measured data, the EHA-HP mostly worked in the first two ventilation levels. When the unit functions in the HRV mode, the ventilators work in the first ventilation level as soon as the unit activates heating (heat pump switches ON) the fans generate the second ventilation level.

Table 1: Summary of measurements on air volume rates

Ventilation level	Dynamic pressure difference [Pa]	Air volume rates [m³/h]	Ventilators' power usage [W]
1	1.5	91.2	20
2	5	184.6	50
3	12	249.4	123
4	26	341.1	334

The measurements were conducted in a KfW 55 efficient house[2] with a living area of 235 m² in Stuttgart, Germany.

The EHA-HP was installed in a hybrid heating system as the central heating source of the building. The remaining heating energy demand of the building was covered using several decentralized electrical heaters, which were installed in the air supply ducts and in several extract air zones of the building (utility room, bathroom, WC and staircase).

Table **2** gives an overview of the applied measurement methods and used sensors. Logging and conversion of signals were conducted using a modular I/O system; the fieldbus controller Ethernet 750-881 from WAGO Kontakttechnik.

[2] A building standard in Germany; specific heating energy demand around 35 kWh/m^{2}a

Table 2: Overview of applied measurement methods and sensors

Value	Sensor/measurement method	Measurement accuracy	Manufacturer
Temperature	Type K Thermocouple	±0.004×|t| [°C]	RS PRO
Humidity	STPH-2-1-05	±2 [%]	NodOn
Dynamic pressure difference	Pitot tube	±2 [%]	Mueller Messinstrumente
Pressure difference transmitter	Testo 6351	±0.3 [Pa]	Testo
Power	Current transformer	±1 [%]	WAGO Kontakttechnik
Air volume rate	Tracer gas	±5 [%]	LumaSense Technologies

Figure 3 shows the measured air temperatures and electrical power of the unit on a sample winter day. The measured data shows that the heat pump works mainly in three modes (Figure 3):

- Part load heating (till around 06:00)
- Full load heating (from around 06:00 to 22:00)
- Defrost (approximately every hour)

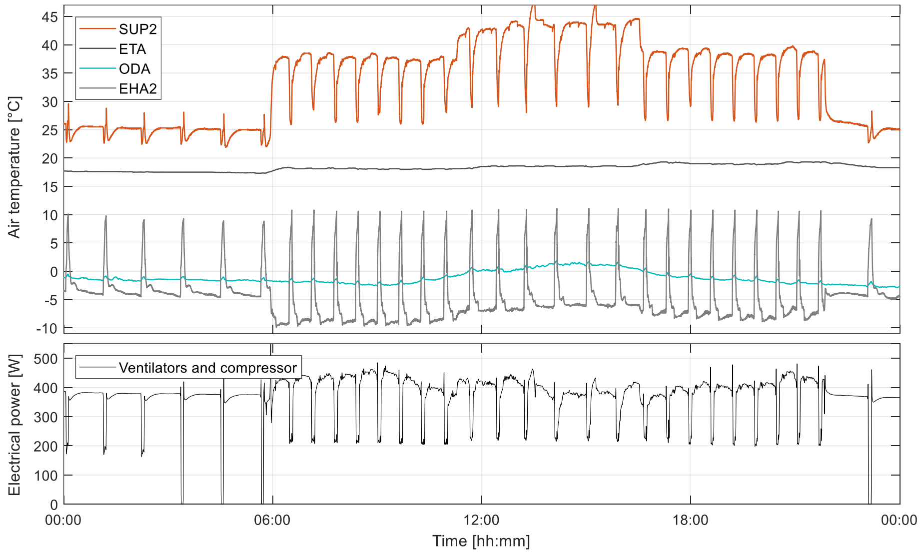


Figure 3: EHA HP behavior on a sample cold winter day

Based on measurements, the heat pump works most efficiently in the full load heating mode with a COP between three and eight; depending on the inside and outside air conditions. In the part load heating mode, the heating COP was measured between one and three. Moreover, it could be seen that there was no active heating taken place while defrosting and only the heating capacity of the condenser was discharged. This will be described in detail later in the defrosting section.

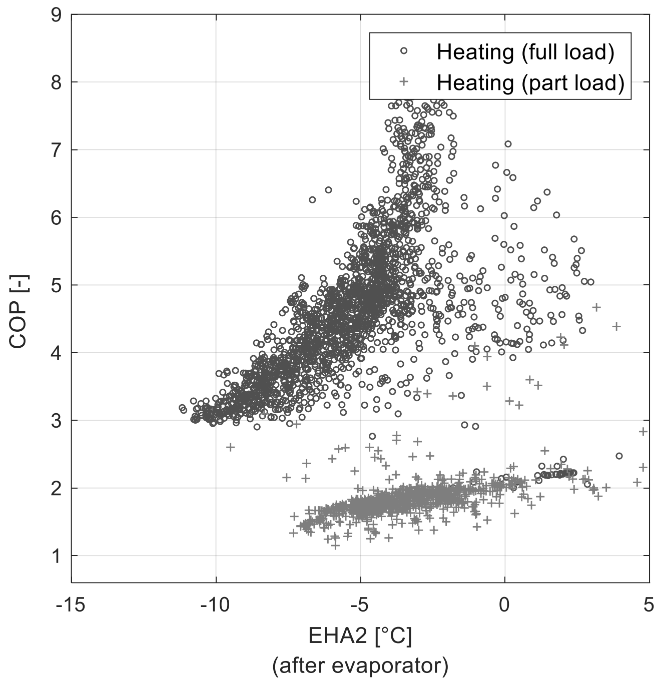


Figure 4: Average measured COP of the EHA-HP in two modes

Almost every night during heat pumps' function, it works in the part load heating mode (possibly due to conventional nighttime temperature reduction strategy). Based on the measured data, in the part load heating mode the heat pump worked approximately with a relative power of 60%.

Figure 4 shows the measured COP of the EHA-HP on around 4000 data points; every data point is a mean value over a ten minutes period of steady state heating mode. Figure 5 shows an example of a steady state period. It could be seen, that the partial derivatives of the air outlet temperature and input power of the unit with respect to time are approximately zero in this period. Additionally, a direct correlation between EHA2 temperature and COP of the HP could be seen. This correlation was applied to model the behavior of the measured EHA HP, which will be explained later in the heat pump simulation model section.

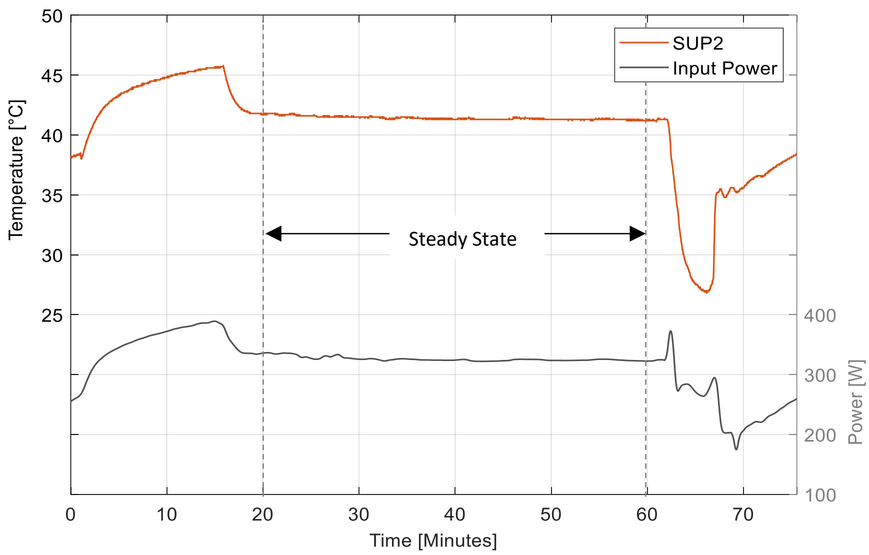


Figure 5: Example of a steady state period

During the HP's run time, maximum and average SUP2 temperature were measured 46.8 °C and 34.8 °C respectively. In addition, HP's maximum and average heating power were calculated 2.02 kW (10.94 W/m³/h) and 1.20 kW (6.5 W/m³/h) respectively. It is worth mentioning, the HP's total heating output in the measured period was surprisingly low (on average total 14.9 kWh/day) for a KfW 55 building. **Fehler! Ungültiger Eigenverweis auf Textmarke.**Table 3 summarizes the EHA HP measurements.

Table 3: Summary of measured data, exhaust air heat pump

Description	Value	Unit
Total thermal energy output	828.41	[kWh]
Total electrical energy input	261.33	[kWh]
Average COP	3.17	[-]
Average air-to-air HEX efficiency (at 184 m³/h)	87.5	[%]
Number of defrost cycles	667	[Cycles]
Average source temperature (EHA1)	4.45	[°C]
Average sink temperature (SUP2)	34.8	[°C]
Heat pump run duration	680.8	[Hours]
Measurement duration	1325.9	[Hours]

DYNAMIC SIMULATION

AIR-TO-AIR HEAT EXCHANGER

The total effectiveness of heat exchange depends on: exchanger construction (including configuration), heat transfer material, moisture transfer properties, transfer surface area, airflow path, distance between heat transfer surfaces, and overall size; and of course inlet conditions for both air-streams, including pressures, velocities, temperatures, and humidity **[14]**. The efficiency of the modelled air-air heat exchanger is obtained from measurements data from the online database of the manufacturer **[15]**. The implemented algorithm for calculation of the air-to-air sensible heat exchange is illustrated in Figure 6.

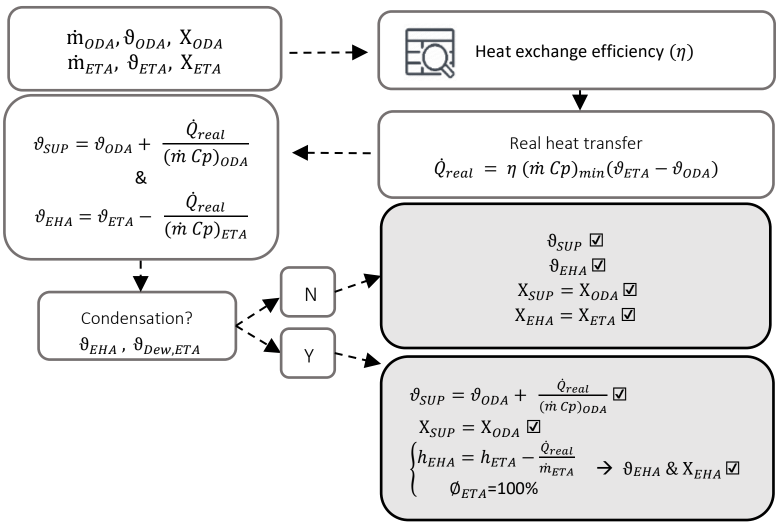


Figure 6: Implemented algorithm for calculation of sensible heat exchange in the air-to-air heat exchanger

Based on the measured data the EHA-HP unit worked in the HRV mode for approximately 645 hours of the 1326 hours total measurement period. The measured air conditions in this period were used to parametrization and validation of the air-to-air heat exchanger simulation model. Figure 7 shows a sample of simulated and measured air temperatures and relative humidity.

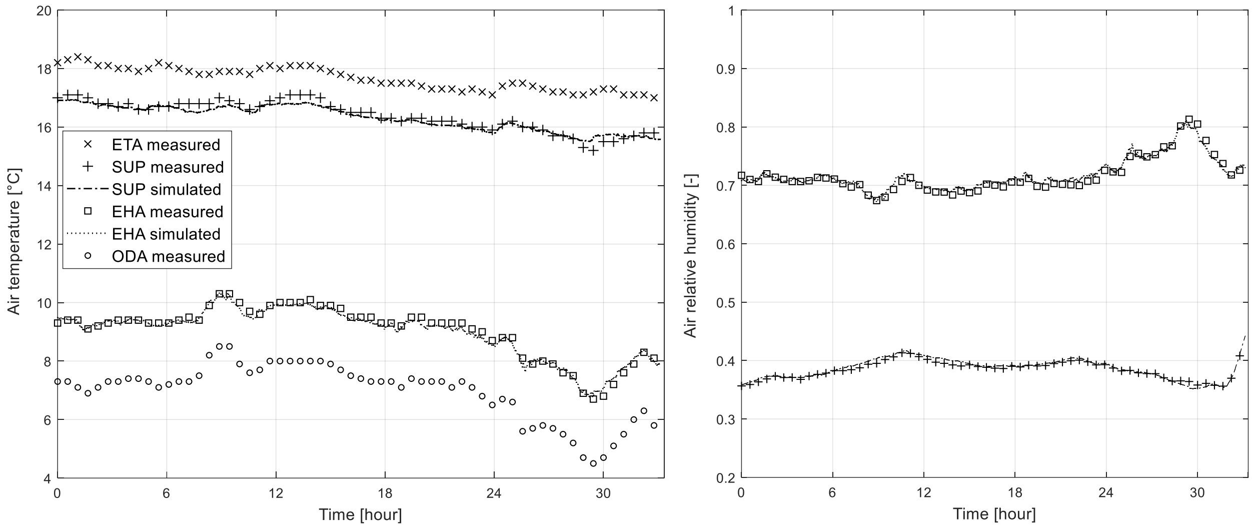


Figure 7: Sample of measured and simulated air conditions in HRV mode (heat pump switched OFF)

Table 4 summarizes the quality of the modeled air-to-air heat exchanger through the calculated root mean squared error between the measured and simulated data. The calculated errors might be the result of measurements' errors and neglection of the ventilators' heating losses. Moreover, the air leakages and volume rate unbalances between the air sides are also neglected in this study, which could also have an impact on the effectiveness of the heat exchanger.

Table 4: Summary of air-to-air heat exchanger model quality

Value	Root mean squared error
EHA temperature [K]	0.7206
EHA relative humidity [%]	3.56
SUP temperature [K]	0.7845
SUP relative humidity [%]	1.90

It was assumed that as soon as outdoor air temperature falls below -2 °C, the condenser II heats up the ODA to -2°C, via the provided valve (V1, see Figure 1). The assumption was made to simulate the frost protection of the air-to-air heat exchanger. The needed energy is extracted from the heating power of condenser I. The heat losses and possible COP reduction were neglected.

HEAT PUMP

The developed model in the current study is based on the static characteristic curves of the measured HP (black box model). The model calculates the power consumption of the unit and the COP of the HP statically. The calculation is based on the air temperature after evaporator (EHA2) from the last simulation step (interval 1 second). EHA2 is calculated by means of adapted curves equations, which are derived from measured data.

Using statically calculated COP and electrical power, thermal powers of the condenser and the evaporator are calculated. Afterwards, the calculated thermal powers are applied to the dynamic models of condenser and evaporator, which were characterized based on their measured dynamic behavior. Figure 8 shows the used algorithm for simulating the HP behavior.

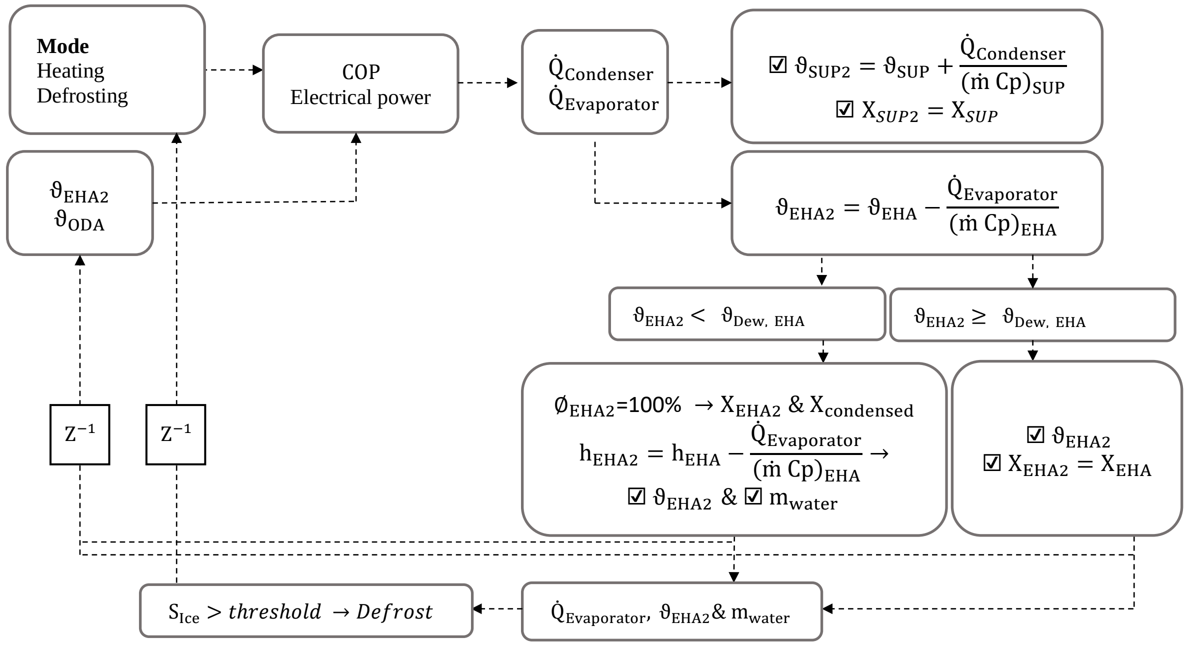


Figure 8: Implemented algorithm for simulating the heat pump behavior

HEATING

Figure 9 illustrates the adapted curves to calculate the COP and the electrical power of the HP. The averaged values (steady state) in the full load heating of the EHA2 temperature were calculated as an indication of the electrical power and COP (Figure 9).

It was expected that with reducing the compressor speed a considerable improvement in the heat pump’s performance would be seen; similar to **[8]**. Therefore, the measured low performance of HP in the part load heating mode was not considered. As a result, it was assumed that the performance of the HP is dependent only on the calculated temperature after the evaporator (EHA2) and the HP works without modulation steps. In **[16]** is shown that this simplification does not significantly affect the annual electricity use of the heat pump.

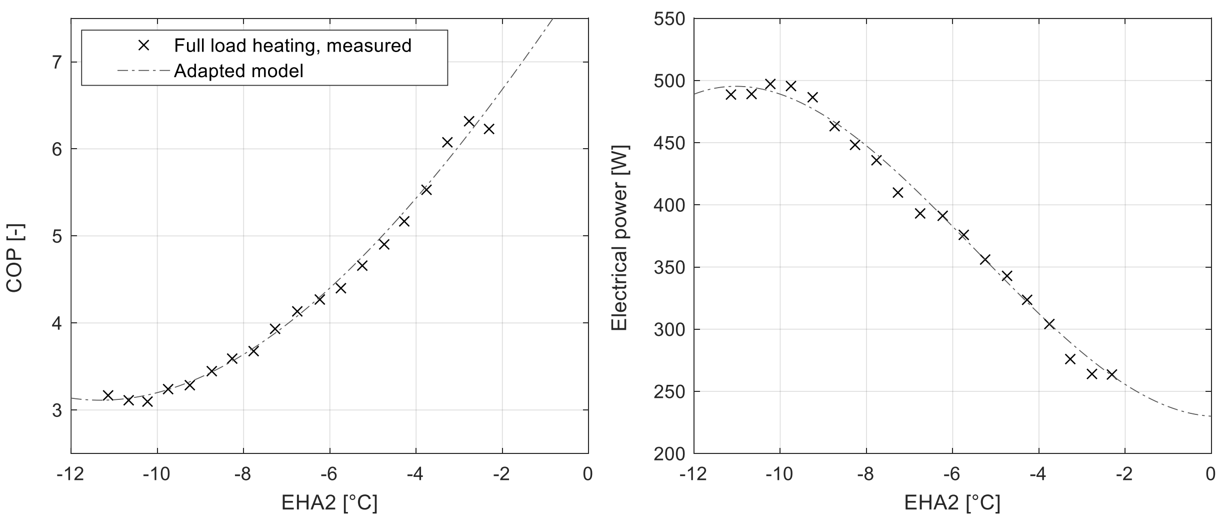


Figure 9: Average measured COP and electrical power of EHA HP and adapted model based on EHA2 temperature

DEFROSTING

Frost formation on the surface of evaporator in the air refrigeration systems is a known and widely discussed issue. There are several available studies on the frost formation and its effects on the energy performance **[17, 18, 19]**. When surface temperature of the evaporator (or any cold surface) falls lower than the dew point of the moist air, water droplets would be formed on the surface. If the surface temperature becomes under the freezing point, water starts to freeze. However, for an accurate simulation of the frost formation behavior, complicated correlations between operating conditions should be investigated **[20]**. Frost formation on the evaporator surface leads to two main difficulties for the heat exchanger. In one hand, ice layer acts as a thermal insulation and reduces the heat exchange efficiency; on the other hand, it reduces the evaporator fin spacing, increases the air pressure drop, and therefore increases the needed power of ventilators. Due to these problems, a defrosting process is necessary to regain normal performance **[21]**.

There are several approaches for defrosting the evaporator in the air refrigeration systems; for instance, the compressed and heated refrigerant after the compressor could be brought to the evaporator via a bypass pipe to defrost it (heated gas defrost method). Alternatively, the refrigerant cycle could be reversed so that the condensation of the refrigerant happened in the evaporator. In this case, the needed evaporation energy should be derived from heating circuit. There are also different methods to control and find the optimum stage to start the defrosting process, from periodic defrosting algorithms to implementation of a defrost algorithm based on pressure and temperature signals **[22]**.

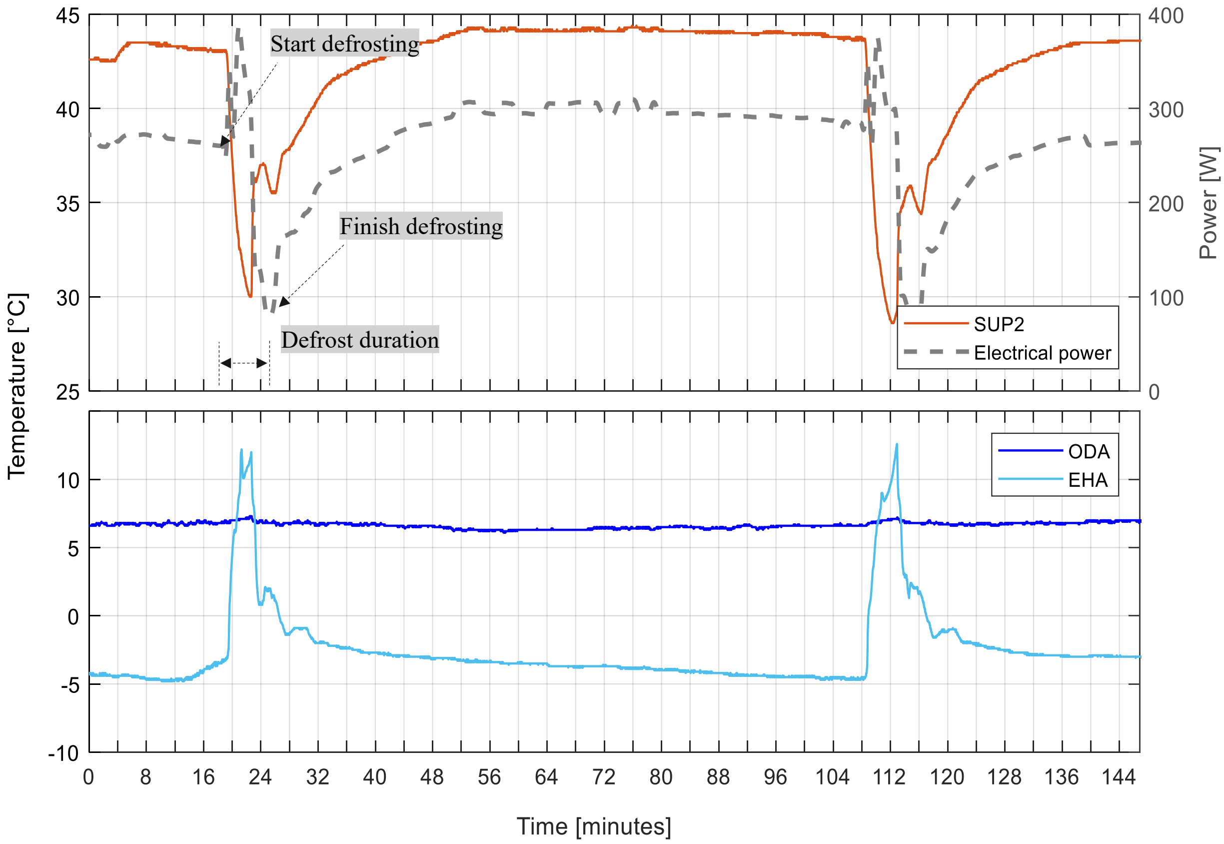


Figure 10: Air temperatures and unit electrical power during defrost process

Defrosting the evaporator in the investigated EHA HP is carried out using the heated gas defrost method. The process starts by heating up the evaporator via the provided bypass valve (V1) in the refrigerant circuit (see Figure 1), for a duration of around five minutes. As a result, with defrosting the evaporator, an increase of EHA2 temperature can be seen, which is recognizable in Figure 10. This method seems to be an appropriate method for defrosting the evaporator in the EHA-HPs. Because in such systems the defrost energy should be gained from the supply air and this could lead to very low air supply air temperature.

In this work, the heating behavior of the EHA-HP during the defrosting process is concerned. It was assumed that during every defrost process the total calculated heating power of the HP would be used for defrosting the evaporator. In order to predict the start of defrost process, a simple method for calculating the ice thickness on the evaporator was implemented. Calculation of the ice mass is based on an assumption that, the surface temperature of the evaporator is 4 K lower than the outlet air temperature (EHA2). This assumption was suggested by **[22]**. The implemented method will be briefly described below.

With $\boldsymbol{\vartheta}_{\mathbf{Evap.Surf.}} = \boldsymbol{\vartheta}_{\mathbf{EHA2}} - \mathbf{4\ K}$; in case of $\boldsymbol{\vartheta}_{\mathbf{Evap.Surf.}} < \mathbf{0\ °C} \rightarrow \boldsymbol{m}_{\mathbf{ice}} \cong \boldsymbol{m}_{\mathbf{water}}$

And with assumption of 6 m² heat transfer surface and ideal sharing of ice on the evaporator surface, the ice thickness is calculated as follows:

$$\mathbf{s}_{\mathbf{ice}} = \frac{\mathbf{m}_{\mathbf{ice}}}{\boldsymbol{\rho}_{\mathbf{ice}} \times \mathbf{A}_{\mathbf{Evap.Surf.}}}$$

The density of the ice, $\boldsymbol{\rho}_{\mathbf{ice}}$, is calculated using the following correlation based on **[23]** and the assumption of $\boldsymbol{\vartheta}_{\mathbf{frost}} = \boldsymbol{\vartheta}_{\mathbf{Evap.Surf.}}$:

$$\boldsymbol{\rho}_{\mathbf{ice}} = \frac{\mathbf{650\ exp(0.277} \times \boldsymbol{\vartheta}_{\mathbf{frost}})}{(\mathbf{1} - \boldsymbol{\varepsilon})} - \boldsymbol{\rho}_{\mathbf{air}}\ \boldsymbol{\varepsilon}$$

Where $\boldsymbol{\vartheta}_{\mathbf{frost}}$ is in °C and $\boldsymbol{\varepsilon}$ is defined as ratio between moist air and the total volume of the porous medium, which was measured by **[23]** between 0.72 and 0.91 and here assumed as a constant value of 0.8.

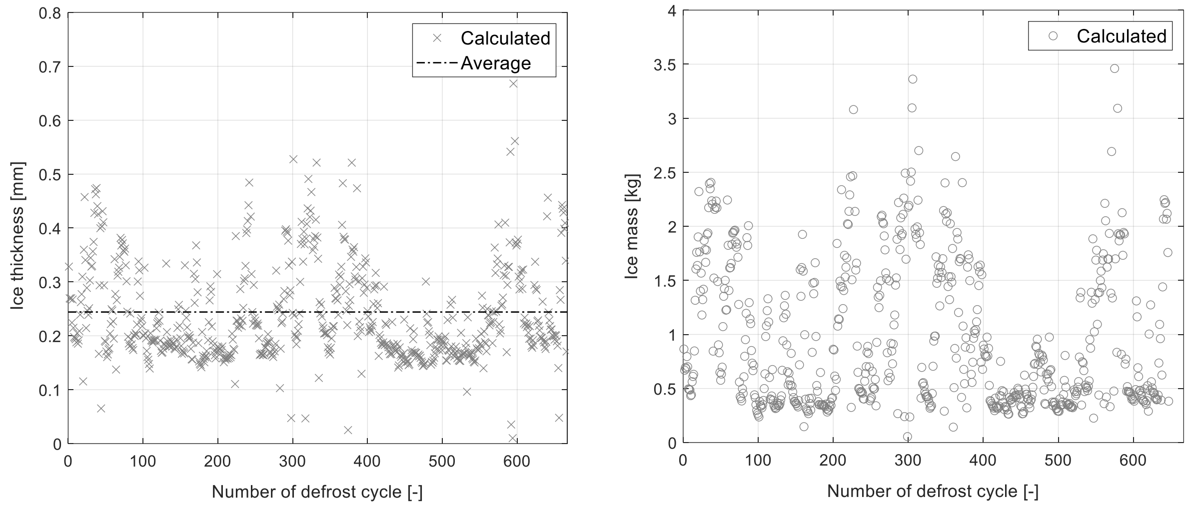


Figure 11: Calculated iced thickness and mass based on measured data

Figure 11 shows the calculated ice thickness and mass using the described method for the 667 measured defrost cycles. The average ice thickness value of 0.2441 mm was assumed as a threshold for starting the defrost process which takes approximately 300 seconds in average based on measured data. An example of two defrosting cycles could be seen in Figure 12.

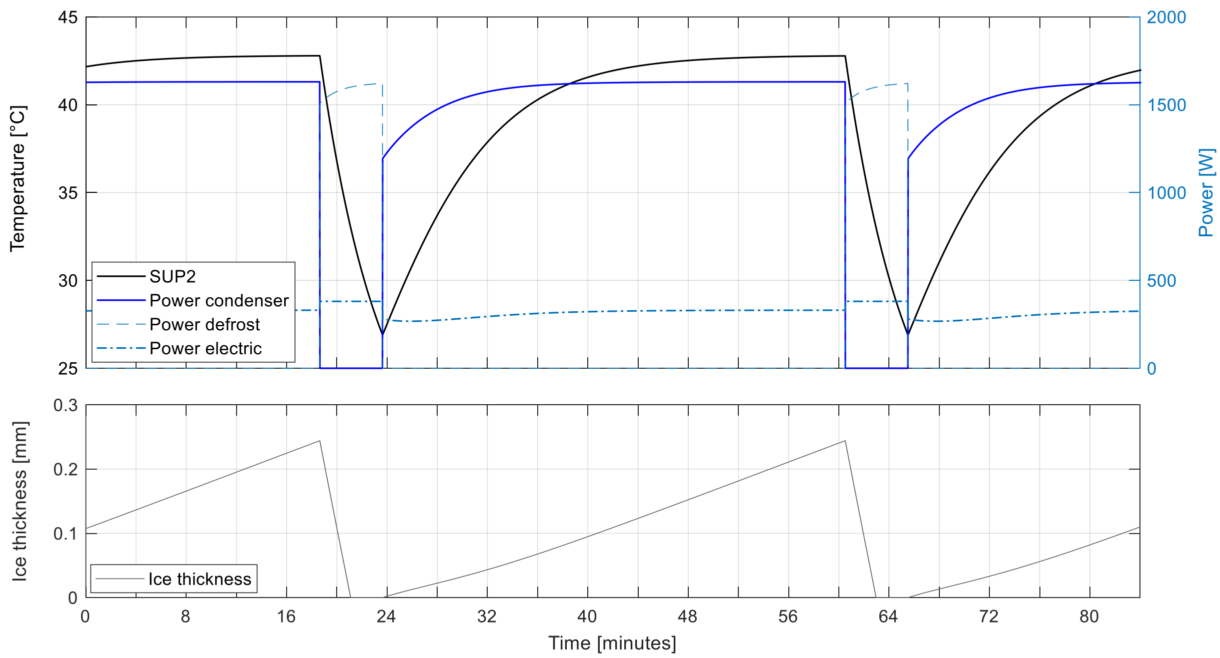


Figure 12: Example of defrost process simulation (ODA: 4 °C& 30 %, ETA: 20 °C & 35 %)

Figure 13 shows the effect of extract air temperature and humidity on the average HP performance. Including the effects of defrosting on the average COP, the performance was calculated within 12 hours simulation under the same air conditions. The significant impact of ETA moisture content on the performance is seen in Figure 13. Based on the modeled EHA-HP, 10% increase in air humidity could increase the HP COP up to 15%.

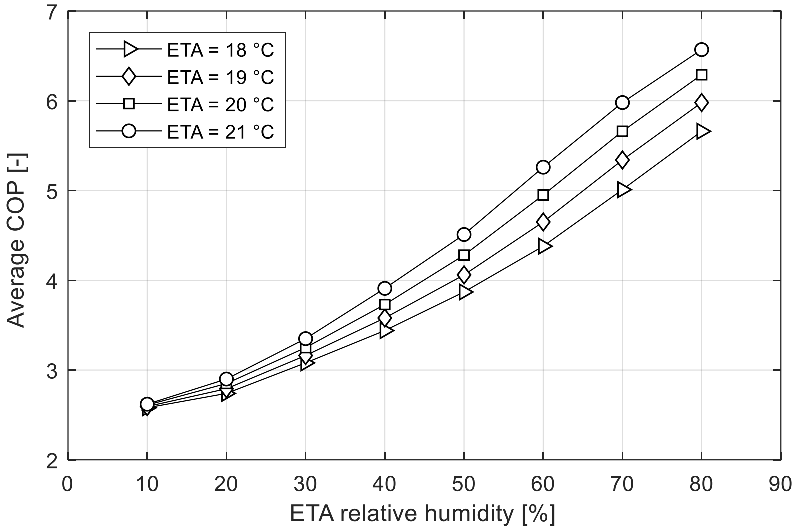


Figure 13: Effect of ETA temperature and humidity on HP performance (ODA: 4 °C & 30 %)

12-hour simulations at different ODA temperatures with the same ETA conditions and ODA humidity were carried out, in order to find out the effect of outdoor air temperature on the HP power and performance. The results confirm the previous findings in **[8]**, where an overall performance for an EHA HP of 2.5 at -7 °C and 4.5 at 10 °C outdoor air temperature was calculated.

The evaluated EHA HP in **[24]** has shown a system COP of 2.8 at -7 °C and 2.5 at 5 °C. The lower calculated COP at 5 °C was referred to high share of electrical power of the utilized ventilators.

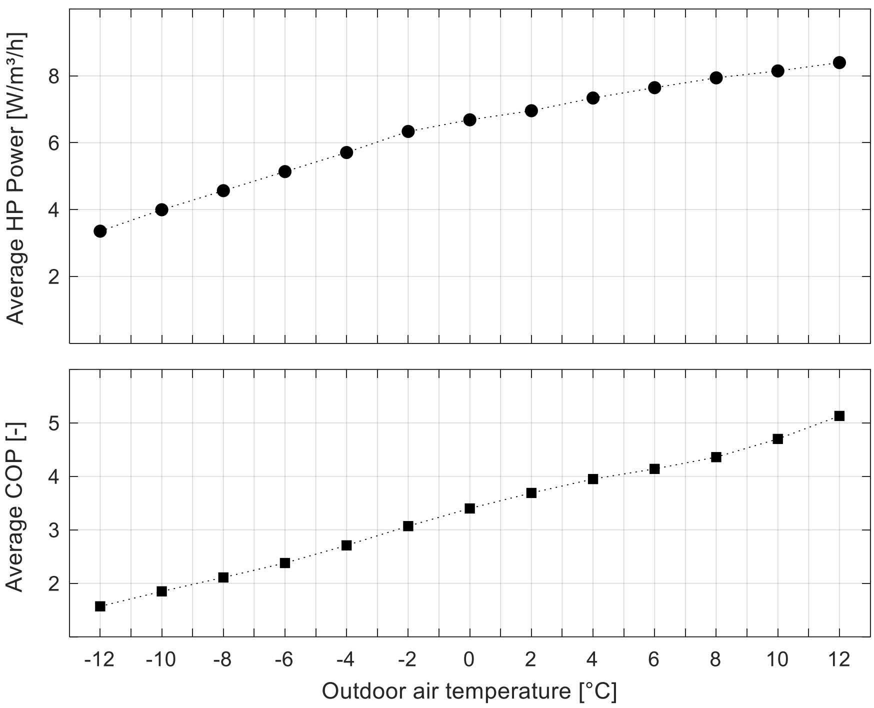


Figure 14: Effect of outdoor air temperature on HP performance and Power (ETA: 20 °C & 30%, ODA: 30%)

SIMULATION BOUNDARY CONDITIONS

The simulations in this work are carried out using Building Technology Simulation Library (BTSL); a Bosch Thermotechnology internal MATLAB Simulink library **[25]**. This library contains different blocks for dynamic simulation of the different parts of heating, venting and hot water providing in a building. From models of basic components like valves, pipes and air ducts to more complicated models like heat pumps. Every single model is validated against measurement data. There are also different blocks to simulate the boundary conditions, such as different user behaviors and weather data.

Figure 15 shows a sample of a reference model in BTSL environment. The developed model of EHA HP in Simulink was also added to the library. For simulation of the building behavior in BTSL, the TRNSYS Type 56 is used. Since both TRNSYS and MATLAB components could be generated as Windows DLL files, the integration of TRNSYS components in MATLAB or vice versa is feasible. The suggested integration method by **[26]** is implemented to adapt the TRNSYS Type 56 to the BTSL Simulink library.

Type 56 models the dynamic thermal behavior of the building using the multi-zone building modelling approach. In this modelling approach, every zone modeled with one air node, which represents the thermal capacity of the zone air volume and its belongings **[27]**.

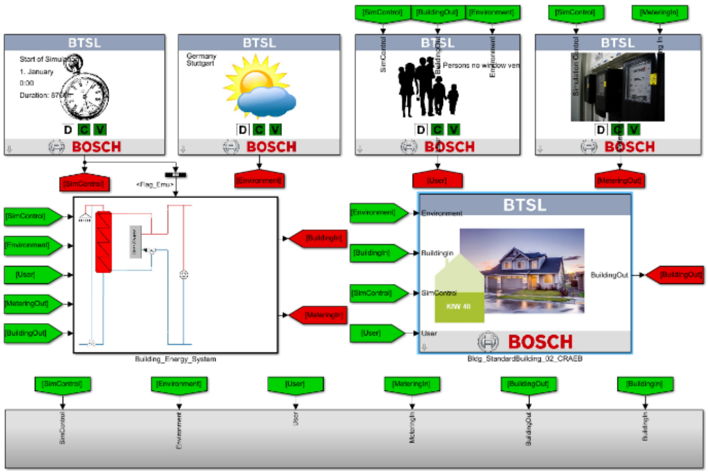


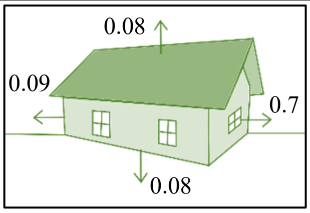
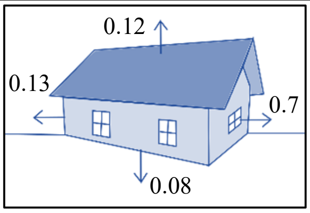
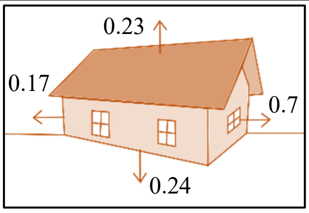
Figure 15: Sample of BTSL environment

Selected buildings category

Table 5 indicates the selected specified U-values of external walls, ceiling, rooftop and windows for the selected buildings. The U-values were selected based on three conventional efficient building standards in Germany: HD15, HD 25 and HD35 (respectively similar to Passive House, KfW40 & KfW55).

The EHA HP model was created based on the measured air volume rate of 184 m^3/h. In order to have an air exchange ratio of 0.4 [h^{-1}] in all the simulated buildings, the living area was assumed 184 m^2 (with room height of 2.4 m). The simulation weather data was selected based on Stuttgart, Germany **[28]**. The selected user profile was based on a three-person household with average internal gains of 7.69 kWh/m^2a. It was assumed that the users do not open the windows and the entire ventilation of the building is conducted using the EHA HP unit.

Table 5: Selected buildings

Building name	**HD15**	**HD25**	**HD35**
Building U-values in [W/m²k]			
Ideal heating demand with HRV	16 [kWh/m²a]	23 [kWh/m²a]	34 [kWh/m²a]
HRV air change ratio	0.4 [h^{-1}]	0.4 [h^{-1}]	0.4 [h^{-1}]
Infiltration ratio	0.08 [h^{-1}]	0.12 [h^{-1}]	0.12 [h^{-1}]

Each simulated building was selected to have five zones, which contains four heated and one unheated zone. Table 6 summarizes the buildings' zones and their floor area. The supply air amount was selected based on the floor area of the supply zones; the proportion of air volume flow for the zone one, three and four were selected respectively 33%, 40% and 27%. All air supply ducts are located in the building envelope and wherever possible in the floor of the extract air zone, in order to utilize the heat losses of the supply air ducts in the extract air zone (not directly air heated). As auxiliary heaters, in every zone of the building an electrical heater with 1 kW power was installed.

Table 6: Specifications of the modelled building

Zone number	Air zone	Zone description	Heated floor area [m^2]
Zone 1	Supply	Living/Dining Room	46
Zone 2	Extract	Kitchen/Bath/WC	46
Zone 3	Supply	Office/Children's rooms	56
Zone 4	Supply	Bedroom	36
Zone 5	Not vented	Attic	(unheated)

HYBRID SYSTEM HEATING CONTROL

Figure 16 shows a schematic illustration of the main principle of the applied hybrid system control. In all system the electrical heater could be activated only in case HP is also running. In addition, the extract air zone's electrical heater could only be activated through user presence (only during the day, in average 2.7 hours per person). The EHA HP was controlled by using the weighted average temperature (based on the floor area) of the supply air zones. Except in Living Room control, that the HP control unit uses the measured living room temperature as its control value.

In all systems the electrical heaters and heat pump were controlled by hysteresis controllers with upper and lower dead bands of 0.25 K. To control the electrical heaters and the EHA HP, different strategies were implemented and compared together. Table 7 summarizes the six control strategies, which were compared with each other in the three selected building standards.

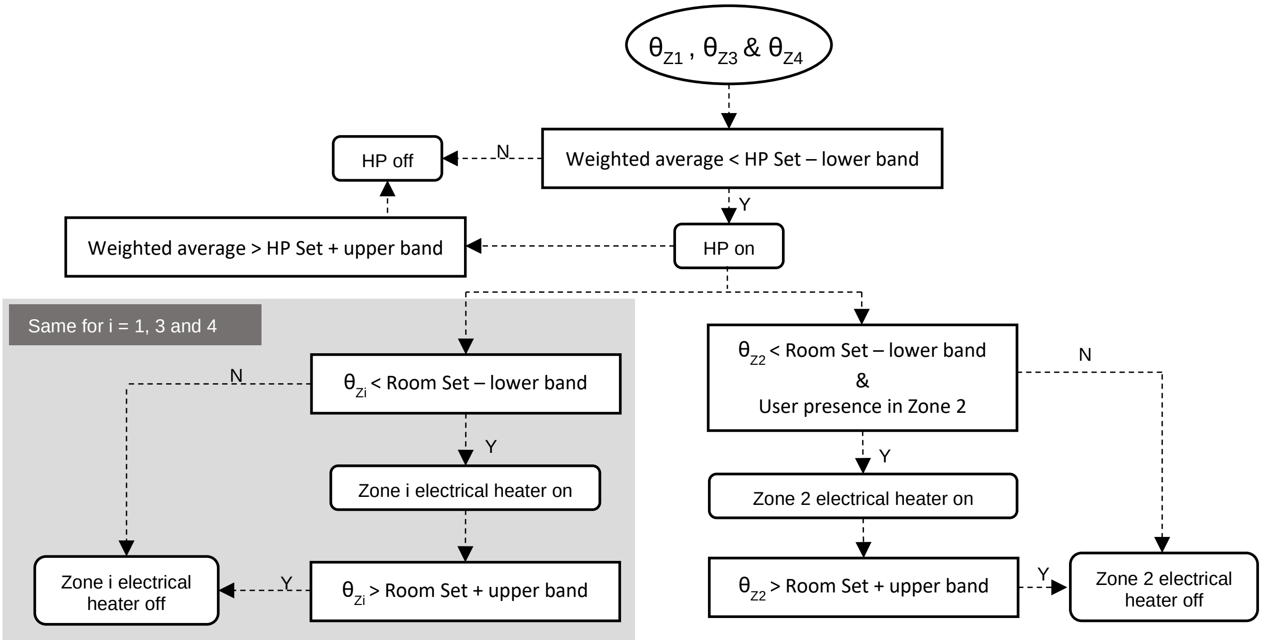


Figure 16: Main principle of the hybrid control system

In the Set Normal and Set Low control strategies, the electrical heaters could only operate if the heat pump operates in full load. In other words, room electrical heaters could be activated only in case the average air temperature of all three supply rooms falls lower than the set temperature. The set temperatures in all building zones were selected equal; in Set Normal and Set Low respectively, 21 °C and 20 °C.

Table 7: Overview of the compared control strategies

Strategy name	Set temperature	HP control temperature	Electrical heater control	Nighttime reduction
Set Low	20 °C	Average of supply zones	Same set in rooms, if HP runs (Extract room only if user presence)	No
Set Normal	21 °C	Average of supply zones	Same set in rooms, if HP runs (Extract room only if user presence)	No
Bivalent Point	21 °C	Average of supply zones	Same set in rooms, If outdoor air < bivalent point (Extract room only if user presence & room temp. under biv. point)	No
Nighttime Reduction	21 °C (18 °C)	Average of supply zones	Same set, if HP runs (Extract room only if user presence)	06:00 – 22:00 (22:00 – 06:00)
Lower Room Set	21 °C (Room: 20°C)	Average of supply zones	If rooms < set room (Extract room only if user presence & room temp. under set point)	No
Living room control	21 °C	Sensor in living room	Same set in rooms, if HP runs (Extract room only if user presence)	No

To determine the bivalent points of the hybrid system in the selected buildings, the heating demand of buildings was calculated using the ideal heating function of TRNSYS Type 56. This function calculates the needed heating power keeping the air temperature of all heated zones above the selected temperature of 21°C.

The average heating demand of each building at different outdoor air temperatures was calculated and compared to the maximum heating power of the EHA HP (see Figure 17). The Bivalent Point was defined as the outdoor air temperature, which the power of EHA HP meets the heating energy demand of the building. For HD15, HD25 and HD35 the bivalent temperatures were calculated -4.7 °C, -0.55 °C and 2.75 °C respectively.

In the Bivalent Point control strategy, the electrical heaters in all of the zones are allowed to be activated only if the outdoor air temperature falls below the bivalent point. Furthermore, in the extract air zone the presence of a user is necessary to activate the heater.

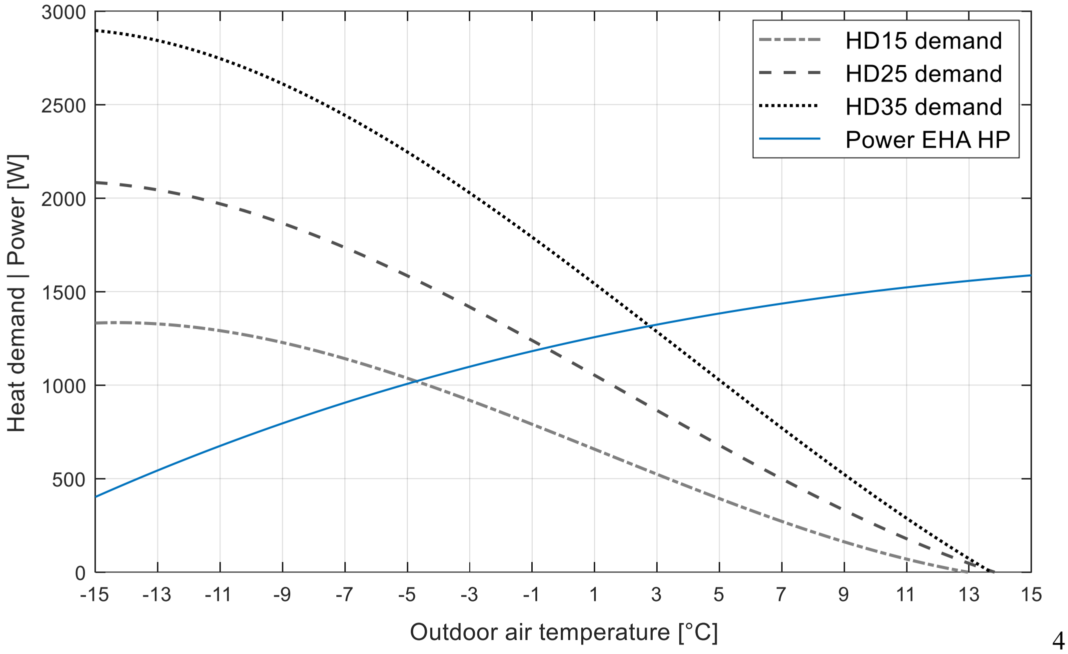


Figure 17: Heating demand of the selected building standards and power of EHA HP versus outdoor air temperature

In the Nighttime Reduction control strategy the set temperature of the building was set back to 18 °C from 22:00 to 06:00 o'clock in the morning. The HP and auxiliary heaters were controlled similar to Set Low and Set Normal control strategy.

In the Lower Room Set control strategy, the set temperature of the building rooms was selected 1 K lower than the HP set temperature (20 °C). In addition, the auxiliary heater of the extract air zone requires similarly user presence to be activated.

In the Living Room Control, electrical heaters were controlled similar to Set Low and Set Normal control strategy.

RESULTS AND DISCUSSION

For comparing the performance of the systems, the Seasonal Coefficient of Performance (SCOP) for the hybrid system was defined. It is calculated as the total heating power of condenser and electrical heaters divided by electrical power of the compressor, ventilators (during heating) and electrical heaters over a year.

Figure 18 summarizes the effect of set temperature of the building on the heating energy demand, the HP and hybrid system SCOP. The white bars show the supplied heating energy amount, which was covered by the HP. The gray bars show the amount of heating energy covered by the electrical heaters. Moreover, the HPs' and the hybrid systems' SCOP are also illustrated on the right y-axis.

Based on the simulations, 1 K lower set temperature in the Set Low strategy reduces the heating demand around 15 % on average. In addition, the SCOP of the HP increases with an increasing building set temperature due to higher temperatures of the exhaust air. Therefore, no significant changes in the hybrid seasonal performance were seen. The average SCOP of the hybrid system is reduced from HD15 to HD25 and from HD25 to HD35, respectively 14% and 22%, along with the decreasing building efficiency.

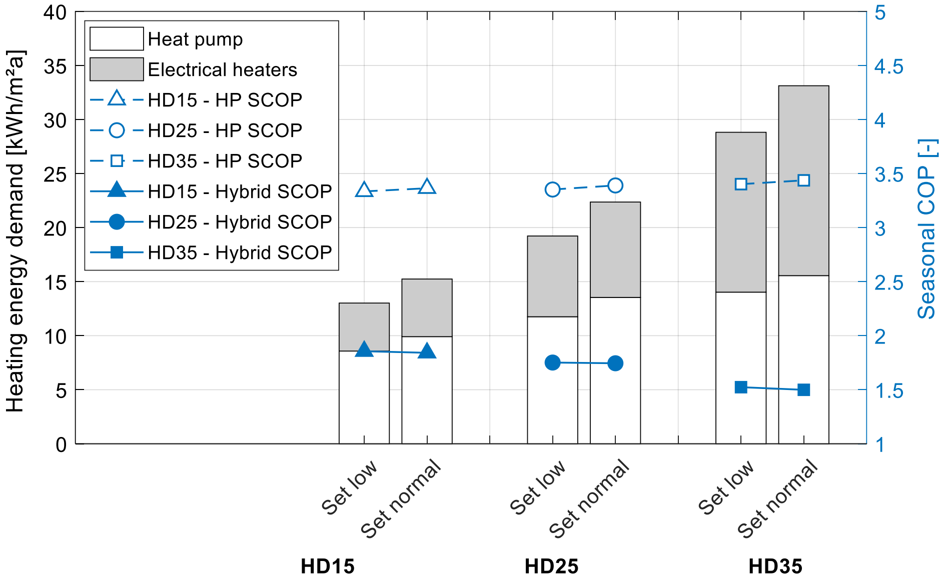


Figure 18: Comparison of set low and set normal control strategies

Figure 19 shows the overview of the selected buildings' heating energy demand, controlled with different strategies. It could be seen, that the heating energy demand of none of the buildings exceed the ideal heating demand of the corresponding building (see Table 5). This shows that the yearly average air temperature was under the set value, 21 °C.

The control strategy has not shown a significant impact on the heat pump seasonal performance. In all the simulated systems, the HP's performances were calculated between 3.4 and 4 [-], which match with the calculated potential seasonal performance of the EHA HP studied in **[29]**.

By implementation of the most control strategies, the hybrid system SCOP was decreased significantly with decreasing the building efficiency (increasing heating demand). This could be referred to the significant impact of limited power of the EHA HP. Except for Nighttime Reduction control strategy, where building energy efficiency has shown no major impact on the seasonal performance of the hybrid system.

Nighttime Reduction strategy has shown the worst average system performance among the compared strategies. In other words, the calculated share of direct electrical heaters' consumption was the highest in the buildings with Nighttime Reduction strategy. Although by using this strategy, the total heating energy demand of the building compared to ideal heating could be reduced in average about 8%, but the electrical heaters has covered approximately half of the total heating demand. The main reason for this occurrence was the needed higher heating power after every cool down, in order to warm the building up again.

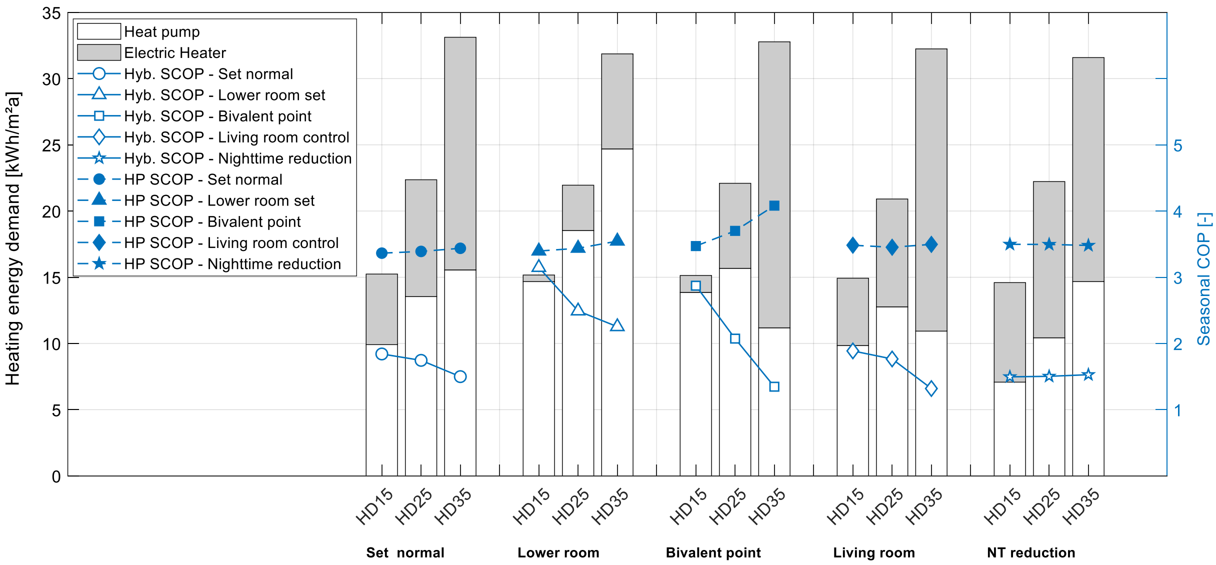


Figure 19: Overview of heating energy demand of buildings and performance of heating system through different control strategies

Furthermore, it could be seen that except Nighttime Reduction and Living Room control strategies, the heat pump performance has increased slightly with the increasing heating demand due to the lower number of needed HP cycles in the run time. This trend would be reversed by using a heat pump, which functions efficiently in its modulation steps

Using an installed sensor in the living room to control the central heating source is a conventional approach of controlling heating systems. The simulations' results have shown that this strategy could not provide an acceptable performance in such heating system. The heating demand of different rooms in an efficient building is highly dependent on the internal gains of the zones. The sensor in the living room could not provide sufficient information to control the central heating system of the building.

The best HP performance was seen in HD35 building by means of the Bivalent Point control strategy. Since the HP in this strategy was the mono heating source for the outdoor air temperatures higher than 2.7 °C, its average source temperature was higher than other strategies. In contrast, it was shown that the Bivalent Point control strategy performance in the HD35 building was one of the worst strategies (beside Living Room control) in this building standard. It could be seen, that most likely the selection of bivalent point in the HD35 building was not correctly done. The HP heating coverage in this building was remarkably lower than other buildings with this control strategy. Through selecting a lower bivalent temperature, the performance of the hybrid system could be improved.

However, by comparing the results of Bivalent Point strategy in HD15 and HD25 buildings, a SCOP value of higher than 2 [-] could not be imagined in the HD35 building. It could be said that the bivalent point control seems to be an appropriate strategy for the HD15 building with the lowest heating energy demand among the compared buildings.

The best hybrid system efficiency was achieved by means of Lower Room Set control strategy. This strategy was the only strategy, which SCOP of the hybrid systems of all three building standards was calculated above 2 [-].

Figure 20 shows the electrical energy demand of hybrid systems in the compared building standard by different control strategies. The ventilation electrical energy demand shows the consumed electrical energy by the ventilators during times the unit works in the HRV mode and the HP was switched off. The lowest total electrical energy consumption in HD15, HD25 and HD35 was calculated respectively; 7.03 kWh/m²a with Bivalent Point strategy, 10.34 kWh/m²a and 15.55 kWh/m²a with Lower Room Set control strategy.

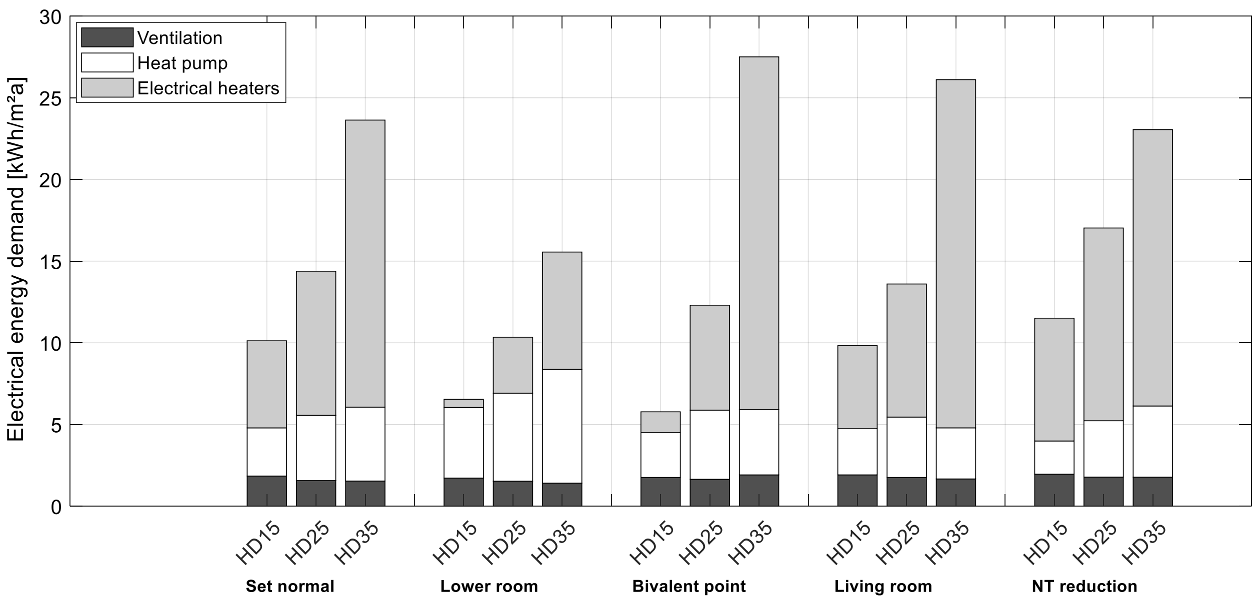


Figure 20: Electrical energy demand of hybrid systems in different buildings and through different control strategies

For a better comparison between the simulated control strategies the number of hours, that the average air temperature of the three supply zones of the buildings falling below 20.5°C are calculated and illustrated in Figure 21. It could be seen that the Living Room control approach not only showed an unacceptable system performance but also an undesirable comfort for the users.

Among the compared strategies, the Set Normal control strategy has shown the best user comfort performance. From comfort point of view, similar to energy performance, the Lower Room Set strategy seems not to be an appropriate method for control the hybrid system in HD35 building standard. Therefore, it could be said, that none of the compared strategies has shown an adequate performance in the HD35 building standard.

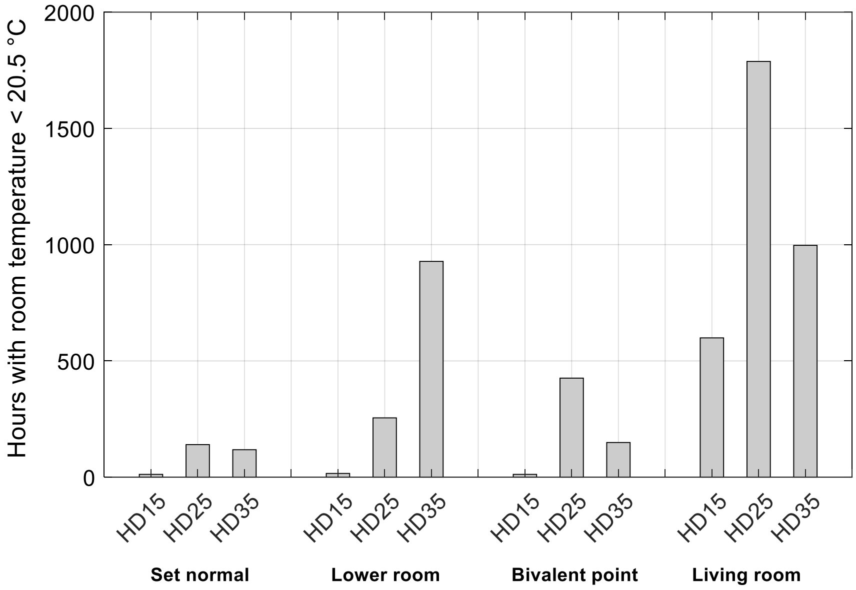


Figure 21: Number of hours with average supply rooms' temperature under 20.5°C

CONCLUSION

Performance evaluation of exhaust air heat pumps in air based heating systems based on field monitoring and dynamic simulations has been presented in this study. A simple approach for modeling the dynamic behavior of EHA HPs as a black box model using MATLAB Simulink has been developed and utilized. Furthermore, a physical approach for modeling the defrost process and its occurrence has been developed based on the measured data. The modelled EHA HP were utilized to study the impact of exhaust air temperature and humidity on the heat pump energy efficiency. The calculated COP and seasonal performance of EHA HPs were similar to the previous findings and works.

Six different control strategies were defined and their impact on the building heating demand in three different efficient building standards was compared with each other. It has been shown, that conventional control approaches in the air based hybrid heating systems containing EHA HP and electrical heaters could lead to high electrical energy demand in efficient buildings. For instance, the calculated share of direct electrical heaters' consumption was the highest in the buildings, which used the conventional nighttime temperature reduction strategy. Because of the needed higher heating power after every cool down of the building.

The Bivalent Point control strategy has shown an acceptable comfort and energy performance in HD15 building standard. However, although the bivalent point was selected using simulation, performance of this strategy was not adequate in buildings with higher heating demand.

Among compared control strategies, only by means of selecting a lower set temperature (1 K) for electrical heaters than that for HP, a hybrid SCOP of higher than 2 [-] could be achieved in all of the selected building standards. However, in HD35 building standard this has led to high number of hours with uncomfortable indoor air temperature in this building standard. Among the compared control strategies, none of them was found suitable for the HD35 building standard.

Overall it can be said, that in designing air heating systems, containing EHA HPs and electrical auxiliary heaters, the building standard and the control strategy of the system must be studied accurately and wisely. Such hybrid systems could provide an efficient heating system with acceptable user comfort, in buildings with very low heating demand and by means of an appropriate control strategy. The higher the energy demand of the building, the greater the possibility of inadequacies in the hybrid system. Utilizing conventional heating control methods or designing EHA HPs in buildings with high heating energy demand could lead to high electrical energy consumption and/or undesirable user comfort.

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