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Simulation study on supply temperature optimization in domestic heat pump systems

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ABSTRACT

An air-to-water heat pump system for the heating of a one-family home is numerically analysed. The influence of the supply temperature on the seasonal performance factor of the heating system is examined by varying the heating curve. Furthermore, an adaptive control algorithm is studied which lowers the supply temperature according to the actual heating demand. The study includes a variation of control parameters. The different configurations are evaluated with respect to their efficiency (seasonal performance factor) and the comfort (room temperature).

In systems with correctly parametrized heating curve controlling the room temperature is likely to be too high because of inner loads and solar gains. Instead of dealing with these gains by lowering the mass flow using thermostatic valves, the supply temperature can be dropped. This has a positive effect on heat pump efficiency because it decreases the total temperature lift. The control algorithm adapts the supply temperature in discrete time steps depending on the position of the thermostatic valve. Special attention has to be paid for the resulting room temperature and its deviation.

With the control algorithm presented in this paper, the seasonal performance factor can be increased by up to 0.19, depending on the allowed variability of room temperature. Savings in annual primary energy demand compared to a standard controlling are up to 6.8%.

1. Introduction

Electrically driven compression air-to-water heat pumps (AWHP) in existing buildings often have an insufficient efficiency with respect to primary energy demand when compared to condensing gas boilers. This was demonstrated by the evaluation of a field test in Germany [1]. Nevertheless heat pumps do offer the potential to save primary energy compared to condensing gas boilers. This has been shown in the same study evaluating brine-to-water heat pumps.

The efficiency of heat pumps achieved in tests with static boundary conditions is hardly attained under real operation conditions, especially in existing buildings with radiator heating systems. Apart from the option of optimizing the component heat pump, meaning the refrigerant cycle, there is an option to optimize the heat pump system. In this paper we focus on the second option, the optimization of the system consisting of heat source, heat pump and heat sink. An AWHP system is analysed in detail and we only study the heating of the building excluding domestic hot water generation.

The temperature lift directly influences the efficiency of the heat pump. Ground coupled or brine-to-water heat pump systems have higher mean source temperatures than AWHP systems. This results in lower temperature lifts and higher annual efficiencies. Analogously, heating systems such as floor heating with lower supply temperatures than radiator heating systems, provide smaller temperature lifts for the heat pump leading to higher efficiencies.

In the case of retrofit, buildings are often equipped with radiator heating systems that need relatively high supply temperatures. This is one reason for low heat pump efficiencies. In Germany, the supply temperature of the heating system is normally controlled according to the ambient air temperature. It is set in a way that with nominal mass flow rate and temperature spread the chosen heating system delivers a heat flow that matches the room's heat load at the according ambient temperature. The heating curve usually is set once for nominal conditions when the heat generator system is installed. Thermostatic valves adapt the mass flow rate according to differing load conditions.

More advanced control strategies adapt the supply temperature according to the load conditions and are generally described in Ref.

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[2]. The algorithm proposed by Kähler and Ohl [3] works in discrete time steps and load conditions are taken into account by measuring the position of thermostatic valves. A detailed description of this control method is given by Kraft [4]. A study evaluates these systems [5], where energy savings through lower heat losses in pipes are analysed. These heat losses can be reduced, but attention has to be paid to higher mass flow rates leading to higher return temperatures.

The effect of influencing the efficiency of a heat pump has been studied, too. Seifert et al. shows that an adaptive control of the supply temperature can increase heat pump efficiency [6]. According to his results, for a well-insulated building the SPF can be increased by approximately 0.2. Focussing on cooling applications, Beghi and Cecchinato present an adaptive supply temperature control [7]. They use the chiller supply and return temperature for load estimation. This estimation is low-pass filtered to avoid unstable control.

In this paper, an adaptive supply temperature control is applied to a heat pump system in an existing one-family house.

2. Background

2.1. Evaluation of electrically driven compression heat pumps

The type of heat pump analysed in this paper is the electrically driven compression heat pump. It can be considered as the standard heat pump built into one family homes. Its actual efficiency can be described by the dimensionless value COP (Coefficient of Performance). The COP is defined as the quotient of the usable heat flow rate $\dot{Q}_{\text{use}}$ and the total electrical power input $P_{\text{el,tot}}$.

As widely known, the Carnot efficiency describes the maximum efficiency of a working fluid cycle. It also describes the general dependence of the efficiency on the temperature lift, meaning the temperature difference between the temperature of heat gain T_{source} and temperature of heat output T_{use}. It assumes isentropic pressure changes and isobaric and isothermal heat transfers. The definition for a heat pump is as follows:

$$\text{COP}_\text{C} = \frac{T_\text{use}}{T_\text{source} - T_\text{use}} \tag{1}$$

In an AWHP system with a radiator heating system the temperature lift is

$$(T_\text{source} - T_\text{use}) = T_\text{su} - T_\text{amb} \tag{2}$$

where T_su is the supply temperature of the heating system and T_amb is the ambient air temperature.

The quality grade η_C is the COP dived by the COP_C. It lies in a range of 0.35–0.50 today, depending on the type of heat pump and the application (cf. [8]). In the already mentioned field test the quality grades for heating operation give mean *annual* values of 0.30 for AWHP [9]. The reason for the differences between COP and COP_C are the non-isentropic processes of compression and expansion, pressure losses in heat exchangers, the effective temperature lift being higher in real processes because of heat exchange to the source and sink, and the necessity of using auxiliary appliances. AWHP need a defrosting procedure which lowers the COP at low ambient air temperatures and high air humidities. In real working fluid cycles, the working fluid also limits the possible efficiency.

Fig. 1 shows the characteristic of the quality grade of the AWHP analysed in this paper. It is a heat pump that was, among others, used in the field test. It uses the working fluid R404A. The COP values are taken from the heat pump test centre in Buchs, Swiss [10]. They are extrapolated respectively interpolated linearly. The values are determined according to the European standard EN 255 [11], which means that the power for defrosting is included. At ambient air temperatures lower than 5 °C, the quality grade decreases with higher supply temperatures.

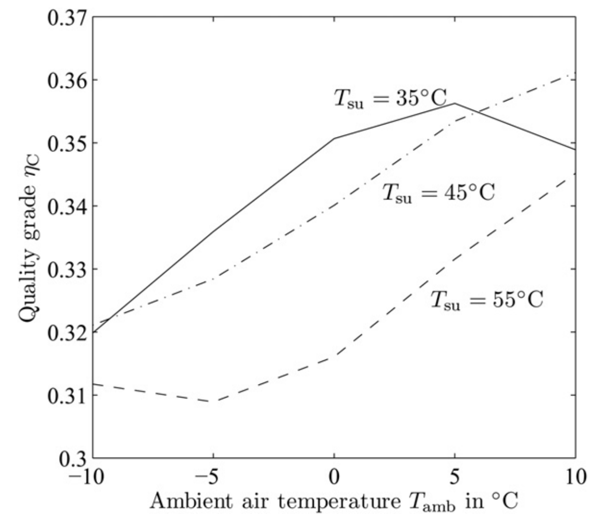

Fig. 1. Dependence of the quality grade on the ambient air and supply temperature of the analysed AWHP.

In usual applications of heat pumps, the efficiency at one operation point does not predict its efficiency for a longer operation. The source temperature as well as the sink temperature are changing (example: ambient air temperature in summer and winter, supply temperature control by a heating curve). Therefore, the heat pump is evaluated by the performance factor PF. It is the quotient of the usable heating energy Q_use and the total electric energy $W_\text{el,tot}$. The calculation of the PF for one year is called SPF (seasonal performance factor). The control volume for the COP and PF is defined in Fig. 2. The energy used for loading pumps for buffer storage is included. This control volume assumes that the storage is located outside of the thermal building envelope. This means that the storage heat losses are not used for heating. In Germany, the heat pump and storages are often located in a part of the basement which is not heated. The electric power consumption of drives on the source side of the heat pump (such as ventilators for AWHP) and an electrical heating rod are included in the control volume as well.

Comparing only heat pump systems that use electricity with the same primary energy factor and considering the same amount of heat delivery, savings in primary energy can be calculated by:

$$\frac{\Delta\text{PE}}{\text{PE}} = \frac{\text{PE}_\text{ref} - \text{PE}}{\text{PE}_\text{ref}} = 1 - \frac{\text{SPF}_\text{ref}}{\text{SPF}} \tag{3}$$

where PE_ref and SPF_ref are the annual primary energy demand respectively SPF of a reference system. PE is the annual primary

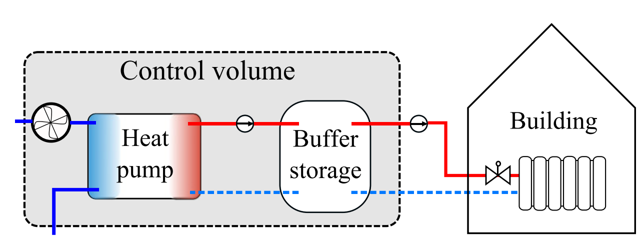

Fig. 2. Scheme of the analysed AWHP and control volume for its evaluation.

energy demand of the system that is compared to the reference system. In the results of this paper the annual heat delivered by the heat pump Q_{use} differs between the configurations because of slightly differing room temperatures depending on control set-ups.

2.2. Determination of supply temperatures

Radiator heating systems are generally designed to deliver a heat flow rate matching the nominal heat load of the building. The nominal heat load is the heat load at the nominal ambient temperature which lies between −16 and −12 °C in Germany, depending on the region. During the design process, the nominal heat load is determined for every room of the building and the radiator for each room is chosen accordingly. The radiator has a sufficient power to deliver the nominal heat load of the room using the nominal supply temperature $T_{su,nom}$ and the nominal temperature spread [$T_{su,nom} - T_{re,nom}$] (with $T_{re,nom}$ as the nominal return temperature). The nominal temperature spread implicates a nominal mass flow, fitting the design conditions of the radiator. The heat generator power is the sum of all nominal radiator heat loads and the circulating pump can be dimensioned by summing all nominal radiator mass flows (considering the pressure difference needed).

Commercially available heat pump control systems, which are often implemented into the heat generation unit itself, usually contain an ambient air temperature control of the heating water supply temperature. The so called *heating curve* copes with the fact that the building's heat load decreases with rising ambient temperatures. By lowering the supply temperature the mean logarithmic temperature difference between the heating fluid inside the radiator and the room air volume ΔT_{log} decreases. This results in a lower heat flow rate which is transmitted to the room (assuming a constant mass flow rate in the radiator) and can be described by the following equation [12]:

$$\frac{\dot{Q}}{\dot{Q}_{nom}} = \left(\frac{\Delta T_{log}}{\Delta T_{log,nom}}\right)^n \tag{4}$$

$\dot{Q}$ is the radiator heat flow rate, n is the radiator index, which is approximately 1.3 for standard radiators. The subscript 'nom' indicates the nominal value. The mean logarithmic temperature difference is defined by

$$\Delta T_{log} = \frac{T_{su} - T_{re}}{\ln\left(\frac{T_{su} - T_{room}}{T_{re} - T_{room}}\right)} \tag{5}$$

Here T_{su} and T_{re} are the supply and return temperature of the heating fluid and T_{room} is the room air temperature.

Besides the ambient temperature, inner loads and solar gains influence the heat load of a building. Thermostatic valves adapt the mass flow rate to control the heat flow rate of each radiator avoiding an increasing room temperature. For heating of German one family homes with water based heating system and a central heat generator, this type of control is usually preferred to a room temperature control, because still each room temperature can be controlled individually.

Fig. 3 shows a typical control characteristic of a thermostatic valve. At the room set temperature, the valve position is $x_{nom} = 0.5$ which allows the nominal mass flow rate to pass the valve. Above and near to the position 0.5 the characteristic is linear. The proportional range is the difference in room temperature between the point at which the valve is totally closed, $x = 0$, and the nominal position $x_{nom} = 0.5$.

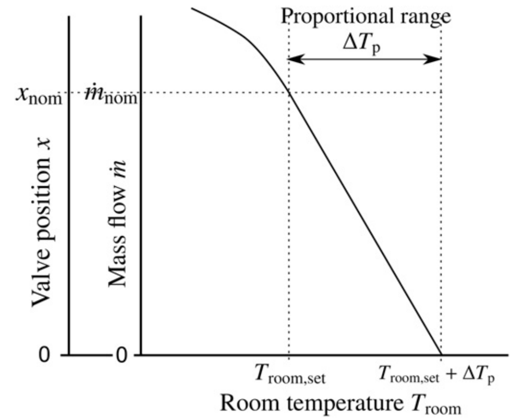

Fig. 3. Characteristic of the thermostatic valve.

2.3. Supply temperature adaption

Another possibility to adapt to changing heat loads is to adapt the supply temperature instead of the mass flow. An overview to such control strategies has been given in Section 1. Their key part is the load estimation. The algorithm studied in this paper uses the positions of the thermostatic valves, which are an indication for the heat demand of each room.

When all thermostatic valves in a heating system are in a position below x_{nom}, the supply temperature can be lowered down to a value at which one valve position gets x_{nom}. This kind of system needs a communication interface with the (electronic) thermostatic valves [2]. To ensure all rooms of the building to be heated properly, the maximum valve position is taken as reference to the supply temperature adaption. The adaption process has to be designed carefully to prevent instabilities [3]. This is achieved through a delayed adaption process (cf. [2]) implemented either through a filter element or through a discrete algorithm (as described by Kraft, who proposes an adaption of 1 K in 20 min intervals [4]). A controlling strategy as described here will be analysed in Section 4.2 of this paper.

3. Modelling and parametrization

Modelling and simulation are done using the object-orientated language Modelica in combination with the software Dymola [13,14]. The Modelica model libraries developed at the Institute for Energy Efficient Buildings and Indoor Climate are used for this work (cf. [15]).

The building models represent the building physics, weather and user influences. The library contains multilayer walls, windows and doors including the phenomena involved such as heat conduction, convection and radiation. The air volume of the room is calculated by the medium models of the Modelica.Media library.

The building services installation library contains components, such as pumps, pipes, boilers and valves. All components use medium models of the Modelica.Media and models of the Modelica.Fluid libraries [14].

The libraries allow a detailed modelling of the whole thermo-hydraulic system including the heat source, heat pump, water storages and heat sink – the building. The modelling is acausal which means that the interaction of all system components in the heat pump system are considered. The models were validated with data of the already mentioned field test where single model components were tested with time series from measurements and could well reproduce the behaviour of the real component [16].

3.1. Heat pump

The heat pump model is implemented as a black box model consisting of two heat exchangers that are connected to a module that calculates the heat flows and compressor power by look-up tables using manufacturer data. Generally this data is given at working points standardized by European standards EN 255 or EN 14511 [11,17]. Here, the heat pump model is parametrized with data available from static standardized tests (cf. [18]). The data is taken from a heat pump that was part of the aforementioned field test and that the model could be validated for. The characteristic concerning the quality grade of the heat pump has already been shown in Fig. 1 and the characteristic regarding the heat output at different temperature lifts is shown in Fig. 4.

The temperature spread of the heating water ($T_{su} - T_{re}$) in the condenser is usually 5 K or 10 K at test conditions. Temperature spreads differing from those at test conditions are implemented by linear functions according to the German guideline VDI 4650 [19].

The heat output of a heat pump decreases with higher temperature lifts. With low ambient temperatures and high supply temperatures the heat output is the lowest. Here, a bivalent layout with a heating rod as additional heat source is chosen. Fig. 4 shows the heat load of the building and the heat pump heat output at different supply temperatures. The bivalent point, which is the point where the characteristic of the heat pump output crosses the characteristic of the building heat load, is −6 °C.

3.2. Buffer storage

As the heat pump is on-off controlled and cannot modulate its heat output according to the actual heat demand, a buffer storage with a volume of 500 l is included. It increases the inertia of the heating system and reduces the necessary operating intervals of the heat pump to avoid wear-out of the compressor. Most heating is done at ambient temperatures higher than 0 °C. Fig. 4 shows that especially AWHP strongly operate in partial load conditions.

The buffer storage model consists of several fluid volumes representing fluid layers. The layers are connected to each other allowing heat and fluid flow. Buoyancy effects are taken into account by an effective heat conductance depending on the temperature differences between the layers. The turbulent heat conductance is a function of the layer thickness, its temperature and the temperature difference to the above layer. It is based on the work of Viskanta et al. [20]. The buffer storage model has fluid inlets and outlets in the top and bottom layers. The number of fluid layers in the model can be calibrated to match the quality of stratification in the real storage.

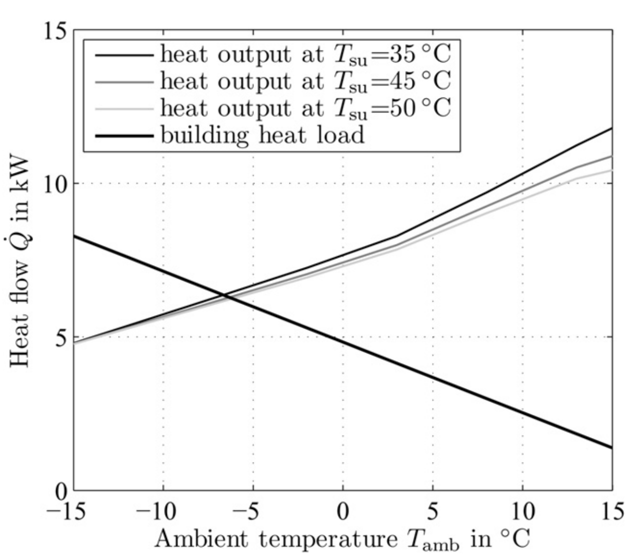

Fig. 4. Heat load of the building depending on the ambient air temperature, heat output of heat pump at different supply temperatures.

3.3. Building

A four zone building model is analysed. Each zone has two windows sighting to two directions and the windows to the south side are bigger than the other windows. The air exchange ratio is 0.5 1/h. All calculations use the German Test Reference Year no. 5 as weather data (geografic region of the city of Essen) [21]. The thermal data of walls and windows corresponds to the German heat conservation ordinance of 1984. Main thermal properties of the building are summarized in Table 1.

The building has an annual heat demand of 14,342 kWh and a nominal heat load of 7.67 kW at an ambient temperature of −12 °C. The heat load characteristic is shown in Fig. 4. A standard heat load calculation for each room results in nearly equal values for each of the four zones. Each zone is equipped with the same radiator type though. Each radiator respectively zone is equipped with a thermostatic valve. As solar gains and heat losses due to wind convection differ in each room, dynamic effects are studied in the hydraulic heating system. Also, this is why the annual heat demand and heat loads will differ from the nominal values in the results of dynamic simulations (cf. Section 4).

3.4. Overall system model and its controlling

The system is set up as shown in Fig. 2 consisting of the heat pump, a buffer storage and the building. It is completed by a loading pump, a heating circuit pump and the heating system with pipes, valves and a radiator in each zone of the building. An electrical back-up heating delivers the heating power lacking at ambient temperatures below −6 °C when the characteristic of the heat pump is below the characteristic of the building (cf. Fig. 4).

The heating curve is determined according to the heat load characteristic using Equations (4) and (5). It is shown in Fig. 5. For daytime, which is set from 6 am to 10 pm, the room temperature is calculated as 21 °C, at night 17 °C. Above temperatures of 15 °C (day) and 7 °C (night) the heat pump is not operating. The heating period is from 1st of September until 15th of May. The rest of the year the heat pump is set off, too.

The heat pump starts to operate when the buffer storage top temperature falls below the supply set temperature $T_{su,set}$ by a temperature difference ΔT_{ctrl}. It turns off when the bottom temperature reaches $T_{su,set}$. The buffer storage being loaded from top to the bottom the top temperature is higher than $T_{su,set}$ at the end of a loading cycle.

Table 1
Thermal data of the building model, static calculation for a nominal ambient air temperature of −12 °C, room temperature of 21 °C and a degree days figure of 84.4 kKh/a.

Air exchange rate	0.5	1/h
Overall heat transfer coefficient of wall	0.42	W/(m^{2}K)
Heat loss coefficient of walls	86.2	W/K
Heat loss coefficient of windows	44.2	W/K
Total heat loss coefficient of ventilation	102	W/K
Total heat loss coefficient	232.4	W/K
Nominal heat load	7.67	kW
Annual inner gains	3485	kWh/a
Annual solar gains	1799	kWh/a
Annual heat demand	14,342	kWh/a

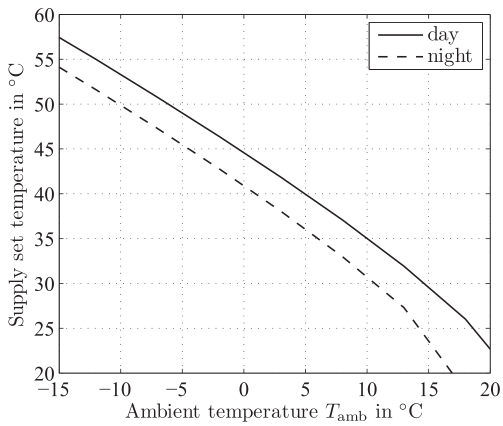

Fig. 5. Heating curve for room temperature of 21 °C (day) and 17 °C (night setback).

4. Results

4.1. Standard AWHP system configuration, variation of the heating curve and control hysteresis

The heat pump system is simulated for the heating curve described in Fig. 5 with a ΔT_{ctrl} of 2 K. This standard AWHP system configuration (configuration 1) has an SPF of 2.65. The monthly performance factors MPF, usable heating energy Q_{use} and electric work W_{el} are shown in Fig. 6. During heat pump operation the supply temperature rises above the supply set temperature given by the heating curve $T_{su,set,hc}$. And it falls below $T_{su,set,hc}$, when it is not operating. The actual supply temperatures simulated are shown in Fig. 7 as hourly mean values for hours from 7 am to 10 pm, compared to $T_{su,set,hc}$.

The heating curve then is dropped by 1 K (conf. 2) and raised by 1 K in another calculation (conf. 3). Table 2 shows the results for these two calculations compared to the system with the standard heating curve (conf. 1). The mean room temperature $T_{room,mean}$ is the mean value for all four rooms, whereas σ_{Troom} is the according standard deviation. Fig. 8 shows the monthly mean supply temperatures being dropped respectively raised by approximately 1 K. It is not exactly 1 K as dynamic effects do occur. The supply temperature varies according to the heat pump turning on and off. The supply temperature being higher (conf. 3), the return temperature slightly decreases.

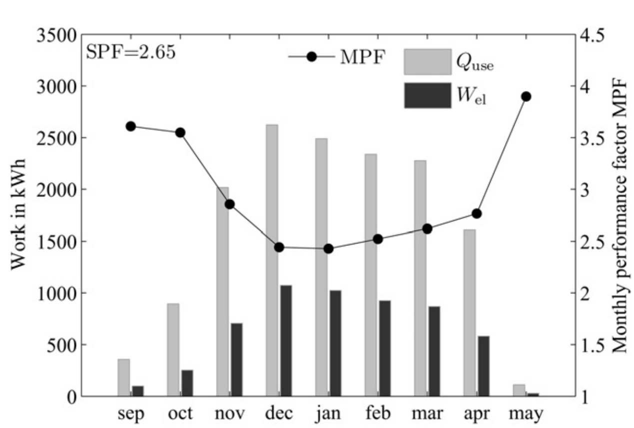

Fig. 6. Configuration 1: Usable heating energy Q_{use}, electric work W_{el} and monthly performance factors MPF within the heating season.

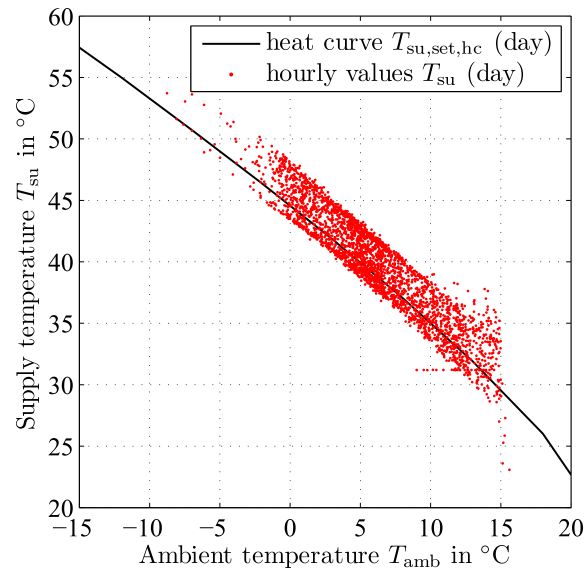

Fig. 7. Configuration 1: Heating curve for daytime and actual supply temperature hourly values for daytime (7 am–10 pm).

For conf. 2 it is vice versa. The temperature spread ($T_{su} - T_{re}$) is smaller, caused by the higher mass flow rate in the heating system, being controlled by the thermostatic valves. At the beginning of each heat pump operating interval the thermostatic valves are fully opened, meaning a high mass flow rate. When operating intervals are as short as it is in transition season, the effect described above is weaker. This can be seen in the diagram for the months of September and May.

The actual heat load of the building is lower than the calculated heat load in most of the time because of internal loads and solar gains. The heating system is designed to supply the calculated heat load with the supply temperature set by the heating curve. This leads to higher room temperatures and lower valve positions. Even with the heating curve dropped by 1 K the mean room temperature is higher than the room set temperature. Fig. 9 shows that the thermostatic valves don't control the room temperature exactly to the set temperature because of their linear control logic. The figure describes the influence of the supply temperature on the room temperature. The mean room temperature being different in each configuration results in different annual heat demands. The SPF for conf. 2 and 3 is approximately 0.05 higher respectively lower than the one of conf. 1. This verifies the conclusion made in Section 2.1 about the efficiency depending on the supply temperature.

An important parameter for the controlling of the heat pump is the temperature difference ΔT_{ctrl}. With ΔT_{ctrl} set to 5 K instead of 2 K in conf. 4 (cf. Table 2) the mean supply temperature drops approximately 1 K below the value of configuration 1 and thus the SPF equals the one of conf. 2. Fig. 10 shows, that the variation of supply temperature is higher than in conf. 1. The number of annual operating intervals depends on ΔT_{ctrl}. It is approximately 1500 for the systems with $\Delta T_{ctrl} = 2$ K and approximately 1000 for the configurations with $\Delta T_{ctrl} = 5$ K. It slightly differs with other control parameters being changed as described in the following.

4.2. Simulation of adapted supply temperature

An adaptive supply temperature controller is analysed in the configurations 5–13. The control algorithm is implemented as described in Section 2.3. Though, to ensure a fast heating up after

Table 2
Control configuration (left) and results (right side of the table) for systems 1–13. Room temperature is analysed between 7 am and 10 pm and in heating season.

Conf.	Δt in s	$\Delta T_{drop,max}$ in K	ΔT_{drop} in K	ΔT_{rise} in K	x_{low}	x_{up}	ΔT_{ctrl} in K	SPF	Q_{use} in kWh	$T_{room,mean}$ in K	σ_{Troom} in K
1	Standard heating curve as in Fig. 5						2	2.65	14,711	21.66	0.46
2	Standard heating curve as in Fig. 5 − 1 K						2	2.70	14,560	21.56	0.49
3	Standard heating curve as in Fig. 5 + 1 K						2	2.59	14,823	21.72	0.43
4	Standard heating curve as in Fig. 5						5	2.70	14,465	21.49	0.54
5	1200	5	1	2	0.30	0.50	2	2.79	13,948	21.10	0.63
6	1800	5	1	2	0.30	0.50	2	2.80	13,972	21.11	0.63
7	3600	5	1	2	0.30	0.50	2	2.80	13,985	21.13	0.61
8	1200	5	1	2	0.40	0.60	2	2.83	13,698	20.92	0.70
9	1800	5	1	2	0.40	0.60	2	2.83	13,719	20.93	0.70
10	3600	5	1	2	0.40	0.60	2	2.84	13,738	20.95	0.70
11	3600	5	1	2	0.30	0.50	5	2.78	13,866	21.04	0.64
12	3600	6	1	2	0.30	0.50	2	2.80	13,907	21.07	0.62
13	3600	5	1	1	0.30	0.50	2	2.81	13,938	21.09	0.62

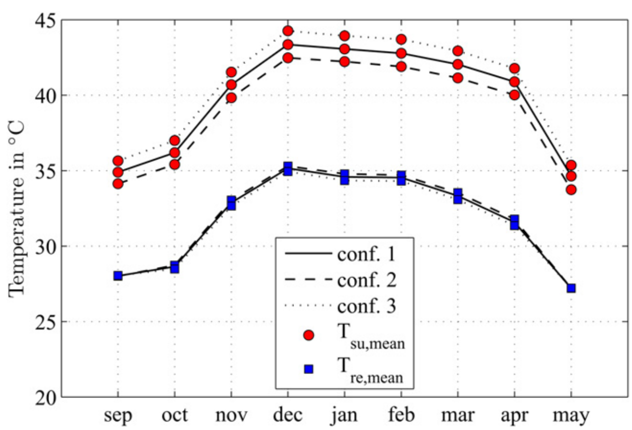

Fig. 8. Configurations 1–3: Mean supply temperatures $T_{su,mean}$ and return temperatures $T_{re,mean}$ at heat pump operation within the heating season.

the night temperature setback, from 6 to 8 am the set temperature is always set to the heating curve value.

The inputs to the controller are the positions x_i of all thermostatic valves in the system, which are four in this case. The supply set temperature is adapted according to the maximum value:

$$x_{max} = \max(x_i) \tag{6}$$

The controller works with several control parameters, which are described below: At a fixed time interval Δt the following logic is implemented: The supply set temperature $T_{su,set}$ is lowered by the value ΔT_{drop} if x_{max} is below the threshold x_{low}:

$$x_{max} < x_{low} \Rightarrow T_{su,set}(t + \Delta t) = T_{su,set}(t) - \Delta T_{drop} \tag{7}$$

The supply set temperature $T_{su,set}$ is increased by the value ΔT_{rise} if x_{max} is higher than x_{up}:

$$x_{max} > x_{up} \Rightarrow T_{su,set}(t + \Delta t) = T_{su,set}(t) + \Delta T_{rise} \tag{8}$$

If x_{max} is between the two thresholds x_{low} and x_{up}, the value of $T_{su,set}$ is not changed. Additionally $T_{su,set}$ is controlled to be set inside a range bounded through the heating curve and a value $\Delta T_{drop,max}$ below the heating curve value $T_{su,set,hc}$:

$$\begin{aligned} x_{low} < x_{max} < x_{up} \vee T_{su,set}(t) < T_{su,set}(t) - \Delta T_{drop,max} \\ \vee T_{su,set}(t) > T_{su,set,hc} \Rightarrow T_{su,set}(t + \Delta t) = T_{su,set}(t) \end{aligned} \tag{9}$$

Different combinations of these control parameters where studied and are listed in Table 2 including the results of the annual simulations. It can be seen that the adaptive control leads to higher SPF values than the standard heating curve control, but the mean room temperatures are lower and their standard deviations higher.

First, the interval Δt is varied from 20 min to 60 min for two boundaries of x_{max} (conf. 5–10). The boundaries are first set to $x_{low} = 0.3$ and $x_{up} = 0.5$ (conf. 5–7) which means that the valve

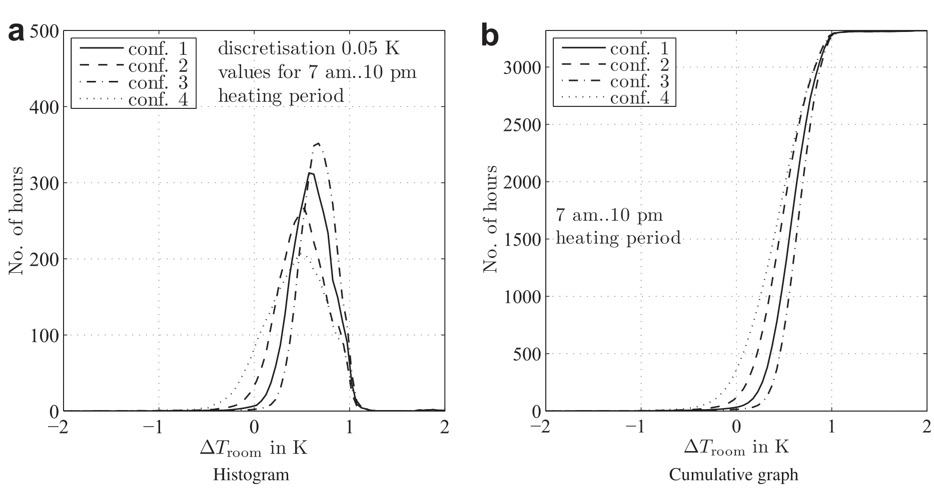

Fig. 9. Deviation of room temperatures for the standard heating curve (conf. 1), the heating curve dropped (conf. 2) and raised (conf. 3) by 1 K and the hysteresis control temperature difference set to 5 K instead of 2 K (conf 4). Values for daytime (7 am–10 pm) in heating season.

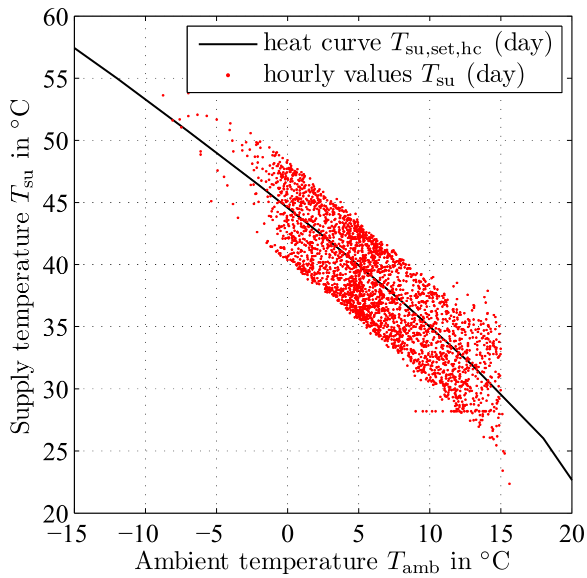

Fig. 10. Configuration 4: Heating curve for daytime and actual supply temperature hourly values for daytime (7 am–10 pm) in heating season.

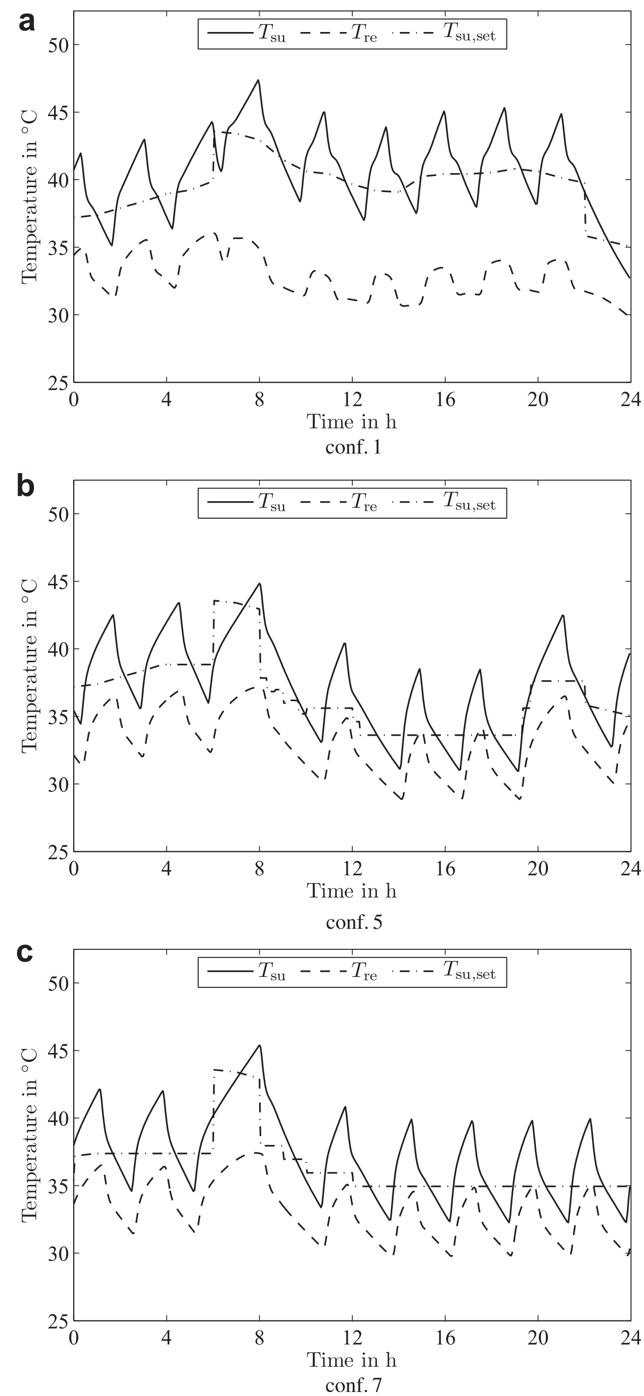

Fig. 11. Comparison of a daily progress of the supply temperature T_{su} and return temperature T_{re} relating to the supply set temperature $T_{su,set}$.

position shall stay within these boundaries corresponding to room temperatures above the room set temperature. In configurations 8–10 the boundaries are $x_{low} = 0.4$ to $x_{up} = 0.6$ which means that the valve operates around its nominal position and thus the mass flow in the system is approximately at its nominal value, too (cf. Fig. 3). This leads to the highest SPF values but the mean room temperature is below the set value.

When the room temperature falls too low, meaning the thermostatic valves opening to a position above x_{up}, the supply temperature is increased. To ensure a fast heating up, ΔT_{rise} is set to 2 K, i.e. it is higher than ΔT_{drop}. Simulations have shown, that setting ΔT_{rise} to 1 K leads to a lower mean room temperature and higher deviation within the heating season (cf. Table 2, compare conf. 13 to conf. 7). However, conf. 13 has only a slightly different room temperature.

The maximum drop of the supply temperature compared to the value given by the heating curve $\Delta T_{drop,max}$ is set to 5 K. Allowing higher drops leads to a higher SPF but it also leads to a lower mean room temperature and a higher deviation (cf. Table 2, compare conf. 12 to conf. 7). Just as in Section 4.1 one configuration is simulated using a ΔT_{ctrl} of 5 K (conf. 11). Comparing it to conf. 7, it has a lower SPF and a lower mean room temperature. It shows that especially with the adaptive control ΔT_{ctrl} is important. It is linked with the accuracy of the supply temperature.

Fig. 11 shows the daily run of the supply, the return and the supply set temperature. In the first chart for conf. 1 it can be seen that the night temperature setback ends at 6 h and begins at 22 h. Each peak of supply temperature corresponds to an operating interval of the heat pump. It generally begins, when the supply temperature is approximately 2 K below the supply set temperature (not exactly, as it is the buffer storage temperature, which is controlled accordingly).

The supply temperature deviates strongly with each operating interval of the heat pump. This is the reason for eventual instabilities of the controlling, if Δt is too small. It is shown in Fig. 11(b) for conf. 5. As mentioned before, the supply set temperature is set to the heating curve values after the night setback for two hours. Afterwards the adaptive controlling operates. At about 19 h the decisive valve position shortly rises above the upper boundary. The supply set temperature increases for two control intervals Δt. For Δt being set to 60 min, conf. 7 in Fig. 11(c), the controlling is more stable.

The highest SPF combined with good values for the room temperature is reached with the control interval of $\Delta t = 60$ min and the boundaries of the valve position set to 0.3 and 0.5 as in conf. 7. Fig. 12 shows hourly values of the supply temperature of conf. 7 compared to the heating curve. In contrast to the characteristic of conf. 1, a much more deviating supply temperature is attained, whereat only a small amount of the values lies above the heating

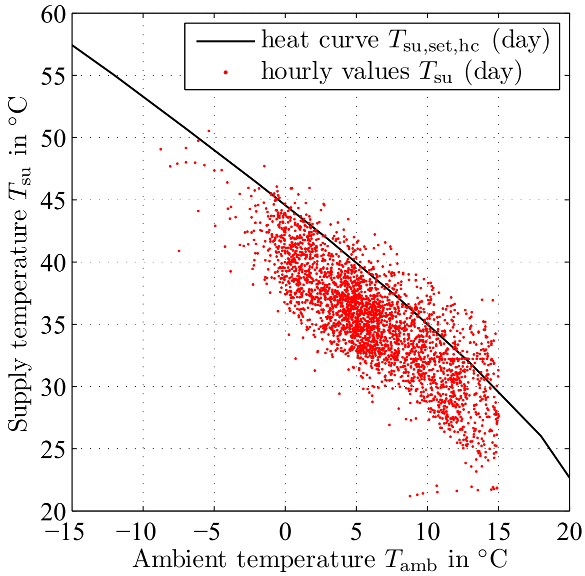

Fig. 12. Configuration 7: Heating curve for daytime and actual supply temperature hourly values for daytime (7 am–10 pm) in heating season.

curve. This causes the room temperature to vary much more than in the standard control configuration, which can be seen in Fig. 13.

5. Discussion and outlook

The distinct influence of the mean supply temperature on the efficiency of an air-to-water heat pump system with radiator heating has been demonstrated. A heating curve decreased by 1 K leads to savings in primary energy of 2.0%.

The adaptive supply temperature control for domestic heat pump systems presented in this paper can save up to 6.8% of primary energy compared to an ideally set standard controller. The SPF is respectively augmented by 0.19. In a standard case the room temperature is likely to be higher than the room set temperature because of solar gains and inner loads. The controller presented here leads to a mean room temperature which is near the room set temperature. As in reality heating curves are often not ideally set in heat pump systems, the controller might achieve higher savings.

To avoid controlling instabilities which effect the room temperature of the heated zones, the adaptive controller has to be parametrized properly. Governing control parameters are the time step of the adaption algorithm and the thresholds of the thermostatic valve positions that determine whether and in which direction to adapt the supply temperature. The supply temperature hysteresis which controls the heat pump to start and stop its operation has a more important role than in a system with standard control.

An adaption of the supply temperature in the analysed air-to-water heat pump system shows good results for 1 h-time steps of 1 K for the supply temperature drop and 2 K for the increase. It leads to an SPF augmented by 0.15 (compared to the standard case) which means savings in primary energy demand of 5.6%. The mean room temperature at daytime is 0.1 K above the room set temperature instead of 0.7 K in the case of the standard heat pump controlling. By tuning the control parameters, the SPF of the heat pump system can be increased even more. But that cannot be done without lowering the comfort.

Further analyses have to cope with the fact that real buildings and heating systems are more complex than the building model analysed in this work. Here, the building model is a four-zone-model with a basic hydraulic system. In further works a more realistic heating system will be analysed.

A combination with ground coupled heat pumps shall be analysed, whereat effects are expected to be comparable to those found here. Systems that operate without a buffer storage are expected to benefit from an adaptive supply temperature control, too. These are either systems with an inert heating system such as floor heating systems or capacity controlled heat pump systems e.g. heat pumps with a speed controlled compressor. These systems will be part of further research and are expected to have a smaller deviation of room temperature because of less varying supply temperatures.

An intelligent control system learning from and adapting to user presence is to reveal higher savings than with the control system presented in this paper. A controller which uses a continuous mean value of the valve position is also going to be studied.

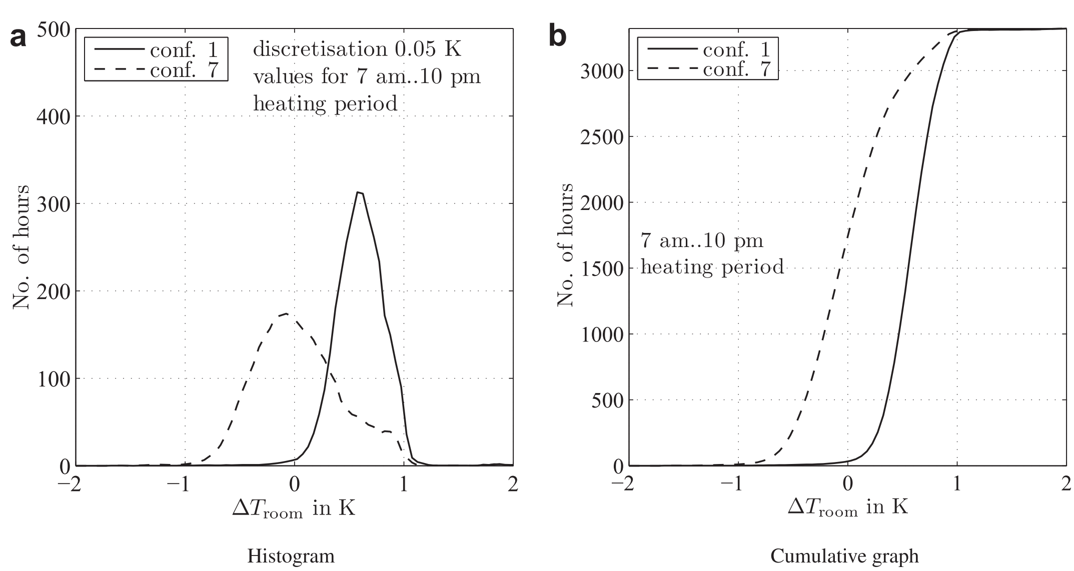

Fig. 13. Deviation of room temperatures for the standard configuration with heating curve (conf. 1) and the adaptive supply temperature control (conf. 7). Values for daytime (7 am–10 pm) in heating season.

6. Conclusion

In this paper the calculation of an air-to-water heat pump system consisting of heat source, heat pump and heat sink has been presented. The model libraries of the Institute for Energy Efficient Buildings and Indoor Climate have been successfully used for the purpose of comparing different control strategies and their influence on the hydraulic heating system and the indoor comfort. A control strategy has been presented that adapts the supply set temperature according to the actual load. Load estimation is done using the positions of the thermostatic valves.

Compared to an ideally adjusted air-to-water heat pump system with radiator heating and a standard controlling, up to 6.8% of primary energy can be saved with the new control strategy. The simulation of different parametrizations showed the distinct influence of the controller setting on the heat pump efficiency. But also the thermal comfort in the heated rooms is influenced. Further analyses will study the combination of the adaptive control combined with other heat pump technologies, system optimizations and control strategies.

Nomenclature

Abbreviations

AWHP	air-to-water heat pump
conf.	configuration
COP	coefficient of performance
MPF	monthly performance factor
PE	primary energy demand
PF	performance factor
SPF	seasonal performance factor

Symbols

$\dot{Q}$	heat flow rate, W
η	efficiency
σ	standard deviation
n	radiator index
P	power, W
Q	heat, J
T	temperature, K
W	work, J
t	time

Subscripts

act	actual
C	Carnot
ctrl	control (temperature difference)
drop	dropping
el	electrical
hc	heating curve
log	logarithmic
low	lower boundary
mean	mean
nom	nominal
re	return
ref	reference system
rise	rising
room	Room
room	room
set	set value
source	source
su	supply
tot	total
up	upper boundary
use	usable

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