ADAPTIVE SUPPLY TEMPERATURE CONTROL FOR DOMESTIC HEAT GENERATORS

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ABSTRACT

Domestic heating systems often work with too high supply temperatures. This means that heat generators and especially heat pumps work with a lower efficiency than possible.

The influence of the supply temperature on the efficiency of an air-to-water heat pump is discussed. An adaptive supply temperature control is presented that operates according to the heat load of the building whereas information of electronic thermostatic valve heads are used.

The control algorithm is studied through parameter study with annual simulations of an air-to-water heat pump system. A building model with four zones, each equipped with a radiator and a thermostatic valve, the hydraulic system and the heat generator are implemented. The acausal modeling with Modelica considers interaction of the different components. The simulations are carried out using the software Dymola.

The results show that the efficiency of the heat pump can be increased through the usage of adaptive control.

INTRODUCTION

Advanced control strategies for buildings in recent years have become one key element of research in the energetic optimization of HVAC (heating, ventilation, air conditioning) systems. Most of the presented control strategies require a deep understanding and knowledge of the particular system.

In new buildings and newly designed systems, this knowledge often exists which allows for exactly fitted control parameters and for model based predictive control (cf. Prívara et al. (2011); Kim and Braun (2012); Schmidt et al. (2010)). However, the application to existing buildings often is difficult, as many parameters are not known or can not be attained by reasonable effort. This is why Bianchi (2006) presents a predictive control for heat pumps that works with a simple building model whose parameters are found automatically.

Typical German heating systems work with heat generators that operate supply temperature controlled by a heating curve as shown in figure 1 (without the part in dashed lines). The heating curve is a feed-forward control that works with the ambient temperature as input (Thomas et al., 2005). Since there are more influences on the heat load than the ambient temperature (see the disturbances in the functional diagram), in typical German heat distribution systems the mass flows in radiators are controlled with thermostatic valves. They use a feed-back control of the control variable, the room temperature. This kind of control allows for a control of multiple heated zones. A feed-back of the room temperature to control the supply temperature usually is implemented for one zone or a building with a main zone (cf. Seifert and Knorr (2011)). Liao and Dexter (2005) developed a control using boiler information and measurement of solar radiation to estimate the room temperature in the building.

Thus, even in systems with correctly parametrized heating curve controlling and thermostatic valves for each heated zone 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 could be dropped. This can have a positive effect on the efficiency of heat generators. Considering that often heating curves are not correctly parametrized, the effect of controlling the supply temperature in line with the heating demands might bring additional benefits.

Especially heat pumps do benefit from low supply temperatures as their efficiency strongly depends on the temperature lift. This applies for air-to-water heat pumps (AWHP), which are commonly applied for retrofit (Brugmann, 2006). In an AWHP system with a water based radiator heating system the temperature lift is

$$(T_{\mathrm{use}} - T_{\mathrm{source}}) = \vartheta_{\mathrm{su}} - \vartheta_{\mathrm{amb}} \tag{1}$$

where ϑ_{su} is the supply temperature of the heating system and ϑ_{amb} is the ambient (outdoor) air temperature. The Carnot efficiency describes the maximum efficiency of a working fluid cycle by

$$\mathrm{COP_C} = \frac{T_{\mathrm{use}}}{(T_{\mathrm{use}} - T_{\mathrm{source}})} \tag{2}$$

The actual COP of a heat pump can be described by a share of $\mathrm{COP_C}$ using the quality grade η_{C} or by the quotient of used heat flow $\dot{Q}_{\mathrm{use}}$ and electric power input P_{el}:

$$\mathrm{COP} = \mathrm{COP_C} \cdot \eta_{\mathrm{C}} = \frac{\dot{Q}_{\mathrm{use}}}{P_{\mathrm{el}}} \tag{3}$$

η_{C} depends on the source and sink temperatures, too.

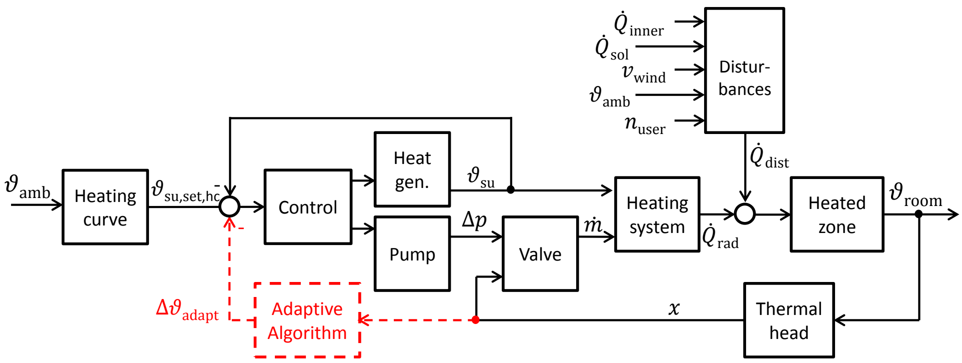

Figure 1: Functional diagram of a simplified heating system with one heated zone. The additional adaptive algorithm is marked by dashed lines.

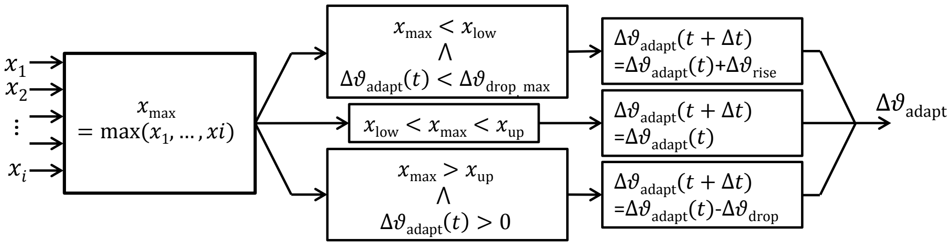

Figure 2: Functional diagram of the adaptive algorithm.

However, the effect in equation 2 dominates the heat pump efficiency.

Combining these considerations lead to the development of a control, that adapts the supply temperature according to the actual heating load of the building. The load estimation can be done by using the information of valve positions. This type of control is shown in figure 1 in dashed lines.

SUPPLY TEMPERATURE CONTROL

Several publications dealt with an adaptive supply temperature control (ASTC). Some describe the possible savings that can be achieved by theoretical evaluations (cf. Seifert and Knorr (2011)). Others describe implementations of ASTC algorithms in simulations (cf. Kraft (2002)) or even implementations in real controllers (cf. Kähler and Ohl (2009)).

The authors presented an adaptive supply temperature control that decreases the supply temperature according to the position of the thermostatic valves (see Huchtemann and Müller (2013)). 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.

The ASTC makes use of the following correlations: By lowering the supply temperature the mean logarithmic temperature difference between the heating fluid inside the radiator and the room air volume $\Delta\vartheta_{\text{log}}$ is decreased. 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 (Glück, 1990):

$$\frac{\dot{Q}_{\text{rad}}}{\dot{Q}_{\text{rad,nom}}} = \left(\frac{\Delta\vartheta_{\text{log}}}{\Delta\vartheta_{\text{log,nom}}}\right)^e \tag{4}$$

$\dot{Q}$ is the radiator heat flow rate, e is the radiator exponent, which is approximately 1.3 for standard radiators. The subscript 'nom' indicates the nominal value at the design ambient temperature.

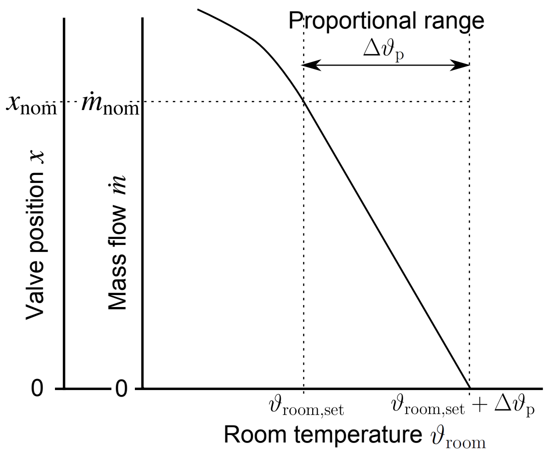

Figure 3: Characteristic of the thermostatic valve.

Figure 3 shows a typical control characteristic of a thermostatic valve. At the room set temperature, the valve position is $x_{\text{nom}} = 0.5$ 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 $x_{\mathrm{nom}} = 0.5$.

With the adaptive supply temperature control, in the functional diagram the part in dashed lines is added to the existing system (cf. fig. 1). In reality the system should feature electronic thermostatic valve heads to allow for a signal processing to the heat generator control. Besides that, no additional measurement of control variables is necessary. By using the valve position, an indirect feed-back of the control variable (the room temperature) is implemented.

As can be seen from the functional diagram, the control by the valve is still existing and it is necessary as multiple zones shall still be controlled. With this configuration there are two variables influencing the control variable. This can lead to an unstable behavior of the control which should be prevented by the adaptive algorithm.

The valves shall still deal fast with disturbances. In addition to this, the adaptive algorithm shall recognize possibilities to reduce the supply temperature. Though, it shall act slower than the valves.

It has to be considered, that even with capacity controlled heat generators at least in certain times of the year heat generators are operating intermittently. Depending on the thermal inertia of the heating system and the building, this causes the room temperature to oscillate and with that the thermostatic valves to oscillate.

This is why the algorithm works at discrete time steps Δt that are long enough to ensure a stable control behaviour. The adaptive algorithm presented here is described in figure 2. The inputs to the controller are the positions $x_1 \ldots x_i$ of all thermostatic valves in the system. The supply set temperature is adapted according to the maximum value of the positions to ensure this critical zone to be heated properly.

At each time interval the algorithm decides according to the valve position boundaries x_{low} and x_{up} if the adaption temperature difference $\Delta\vartheta_{\mathrm{adapt}}$ is lowered by $\Delta\vartheta_{\mathrm{drop}}$ or raised by $\Delta\vartheta_{\mathrm{rise}}$. A maximum adaption of $\Delta\vartheta_{\mathrm{adapt,max}}$ is allowed and $\Delta\vartheta_{\mathrm{adapt}}$ can not get negative, which means that an adaption towards supply temperatures higher than the one set by the heating curve is not allowed.

The algorithm has been studied with one heating system in the aforementioned study (Huchtemann and Müller, 2013). Here, the effect of the inertia of the heating system (by varying the buffer storage volume) and the speed of supply temperature adaption in an AWHP system is studied.

MODELLING, PARAMETRIZATION

Modelling and simulation is done using the object-orientated language Modelica in combination with the software Dymola (Dassault Systems, 2011; Modelica Association, 2011). The Modelica model libraries developed at the Institute for Energy Efficient Buildings and Indoor Climate are used for this work (cf. Müller and Badakhshani (2010)).

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. User influences are described by internal heat loads and variable air exchange.

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 Modelica Association (2011).

Cross correlations of weather, control, heating system and building structure are considered. E.g. the room air temperature is calculated dynamically according to the heat flows of the radiator and the ”disturbances” such as weather and user influences. This kind of modeling allows for an analysis of all parts of the building services system and their interaction and thereby enable the study of the control presented in this paper. However, slightly different room temperatures caused by different control strategies lead to different calculated heat demands.

The heat pump cycle and auxiliary appliances of the heat pump are represented by polynomial functions in the system simulation, whereas on the hydraulic side of the condenser the heat flow and pressure drop is implemented by models of the Modelica.Fluid library.

The polynomials are calculated using a static working fluid cycle calculation with data for R410a using medium models of Lemmon et al. (2002). Constant heat transfer coefficients in the evaporator and condenser are assumed going together with 5 K superheating and 5 K sub-cooling. The compressor characteristic is taken from manufacturer's polynomials. The electric power for the ventilator on the air side is implemented according to Gasser et al. (2008).

As the heat pump is on-off controlled and cannot modulate its heat output according to the actual heat demand, a thermal storage, usually called 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. Figure 4 shows that especially AWHP strongly operate in partial load conditions.

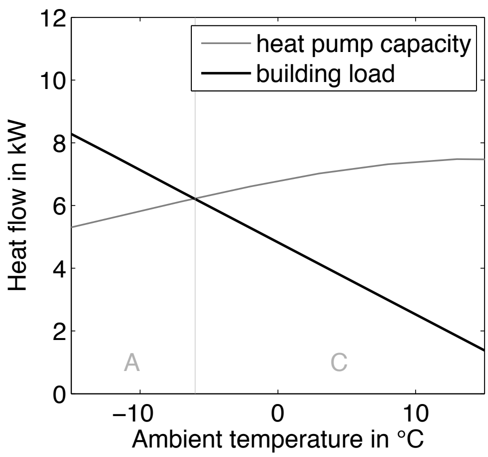

Figure 4: Dimensioning of the system with the on/off-controlled heat pump. The vertical line represent the boundary of operating domains: A: Bivalent operation; C: on/off-operation.

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. For the variation of the volume, in this paper, the storage is proportionally scaled (hydraulic connections and heights of temperature sensors included).

The heat sink, meaning the hydraulic heating system and the building physics are modeled as follows. A one family home is implemented with four zones. The total heat loss coefficient is 232.4 W/K. The building is located in the west of Germany (the German test reference year of region 5, City of Essen is used, cf. Christoffer et al. (2004)). The air exchange rate is set to 0.5 1/h. The nominal heat load is 7.67 kW at a nominal ambient temperature of -12 °C.

The heating curve is calculated according to the nominal supply temperature of 55 °C, the radiator exponent and the heat load characteristic of the building. The thermostatic valves are set to a target room temperature of 21 °C. The heating season is set from September 15th to May 1st. A heating boundary at an ambient temperature of 15 °C is implemented as well as a night setback between 10 pm and 6 am.

The system is equipped with an electrical back-up heater that works below the bivalent temperature. It is placed between the heat pump and the buffer storage. A rough scheme of the total system is given in fig. 5.

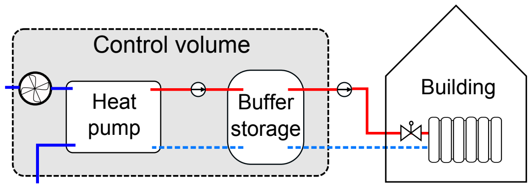

Figure 5: Scheme of the modeled heat pump system and control volume for calculation of performance factor.

RESULTS

For the reference system with the buffer storage of 500 l, similar results as in previous studies could be achieved (cf. Huchtemann and Müller (2013)), even though a different heat pump model was used, that achieves seasonal performance factor (SPF) values on a higher level. Table 1 shows the main results for the reference system. The control volume for SPF calculation includes the buffer storage and electrical pump energy for the loading cycle of the buffer storage. Q_{use} is the thermal energy transferred in the heating circuit. The SPF is calculated by

$$\mathrm{SPF} = \frac{Q_{\mathrm{use}}}{W_{\mathrm{el}}} \tag{5}$$

Table 1: Results of the reference system for one heating period.

SPF	Q_{use}	$\bar{\vartheta}_{\mathrm{room,day}}$
3.05	14665 kWh	21.6 °C

The results for similar systems with smaller (150 l) or larger (1000 l) buffer storage volumes do only differ slightly from the reference system. The smaller buffer storage leads to lower heat losses and slightly higher SPF. The main difference can be recognized in the number of operating intervals. A higher number of operating intervals would lead to a lower efficiency through cycle losses of a heat pump with thermal inertia in the working fluid cycle, which is not implemented here.

The focus here is on the implementation of the ASTC. With the adaptive control the total number of operating intervals is 10 to 15% lower than without. The number of operating intervals within one heating period is shown in figure 6 for different storage sizes. The adaptive algorithm is parametrized with the values shown in table 2. At the end of the night setback, the adaption is set to zero for two hours, to assure the heating up.

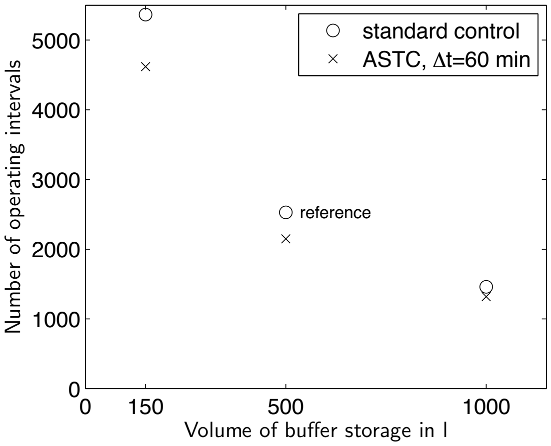

Figure 6: Number of operating intervals within one heating period.

Table 2: Parametrization of the adaptive algorithm (cf. figure 2).

x_{low}	0.3
x_{up}	0.5
$\Delta\vartheta_{\mathrm{drop}}$	1 K
$\Delta\vartheta_{\mathrm{rise}}$	2 K
$\Delta\vartheta_{\mathrm{adapt,max}}$	5 K

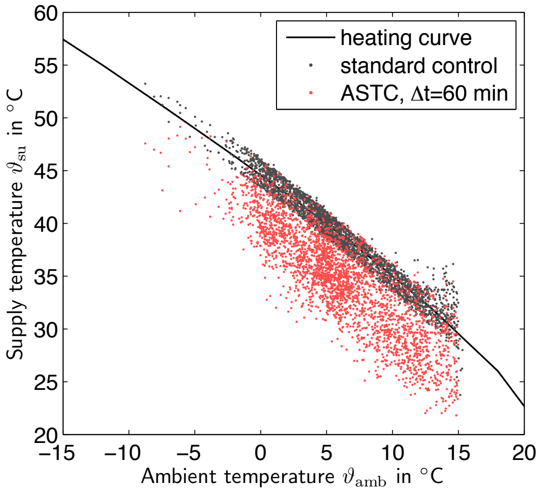

Figure 7: Hourly values of supply temperature.

Figure 7 shows the heating curve and the hourly values of supply temperatures with the reference system and the system with ASTC. The usage of ASTC leads to lower supply temperatures at many times of the heating period. According to equation 2 this leads to a higher efficiency of the heat pump:
Figure 8 that the SPF of systems with ASTC is about 0.15 higher than the systems without.
Figure 8 also shows that the adaptive algorithm works the best with small buffer storages or systems with low thermal inertia. The large buffer storage is loaded less frequently. This gives less probability for the storage to be loaded at adapted supply temperatures, as often one loading interval a day (the one after the night set-back) is done at non-adapted heating curve set temperature. However, the effect is not very strong and as mentioned above the start-up losses of a heat pump could influence these results.

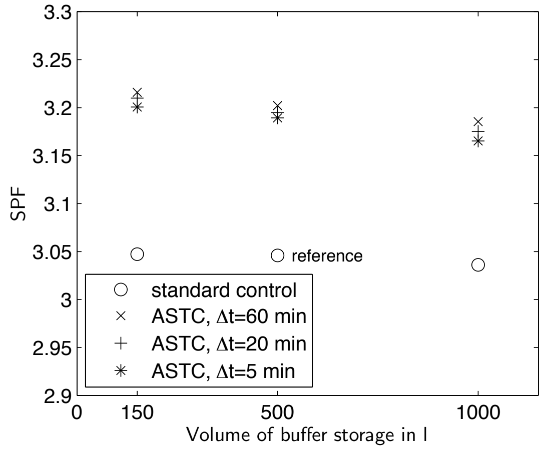

Figure 8: Comparison of seasonal performance factor

Systems with shorter adaption intervals Δt have slightly lower efficiencies that nevertheless lead to similar savings compared to the systems without ASTC. These results show that the presented algorithm works robustly.

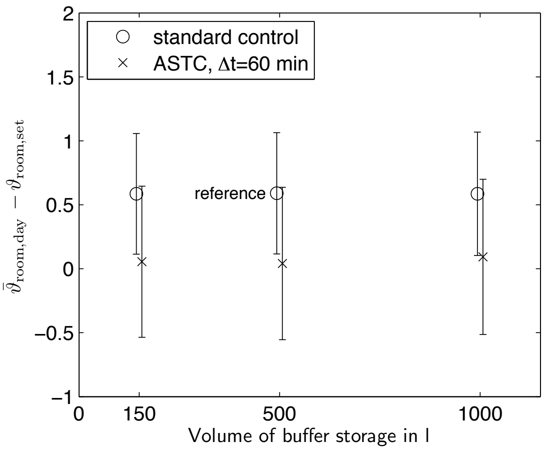

Figure 9: Difference of mean room temperature at daytime and room set temperature. Standard deviation of room temperature.

Figure 9 shows that the mean room temperature at daytime are about 0.5 K above the room set temperature which can be explained by the linear (proportional) control characteristic of the thermostatic valve (see fig. 3). The temperatures in systems with ASTC are closer to the set value and have similar standard deviations.
Comparing the time series of supply and return temperature in figures 10 and 11 it gets apparent how the ASTC works. This is a sunny day with relatively high

ambient temperatures in the afternoon. When the ambient temperature decreases in the evening, the heating curve increases the set temperature, which is not necessary as the building has stored thermal energy. In the system with ASTC, the set temperature stays on a lower level.

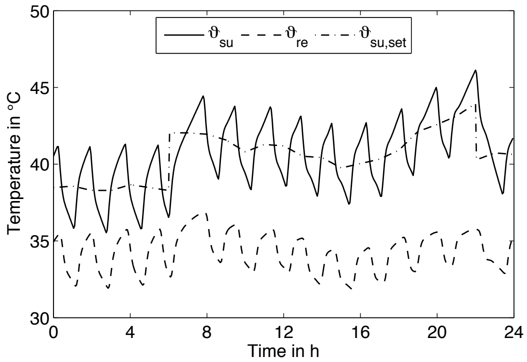

Figure 10: Reference System: Supply and return temperatures on 7th of March.

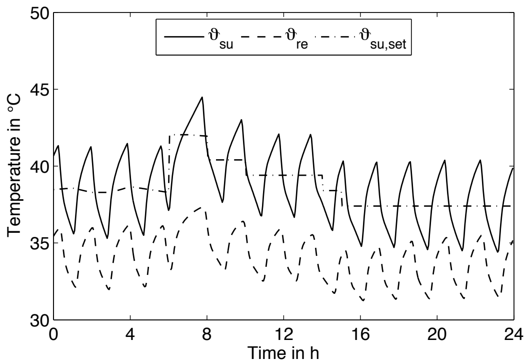

Figure 11: System with ASTC, $\Delta t = 60\,\mathrm{min}$*: Supply and return temperatures on 7th of March.*

CONCLUSION AND OUTLOOK

An AWHP system with buffer storage and heating curve control and the same system with ASTC were numerically analyzed and compared. Different storage volumes and adaption intervals of the ASTC algorithm were studied. It was demonstrated that the ASTC could decrease the supply temperature independent of these parameter variations. This leads to a higher seasonal efficiency of the AWHP. A higher inertia of the heating system (achieved through a larger buffer storage) slightly diminishes the effectiveness of the ASTC because fast changes in heating load cannot be considered for reduction of supply temperature. The adaption interval has a small effect on the efficiency.

Thus, the ASTC presented in this paper seems to be an effective measure to improve the efficiency of heat pumps in buildings. There is no need for a detailed knowledge about the building structure and its behavior towards thermal disturbances. However, the adaptive control only works with a feed-back of the room temperature. Here, the load estimation is implemented with electronic thermostatic valve heads that transmit the thermostatic valve positions to the adaptive controller. It is also implied that the heating system is hydraulically balanced. If it was not, the valve positions eventually would not include the information on the load of the heated zones.

Further research is going to deal with different building and user types and their influence on the efficiency of the AWHP system when being operated with an ASTC. Modeling is going to deal with start/stop behavior of the heat pump. It shall also be analyzed how to deal effectively with the heating up after the night setback. It was found that it is not always necessary to therefore reset the adaption as it has been done in the investigated configurations. However, the results presented in this paper showed that the ASTC is robust relating to the thermal inertia of the heating system.

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Nomenclature

Abbreviations

ASTC	Adaptive supply temperature control
AWHP	Air-to-water heat pump
COP	Coefficient of performance
SPF	Seasonal Performance Factor

Greek Symbols

$\bar{\vartheta}_{\mathrm{room,day}}$	Mean room temperature at daytime
ϑ	Temperature

Symbols

$\dot{m}$	Mass flow
$\dot{Q}$	Heat flow
e	Radiator exponent
n	Ventilation rate
Q	Thermal energy
T	Thermodynamic Temperature
t	Time
v_{wind}	Velocity of wind
x	Valve position
p	Pressure

Subscripts

C	Carnot
dist	Disturbances
drop	Dropping
hc	Heating curve
inner	Inner gains
log	Logarithmic
low	Lower boundary
nom	Nominal
rad	Radiator
rise	Rising
room	Room
set	Set value
sol	Solar gains
source	Source
su	Supply
up	Upper boundary
use	Usable

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