

MODEL-BASED OPTIMIZATION OF CONTROL STRATEGIES FOR LOW-EXERGY SPACE HEATING SYSTEMS USING AN ENVIRONMENTAL HEAT SOURCE

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ABSTRACT

This paper focuses on the optimization of control strategies with regard to the end energy consumption and the thermal comfort of low-exergy systems for space heating in non-residential buildings. The survey building uses a small-scale heat pump system (2.8 kW_{therm}) with a floor heating system. Firstly, the thermal as well as the hydraulic system was modeled in Modelica. The control parameters of the applied conventional control strategy were investigated concerning their impact on the end energy consumption and the thermal comfort. For the most sensitive control parameters a model-based optimization was performed to obtain the parameter set that achieves an optimum for the objective criteria.

INTRODUCTION

Low-exergy concepts for space heating and cooling offer the potential for a drastic reduction in end and primary energy consumption of buildings. These systems utilize environmental heat sources/sinks such as surface-near geothermal energy, ground water or ambient air for space heating and cooling applications. In addition, end energy has to be invested on the one hand for heat transformation processes to obtain temperature levels that are suitable for space heating and cooling and on the other hand for fluid transport processes in the hydronic circuits on the environmental heat source/sink and building side. High system efficiencies can be achieved if the temperature lifts between environmental heat source/sink and building heat/cold delivery system are kept low and pressure drops respectively volume flow rates in the hydronic circuits are reduced. In many cases, an optimum between system temperature levels, i.e. energy consumption for the heat pump compressor and volume flow rates, i.e., energy consumption for circulating pumps exists that has to be found to achieve minimum end energy consumption. Unfortunately, in existing low-exergy systems the potential for high system performance is only hardly exploited due to deficiencies in the planning, design and operation phases. Among other things, reasons for poor system performance are:

- Poorly designed hydronic circuits produce high pressure drops that need to be overcome by the circulating pumps. In the planning phase, often not enough attention is paid to the selection of hydraulic components such as pipes, fittings and valves. Hydronic circuits in low exergy systems (that include e.g. borehole heat exchangers, floor heating systems) are usually much longer than in conventional heating and cooling systems. Therefore, the auxiliary, electrical energy consumption of the heat and cold distribution system has a much higher impact on the total end energy consumption.
- Incorrectly dimensioned circulating pumps lead to unnecessary high volume flow rates under nominal conditions. The electric power consumption of circulating pumps increases with the third power of the volume flow rate and hence has a high impact on the energy consumption.
- Unnecessary high supply temperature setpoints for the heat delivery system (respectively heat storage or distribution) cause an increasing energy consumption of the heat pump compressor due to decreasing COPs with higher temperature levels on the condenser side.
- Furthermore, often high temperature levels are supplied by the heat pump but later on reduced in the further system by mixing processes for example in buffer storages or by return line addition in mixing valves.

This paper focuses on the optimization of control strategies for low exergy systems. For an example building it is shown which control parameters have a high impact on the system performance and how an optimal compromise between the energy consumption of circulating pumps and the energy consumption of the heat pump compressor can be achieved in order to decrease the total energy consumption and increase the overall system efficiency.

Because of the strong interaction and the highly dynamic behaviour of the thermal and hydraulic system components with regard to system control, a simulation-based evaluation of control strategies was applied. The dynamic simulation environment Dymola and the prescription language Modelica was chosen

for the modeling of the thermo-hydraulic system. The simulation model was used to identify optimized control parameters by a model-based optimization.

DESCRIPTION OF SURVEY BUILDING, PLANT AND CONTROL STRATEGY

The survey building is a floating house on a lake located in Kalkar, Germany as shown in Figure 1. It is a demonstrator for a residential building floating on water but is at that time not in use. The total heated net useful area is approximately 93 m². The building envelope is built in passive house standard, so the thermal insulation of opaque constructions and windows, with heat transfer coefficients of 0.09 W/m²K and 0.91 W/m²K, respectively, is very good. It is intended to achieve high passive solar heat gains by two big south windows with a surface area of 18.75 m² and two big east and west windows with a surface area of 8.5 m² each.



Figure 1 Photo of survey building

The energy supply system for space heating as shown in Figure 2 consists of a heat pump system that utilizes the lake water as heat source by a coiled tube heat exchanger.

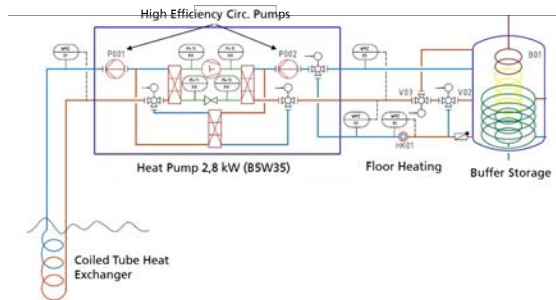


Figure 2 Schematic representation of the supply system for space heating and cooling

In the system a small-scale electric driven, non-speed-controllable heat pump with a nominal heating power of 2.8 kW_{therm} (B5W35) is implemented. The heat delivery to the rooms is realized by a floor heating system with four parallel circuits. Between the heat pump and the floor heating system a water buffer storage with an optional bypass (only used in cooling operation) is integrated in serial connection in the supply line of the hydronic circuit to reduce the switching cycles of the heat pump. For the heat distribution, speed-controllable high efficiency circulating pumps are integrated in the primary (lake heat exchanger and heat pump evaporator) and secondary

(heat pump condenser, buffer storage and floor heating system) hydronic circuits.

Furthermore, on the one side passive cooling to the lake water by a heat exchanger and on the other side domestic hot water production by charging the upper layers of the buffer storage by the heat pump is provided. Nevertheless, this work focuses only on space heating, hence in the following neither cooling operation nor domestic hot water production is further considered.

For the evaluation of the energy performance of the energy supply system, a data acquisition system is implemented in the real building with data points for volume flow rates, supply and return water temperatures, room temperature, electric meters for heat pump, circulating pumps and controller. In addition, environmental conditions like solar radiation and ambient air temperature are measured. Some of the measurement data could later on be used for the modeling and the validation of the system components and building.

The building consists of one thermal zone that is supplied. For the room temperature control, a conventional control strategy based on a heating curve for the supply water temperature of the floor heating system is applied. Figure 3 shows schematically the heating curve for the supply water temperature setpoint dependent on the moving mean value of the ambient temperature of the past 24 h. The course of the heating curve is characterized by the nominal supply water temperature setpoint ($T_{sup_nominal}$) at the nominal ambient temperature of -12°C according to DIN 12831 and the gradient of the heating curve with increasing ambient temperatures (*Gradient*).

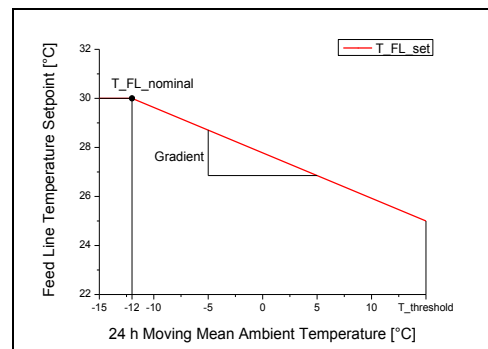


Figure 3 Schematic heating curve for supply water temperature setpoints dependent on moving mean of ambient temperature

A hysteresis controller switches the heat pump on and off dependent on the deviation of the measured supply water temperature and the supply water temperature setpoint. The hysteresis shall prevent high switching frequency of the heat pump compressor and is chosen to be ±0.5 K. In order to be able to provide the controller the actual supply water temperature of the floor heating system the secondary circulating pump is continuously running during the

heating season with the nominal volume flow rate (moving mean ambient temperature below a heating threshold of 15 °C).

By default, neither a room temperature feedback loop nor a thermostatic control valve in the secondary hydronic circuit is used. Because the floor heating system is a slow reacting system and provides a high thermal capacity, an adaption of control parameters at this time would not be useful and could lead to an unstable control. For the compliance with the thermal comfort requirements, the self-regulating effect of the floor heating system is used. Generally, floor heating systems are operated with temperatures near the room temperature setpoint for the heat delivery. Thereby, it can be taken advantage of the effect that with decreasing room temperatures and, by this, increasing temperature difference between the surface of the floor heating system and the room, the heat flow rate automatically increases significantly. At the same time, if internal or solar heat gains increase the room temperature, the decreasing temperature difference between the floor heating surface and the room decreases also automatically the heat flow rate from the floor heating system.

THERMO-HYDRAULIC SIMULATION MODEL

The modeling and simulation was performed in the dynamic simulation environment Dymola (version 2013) using the prescription language Modelica. The thermal as well as the hydraulic system components were modeled. The modeling of the system is based on the fluid and thermal connectors that are provided by the *Modelica Standard Library (MSL)* (Franke et al., 2009). This made it possible to use as far as available already existing models for building technology that are provided on the one hand by the *MSL (Modelica.Fluid, Modelica.Thermal and Modelica.Media)* and on the other hand by the *Modelica library Buildings* that is developed and distributed by the Lawrence Berkley National Laboratories in Berkeley (Wetter, 2009). Not yet existing models or models that were not suitable for application were developed within this work. In the following, a short description of the used simulation models for the system components and building is presented. Further information regarding the system model can also be found in Burhenne et al. (2013):

Heat generation/transformation and heat source

In the survey building a constant compressor speed heat pump is implemented and also modeled for this work. The heat pump model is built up as a black-box model using characteristic curves for COP and heating power dependent on the brine and water side temperature levels of the evaporator and condenser of the heat pump. For the generation of the characteristic curves, firstly, the measurement data of the real plant of the heating season 2011/2012 was filtered for relevant data points of the heat pump operation.

In Figure 4 every black dot represents the COP of the heat pump at different operating points. Secondly, a plane equation according to Equation 1 was chosen for the characteristic curve.

$$COP = a \cdot T_{evap,mean} + b \cdot T_{cond,mean} + z_0 \quad (1)$$

$T_{evap,mean}$ and $T_{cond,mean}$ represent the arithmetic mean temperatures of the inlet and outlet of the evaporator and the condenser. The equation parameters a , b and z_0 were identified by a plane surface fit and provide a coefficient of determination for the COP of 0.88. The resulting characteristic curve for the COP can also be seen in Figure 4. The characteristic curve for the heating power was generated analogous to the prescribed procedure for the COP.

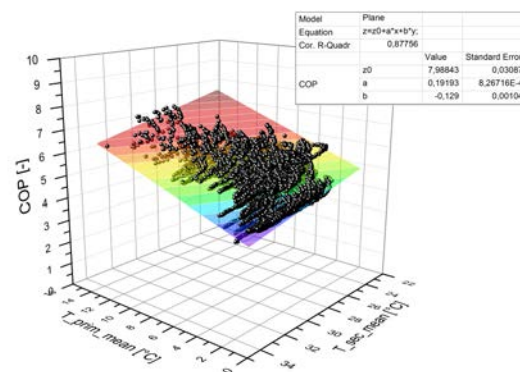


Figure 4 Filtered data points of COP of heat pump dependent on evaporator and condenser temperature level as black dots and fitted regression curve

In the heating season 2011/2012 the secondary circulating pump was operated with a constant volume flow rate. The derived characteristic curves from these measurement data are therefore only valid at this volume flow rate in the first place. Measurements at our test facility with similar sized heat pumps and the same refrigerant have shown that the COPs at operating points with volume flow rates between 38 and 150 % of the nominal volume flow rate deviate only in a range of ± 1.8 % from the COPs at nominal volume flow rates. So for this work the characteristic curves of the heat pump model are assumed also to be valid with variable volume flow rates.

The lake water heat exchanger is a coiled tube heat exchanger and is modeled by a physical approach using an empirical correlation for the heat transfer between lake water and brine. The heat transfer consists of a forced convection between brine and coil inner wall, heat conduction through the coil and free convection between lake water and coil surface. It is assumed that the latter dominates the heat transfer process as it builds the highest thermal resistance. For the calculation of this heat transfer coefficient a Nusselt-correlation as in Equation 1 from Incropera (2007) for free convection on a horizontal tube is used.

$$Nu = \frac{h \cdot L}{k_f} = \left(0.75 + 0.387 [Ra \cdot f(Pr)]^{1/6}\right)^2 \quad (2)$$

$$\text{with } f(Pr) = \left[1 + \left(\frac{0.559}{Pr}\right)^{9/16}\right]^{-16/9}$$

Nu is the Nusselt-number, h the heat transfer coefficient, L the characteristic length so the coil length, k_f the heat conductivity of water, Ra the Rayleigh-number and Pr the Prandtl-number.

Unfortunately, in the data acquisition system no sensor for the undisturbed lake water temperature is implemented so a validation of the lake water heat exchanger with measurement data was not possible. Up to now, only plausibility checks were performed for this sub-model.

Heat storage

The water buffer storage model from the Modelica library *Buildings* (Wetter, 2009) is used. The fluid volume is discretized in axial direction into five layers. Heat transfer processes between adjacent layers occur by fluid transport initiated by direct loading and unloading of the storage, heat conduction through the water and temperature inversion induced fluid flow. Furthermore, a thermal resistance between each fluid volume and the ambient air accounts for heat losses by heat conduction through the storage envelope.

Heat distribution

The hydraulic heat distribution system of the plant was also modeled in this work. The Modelica library *Buildings* (speed controlled pump, two-way- and three-way-valves) and the *MSL* (pipes and fittings) provide hydraulic component models.

For the modeling of the hydraulic network, firstly, a detailed on-site measurement of the hydraulic components was conducted to obtain pipe diameters, pipe lengths, number and geometry of fittings, type labels of pumps, etc. In a next step, the models could then be parameterized with these data based on pressure loss coefficients taken from literature and flow coefficients and characteristic lines from manufacturer data sheets. In one section of the hydraulic network all pipes, pressure loss coefficients for fittings and orifices and flow coefficients for other hydraulic resistances (like e.g. in a heat exchanger) were each aggregated to one hydraulic resistance (see also Figure 5: hydraulic resistances between two fluid nodes represented by white squares). By this, the number of equations in the whole system should be kept small.

The models of the speed-controlled circulating pumps were parameterized with operating points on the one hand for pressure drops depending on the volume flow rate and on the other hand for electrical power consumption depending on the volume flow rate under nominal conditions (i.e. 100 % RPM) taken from characteristic lines from manufacturer

data sheets. The operating points are entered in form of a table and values in between are interpolated with a cubic hermite spline. Part load operating points with reduced speed are calculated applying affinity laws for the correlation of volume flow rate, pressure drop and electrical power consumption with the relative speed of the pump. They are assumed to correlate according to Equations 3 to 5:

$$\frac{Q_1}{Q_{nom}} = \frac{N_1}{N_{nom}} \quad (3)$$

$$\frac{\Delta p_1}{\Delta p_{nom}} = \left(\frac{N_1}{N_{nom}}\right)^2 \quad (4)$$

$$\frac{P_1}{P_{nom}} = \left(\frac{N_1}{N_{nom}}\right)^3 \quad (5)$$

Q is the volume flow rate, Δp is the pressure drop, P the electrical power consumption and N the speed of the pump in part load (index 1) or under nominal conditions (index *nom*).

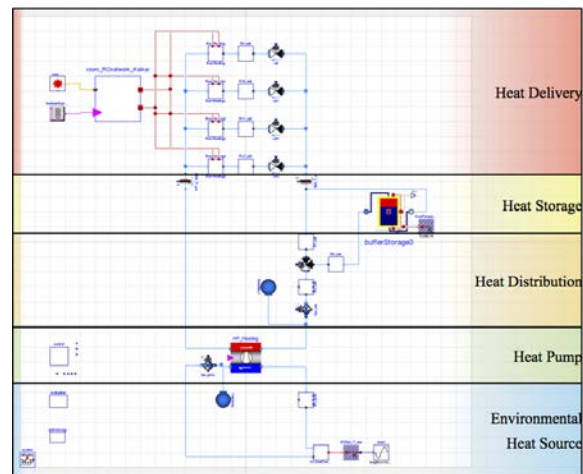


Figure 5 System model in Dymola/Modelica

Heat delivery

For the floor heating system, an implementation for a thermo-active building system (TABS) in Modelica by Jacob (2012) was used. The transient 3-dimensional heat flow from the heat carrier medium to the surface is herein simplified to a 1-dimensional transient heat flow. The model was validated in Jacob (2012) with the TABS model in TRNSYS.

Building

As the aim of application of the system model was the optimization of control strategies and, therefore, a fast simulation performance should be achieved, a very simplified RC-network as proposed in the ISO 13790 (2008) was implemented and used for the modeling of the dynamic heat demand of the building. The RC-network as shown in Figure 6 only contains one air, surface and mass node each as well as only one thermal capacity for the building mass. The

thermal resistances between the nodes were parameterized in accordance to the ISO 13790 based on planning documents of the building.

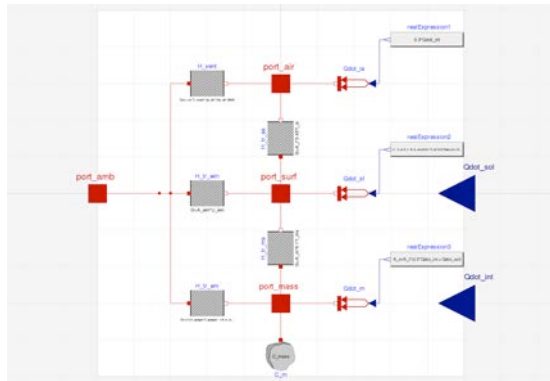


Figure 6 Simplified RC-network of the ISO 13790 room model in Dymola/Modelica

Calibration and validation of coupled floor heating system and building model

To validate the strong simplifications within the room model and to check the applicability, a calibration and validation was carried out for the coupled floor heating system and room model. During the measurements of the survey building the boundary conditions e.g. fluctuating ambient air temperature and solar radiation complicate a validation because of permanent changes of the influencing factors. Therefore, it was decided to validate the Modelica model with a detailed simulation model of the building coupled with a floor heating system in IDA ICE. The IDA ICE models are validated in reference to the requirements of the IEA SHC Task 12 and 22 (Crawley et al., 2005). For the validation in this work both models, in Modelica and IDA ICE, were parameterized equally.

The influencing parameters on the dynamic behaviour of the ISO 13790 Modelica model are the internal thermal capacity of the building C_m and the effective heat transfer area between room mass and surface node A_m . So firstly, a calibration of these parameters was carried out. In this calibration simulation the room was heated by the floor heating system with an ideal PI-controller to exactly 20 °C for a period of 7 days so that the room reached a steady-state condition. The ambient air temperature was held constant at 0 °C. After that, heating was abruptly stopped so the building cooled constantly down. Afterwards, the dynamic parameters C_m and A_m of the ISO 13790 Modelica model were calibrated so that a minimum deviation of the cooling-down curve of the room air temperature in reference to the IDA ICE cooling-down curve was achieved.

As validation test case, a periodically heating of the building was chosen with a periodic time of 24 h. The floor heating system heats the room for 12 h with a defined mass flow rate of 0.2 kg/s and a fluid inlet temperature of 30 °C. Afterwards it is shut off for 12 h.

Figure 7 shows the room air temperatures of the Modelica and IDA ICE model as time series and as well in direct comparison (points on the diagonal have a perfect match of Modelica and IDA ICE room temperature at the same time step).

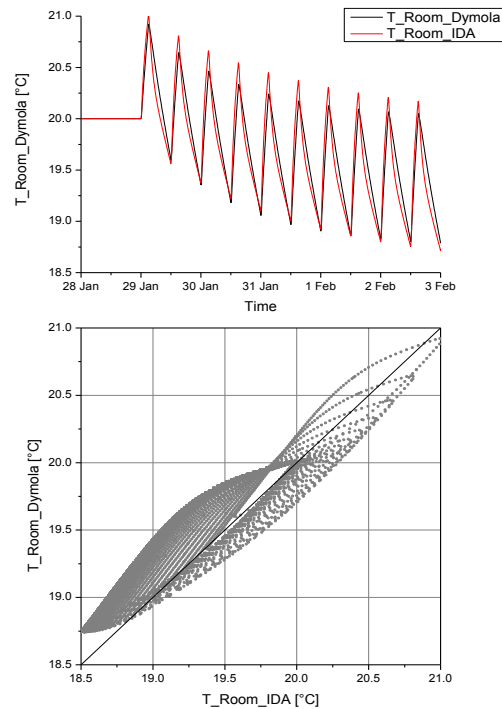


Figure 7 Room temperature of Modelica and IDA ICE model as time series and in direct comparison

The time series indicate a relatively good match. In the diagram of the direct comparison it can be seen more clearly that the maximum deviation of the room temperature is about +0.45 K (at $T_{Room_IDA} \approx 19.25$ °C) and -0.3 K (at $T_{Room_IDA} \approx 20$ °C). The mean deviation is 0.23 K.

Figure 8 shows analogous the comparison of the heating power of the floor heating system of the Modelica and the reference IDA ICE model. The time series also indicate a good match.

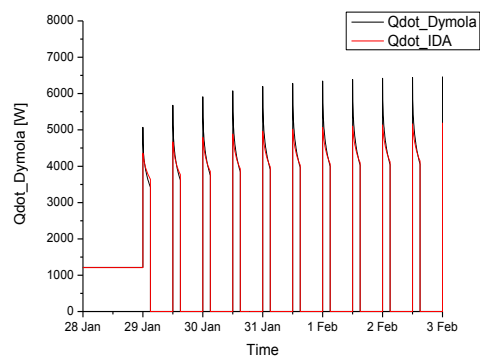


Figure 8 Heating power of Modelica and IDA ICE room model as time series

The direct comparison in Figure 9 shows that especially at the change-over points of the floor heating system between off and on the deviations rise up to

about +1250 W respectively about +25 % (at $Q_{dot_IDA} \approx 5250$ W). Nevertheless, the majority of the values are in good agreement so a total mean deviation of 44 W is achieved.

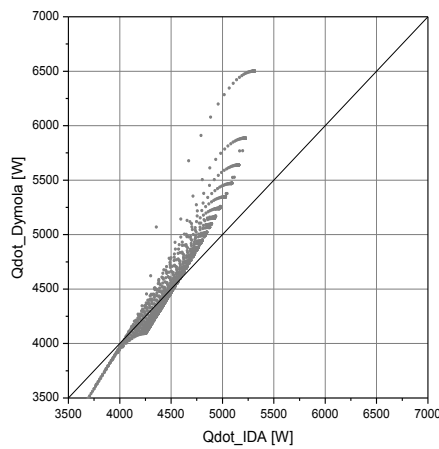


Figure 9 Heating power of Modelica and IDA ICE room model in direct comparison

As conclusion, the Modelica model is considered sufficiently precise with regard to the application and aims of this work.

EVALUATION OF SYSTEM PERFORMANCE

For the optimization of control strategies, annual simulations carried out. The system performance is evaluated concerning the following system parameters:

Annual end energy consumption

The electrical power for the circulating pumps and the heat pump compressor is integrated over a whole year and delivers the total end energy consumption $W_{el,pump,prim}$, $W_{el,pump,sec}$ and $W_{el,compressor}$.

Annual thermal energy consumption

The integrated heat flow rate that is transferred to the building thermal zone by the floor heating system provides the total annual thermal energy that it consumed by the room Q_{room} .

Seasonal performance factor

The seasonal performance factor is defined as the ratio of the annual thermal energy consumption of the building related to the annual end energy that had to be invested as in Equation 6:

$$SPF = \frac{Q_{room}}{W_{el,pump,prim} + W_{el,compressor} + W_{el,pump,sec}} \quad (6)$$

Thermal comfort

As a floor heating system is a slow reacting heat delivery system, the control to a user defined room temperature is very difficult. Hence, under- and oversupply may occur and can lead to a violation of the thermal comfort requirements especially when

internal or solar heat gains arise. In this work, a penalty function $\varphi_{comfort}$ according to Equation 7 is used. It represents the duration of under- and overshooting of the room temperature thresholds $T_{threshold,low} = 20$ °C and $T_{threshold,high} = 24$ °C. Furthermore, it is weighted with the degree of under- and overshooting.

$$\varphi_{comfort} = \begin{cases} \int \min(0, T_{room} - T_{threshold,low}) dt \\ \int \max(0, T_{room} - T_{threshold,high}) dt \end{cases} \quad (7)$$

T_{room} is the actual room air temperature.

SENSITIVITY ANALYSIS OF CONTROL STRATEGY PARAMETERS

The applied control strategy is based on a heating curve for the supply water temperature of the floor heating system as prescribed in the previous chapters. Firstly, a sensitivity analysis of control parameters shall identify the parameters with a high impact on the total annual end energy consumption. The following parameters were evaluated (see also Figure 3):

Table 1 Evaluated control parameters

	DESCRIPTION	DEFAULT
<i>Normed_RPM</i>	relative speed of sec. pump, i.e. nominal volume flow rate	0.7
<i>T_sup_nom</i>	nominal supply water temperature	29 °C
<i>Gradient</i>	gradient of heating curve	-0.15
<i>T_threshold</i>	heating threshold temperature	15 °C

Figure 10 and Figure 11 show the total annual end energy consumption for heat pump compressor and circulating pumps dependent on the values of the four evaluated control parameters.

Relative speed

Figure 10 shows that the relative speed of the secondary circulating pump has a high impact on the total end energy consumption. The regression curve in the figure shows that the pump energy consumption increases with a quadratic function dependent on the relative speed. The annual energy consumption varies between 618 and 846 kWh/a with a portion of the circulating pumps of 2.7 and 29 %. The energy consumption of the heat pump compressor is almost uninfluenced because it depends mainly on the COP of the heat pump. The COP is in this case for all pump speeds almost equal because it depends mainly on the heating curve that is kept constant (T_{sup_nom} and $Gradient$ are set to default).

Nominal supply water temperature setpoint

The nominal supply water temperature setpoint has also a high impact on the total end energy consumption that varies between 559 and 758 kWh/a.

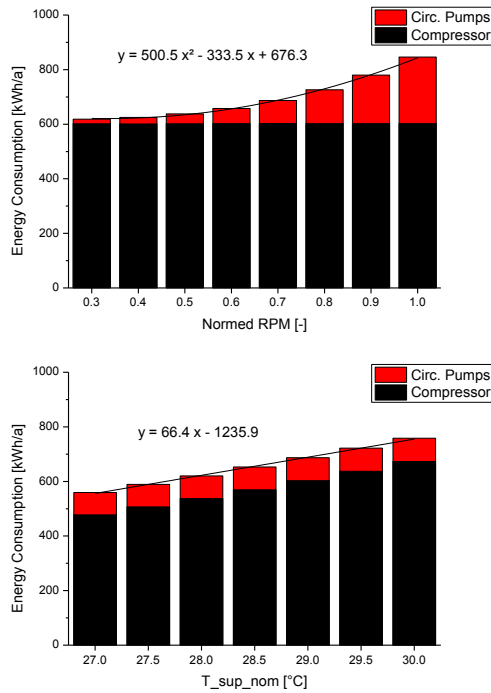


Figure 10 Annual end energy consumption dependent on relative pump speed and nominal supply water temperature

With higher supply water temperatures the COP of the heat pump decreases and causes a higher energy consumption of the compressor. The sensitivity on the energy consumption is 66.4 kWh/a per Kelvin variation of the supply water temperature setpoint. Furthermore, it has a high impact on the thermal comfort. The annual thermal energy that is transferred to the rooms varies between 2538 and 3406 kWh/a. Therefore, too high setpoints may lead to an oversupply of the building.

Gradient of heating curve

Analogous to the nominal supply water temperature setpoint the gradient of the heating curve affects the supply water temperature setpoints. Therefore, it also has a high impact on the total end energy consumption because of lower COPs of the heat pump with lower gradients of the heating curve. The linear regression curve shows a sensitivity of -123 kWh/a per change of the gradient of 0.1 K/K. The thermal comfort is similarly affected by the heating curve gradient.

Heating threshold temperature

The heating threshold temperature defines at what ambient temperature the space heating system is switched off. By this, the running time of the secondary circulating pump can be reduced. Nevertheless, the sensitivity on the total annual energy consumption is only 7.1 kWh/a per Kelvin.

OPTIMIZATION OF CONTROL

An optimization for the identified most sensitive control parameters *Normed_RPM*, *T_sup_nom* and

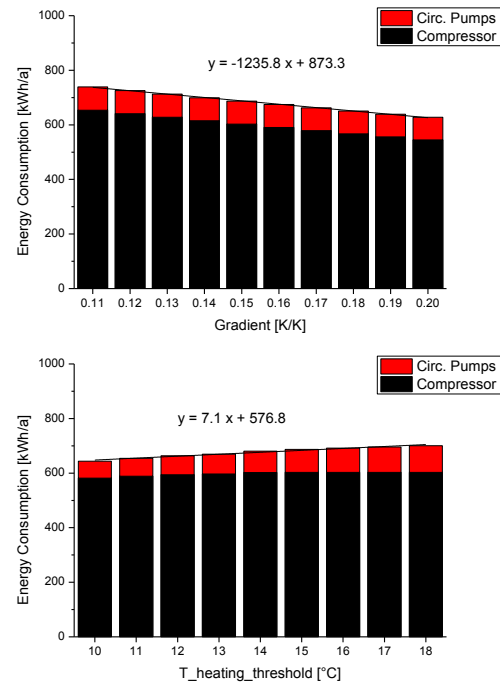


Figure 11 Annual end energy consumption dependent on gradient of heating curve and heating threshold temperature

Gradient was performed to obtain an optimized parameter set that achieves a minimum for the objective function according to Equation 8.

$$x = \arg \min \left\{ \begin{array}{l} W_{el,pumps}(x) + W_{el,compressor}(x) \\ \varphi_{comfort}(x) \end{array} \right. \quad (8)$$

The control parameters are merged to a vector x . The objective criteria are the sum of the end energy consumption for pumps and heat pump compressor and the penalty function for the thermal comfort. The compliance with the thermal comfort was chosen to have the highest priority so every x that violates the thermal comfort requirements is excluded from the solution set (constraint: $\varphi_{comfort} = 0$).

The Modelica library Optimization 2.1 that is provided by the Dymola 2013 release was used for the solution of the multi-criteria, multi-parameter optimization problem. In this work, the optimization method *pattern search* was chosen as it is supposed to be robust against non-smooth objective functions (Pfeiffer, 2012). Non-smoothness may occur in this optimization task due to the constraint for the thermal comfort that may lead to an abrupt drop-off of the objective function in the parameter search space.

The *pattern search* is a local convergent optimization method (Pfeiffer, 2012). The found optimum may strongly depend on the chosen start values for the tuner parameters. Therefore, it has to be examined whether the global optimum was found. In this work, a brute-force method was performed for the convergence verification. The whole parameter search space was scanned for minima by excessive simulation runs

and afterwards compared to the found minimum by the *pattern search*.

The following optimal control parameters were found by the *pattern search*:

- $Normed_RPM = 0.488$
- $T_{sup_nom} = 27.48\text{ }^{\circ}\text{C}$
- $Gradient = 0.1854\text{ K/K}$

Figure 12 shows the total end energy consumption dependent on the control parameters $Normed_RPM$ (step size: 0.02) and $Gradient$ (step size: 0.004 K/K) while T_{sup_nom} is kept constant at the found optimal value. The white areas indicate that the thermal comfort requirements have been violated. The optimum lies in between the grid points.

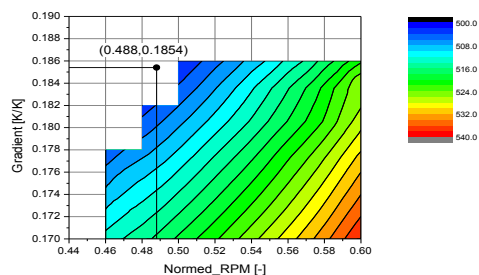


Figure 12 Total annual end energy consumption dependent on $Normed_RPM$ and $Gradient$ and found optimum by *pattern search* ($T_{sup_nom} = const.$)

Using the optimal parameter set, the total end energy consumption can be minimized to 500.4 kWh/a with a portion of 93.6 % for the heat pump compressor and 6.4 % for the circulating pumps. The annual thermal energy that is transferred to the rooms by the floor heating system is 2461 kWh/a. A seasonal performance factor of 4.92 is achieved.

CONCLUSION

In this paper a model-based optimization of the applied conventional control strategy of a small-scale heat pump system with a floor heating system was performed. The control strategy is based on a heating curve for the supply water temperature dependent on the ambient temperature. Because a floor heating system is a slow reacting heat delivery system, it is difficult to control the room temperature to a defined setpoint. In a sensitivity analysis it has been shown what impact the different control parameters have on the end energy consumption resp. over- and under-supply of the building. Based on the conventional control strategy an optimal parameter set was identified that minimizes the annual end energy consumption with compliance to the thermal comfort requirements.

FUTURE WORK

The aim of future work will be the development of new control strategies for low-exergy buildings. The results in this paper will represent the reference case as optimized *conventional* control strategy. In con-

ventional control strategies it is tried to account for the bad controllability of slow reacting heat delivery systems and the reduced heat demand in part load operation by applying heating curves e.g. for supply water temperatures dependent on the ambient temperature. In further studies it will be focused to investigate if further dependencies on the heat demand than only the ambient temperature exist. A model predictive control (MPC) will be conducted that accounts e.g. for weather predictions to determine optimal supply water temperatures and volume flow rates for a prediction horizon. From the results of the MPC, rules as simple if-then-statements shall be extracted by statistical methods (May-Ostendorp et al., 2011). These rules could be easily implemented in buildings and enable an energy efficient operation.

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