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# MODELLING OF HYDRAULIC CIRCUITS FOR DIFFERENT HEAT PUMP SYSTEMS IN LOW ENERGY BUILDING

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## ABSTRACT

The present paper describes heat pump systems and their energy efficiency. In the first part of the paper the modelling of heat pump system is presented. In a second part a parameter study is discussed which takes into account the hydraulic circuit (serial / parallel hydraulic circuit of the storage tank) of the heat pump system and control strategies used for its operation.

## INTRODUCTION

The increase in energy prices especially for oil and gas in the last years has resulted in the renaissance of heat pump systems. Because of their rising importance in the heating appliances market the question of the choice of the heating system (boiler vs. heat pump), the comparison of the different heat pump systems, their interaction with the building, best circuit configuration and optimal control strategies are also gaining in importance. Such general questions are best investigated by means of the computer building simulation coupled with the heating surfaces, heat pump system and relevant outside components such as ground heat exchanger or weather conditions.

The modelling of the heat pump system is a prominent issue of its own that greatly influences the obtained results. Heat pump systems can work effectively only if the complete system is well designed which means that the

- hydraulic circuits,
- the temperature level of the system
- and the control strategy

must be well balanced.

In Germany the typical heat pump system contains a hydraulic circuit with parallel or serial storage tank. The temperature levels of heat pump system typically range between 35°C and 55°C supply temperature. A heating curve is the most popular solution for the control. Countless control strategies have been developed, that focus on adaptation of the heating power to the instantaneous heating demand of the building. The energy effects of all this issues have been partly discussed. This paper presents investigations which complement the former discussion by using a detailed numerical simulation of the whole building and all consequential components of the heating system.

## SIMULATIONS

**Simulation Software**

The numerical investigations are carried out with help of the version 14.2 of TRNSYS® [Klein 1976] which has been further developed and validated at the TU Dresden. With this software the

- thermal behaviour of the house,
- the heating distribution system
- the heat pump
- all boundary conditions (weather, internal loads)

are modelled.

The details of the mathematical models of the house and the heating distribution system are presented in [Perschk 2000] and [Felsmann 2002]. Those models have now been complemented with the model of the heat pump based on an equation-fit model from [Afjei and Wetter 1996].

This model has been further developed so that it now models the following phenomena:

1. the energy consumption by the auxiliary units (circulation pump, storage tank pump, etc)
2. operation of the heat pump in the cooling mode during the warm period
3. the dynamic behaviour of the heat pump in the start phase

It can also take the following boundary conditions into account:

1. minimal and maximal run times as specified by the manufacturer of the heat pump
2. stoppage times as specified by the provider of electricity.

The described model is simple but robust and easily parameterized and can be used for long calculation times.

**Building characteristics**

A well-insulated house that meets the requirements of the German standard [EnEV 2004] is being investigated. The annual predicted energy consumption for this house is lower than 45 kWh/a/m$^2$. It is heated with a radiator heating system. The investigated systems were constructed for two different system temperatures: 45°C/35°C/20°C and 55°C/45°C/20°C. This description means that for example for the system with temperatures 45°C/35°C/20°C a supply temperature of 45°C and a return temperature of 35°C keeps a room at the temperature of 20°C under design conditions (outside temperature of -14°C). In the bathroom, a set point room temperature of 24°C is assumed. Every radiator is controlled by an electronic PI-controller. The set point temperature is the operative room temperature.
For the investigations, we use the weather-conditions from the TRY-04 [Christoffer 2004] that are typical for Germany. The presence and activity of people has been modelled as internal, time depended thermal loads. The detailed description of all boundary conditions can be found in [Richter et al. 2008].

**Facilities**

Two different hydraulic circuits were investigated. The first is a typical circuit with a serial storage tank depicted in figure 1.

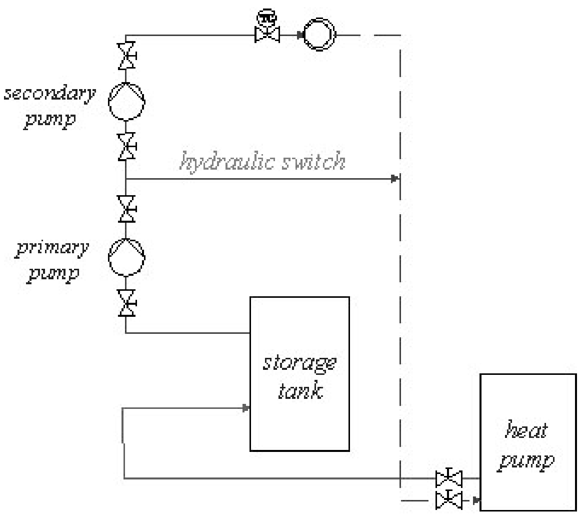


*Figure 1: Serial hydraulic circuit of the storage tank*

In a second circuit a parallel storage tank is used as pictured in figure 2.

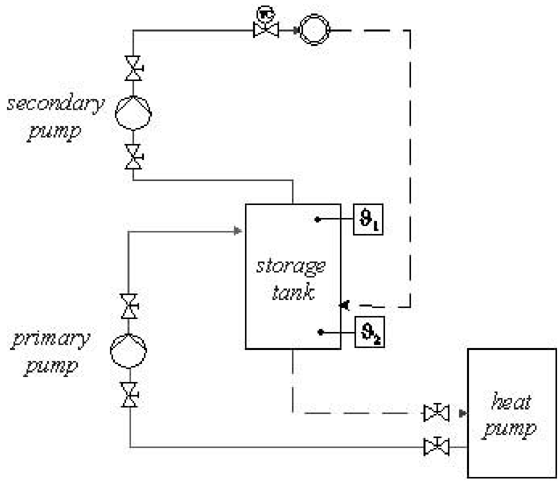


*Figure 2: Parallel hydraulic circuit of the storage tank*

**Energetic evaluation**

A plethora of studies on efficiency of heat pumps exists. They all use the term "coefficient of performance" as the ratio of the amount of heat gained from the heat pump system to the electrical energy used up for heat pump's operation. The last two concepts are almost never defined of defined in a number of different ways. For this reason the following definitions are proposed. The annual integral coefficients of performance are defined as internal ($\beta_i$), external ($\beta_a$) and total system's ($\beta_{sys}$) coefficients of performance. The first characteristic is named internal integral coefficient of performance ($\beta_i$). It is the ratio of the amount heat won from the heat pump over the whole heating period to the electrical energy used up by the compressor in the same time:

$$\beta_i = \frac{\int_{t_1}^{t_2} \dot{Q}_{HP}\, dt}{\int_{t_1}^{t_2} P_C\, dt} \qquad (1)$$

The second characteristic (value) is the external integral coefficient of performance ($\beta_a$). This value accounts for the electrical energy used by auxiliary units, especially pumps and for the control appliances in and around the heat pump system. It does not account for the electrical energy consumption of the circulation pump of the load circuit:

$$\beta_a = \frac{\int_{t_1}^{t_2} \dot{Q}_{HP}\, dt}{\int_{t_1}^{t_2} (P_C + P_A + P_{Cont.})\, dt} \qquad (2)$$

The third characteristic value is the total system's integral coefficient of performance ($\beta_{sys}$). Unlike the external ICOP it takes the heat losses of the storage tank into acount:

$$\beta_{sys} = \frac{\int_{t_1}^{t_2} (\dot{Q}_{HP} - \dot{Q}_S)\, dt}{\int_{t_1}^{t_2} (P_C + P_A + P_{Cont.})\, dt} \qquad (3)$$

## DISCUSSION AND ANALYSIS

**Basic system analysis**

For the given circuit configuration, volume of the storage tank and traditional control strategy the relationship among the integral coefficients of performance are like those shown in figure 3.

Regardless of the configuration of the storage tank (parallel or serial) the values of the system's coefficient of performance are smaller than those of the external system coefficient of performance and they are both much smaller than the values of the internal coefficient ($\beta_{sys} < \beta_a \ll \beta_i$ ).

The differences between $\beta_a$ and $\beta_{sys}$ are very small. Large differences can be ascertained between the investigated hydraulic circuits. The values for a parallel storage tank are approximately 0.5-point higher than the values for the serial storage tank.

These big differences between the serial and parallel storage tank can be explained by the differences in the temperature of water entering the heat pump during the whole heating period. The heat pump runs during some 1300 hours of the 6000 hours that make up the whole heating period. The number of hours with certain supply or return temperature that occur in the system with serial storage tank has been presented in figure 4.

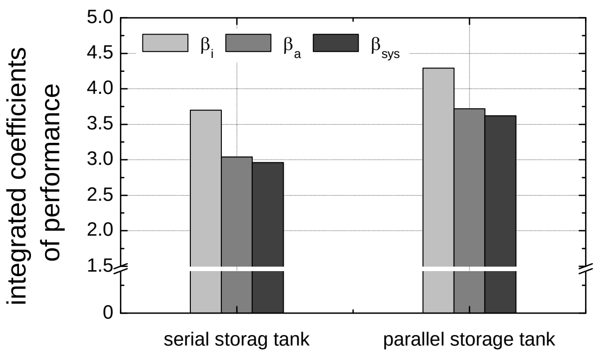


*Figure 3: Integrated coefficients of performance for different hydraulic circuit of the storage tank (system temperature 45/35/20°C; storage volume 200 l)*

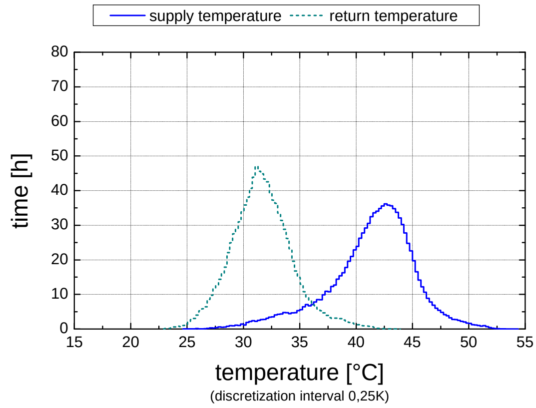


*Figure 4: Occurrence of the supply and return temperature for a system with serial storage tank (system temperature 45/35/20°C; storage volume 200 l)*

The temperature distribution in this case is very symmetrical. The average supply temperature can be fixed around 42°C and the average return temperature at 32°C. The situation is completely different for the parallel storage tank as depicted in figure 5.

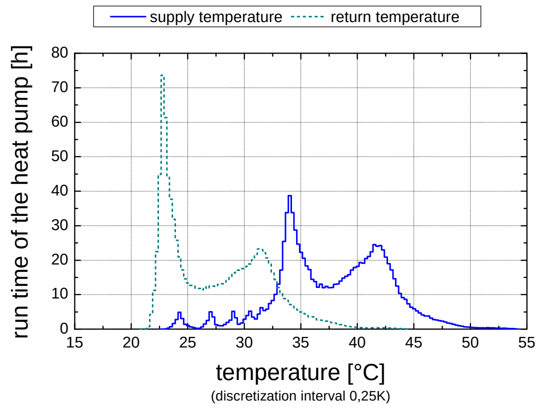


*Figure 5: Occurrence of the supply and return temperature for a system with parallel storage tank (system temperature 45/35/20°C; storage volume 200 l)*

The temperature distributions in system with the serial storage tank are not symmetrical The average supply temperature is approximately 37°C and the return temperature around 27°C, also on average 5°K lower than in the system with the serial storage tank. These big differences in the temperature level of the heat pump explain the big differences in the efficiency coefficients.

The interesting point is why are the temperature levels so different for systems with serial and parallel tank. The distributions of temperature for serial storage tank system are no surprise. They seem to have an almost Gaussian distribution around the average value for the heating period at 32 and 42°C. The system with parallel tank also has those maxima

but they are only local. The real maxima are at 23 and 33°C. In order to understand the occurring phenomena, one must look closely at the configuration and control of both systems and the role that the storage tank plays in them.
The serial system is controlled to keep the demanded supply temperature after the primary pump. For this scheme to work, the primary pump must be turned on all of the time. If there is no demand in the secondary circuit, it only circulates the hot water through the inactive heat pump and storage tank. When demand is there, the secondary pump turns on. The temperature of water returning to the heat pump is the temperature of a mixture of return water from secondary circuit and of the water circulated in the primary circuit. After prolonged period of demand, the water in primary circuit will cool down (the heat pump is still off) below the needed supply temperature but still well above the return temperature of the secondary circuit. The cooling below a certain temperature will turn the heat pump on until the supply temperature increases above the set temperature and the heat pump turns off again. This control differential for the heat pump is usually set around 4 K. This means that the temperature of water entering the heat pump is only some 2 K lower than the needed supply temperature. The heat pump efficiency could be increased by lowering the temperature of water entering the heat pump and that is exactly what the parallel circuit system does.
The parallel circuit (see figure 2) does not mix the water returning from secondary circuit with that of primary one. It simply collects the returning cold water until the storage tank is filled with it and the cold water reaches the upper temperature sensor. Then the heat pump turns on and heats up the much colder water from the tank therefore gaining higher efficiency. At lower outside temperatures the water has to be heated twice in order to gain the required supply temperature as discussed earlier. During the second run through the heat pump, the temperature levels and the efficiency are like that of a serial tank system.
That explains the similarities and differences between the shown temperature levels and respective efficiencies of the two systems.

**Influence of the throttle range of the supply set point temperature in systems with serial storage tank**

It is possible to gain higher efficiency coefficients even in a system with serial storage tank. One way is to set bigger difference between the temperatures at which heat pump switches on and off (bigger control differential). The dependence of the integral coefficients of performance on the control differential of the heat pump control appliance is shown in figure 6. It is clear that bigger control differential increases the efficiency of the system. However, it comes at some cost. The figure 7 shows the occurrence of operative room temperatures.

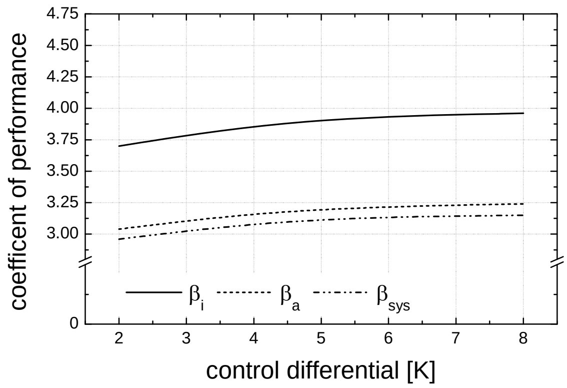


*Figure 6: Coefficient of performance depending on the control differential for a heat pump system with serial storage tank 200 l (system temperature 45/35/20°C)*

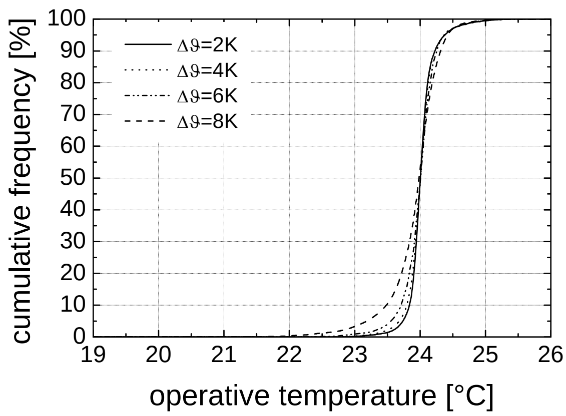


*Figure 7: Operative temperature depend on the control differential for a heat pump system with serial storage tank 200 l (system temperature 45/35/20°C)*

The bigger the control differential the more difficult it is to keep the operative room temperature in the required range. The user senses the divergence from the comfort temperature if it is greater than 0.5 K. Especially in the bathroom the thermal comfort situation is critical as the divergence from the comfort temperature is bigger than 0.5 K for 10% of the time. For a system with a control differential of 8K it is not possible to hold the operative room temperature of 24°C in the bathroom during the whole heating period.

**Influence of the volume of the storage tank and temperature level of the system**

The efficiency of the heat pump system can be increased not only by changing the control differential but also by adjusting the volume of the storage tank or by changing the temperature levels of the heating system. Figure 8 shows the total system's coefficients of performance for a serial and parallel tank as a function of the tank volume. The results are plotted for two different temperature levels of the heating system.

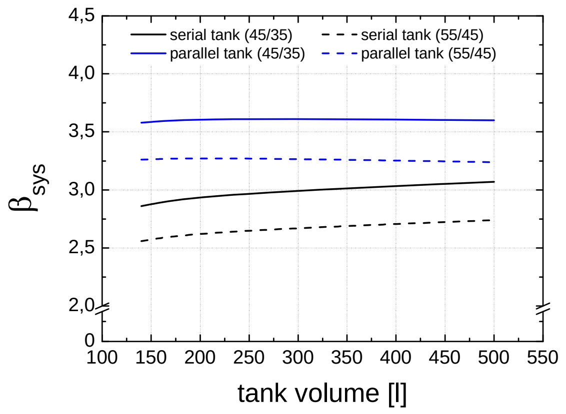


*Figure 8: $\beta_{sys}$ as a function of the tank volume (140 l-200 l-500 l) and the system temperature for serial and parallel storage tank*

The plotted curves show, that the temperature level of the load system has a big influence on the system's efficiency. Between the a temperature level of 55°C/45°C and 45°C/35°C the $\beta_{sys}$ values increase by 0,3-points. It is worth noting, that the change is nearly the same for both hydraulic systems. A closer look at the curves reveals that a change of the tank volume and therefore the thermal capacity has some influence on the system with a serial storage tank. For a system with a parallel storage tank, the values of $\beta_{sys}$ are practically independent of the volume of the storage tank.

**Modern control strategies**

Hydraulic circuit, control differential and temperature level of the system have a big influence on the energy efficiency of the system. However, in practice it is not always possible to change the complete configuration of a system even if it could rise the energy efficiency. In such a case, the solution lies in the change of the control strategy. In the past, many proposals regarding the control strategy were made. The standard solution in the today's heating systems is to shape the supply temperature according to the theoretical heating curve calculated as a function of the outside temperature.

As can be seen in the figure 9, the real supply temperatures in the heat pump system controlled in such a way are higher than envisaged by the theoretical heating curve. The reason for this is that the heat pump cannot modulate its heat production. Especially in the situation of the partial load, this leads to much higher temperatures of water leaving the heat pump. The temperature growth produced by the heat pump is about 10 K. The compressor turns on when the storage tank is filled with the cold water returning from the heating system. In order to heat the water by say 15 K the whole volume of the storage tank must be pumped two times through the heat pump and regardless of the demanded set temperature it will be heated up by 20 K. Such Situation occurs especially at lower outside temperatures when the required heating between return and supply is high. During the second pumping cycle the temperature of the water entering the heat pump is higher which has negative consequences for its coefficient of performance.

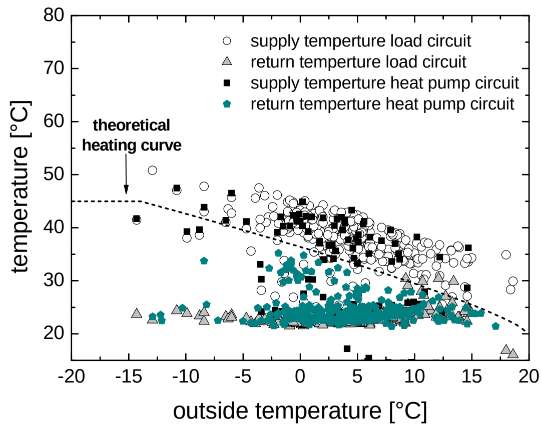


*Figure 9: Hourly temperature for the heat pump an the load circuit (system temperature 45/35/20°C; parallel storage tank 200 l) – heat curve as a function of the outside temperature*

A modern control strategy allows for decrease in the set point temperature of water in the storage tank when the valve lifts in all controlled rooms are smaller than for example 40% of the designed valve lift. In such a system, the heat pump has to be given information on valve lifts in all relevant controlled rooms. Figure 9 shows the distribution of supply and return temperatures in systems using such a modern control strategy. The majority of the supply temperatures lie beneath the theoretical heating curve. For the ambient temperatures above 5°C there is no visible difference, as the difference between the demanded supply and return temperatures no longer exceeds 10 K.

This lower temperature differences have of course a big influence on the energy efficiency of the heat pump. Figure 11 compares the total system's coefficients of performance for a system with parallel storage tank (200 l) with and without an adaptive control strategy.

The values of $\beta_{sys}$ for a system with an adaptive control strategy are approximately 0,2-point higher then for a system with a standard control strategy. In this context, the increase in $\beta_{sys}$ is a little bit lower than for a parallel storage tank than for a serial one. The tank volume has nearly no influence on efficiency of the system. For a serial storage tank the coefficients of performance increase with a growing tank volume.

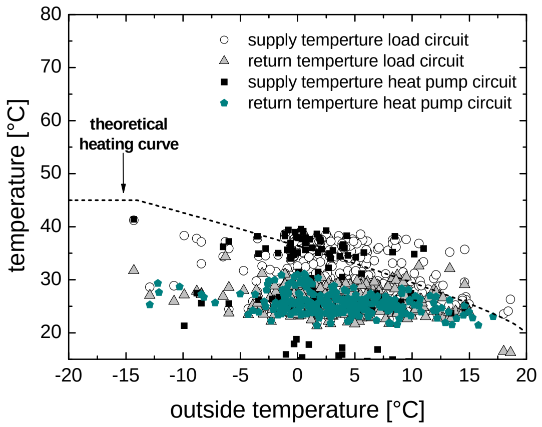


*Figure 10: Hourly temperature for the heat pump an the load circuit (system temperature 45/35/20°C; parallel storage tank 200 l) – supply temperature as a function from the thermal conditions in the rooms*

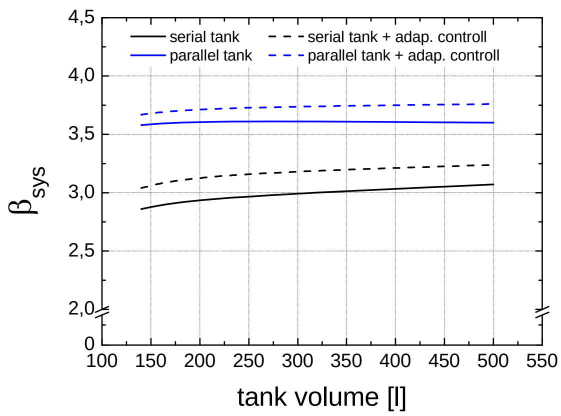


*Figure 11: $\beta_{sys}$ depend on the tank volume (140 l-200 l-500 l) and the control strategy – system temperature (system temperature 45/35/20°C; parallel storage tank)*

## CONCLUSIONS

This paper has shown the influence of different hydraulic circuit control strategies on the energy efficiency of a heat pump system. The results show that a system with an parallel storage tank has a higher energy performance than system with serial storage tank of the same size. The differences between those two systems can reach up to 0.5-points. The volume of the storage tank also has a big influence especially for systems with serial tanks.
The energy efficency can be increased by applying few simple measures. One of them is to set a higher control differential for a system with serial storage tank. In such a case the Constructor must look very carfully whether the system with high control differential can ensure the satisfactory level of thermal comfort in all rooms. A better solution is to use an adaptive control startegy. In this case the room in which the biggest valve lift can force the change in the set point temperature of the heat source if the lift exceeds say 80%. Such configuration of the system can raise the system's coefficent of performance by up to 0,2-points.

## ACKNOWLEDGMENT


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## NOMENCLATURE

**General**

| | |
|---|---|
| $\dot{Q}$ | heat flow |
| P | electric power |
| t | time |

**Greek**

| | |
|---|---|
| β | coefficient of performance |

**Subscripts**

| | |
|---|---|
| A | auxiliary devices except for the pump of the heating system |
| C | compressor |
| Cont. | control units |
| HP | heat pump |
| S | storage tank |
| a | external |
| i | internal |
| sys | global property of the system |

## REFERENCES


Afjei and Wetter 1996, Afjei, Th. and Wetter, M., 1996, Kompressionswärmepumpe inklusive Frost- und Taktverluste, Zentralschweizerisches Technikum Luzern

Christoffer et al. 2004, Christoffer, J.; Deutschländer, T.; Webs, M., 2004, Testreferenzjahre von Deutschland für mittlere und extreme Witterungsverhältnisse TRY, Deutscher Wetterdienst, ISBN-Nr.: 3-88148-398-5

EnEV2004 2004. Energieeinsparverordnung 2004, Bundesregierung von BRD, Berlin, Germany

Felsmann 2002. Felsmann, C., 2002, Ein Beitrag zur Optimierung der Betriebsweise heizungs- und raumlufttechnischer Anlagen, PhD-Thesis, TU Dresden

Klein et al. 1976, Klein, S A, Duffie J A. and Beckman, W A., 1976, TRNSYS - A Transient Simulation Program. ASHRAE Trans 82 (1976), S. 623

Perschk 2000. Perschk, A., 2000, Gebäude-Anlagen-Simulation unter der Berücksichtigung der hygrischen Prozesse in den Gebäudewänden, PhD-Thesis, TU Dresden

Richter et al. 2008. Richter, W., Seifert, J. Knorr, M., 2008, Heizen und Kühlen mit Niedrigexergie (LowEx), TU Dresden