BUILDING SIMULATION BY APPLICATION OF A HVAC SYSTEM CONSIDERING THE THERMAL AND MOISTURE BEHAVIORS OF THE PERIMETER WALLS

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ABSTRACT
This article introduces building simulation in conjunction with a HVAC system, especially a split system, considering thermal and moisture behaviors of the perimeter walls. A description of such a HVAC system is taken from TASK 22/ HVAC BESTEST [4]. Its model had already been validated by analytical methods and verified by comparative tests [5]. In order to simulate the thermal and moisture behaviors of the walls a TYPE 158 [6] has been developed based on the finite element method. Its results are in very good agreement with the humidity response function [7]. The TYPE 158 and the humidity response function are suited for the determination of energy consumption and for the evaluation of control behaviors, respectively.

1. INTRODUCTION
Nowadays, a large portion of glass is installed in office and administrative buildings. In addition to high interior gains (e.g. personal computer, office equipment) it leads to high cooling loads. So, air-conditioning is necessary. On the other hand, the HVAC Split system has became more and more important in the field of air conditioning. The application of such systems has been continuously increased because of lower costs for investment and operations. But the unsteady-state behavior of this system in combination with buildings has not yet been analyzed. For a practical use, this thermal and moisture behavior is very important.

To predict the energy costs for the HVAC operation, a calculation of energy consumption is required. This is calculated according to the purpose of the air-conditioning. In the TASK 22 / HVAC BESTEST [4] the comparative tests for a split system have been carried out at a stationary thermal behavior of the perimeter walls.

In this paper, the derivation of the energy consumption as well as the operating behavior of a split system, considering unsteady-state thermal and moisture behaviors of the walls, is shown. For that, various control strategies were analyzed.

As it is well known, a split system takes sensible and latent energy away from the zone. In general, the control of this system depends on the temperature. The relative humidity floats free in the conditioned zone. The coupled Building and HVAC system simulation determines the zone air temperature, humidity, energy consumption and the control behavior.

2. SIMULATION
2.1 DESCRIPTION OF THE OPERATING BEHAVIOR OF THE SPLIT SYSTEM AND THE VALIDATION OF THE SIMULATION
The description of this applied system is obtainable from [4] (Figure 1). There, the performances of the indoor and outdoor units are dependent on the zone air condition and the weather data.

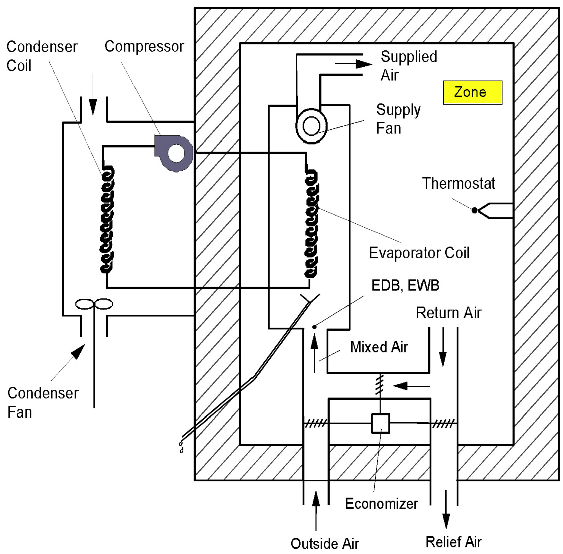

Figure 1: Scheme of the applied split system[4]

The total performance of the evaporator coil is equal to the sum of the sensible and latent portions.

$$\dot{Q}_{coil,tot} = \dot{Q}_{coil,sen} + \dot{Q}_{coil,lat} \quad (1)$$

The total performance and the sensible portion of the evaporator coil are given in the performance map.

They can be determined by using multi-linear approximation as below.

$$\dot{Q}_{coil,tot} = (\vartheta_{AU} * A_1 + A_2) * \vartheta_f + (\vartheta_{AU} * A_3 + A_4) \quad (2)$$

$$\dot{Q}_{coil,sen} = (\vartheta_{AU} * B_1 + \vartheta_{EIN} * B_2 + B_3) * \vartheta_f + (\vartheta_{AU} * B_4 + \vartheta_{EIN} * B_5 + B_6) \quad (3)$$

The latent energy is solved from Eq. (1)

$$\dot{Q}_{coil,lat} = \dot{Q}_{coil,tot} - \dot{Q}_{coil,sen} \quad (4)$$

The energy consumption of the compressor is determined as Eq. (5).

$$\dot{Q}_{comp} = (\vartheta_{AU} * C_1 + C_2) * \vartheta_f + (\vartheta_{AU} * C_3 + C_4) \quad (5)$$

Eqs. (2), (3) and (5) are the characteristic curves of the evaporator coil identifiable from the performance map at full load operation. The coefficients A_i, B_i and C_i are determined from approximation with the given performance map.

The steady-state operating point is solved if the ratio of the sensible to the total zone load equals to the ratio of the sensible to total capacity of the evaporator coil (Eq. (6)).

$$\frac{\dot{Q}_{zone,sen}}{\dot{Q}_{zone,tot}} = \frac{\dot{Q}_{coil,sen}}{\dot{Q}_{coil,tot}} \quad (6)$$

Further details are obtainable in [1]. The coefficient of performance as well as the energy consumption of compressor and fans are determined as described in [4]. The part load ratio (PLR) has been defined as follows.

$$PLR = \frac{\dot{Q}_{zone,tot}}{\dot{Q}_{coil,tot}} \quad (7)$$

The model of the split system had been successfully verified by the application of analytical methods [1], [8] and validated with comparative tests [5]. There are 14 test variants with different cooling loads, varied types of weather data and several temperature set-points.

2.2 MODELLING THE THERMAL AND MOISTURE BEHAVIOR OF THE BUILDING

The thermal behavior can be simulated with the program TRNSYS-TUD. This is TRNSYS with revisions and additional modules from TU Dresden, e.g. revised algorithms for the calculation of exchanges of long-wave radiations as well as the unsteady-state thermal behaviors of the perimeter walls etc. The two models integrated in standard TYPE 56 for the calculation of the moisture behaviors of the walls have not been used in this paper because they are not precise enough [3]. The humidity response function method, which describes the behavior of the interior plasters, the processes of adsorption and absorption of the walls is given in [7]. This method is suited especially for the analyzing of control processes. Results for the control processes are obtainable in [2]. The humidity response function has the following form.

$$g(t) = a * e^{-\left(\frac{t}{\tau}\right)^b} \quad (8)$$

By varying relative humidity (input) the wall per square m absorbs or adsorbs a steam mass g depending on increase or decrease of humidity.

On the other hand, a new developed TYPE 158 [6] for TRNSYS simulation considering the moisture behavior has been approached. This TYPE is based on the finite element method and calculates the moisture flow proceeding from the local difference of the moisture potential. Thereby, the coefficients of the materials depending on temperature and moisture have been considered. Also in [6] several validations of this model were carried out. The steady-state heat transmission will be reached after a few weeks, whereas the stationary moisture transport requires a significantly longer time (years).

The main purpose of this paper is the analysis of the control behavior of the HVAC system. The boundary conditions are cyclic running procedures. Therefore, only the processes of adsorption and absorption of the interior plaster should be considered, but not the steam transmission through the whole wall [7]. The model of the humidity response function is applied to enable a simulation with consideration of the thermal and moisture behaviors of the perimeter walls. In order to realize this simulation a developed TYPE 108 has to be coupled with the TYPE 56. Figure 2 shows the coupling of both TYPES.

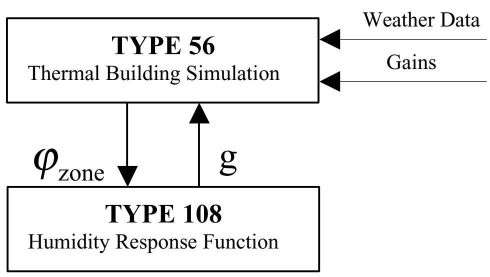

Figure 2: Coupling Scheme in TRNSYS

The results of the two different methods give a very good agreement for a lime plaster (Figure 2). A jump of relative humidity of the zone from 40 to 80 % is valid as the input. The absorption of the lime plaster within 24 hours is represented in Figure 3. This period is especially important for the control behavior.

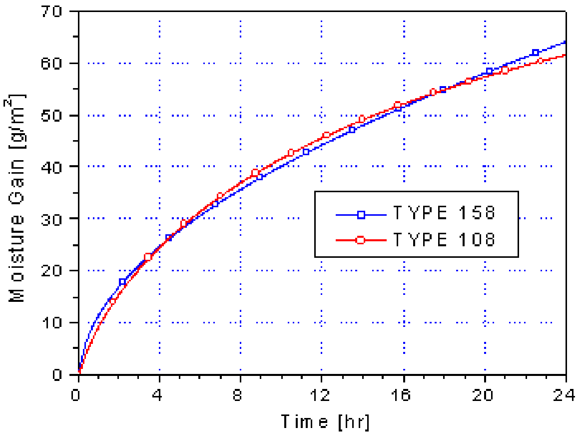

Figure 3: Moisture Gains from Different TYPES

3. ANALYSIS OF THE CONTROL BEHAVIORS

3.1 OPERATING BEHAVIOR OF THE SPLIT SYSTEM

For the examples [4] the near adiabatic wall of the zone was used. The floating zone conditions are dependent on the interior gains, the operating behavior (e.g. performance, control behavior) of the split system, the outdoor dry bulb temperature and the set point of the thermostat.

The number of switching cycles within two hours of simulation with a simulation time step of 90 s depending on the PLR by a two point control is shown in Figure 4. It is remarkable that the HVAC system immediately switches on/off, if the zone dry bulb temperature is greater/less than the set point. This means, the controller works without hysteresis around the set point. With a PLR of about 0.65 the number of switching cycles reaches a maximum. A lot of changes of the control position leads to high abrasion. So, the number of the switching cycles per hour is limited to a certain value by the manufacturers. For a system the following properties are given:

- time step for the controller
- hysteresis around the set point
- minimal time interval for ON and OFF

Therefore, the following frame conditions versus [4] are changed for the analysis below:

- no interior latent loads
- constant outdoor conditions $\vartheta_{AU} = 32\ °C;\ \varphi_{AU} = 35\ \%$
- set point temperature: 24 °C
- hysteresis = ± 1 K
- minimum time for ON position: 5 minutes
- minimum time for OFF position: 3 minutes
- continuous operation or cyclic operation of the supply fan together with the compressor

The number of switching cycles of two different control strategies depending on time step is represented in Figure 5.

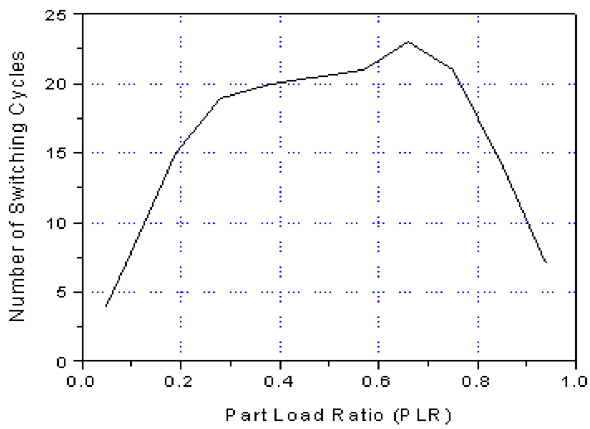

Figure 4: Number of Switching Cycles of the Compressor Depending on PLR at a Time Step of 90 s within 2 Hours of Simulation

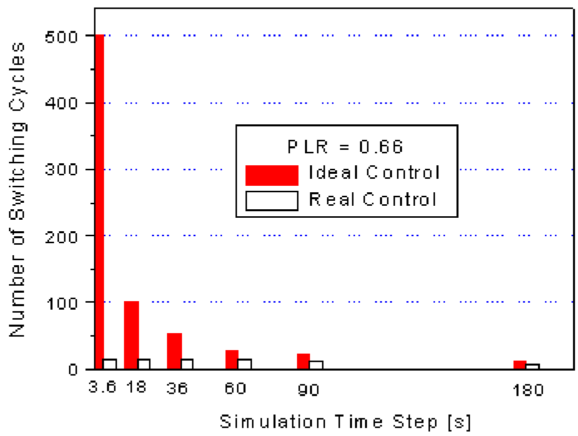

Figure 5: Maximum Number of Switching Cycles of the Compressor at Different Time Steps within 2 Hours of Simulation

The simulation time step is varied between 3.6 s and 180 s. With increasing time step the maximal number of switching cycles strongly decreases by ideal[1] control, whereas the real[2] control has no comparable effect on the switching cycles.

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[1] Designated for a control without hysteresis

[2] Designated for a control with a hysteresis of 1 K and a minimum time interval of 5 minutes for ON; 3 minutes for OFF

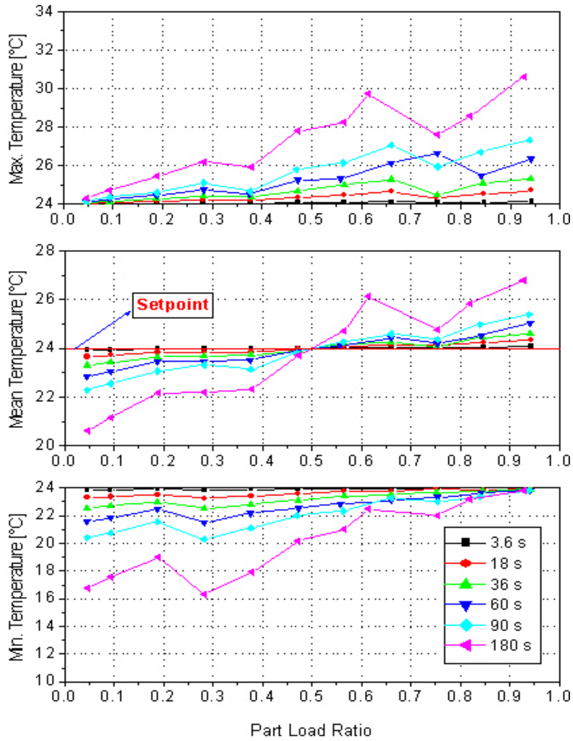

Figure 6: Temperature Dynamics by Ideal Control

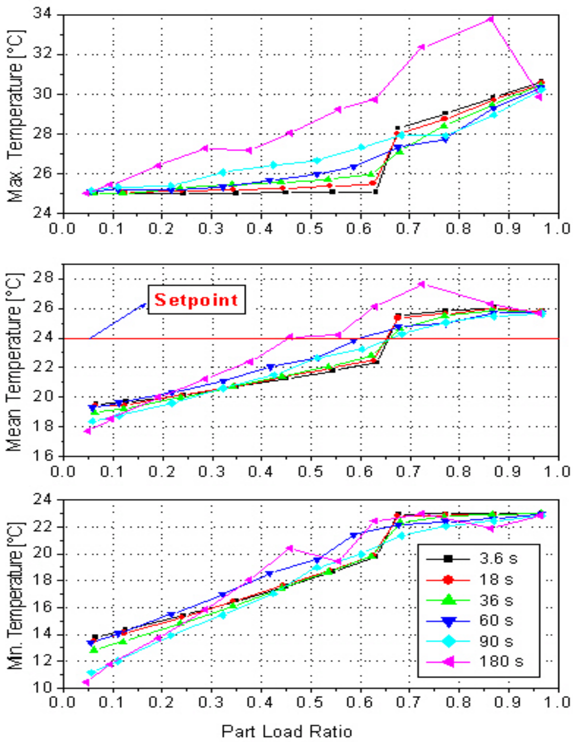

Figure 7: Temperature Dynamics by Real Control

The deviation and minimum and maximum temperatures are especially interesting for the control. Concerning the temperature dynamics the time step is very important (Figures 6 and 7). These figures describe clearly the temperature behavior using the above mentioned control strategies. The temperature dynamics of ideal control are represented in Figure 6 depending on PLR and time step, whereas the real control results are shown in Figure 7.

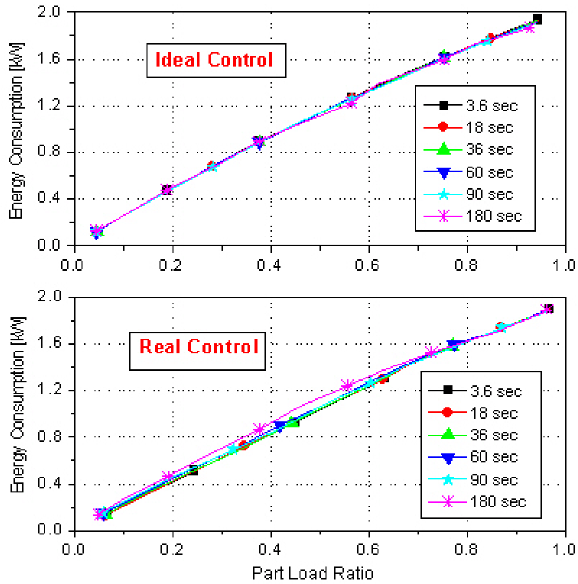

Figure 8: Total Energy Consumption at Different Kinds of Control. Compressor and Supply Fan Cycle Together.

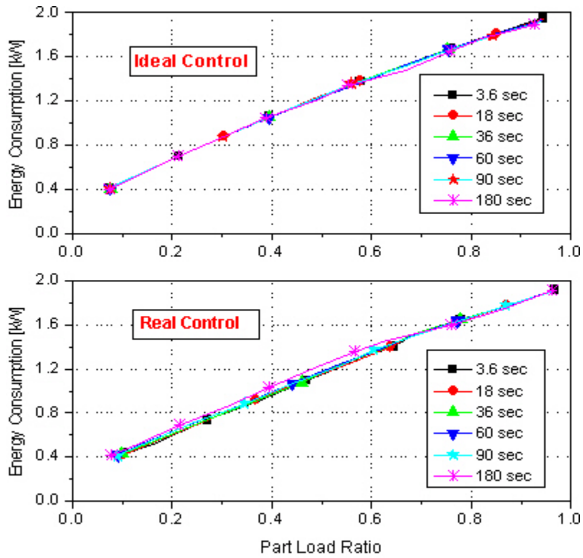

Figure 9: Total Energy Consumption at Different Kinds of Control, the Supply Fan Running Continuously.

At small PLR the zone temperature decreases very quickly. The zone heats up slowly if the HVAC system turns itself off. With increasing time step the minimum zone temperature is lower, whereas the maximum temperature is not significantly greater than the set point (Figure 6). The control deviation is negative, about zero or positive at small, mean or high PLR, respectively. The real control gives a similar correlation (Figure 7) as the ideal control. The control deviation at average PLR is not zero but negative.

The energy consumptions of the HVAC system for the two control strategies are represented in the Figures 8 and 9. The chosen time step contributes insignificantly. The energy consumption at small PLRs in the case of a cyclically operating supply fan is lower compared to a continuously operating supply fan. For further analysis a time step of 60 s or 90 s is reasonable.

3.2 ANALYSIS OF THE REAL BEHAVIOR OF THE BUILDING AND HVAC SYSTEMS

The main point of this section is the analysis of a building with a HVAC system considering the thermal and moisture behavior. For this purpose a zone with the dimensions of 10x7x3 m and a south-facing window of 2x10 m is used. The weather data of a typical meteorological year TRY 05 of the area Wuerzburg, Germany on 14th July is used for the simulation. The HVAC system is controlled daily from 7 AM to 6 PM. At all other times, the zone conditions float freely. At this interval of the operating time the interior sensible cooling load of 3000 W and the interior latent cooling load of 750 W are given. With this weather data, the interior cooling loads and a time step of 90 s the simulation runs 30 days in a transient mode. The results of the 31st day are used for the evaluations. The following points are taken into account for the analysis:

- control behavior
- effects of air exchange at night
- effects of economizer control
- effects of moisture storages in the walls.

The energy consumption, the number of switching cycles per operating day as well as the temperatures (medium, maximum and minimum) are considered to be dimensions for the evaluations of each variant. Of course, the control deviation of the two point controller is derived from that.

3.2.1 CONTROL BEHAVIOR

For the analysis of the control behavior, the aforementioned control strategies are applied. A third control strategy "hysteresis" is added (Table 1).

Table 1: Variants for the analysis control behavior

Case	Name	Hysteresis	T_{ON_min}	T_{OFF_min}
1	Ideal	0	0	0
2	Hysteresis	1 K	0	0
3	Real	1 K	5 min	3 min

The air change per hour at operating time is 1 h^{-1}, else 0.2 h^{-1}. It corresponds to an economizer position of 13.7 % at operating time and 2.75 % at night. The results of these three control strategies are represented in Figure 10.

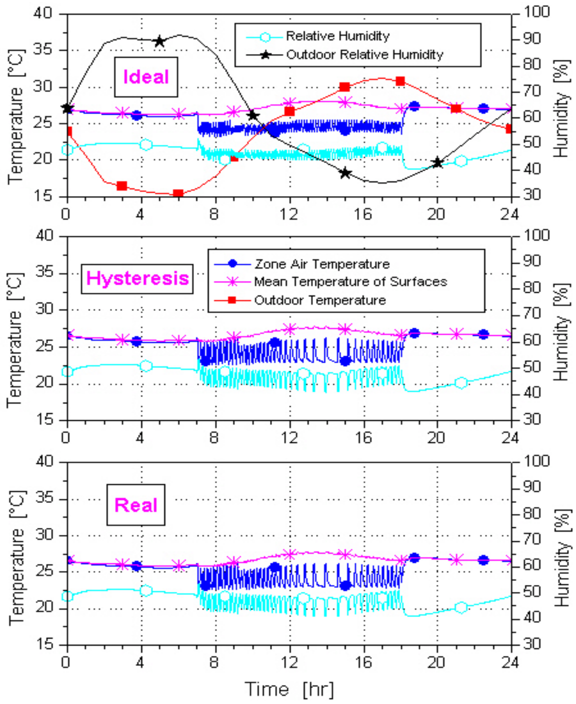

Figure 10: Zone Air Conditions by Various Control Strategies

The vibration of temperature and moisture of the cases "hysteresis" and "real" are remarkably higher than the vibration of the case "ideal". After 6 PM the zone air temperature immediately rises and matches the middle temperature of the surfaces. At this time the outdoor temperature is significantly higher than the indoor temperature and there is also solar radiation. Figure 11 shows the curves of the arithmetic mean values of the temperatures and relative humidities of the three control strategies. The curves of "hysteresis" and "real" have collapsed, so the difference could not be identified. The PLRs are shown in Figure 12. Despite of the same interior gains and the same weather data, the PLR resulting from the case "ideal" is lower in the afternoon. The temperature of this case is higher at this time. As shown in Eq. (3), the sensible capacity of the split system is dependent on the inlet (mixed) temperature that is determined from the outdoor and zone temperatures.

The control deviation of the case "ideal" is positive. Comparisons among Figures 6, 11 and 12 show a connection for the case "ideal". The control deviation at PLR greater than 0.9 is negative for the cases "hysteresis" and "real" (Figures 11, 12), whereas it is positive in Figure 7. The reason for that is the different sensible heat ratio (SHR) of the cooling loads;

SHR = 1 in Figure 7 and SHR = 0.8 in Figure 11. The sensible portion causes a rise of the zone temperature while the HVAC turns itself off. Table 2 gives the results of the three control strategies in one operating day. The energy consumption of the case "ideal" is lower, but the number of the switching cycles is much higher. There is no difference of results between the cases "hysteresis" and "real".

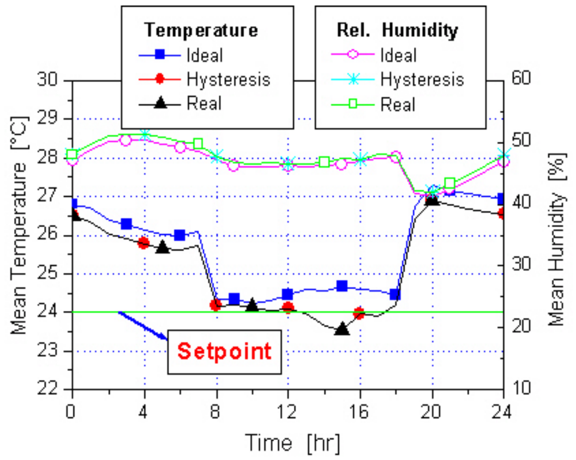

Figure 11: Mean Temperature and Humidity

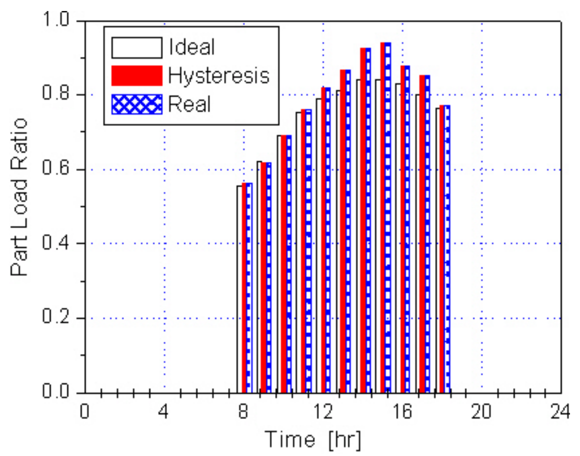

Figure 12: PLR by Various Control Strategies

Table 2: Results of the three control strategies

	ideal	hysteresis	real
Energy Consumption [Wh]	17007	17323	17323
Number of Switching Cycles	95	43	43

3.2.2 EFFECT OF AIR EXCHANGE AT NIGHT

The energy consumption should be lower, if the perimeter wall has been cooled through cool air at night. To realize this study, the case "real" in section 3.2.1 is applied as the basic variant. Another case differs from the basic case in the air exchange per hour of 7.3 h^{-1} at the time interval from 0 AM to 7 AM.

Results in Figure 13 show the increasing cooling for the case with intensive infiltration at night. The difference of the mean temperature of surfaces is about 3 K at 7 AM and decreases during the day, but still remains over 1 K. The deviation of zone temperatures is much smaller, so the operative zone temperature by high air exchange at night is significantly lower. Because of strong cooling at night the PLR is consequently lower (Figure 13).

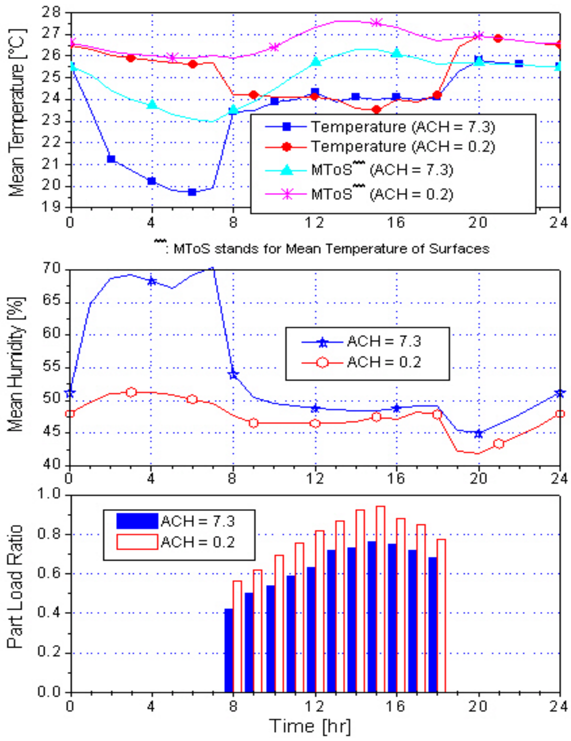

Figure 13: Results of Different Air Exchanges at Night

Table 3: Results of the effect on air exchange

	Basic Variant	High Air Exchange
Energy Consumption [Wh]	17323	14736
Number of Switching Cycles	43	58

Because of small PLR in the case of intensive infiltration, the number of switching cycles is higher and the energy consumption for one operating day is significantly lower, at about 14.9 %. Even if the infiltration is realized with the supply fan of the power of 230 W, moving a volume of 0.425 m^3/s, the energy consumption amounts to 16346 Wh. This cuts the energy consumption by about 5.6 %.

3.2.3 EFFECT OF ECONOMIZER CONTROL

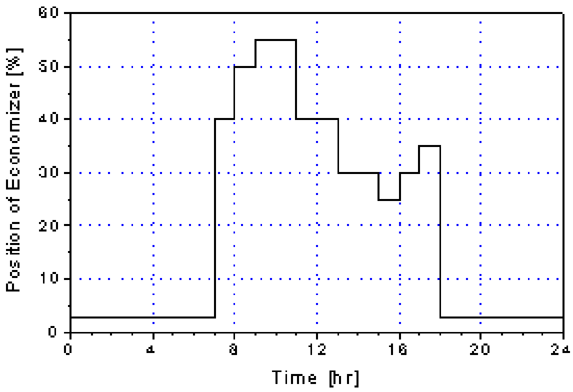

Figure 14: Control of Economizer Position

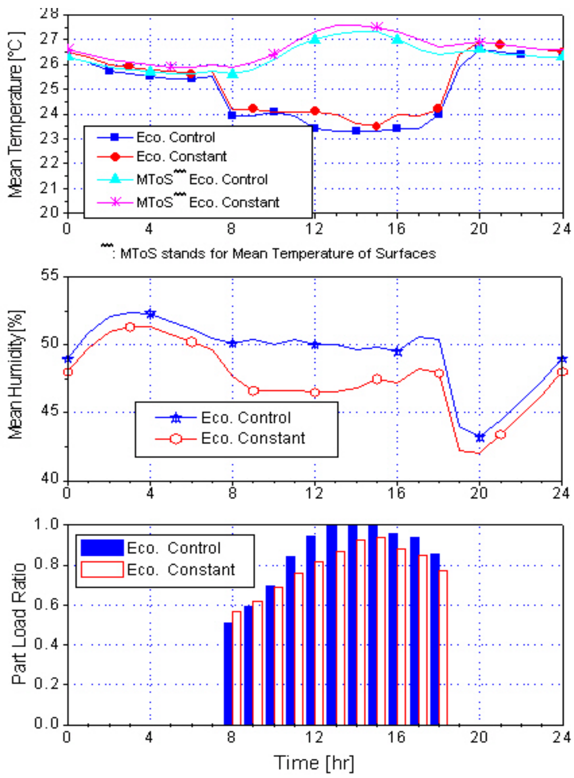

Figure 15: Results of with and without Eco. Control

As mentioned in section 2, the floating humidity has been determined from equation (6). So humidity depends on the SHR of the evaporator performance.

A change of this ratio (e.g. through the economizer control the outside air portion is varied) influences the humidity. In the basic case in subsection 3.2.2 the humidity is less than 50 %. If a humidity of 50% is required, the required position of the economizer has to be adjusted as shown in Figure 14. The PLR at midday (high outdoor temperature) is accordingly greater (Figure 15). The energy consumption is 18734 Wh, about 8.1 % higher, whereas the number of switching cycles amounts to 26. So the humidity in a certain interval can be controlled through the economizer position concerning the outdoor conditions, e.g. with a warm and extremely dry outside air and by a given temperature set point the zone humidity could not be increased, but decreased. At outside air with high moisture content, a rise of zone humidity through economizer control is realizable. Through the economizer control with warm outside air, the influence on the capacity of the HVAC system is remarkable. Thus, the maximum performance of this system has to be considered, otherwise the zone temperature cannot be controlled any more by the HVAC system.

It has been shown that, in a certain interval of outside air, temperature and humidity can be controlled by the evaporator and the economizer, respectively. The less the PLR or the less the cooling loads, the greater the influence of the portion of outside air.

3.2.4 EFFECT OF MOISTURE STORAGE IN THE WALLS

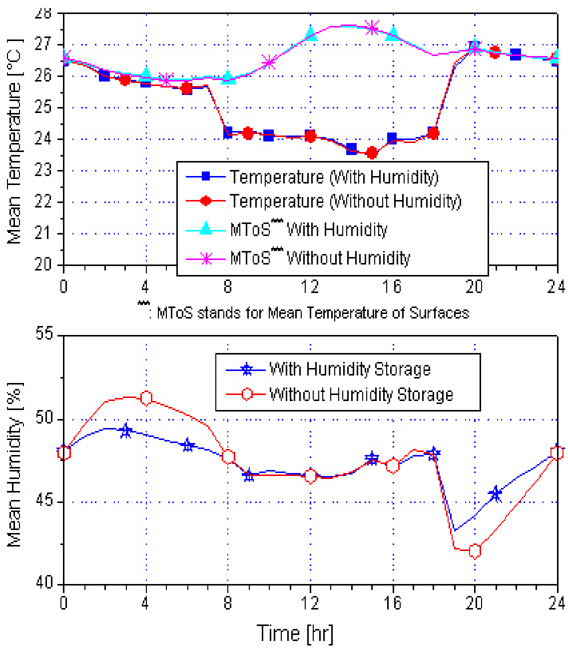

Figure 16: Results of two Cases: With and Without Humidity Storage in the Walls

Up to now, the simulation has considered only the thermal behavior. In reality, thermal as well as moisture processes take place at the same time. Figure 16 shows a comparison between two cases: with and without moisture storage in the perimeter walls. The mean temperatures of the surfaces as well as the zone temperatures are identical. The difference of relative humidity during the operating time is very small. At first, the humidity for the case without consideration of moisture storage decreases steeply immediately after the operating time. After that, it rises again faster because of an exchange with the wet outside air of 0.2 h^{-1}. In contrast, the zone humidity of the moisture storage case rises slower because the walls store moisture. Therefore the amplitude is smaller. The maximum deviation is 2.2 % at 4 AM. Figure 17 shows the curve of the processes of adsorption and absorption of the walls. In this example there are no differences in energy consumption or in the number of switching cycles in one operating day.

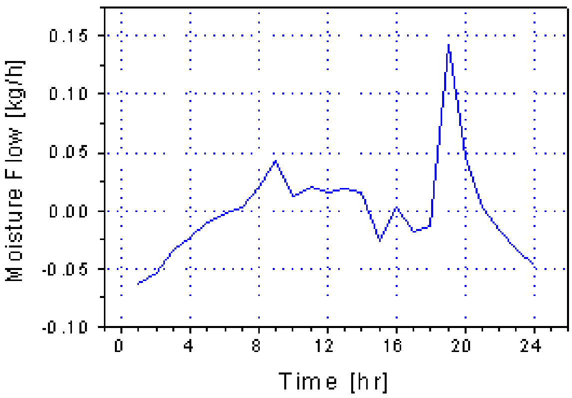

Figure 17: Total Moisture Flow through Adsorption and Absorption of the Interior Plaster

4. CONCLUSIONS

In this article a simulation of a zone was carried out considering the thermal and moisture behavior of the perimeter walls in combination with a split system. The behavior of the split system is validated by an analytical method. With the numerical analysis a suitable time step control behavior has been determined under consideration of the control behavior.

Through a simulation there could be calculated the energy consumption and the control behavior of the HVAC system as a function of the dynamics of temperature and moisture, the perimeter walls, the cooling loads, and the weather data. The results depend on the control strategy, PLR and SHR.

5. REFERENCES

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6. NOMENCLATURE

a, b	Factor for the Humidity Response Function (HRF)
g	Total Moisture Flow through Interior Plasters
ACH	Air Change per Hour
PLR	Part Load Ratio
SHR	Sensible Heat Ratio
$\dot{Q}_{coil, lat}$	Latent Capacity of Evaporator Coil
$\dot{Q}_{coil, sen}$	Sensible Capacity of Evaporator Coil
$\dot{Q}_{coil, tot}$	Total Capacity of Evaporator Coil
$\dot{Q}_{comp}$	Power of Compressor
$\dot{Q}_{zone, sen}$	Sensible Cooling Loads
$\dot{Q}_{zone, tot}$	Total Cooling Loads
φ_{AU}	Relative Humidity of Outside Air
φ_{zone}	Relative Humidity of Zone Air
τ	Time Constant of the HRF
ϑ_{AU}	Outdoor Dry Bulb Temperature
ϑ_{EIN}	Temperature of Mixed Air
ϑ_{f}	Wet Bulb Temperature of Mixed Air