INFLUENCE OF CONTROLLER TYPES IN HVAC SPLIT SYSTEMS ON ZONE COMFORT AND ENERGY CONSUMPTION

by

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ABSTRACT
This article introduces the modelling of the split system (room air conditioner) with a continuous action controller for a building energy simulation. This model shows a way to calculate the energy consumption, the sensible and latent cooling capacity of the split system at part load operation matching the sensible and latent cooling zone loads. As a result, the zone air states can be exactly determined. The model of the split system with a two-point-controller has already been developed and verified (Neymark and Judkoff 1998; Neymark and Judkoff 2000; Knabe and Le 2000a). Results of the simulation show an impact of the two controller types on both the room comfort (temperature and humidity) and the energy consumption.

1. INTRODUCTION
With increasing interior gains because of numerous electrical appliances air conditioning for the removal of the cooling zone loads has become more and more important. Split systems are increasingly used due to lower costs for investment and operation as well as space saving.

According to their configuration, split systems are mainly divided into two groups, i.e. mono- split and multi-split systems. The former is a simpler sort of the split system in which one outdoor unit is connected to one indoor unit. In the latter, one outdoor unit serves several indoor units. In split systems, two types of controllers, continuous action controller (CAC) and discontinuous action controller (DAC), are utilised to control the cooling capacity. The DAC is preferably used in systems with low capacities, whereas the CAC is suitable for systems with high capacities. In the course of technical progress, the CAC becomes more and more important, therefore an analysis of the behaviour of the split system with CAC is absolutely necessary.

At full load operation, the split system runs continuously, so the applied controller types do not have any influence on this system. But at part load operation, the system behaviour depends on the controller types used. As known, the total cooling capacity of the split system consists of a sensible and latent part in wet coil conditions that are normal for the practical operation of such systems. An analysis of the split system with DAC at full load operation as well as at part load operation has already been carried out in the IEA - HVAC BESTEST/ TASK 22 (Neymark and Judkoff 1998) and described in detail (Neymark and Judkoff 2000; Knabe and Le 2000a).

A new model of the mono-split system (Neymark and Judkoff 1998) with a CAC is presented in this paper. This model shows a deterministic method for the calculation of the energy consumption as well as the sensible and latent cooling capacities at part load operation thereby matching the sensible and latent cooling zone loads. Consequently, the zone air states can be precisely calculated. The CAC shows significant differences of the zone air states in comparison with the DAC. Furthermore, using CA controllers saves energy. On this basis, a calculation algorithm for a multi-split system is derivable.

2. MODEL OF THE SPLIT SYSTEM WITH CONTINUOUS ACTION CONTROLLER
2.1 ANALYSIS WITH AN IDEAL CONTINUOUS ACTION CONTROLLER
This paper uses the mono-split system as shown in Figure 1 (Neymark and Judkoff 1998). There is a table of the cooling capacities at full load operation depending on the boundary conditions, i.e. outdoor drybulb temperature (ODB), entering drybulb temperature (EDB) and entering wetbulb temperature (EBW). The behaviour of the split system with an ideal CAC at part load operation is analysed as follows. This means that the cooling capacities are exactly adjusted to the cooling zone loads in accordance with the particular boundary conditions and therefore there is no control deviation.

The part load ratio (PLR) and the sensible heat factor (SHF) of the cooling zone loads continuously change due to the real boundary conditions. Thus, the behaviour under such conditions is difficult to represent and understand. As a result, the analysis below is carried out varying one of the two above-
mentioned variables (PLR, SHF), whereas the other one remains unchanged. Moreover, the economizer is turned off, so that the portion of the outside air is zero. Therefore, it becomes possible to illustrate the behaviour of this system more clearly. The developed approach (Knabe and Le 2000b) can be easily adopted to consider the portion of the outside air due to the control of the economizer.

At first, Figure 2 shows the state change at full load operation and for the boundary conditions (ODB=32°C, EDB=setpoint=24°C, SHF=0.85). That means the state change from entering air (EAfull) to leaving air (LAfull) of the split system is exactly matched with the cooling zone loads to maintain the given temperature setpoint and therefore PLR = 1. The split system runs continuously and the zone air humidity floats freely. The EWB at the steady state is 16.6°C. From now on, the cooling zone loads are reduced to 70% in short time interval, but the so-called boundary conditions remain as before. That means the PLR is now 0.7 and dh/dxzone = 0.7* dh/dxfull. Due to the high cooling capacity of the evaporator coil, the CAC tries to throttle down the refrigerant sent to the evaporator coil. This leads to an increase of the apparatus dew point (ADP) of the evaporator coil and the ADP exceeds the dew point of the entering air EAfull, so dry coil cooling takes place temporarily and the latent cooling capacity is zero. The more the ADP increases the less the sensible cooling zone loads are carried off. Therefore, the latent zone loads are kept in the zone during dry coil cooling. Consequently, the water content of the zone air increases. At the end of the unsteady process, the entering wetbulb temperature levels off at 18.4°C (Figure 2). The steady state is reached if the following conditions are fulfilled:

Exact matching of the cooling capacities at part load operation with the cooling zone loads, which means: $Q_{\text{sen load,part}} = Q_{\text{sen load}}$ and $Q_{\text{lat load,part}} = Q_{\text{lat load}}$

The cooling rating is constant and independent of the PLR because the air mass flow rate passing the evaporator coil is unchanged due to one stage of the indoor fan (Brandemuehl 1993; Le 1996). That means:

$$\eta = \frac{h_{E_{A,full}} - h_{L_{A,full}}}{h_{E_{A,full}} - h_{ADP,full}} = \frac{h_{E_{A,part}} - h_{L_{A,part}}}{h_{E_{A,part}} - h_{ADP,part}}$$

On the basis of Figure 2, one can calculate the sensible and latent cooling capacities at part load operation because the cooling rating and the mass flow rate are known. The values of the former are put into Figure 3 and the process described in Figure 2 is now represented as process 1 in Figure 3.

Now, the SHF is reduced from 0.85 to 0.7 while all other parameters are maintained. This results in pro-
cess 2 in Figure 3. In this way, varying the SHF provides the characteristic curves of the split system at the given PLR as shown in Figure 3.

The characteristic curves for the total field of the part load can be created by repeating the whole process with another PLR and so on. Figure 4 shows the characteristic curves of the split system investigated at PLR=1, PLR=0.7 and PLR=0.5. The following features are identified (Figure 4):

The characteristic curves for the same boundary conditions (ODB, EDB) are a linear function of the entering wetbulb temperature EWB.

The characteristic curves for the total cooling capacity are parallel to each other, and so do the characteristic curves for the sensible cooling capacity.

The ratio of the cooling capacity at the transition cooling point (TCP) (this is an operating point where the coil just becomes dry) of the given PLR to the cooling capacity at the transition cooling point of full load operation is equal to the PLR.

Figure 4: Characteristic curves of the applied split system for PLR=1, PLR=0.7 and PLR=0.5

The transition cooling points of different PLRs form a curved line.

On the basis of this curved line $Q_{N,TCP}$ obtained for the transition cooling points (Figure 4), the transient process 1 in Figure 3 can be illustrated in more detail. As mentioned earlier in this paper, due to the reduction of the zone cooling loads from PLR=1 to PLR=0.7 the evaporator coil temporarily operates at the entering wetbulb temperature of 17.95°C in which the sensible cooling capacity exactly matches the sensible cooling zone loads (Figure 5). At the temporary operating point, the PLR is not 0.7 but 0.63. The actual entering wetbulb temperature of the zone air is still lower than the temporary operating wetbulb temperature so that dry coil operation occurs. More details about this behaviour are given in (Knabe and Le 2000a). During dry coil operation, the latent cooling zone loads remain in the zone and this leads to an increase of the zone air humidity and the entering wetbulb temperature of the zone air because the temperature setpoint is still unchanged. If the entering wetbulb temperature of the zone air exceeds the temporary operating point the coil begins to remove the latent cooling zone loads. The characteristic curves of PLR=0.63 are applied. Consequently, the sensible cooling capacity decreases and the setpoint temperature is not maintained anymore. Therefore, the CAC releases a signal that increases the mass flow rate of the refrigerant sent to the evaporator. This results in a rise of the PLR. Depending on the actual entering wetbulb temperature (17.95°C $\leq$ EWB $\leq$ 18.4°C), the transition cooling point is between point 1 and 2 of the curved line $Q_{N,TCP}$ (Figure 5).

The length of the transient finally depends on the volumes of the applied zone and the PLR. The transient takes longer for larger volumes of the zone or for smaller PLR.

For the operating case with SHF$_{load}$=1, the operating point lies on the curved line for the transition cooling points in accordance with the PLR to match the occurring sensible cooling zone loads. The zone air humidity remains unchanged because the latent cooling capacity as well as the latent cooling zone loads is zero.

Figure 5: Detailed explanation of the transient process from PLR=1 to PLR=0.7 unchanging SHF (SHF=0.85)
Up to now, the analysis of the system behaviour has been carried out for unchanged boundary conditions. To consider the influences of the boundary conditions on the system behaviour at part load operation the whole testing procedure is repeated with other boundary conditions. This result in characteristic curves whose behaviour is similar to those described earlier. Therefore, the conclusions drawn above are valid for all operating cases at part load operation. As described in (Neymark and Judkoff 2000; Knabe and Le 2000a), the characteristic curves at full load operation are multi-linear functions of the ODB, EDB and EWB. Therefore, the characteristic curves at part load operation also depend on these parameters. To determine these curves the TCP is calculated at first. After that, the TCP is used to draw lines parallel to the particular characteristic curves at full load operation. An empirical approach (Eq. 1) is applied for the calculation of the TCP.

\[ \dot{Q}_{N,TCP} = (A_1 \cdot ODB + A_2 \cdot EDB + A_3) \cdot EWB + (B_1 \cdot ODB + B_2 \cdot EDB + B_3) \cdot EWB + (C_1 \cdot ODB + C_2 \cdot EDB + C_3) \]  

(1)

The cooling capacity at the TCP can be calculated (Eq. 2) for a given PLR.

\[ \dot{Q}_{N,TCP,PLR} = PLR \cdot \dot{Q}_{N,TCP,PLR-1} \]  

(2)

\[ \dot{Q}_{N,TCP,PLR-1} \] is determined from the characteristic curves at full load operation and this is detailed explained in (Knabe and Le 2000a). \( \dot{Q}_{N,TCP,PLR} \) is put into Eq. 1 so the EWB\(_{TCP,PLR} \) is solved. At the end, the characteristic curves Eq. 3 and Eq. 4 at a particular part load operation can be drawn up.

\[ \dot{Q}_{N,ass} (PLR) = (E_1 \cdot ODB + E_2) \cdot (EWB - EW_B) + \dot{Q}_{N,TCP,PLR} \]  

(3)

\[ \dot{Q}_{N,ass} (PLR) = (F_1 \cdot ODB + F_2 \cdot EDB + F_3) \cdot (EWB - EW_B) + \dot{Q}_{N,TCP,PLR} \]  

(4)

The coefficients \( A_i, B_i, C_i, E_i \) and \( F_i \) are specified in (Le 2003).

The energy consumption of the indoor fan is calculated from the multiplication of this fan power with the run-time because this fan runs continuously whereas the compressor and the outdoor fan are controlled as the CA controller is considered to be ideal. For this consideration, the energy consumptions of the compressor and the outdoor fan can be calculated with the approach below.

\[ \dot{P}_{PLR} = PLR \cdot \dot{P}_{PLR-1} \]  

(5)

2.2 SIMULATION MODEL OF THE SPLIT SYSTEM WITH A REAL CA CONTROLLER

In section 2.1, the split system is modelled for the steady-state conditions. In practice, the transient conditions occur during the whole operating time. Thus, the modelling of the split system is absolutely necessary on these conditions.

![Coupling of the split system with the TRNSYS-TUD building module Type 56 (TRNSYS-TUD 1996)](image)

At first, Figure 6 shows the scheme how the split system is coupled into the program TRNSYS-TUD (TRNSYS-TUD 1996). As shown above, the economizer is applied in this model. The economizer is controlled by a given strategy to fulfil the hygienic requirements as well as the energy saving during the operating time at low outdoor drybulb temperatures. The mixed air conditions (entering air) are determined from the zone air and the outdoor air as well as their mass flow rates. Depending on the ODB, EDB, and EWB, the split system supplies its cooling capacities to the building module (Type 56). In the Type 56, heat and moisture are balanced. Of course, this Type 56 considers the heat and moisture transfers through the walls and the windows, the radiations of the sun, the infiltration etc. It results in the zone air conditions that are used to calculate the entering air by the economizer. Therefore, iteration is necessary for the calculating procedures in Figure 6. As a result, these are very time-consuming. To avoid this iteration, the results of the previous time-step (zone air conditions) are introduced into the calculation for the current time step. This is equivalent to a digital controller used for the control of HVAC systems in practice. The time step chosen for the simulation should be short to reduce the deviation caused by this consideration.

This model assumes that the zone air temperature is homogenous within the zone. As the CFD model is available in the program TRNSYS TUD one can evaluate the distribution of the zone air temperature. The reasons for not applying the CFD model are long
computing time and the dependence on the location of the supply air etc.

To determine the cooling capacities the characteristic curves at full load operation described in (Knabe and Le 2000a) are used. A PI controller is applied in this model. Its output variable is assumed to be the PLR so that the cooling capacity for a given PLR can easily be calculated (Eq. 1 to Eq. 4). As mentioned above, the total cooling capacity consists of a sensible and a latent part. The sensible cooling capacity is responsible for the maintenance of the setpoint temperature, whereas part of the water content of the zone air condenses due to the latent cooling capacity. After a short runtime, the steady state operating point adjusts and the latent cooling capacity exactly matches the latent cooling zone loads.

As a direct input as latent cooling capacity or latent cooling zone loads in the program TRNSYS-TUD is not allowed, a conversion from latent capacity into mass flow rate of water vapour needs to be done. This input is available in this program. The vapour rate is calculated with the following equation.

$$m_{\text{vapour}} = \frac{Q_{\text{lat}}}{2501 + 1.86 \cdot EDB}$$  (6)

Figure 7: Nondimensional electric power as function of PLR (A-linear regression; B- comparison)

A measurement of compressors at part load operation is described in (Arndt and Jantsch 2002). Figure 7A shows the nondimensional electric power of the compressor. A linear regression is carried out and extrapolated down to 30% (Figure 7B). Thus the energy consumption of the compressor at part load operation is calculated with Eq. 7.

$$\frac{P}{P_{\text{full}}} = K_1 \cdot \text{PLR} + K_2$$  (7)

The nondimensional electric power of the two-point-controller at part load operation is derived from data given in (Neymark and Judkoff 1998). It is also represented in Figure 7B. This Figure illustrates that saving energy becomes possible by using a CA controller.

3. RESULTS

This section compares the simulation results obtained with the DAC as well as the CAC for the same boundary conditions to illustrate the controllers’ influences on the behaviour of the split system. For this purpose, the results for the DAC (two-point-controller) were taken from (Neymark and Judkoff 2000; Knabe and Le 2000a; Knabe and Le 2001; Neymark et al. 2001; Neymark et al. 2002a; Neymark et al. 2002b). In addition, a differential gap of 1K was considered due to the control action of this controller. A zone with the dimensions of 10x7x3m and a south-facing window of 2x10m was used. The wall materials for room type M “medium” of VDI 2078 (VDI2078 1996) were used. The weather data of the typical meteorological year TRY 05 of the area Wuerzburg, Germany on 14th July were applied for the simulation. The split system was controlled daily from 7 AM to 6 PM. Apart from the running time, the zone conditions are self-adjusting. During the operating time, the interior sensible cooling zone loads of 3500W and the interior latent cooling zone loads of 500W are given. It is assumed that the sensible cooling zone loads are 100% convective. Furthermore, an infiltration of 0.1h⁻¹ around-the-clock due to window mortise is taken into account. With these weather data, the interior cooling zone loads and a time step of 36s, the simulation ran for 30 days for the transient process. The results of the 31st day were used for the evaluations.

3.1. OPERATION WITHOUT OUTSIDE AIR

At first, the operation with 100% recirculation air is considered. That means the economizer is turned off. Figure 8 shows the zone air temperature as well as the zone air humidity of the two controllers. Using the CA controller, the temperature setpoint is exactly maintained, whereas the DA controller causes a fluctuation of the zone air temperature and of the zone air humidity. Applying the CA controller, its humidity is generally higher than the humidity of the DAC.
Figure 8: A- zone air temperature; B- zone air humidity for the operating case without outside air

Figure 9: A- cooling capacity; B- COP and C-energy consumption of the compressor for the operating case without outside air

Figure 10: A- zone air temperature; B- zone air humidity for the operating case with outside air

Figure 11: A- cooling capacity; B- COP and C-energy consumption of the compressor for the operating case with outside air
At the hour the operation begins, the latent cooling capacity of the DA controller is significantly higher than that of the CA controller (Figure 9A), and the zone humidity decreases (Figure 8B). After that, the sensible and latent capacities of both controllers are almost the same. The difference between their coefficients of performance (COP) is insignificant (Figure 9B). It should be noted that PLR and COP given in Figure 9 are the average values for the whole hours, whereas the curves for temperature, humidity and energy consumption are represented for every time step. The total energy consumption is reduced by about 2.0% when the CA controller is used.

3.2 OPERATION WITH OUTSIDE AIR
For hygienic reasons, a portion of outside air is necessary. It is assumed that eight people work in the applied zone. Therefore, a mass flow rate of outside air of 480m$^3$/h (60m$^3$/(h*pers.)*8pers.) is required. This is about 31.4% of the entering air. The economizer is not controlled but adjusted such that the required portion of outside air enters.

![Figure 12: Comparison of the zone air humidity of the two operating cases (A- relative humidity; B- absolute humidity)](image)

Analogously to Figures 8 and 9, Figures 10 and 11 show the zone behaviours of the controllers used. Due to the warm outside air the split system operates with higher PLR. The split system that uses the DA controller does not change ON/OFF at 2 PM, but it operates continuously (Figure 10A). Figure 11A shows a significant difference of the latent cooling capacity of both controllers. The COP is almost the same except for the two hours at the beginning of the operation and is general greater than 3.5. The total energy consumption obtained with the CA controller is about 5.6% less than that of the DA controller.

Figure 12 shows a comparison of the zone air humidity for the two operating cases. The high humidity of the outside air leads to a high SHF. As known from (Neymark and Judkoff 2000; Knabe and Le 2000a), the zone air humidity only depends on the SHF of the cooling zone loads when the DA controller is used. As a result, during the operation with the portion of outside air the zone air humidity of the DA controller is significantly higher than without outside air (Figure 12).

The operation with outside air causes high a PLR as well as a high SHF. As already mentioned earlier, the higher the PLR the lower the humidity of the zone air and the higher the SHF the more the zone air humidity increases. For these given cases, the SHF and the PLR eliminate each other when the CA controller is used. Consequently, the zone air humidity is almost the same. Operating with the portion of outside air, the total energy consumption of the CA controller is about 8.4% higher than without outside air, whereas the DA controller needs about 12.2% more energy for the operation with the portion of outside air.

4. CONCLUSIONS
The analysis of the behaviour at full load operation as well at part load operation of the split system using a two-point-controller has already been carried out in the IEA/ HVAC BESTEST TASK 22. On the basis of BESTEST, a new model of the split system with a deterministic calculating method applying a continuous action controller has been developed and is introduced in this paper. The controller types have no influence on the behaviour of the split system at full load operation, but they cause a clearly different system behaviour at part load operation due to their control actions.

The zone air temperature of the CA controller is almost equal to the setpoint temperature. The DA controller causes a fluctuation of the zone air temperature. The zone air humidity of the DAC controller is only a function of the SHF, whereas the SHF as well as the PLR influence the zone air humidity when CA controller is applied. CA controllers allow saving energy which depends on the particular application.
On the basis of this method described an algorithm for the calculation of multi-split systems is derivable.

5. REFERENCES
TRNSYS-TUD. 1996. TRNSYS (A Transient System Simulation Program) is written by Solar Energy Laboratory, University of Wisconsin, USA; Internet: http://sel.me.wisc.edu/trnsys/default.html.
Since applying a license of this program the Dresden University of Technology has changed the program code and has written additional modules, so the TUD has a new program for the simulation of heating system and air conditioning. It is designated TRNSYS-TUD.

6. NOMENCLATURE
ADP  Apparatus Dew Point    [°C]
CAC  Continuous Action Controller   [-]
CFD  Computing Fluid Dynamics    [-]
COP  Coefficient Of Performance [-]
DAC  Discontinuous Action Controller  [-]
EA  Entering Air   [-]
EDB  Entering Drybulb Temperature  [°C]
EWB  Entering Wetbulb Temperature  [°C]
h  Enthalpy of humid air  [kJ/kg]
LA  Leaving Air  [-]
IDB  Indoor Drybulb Temperature  [°C]
IHR  Indoor Humidity Ratio  [kg/kg]
ODB  Outdoor Drybulb Temperature  [°C]
OHR  Outdoor Humidity Ratio  [kg/kg]
P  Electric Power  [W]
PLR  Part Load Ratio  [-]
SHF  Sensible Heat Factor  [-]
TCP  Transition Cooling Point  [-]
\(\dot{m}_{GA}\)  Mass Flow Rate of Entering Air  [kg/s]
\(\dot{m}_{vapour}\)  Mass Flow Rate of Water Vapour  [kg/s]
\(Q_{2,lmt}\)  Total Cooling Capacity  [W]
\(Q_{V,sem}\)  Sensible Cooling Capacity  [W]
\(Q_{l,lmt}\)  Latent Cooling Capacity  [W]
\(Q_{2,lmt}\)  Curved line formed by transition cooling points  [W]
\(\dot{Q}_{\text{Load,nt}}\)  Total Cooling Zone Load  [W]
\(\dot{Q}_{\text{Load,sem}}\)  Sensible Cooling Zone Load  [W]
\(\dot{Q}_{\text{Load,nt}}\)  Latent Cooling Zone Load  [W]