Predictive Functional Control (PFC) of a Heating Ventilation Air Conditioning (HVAC) Process

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Abstract: The development of a better system for a HVAC system promises savings of energy and conservation of resources. Predictive Functional Control is applied for the temperature control in order to achieve a better control performance than with a PI(D) control. The research work is the result of cooperation with the company A. Nattermann & Cie GmbH and the University of Applied Sciences Cologne under the guidance of Dr. J. Richalet. The A. Nattermann & Cie GmbH is a pharmaceutical company in which the largest energy consumer is the HVAC system and therefore has the greatest potential for savings. Modeling, controller design and simulation of the temperature control is presented. Most of the algorithm has been already real-time applied.

Keywords: Predictive functional control, HVAC system, temperature control, process model, simulation, real-time control.

1. INTRODUCTION

Energy saving is very important in industrial plants as heating and cooling costs a lot of money. The classical temperature control with fixed set-point is not optimal because sometime cooling and heating overlap each other. As both peoples and goods are not too sensitive to the room temperature it is enough to control the rooms and halls within a specified temperature range. The consequence is less energy consumption and less switching of the heating and cooling apparatus.

A second saving possibility is to recirculate the room temperature and mix it with the fresh air. As fresh air can be very cold (in winter) or to warm (in summer), running with 100 % recirculation could be optimal from energetic point of view. However, a minimal portion of fresh air is required for ensuring the air quality.

The third possibility is humidity control, which is not dealt with in this paper.

Traditionally climate control is realized by PI(D) controllers. Heating and cooling are realized by means of heat exchangers. Control of heat exchangers is a well investigated topic. For example Bonivento et al. (2001) and Raul et al. (2013) compare classical PID and modern model based predictive control of heat exchangers and show the advantage of predictive control. Bonivento et al. (2001) and Chalupa et al. (2010) apply GPC (Generalized Predictive Control). However PFC (Predictive Functional Control) is an easier realizable predictive control as no numerical optimization is necessary even when considering manipulated variable limits (Richalet and O'Donavan, 2009). As a lot of PFC application has been already used successfully in the industry (see e.g. Abdelghani-Idrissi et al., 2001; Arbaoui et al., 2007), PFC was selected for temperature control in this application.

The A. Nattermann & Cie GmbH is a pharmaceutical company in which the HVAC (Heating, Ventilation and Air Conditioning) systems play a central role. On the one hand these systems guarantee the hygienic standard, which is required by the guidelines of the GMP (Good Manufacturing Practices). On the other hand they allow the people to work under good climatic conditions in the clean areas. Because a clean room is usually 24h active, the air handling units are running full speed and the relevant parameters (temperature, relative humidity and pressure) are monitored permanently. This results in a big amount of energy consumption over the year, with around 60% of the due of the total site energy consumption.

The paper is structured as follows. In Section 2 the HVAC system is introduced and in Section 3 requirements and limits of the site specific air handling unit control are explained. The modeling of the system is presented in Section 4. In Section 5 the control algorithm is dealt with. Section 6 and 7 show the control performance both for simulation and real-time operation. The temperature control with a fixed set-point is compared to the control in a temperature range (floating set-point strategy). Section 8 illustrates the dynamic flap control which will be realized in the near future. In Section 9 the performance of the different control methods are compared.

2. HVAC PLANT

The schema of the HVAC plant is given in Fig. 1.

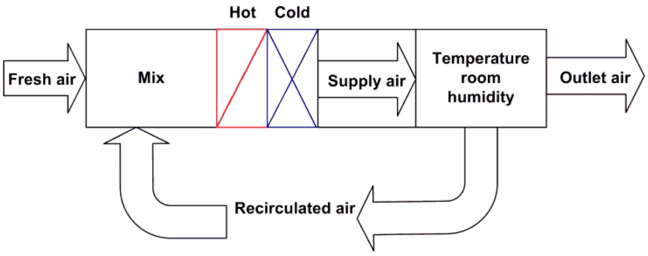

Fig. 1: HVAC plant with recirculation

The ventilation system consists of a mixing chamber in which the fresh air is mixed with the recirculated air. Thereafter, the air is heated or cooled depending on the temperature of the mixed air. Then, the air is directed into the room or workshop. Part of the air is recirculated and the other part is discharged into the atmosphere.

Fig. 2 shows a detailed PI (Piping and Instrumentation) diagram of the HVAC.

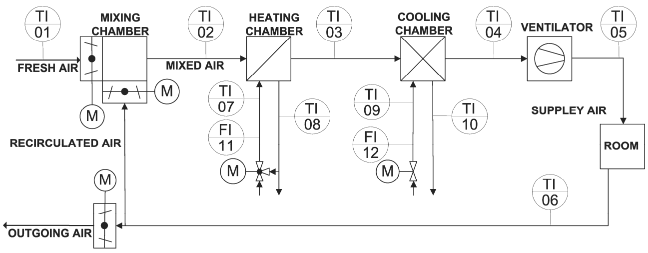

Fig. 2: PI diagram of the HVAC

The measuring points are

- 01: ambient air temperature.
- 02: mixed air temperature.
- 03: air temperature after heater.
- 04: air temperature after cooler.
- 05: air temperature after ventilator.
- 06: room temperature.
- 07: heater fluid inlet temperature.
- 08: heater fluid outlet temperature.
- 09: cooler fluid inlet temperature.
- 10: cooler fluid outlet temperature.
- 11: mass flow heater fluid.
- 12: mass flow cooler fluid.

The temperature 01 of fresh air is measured. The fresh air is mixed with a part of the recirculated air with the temperature 06, and the mixed air has the temperature 02. If the mixed air is too cold then it has to be heated by the heater. The inlet temperature of the heating medium is called THI (07). If the mixed air is too hot then it has to be cooled. The coolant temperature is TCI (09). Then, the air with temperature 05 is fed into the room. The air leaves the room with temperature 06. A ratio of this air is recirculated and the other part is passed into the atmosphere.

Fig. 3 shows the flow, inlet and outlet temperature sensors placed on a heat exchanger in the HVAC system.

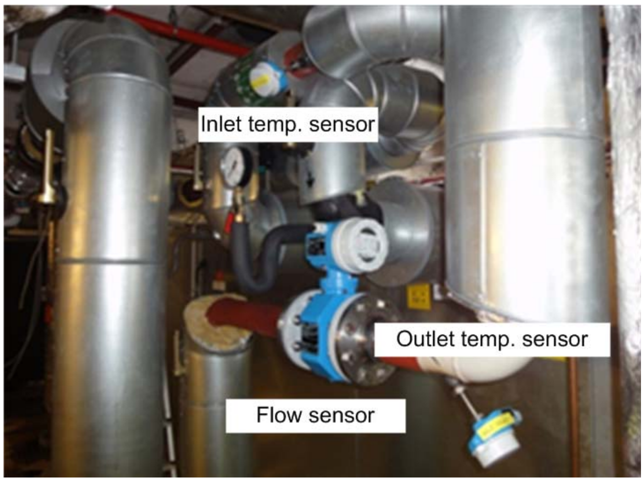

Fig. 3: Flow, inlet and outlet temperature sensors placed on a heat exchanger

For the energy calculations the HVAC system can be balanced over the air side of the heat exchanger using the total air flow and the temperature difference between the air temperature measured before the first heat exchanger (see Fig. 2 with T02) and room-in temperature, but still with some estimation on the heat losses, resulting in an assumed factor. More precise is the measurement by additional sensors on the fluid site of the heat exchanger in order to calculate the actual power, which is shown by (1)

$$P_{HE} = \dot{Q}_{\mathrm{HE}} = \dot{m}_{\mathrm{fluid}} \cdot c_{\mathrm{p,fluid}} \cdot (\vartheta_{\mathrm{fluid\_in}} - \vartheta_{\mathrm{fluid\_out}}) \qquad (1)$$

where

$\dot{m}_{\mathrm{fluid}}$: mass flow of the heat exchanger fluid

$c_{\mathrm{p,\,fluid}}$: specific heat capacity of the fluid

$\vartheta_{\mathrm{fluid\_in}}$: inlet temperature of the heat exchanger fluid

$\vartheta_{\mathrm{fluid\_out}}$: outlet temperature of the heat exchanger fluid

3. TACKLING THE CONTROL OF HVAC SYSTEMS

The basic HVAC (Heating, Ventilation and Air Conditioning) optimization reducing the energy consumption starts with implementing the best practices known as floating set-point, reduced mode (e.g. night set up) and heat recovery (e.g. recirculation). These control strategies can be directly applied for a new HVAC system, after setting the user requirements and realized with many proofed standard solutions. For the optimization of an existing systems it is absolutely necessary in a previous step to get information about the current control situation (especially, when the system was running for years before). A sporadic data recording comparing the current activities of the control system with the desired control from the starting point can deliver the useful information about the quality of control and the space for improvement.

The internal guidelines for the air handling systems and the clean rooms define among other requirements for the set-points and tolerance limits of the relevant room parameters: temperature, relative humidity and pressure, which follows the categories of different zones. The requirement was to keep the controlled room temperature between 19°C till 25°C. In addition to it a minimum due of fresh air is required depending on the maximum amount of people working in the area at the same time, the kind of the product (e.g. powder handling), explosive atmosphere, due of outgoing air, etc. In our case a minimum fresh air due of 30% is defined. The reduced mode, e.g. night set up (e.g. reduced temperature or reduced speed of the ventilators) is not treated in this paper.

Fig. 4 shows the characteristic of a defective control behavior of a PI controlled air handling unit coming from a sporadic data recording. This alternating heating and cooling result in a serious energy waste.

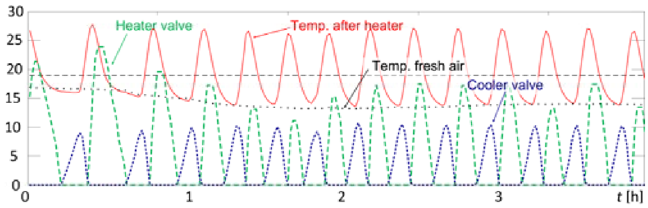

Fig. 4: Alternating heating and cooling as a result of defective control

4. MATHEMATICAL MODEL OF THE PLANT

A HVAC consists of the following parts:

1. Air mixing chamber
2. Cooler/heater
3. Room (whose temperature is to be controlled)

In the following the expression thermal convexity will be introduced (Abdelghani-Idrissi et al., 2001). The aim of introducing the thermal convexity is to eliminate the nonlinear function between the real manipulated signal and the physical variable (mass flow multiplied by thermal capacity coefficients) from a thermal apparatus like the air mixer. In this way linear control algorithm can be used for the nonlinear plant.

4.1 Air mixer

Fig. 5 shows the scheme of an air mixer. In the air mixer the fresh and the recirculated air are mixed. The recirculated air temperature is practically equal to the room temperature.

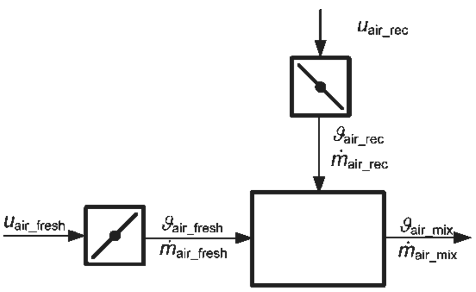

Fig. 5: Air mixer

The output mass flow is the sum of the two mass flows:

$$\dot{m}_{\mathrm{air\_out}}(t) = \dot{m}_{\mathrm{air\_fresh}}(t) + \dot{m}_{\mathrm{air\_rec}}(t) \tag{2}$$

where t is the continuous-time and

$\dot{m}_{\mathrm{air\_fresh}}(t)$: mass flow of fresh air,

$\dot{m}_{\mathrm{air\_rec}}(t)$: mass flow of recirculated air,

$\dot{m}_{\mathrm{air\_out}}(t)$: total mass flow of mixed air.

The total inlet air flow is controlled over a defined pressure difference resulting in a certain power of the inlet air ventilator. The dues of fresh air and recirculated air are delivered over the air flaps which are positioned outside of the mixing chamber. The mixing process can be divided into two parts:

- <u>*static part*</u>:
  symbolizing the stationary mixing process
- <u>*dynamic part*</u>:
  symbolizing the time required for total mixing.

The air mixing is shown in Fig. 6 with these both parts.

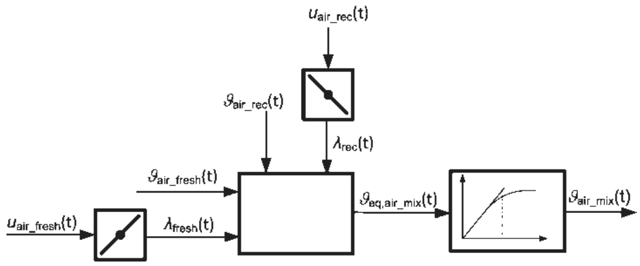

Fig. 6: Air mixing with separating the static and the dynamic processes

The temperature between the both parts is the equivalent temperature $\vartheta_{\mathrm{eq,air\_mix}}$. The following static equation describes the process

$$\vartheta_{\mathrm{eq,air\_mix}}(t) = \lambda_{\mathrm{fresh}}(t) \cdot \vartheta_{\mathrm{air\_fresh}}(t) + \lambda_{\mathrm{rec}}(t) \cdot \vartheta_{\mathrm{air\_rec}}(t) \tag{3}$$

where

$\vartheta_{\mathrm{air\_fresh}}(t)$: fresh air (ambient air) temperature

$\vartheta_{\mathrm{air\_rec}}(t)$: recirculated air (room air) temperature

$\vartheta_{\mathrm{eq,air\_mix}}(t)$: mixed air (so called equivalent) temperature

The factors

$\lambda_{\mathrm{fresh}}(t)$: thermal convexity of fresh air

$\lambda_{\mathrm{rec}}(t)$: thermal convexity of recirculated air

show the effect of the two air flows on the resulting (equivalent) temperature, which can be defined also as functional manipulated variable (Abdelghani-Idrissi et al., 2001). They both depend basically on the air mass flows and approximately they are proportional to them. The sum of the both thermal convexities is one. This means that the output temperature from the air mixer lies between the temperatures of fresh and recycling air

$$\vartheta_{\text{eq,air\_mix}}(t) = \lambda_{\text{fresh}}(t) \cdot \vartheta_{\text{air\_fresh}}(t) + [1 - \lambda_{\text{fresh}}(t)] \cdot \vartheta_{\text{air\_rec}}(t) \quad (4)$$

Consequently the thermal convexity of the fresh air can be calculated form the measured temperatures as follows:

$$\lambda_{\text{fresh}}(t) = \frac{\vartheta_{\text{eq,air\_mix}}(t) - \vartheta_{\text{air\_rec}}(t)}{\vartheta_{\text{air\_fresh}}(t) - \vartheta_{\text{air\_rec}}(t)} \quad (5)$$

The linear dynamic process can be described by a first-order differential equation with the time constant $T_{\text{air\_mix}}$

$$T_{\text{air\_mix}} \cdot \dot{\vartheta}_{\text{air\_out}}(\mathrm{t}) + \vartheta_{\text{air\_out}}(\mathrm{t}) = \vartheta_{\text{eq,air\_mix}}(\mathrm{t}) \quad (6)$$

The gain is unity because the static part is considered by (3).

The task of the air mixing chamber is to control the outlet air temperature by manipulating the air flaps. Generally there is a nonlinear static function between the manipulated signal of the flaps (Fig. 6), the pressure difference on the flap and the mass flow. The thermal convexity is approximately proportional to the mass flow (assuming equal thermal capacity coefficients) and it may depend nonlinearly on the manipulated signal

$$\lambda_{\text{rec}}(t) = f_{\text{rec}}(u_{\text{rec}}) \approx K_{\text{rec}} \cdot u_{\text{rec}}(t) \quad (7)$$

4.2 Heater and cooler

Fig. 7 shows the scheme of a heater or cooler. (For simplicity both cases are treated together in a uniform way.)

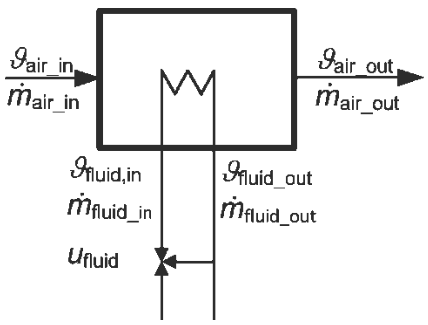

Fig. 7: Heater or cooler

Both the heater and the cooler are heat exchangers. The following notations are used.

$\vartheta_{\text{air\_in}}(t)$: inlet air temperature,

$\vartheta_{\text{air\_out}}(t)$: outlet air temperature,

$\vartheta_{\text{fluid\_in}}(t)$: inlet temperature of heating or cooling fluid,

$\vartheta_{\text{fluid\_out}}(t)$: outlet temperature of heating or cooling fluid.

The heat exchange process can be divided into two parts (Fig. 8):

- <u>static part</u>:
  symbolizing the stationary heat exchange process.
- <u>dynamic part</u>:
  symbolizing the time required for heat exchange process.

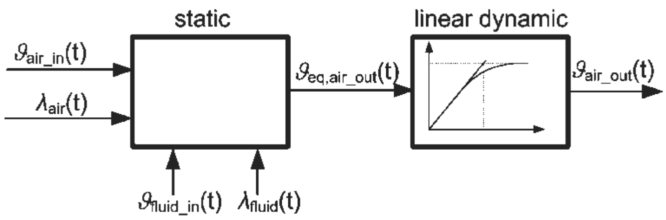

Fig. 8: Static and dynamic parts of the heat exchanger

The following static equation describes the process

$$\vartheta_{\text{eq,air\_out}}(t) = \lambda_{\text{fluid}}(t) \cdot \vartheta_{\text{fluid\_in}}(t) + \lambda_{\text{air}}(t) \cdot \vartheta_{\text{air\_in}}(t) \quad (8)$$

where

$\lambda_{\text{fluid}}(t)$: thermal convexity of (heating or cooling) fluid

$\lambda_{\text{air}}(t)$: thermal convexity of the air

The factors $\lambda_{\text{fluid}}(t)$ and $\lambda_{\text{air}}(t)$ represent the effect of controlling fluid flow and the air flow on the resulting (equivalent) temperature. They both depend approximately linearly on the product of the corresponding mass flows multiplied by the corresponding heating coefficients.

The sum of the both thermal convexities is 1, consequently

$$\vartheta_{\text{eq,air\_out}}(\mathrm{t}) = \lambda_{\text{fluid}}(\mathrm{t}) \cdot \vartheta_{\text{fluid\_in}}(\mathrm{t}) + [1 - \lambda_{\text{fluid}}(\mathrm{t})] \cdot \vartheta_{\text{air\_in}}(\mathrm{t}) \quad (9)$$

The linear dynamic process can be approximated by the first-order differential equation with unity gain

$$T_{\text{HE}} \cdot \dot{\vartheta}_{\text{air\_out}}(t) + \vartheta_{\text{air\_out}}(t) = \vartheta_{\text{eq,air\_out}}(t) \quad (10)$$

T_{HE}: time constant of the heat exchanger

The advantage of introducing the convexity factors is that they influence linearly the output (and the equivalent) temperature. The fluid convexity can be defined between zero and unity.

If $\lambda_{\text{fluid}} = 0$ then the outlet air temperature is equal to the inlet air temperature and if $\lambda_{\text{fluid}} = 1$ then the outlet air temperature becomes equal to the fluid temperature.

When the heating is active then the fluid temperature is larger than the inlet air temperature and the air temperature increases in the heater. When the cooling is active the fluid is colder than the inlet air temperature then the air temperature decreases in the cooler.

In the knowledge of the temperatures the fluid convexity can be calculated from (11)

$$\lambda_{\text{fluid}}(t) = \frac{\vartheta_{\text{eq,air\_out}}(t) - \vartheta_{\text{air\_in}}(t)}{\vartheta_{\text{fluid\_in}}(t) - \vartheta_{\text{air\_in}}(t)} \quad (11)$$

The fluid convexity is a function of the manipulated signal $u_{\text{fluid}}(t)$ of the valve.

4.3 Room

The room inlet temperature is the temperature of the controlled (heated or cooled) air. The fresh air is the disturbance. Fig. 9 shows the room with the inlet and outlet air temperatures.

Fig. 9: Room with the inlet and outlet air temperatures

The following notations are used:

$\vartheta_{room\_in}(t)$: room inlet air temperature,

$\vartheta_{air\_fresh}(t)$: fresh air temperature,

$\vartheta_{room}(t)$: room temperature

The mixing process between room inlet and ambient temperature can be described by the static equation:

$$\vartheta_{eq,room}(t) = \lambda_{room}(t) \cdot \vartheta_{room\_in}(t) + [1 - \lambda_{room}(t)] \cdot \vartheta_{air\_fresh}(t) \quad (12)$$

The dynamic model of the room is described by a first-order model with time constant T_{room} and unity gain

$$T_{room} \cdot \dot{\vartheta}_{room}(t) + \vartheta_{room}(t) = \vartheta_{eq,room}(t) \quad (13)$$

The room outlet temperature outlet is equal to the room temperature. That is the temperature of recirculating air.

4.4 Model parameters

Step responses have been recorded of the cooling, heating and air mixing process of the air handling system. The following parameters were identified based on experiments and used in the simulation program:

- time constant of mixer: T_{mixer}=30 s = 0,5 min
- time constant of heater: T_{heater}=200 s = 3,33 min
- time constant of cooler: T_{cooler}=200 s = 3,33 min
- time constant of room: T_{room} =1400 s = 23,33 min

5. TEMPERATURE CONTROLLER

The control algorithm consists of the following parts:

1. dead zone control
2. cascade control (room vs. heater/cooler control)
3. split-range control (heating or cooling)
4. PFC (Predictive Functional Control)

5.1 Dead zone control

The control actions can be reduced by allowing on the one hand a dead zone around a fixed set-point and on the other hand in the case of temperature control a dead zone which results in a split of one set-point in two set-points, the heating set-point as the lower limit and the cooling set-point as the upper limit. The control is not active within a dead zone around the set-point and between the set-points. The case when neither heating nor cooling is active is called floating. Fig. 10 shows the schema of the dead zone control with the variables:

$y_{r,room}$: set-point of room temperature

u_{HE}: manipulated variable of heat exchanger (heating or cooling)

y_{room}: controlled variable (room temperature)

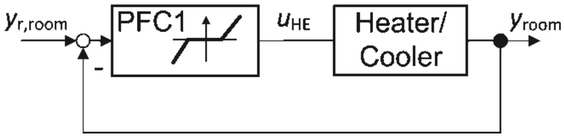

Fig. 10: Control scheme with dead zone

5.2 Cascade control

The manipulated variable of the room temperature is the inlet temperature to the room. This variable is controlled by the heater or by the cooler, therefore a cascade control structure (Richalet and O´Donovan, 2009, 2012) exists (Fig. 11).

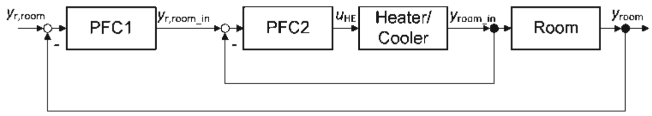

Fig. 11: Scheme of cascade control

The variables of the cascade control are:

$y_{r,room}$: set-point of room temperature
y_{room}: room temperature
$u_{room}=y_{r,room\_in}$: manipulated variable of room temperature equal to the set-point of the room-in temperature
y_{room_in}: temperature of inlet room air
u_{HE}: manipulated variable of actor (heater or cooler)

5.3 Split-range control (heating or cooling)

Depending on control error (i.e. whether the room temperature is bigger or smaller than its reference value) the air leaving the mixing chamber has to be heated or cooled. This kind of control is called split-range (Fig. 12).

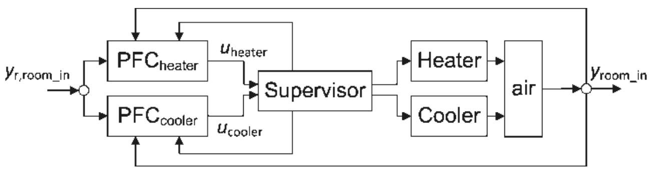

Fig. 12: Scheme of split-range control

The supervisor (Richalet and O´Donovan, 2009) decides about heating or cooling. A trivial condition is that heater and cooler does not work simultaneously. The variables in Fig. 12 are:

- $y_{r,room\_in}$: set-point of room inlet temperature
- y_{room_in}: room inlet temperature

- u_{heater}: manipulated variable of actor heater
- u_{cooler}: manipulated variable of actor cooler

5.4 PFC (Predictive Functional Control)

The control aim is to keep the room temperature within its upper and lower reference signals defined by the dead zone. For this task a controller is necessary which can achieve a set-value without any overshoot. PFC is a predictive control algorithm, which can ensure this control requirement and can be easier tuned than a PI(D) controller. PFC is a very easy realizable predictive control algorithm, which does not require any matrix inversion or require any numerical minimization of a cost function.

The principle of the PFC applied is that the controlled variable y achieves the reference trajectory at the target point using only one change in the manipulated variable u (Fig. 13).

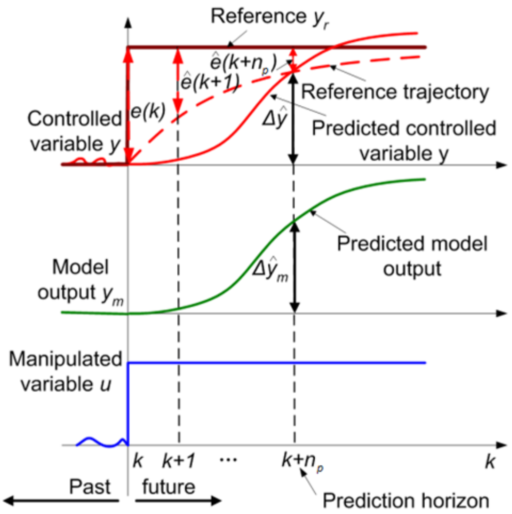

Fig. 13: PFC principle

The desired changes in the controlled output y during n_p steps can be calculated by supposing y reaches the reference trajectory at the target point (n_p step ahead) as follows:

$$\Delta\hat{y}(k+n_p\,|\,k)=\hat{y}(k+n_p\,|\,k)-y(k)=e(k)-\hat{e}(k+n_p\,|\,k) \quad (14)$$

where k is the discrete-time, y_r is the reference signal and the control error is

$$e(k)=y_r(k)-y(k)$$

The reference trajectory can be chosen as an exponential function for simplicity

$$\hat{e}(k+n_p\,|\,k)=\lambda_r^{n_p}e(k) \quad (15)$$

with the reduction ratio λ_r of the control error.

The reference trajectory provides the settling time $t_{95\%}= T_c$ for the closed-loop control system if $\lambda_r=\exp(-3\Delta t/T_c)$, where Δt is the sampling time.

From (14) and (15), the desired change in y becomes

$$\Delta\hat{y}(k+n_p\,|\,k)=\hat{y}(k+n_p\,|\,k)-y(k)=(1-\lambda_r^{n_p})e(k) \quad (16)$$

For simplicity the controlled processes (mixing chamber, heating, cooling) are approximated by first-order processes without dead time. Therefore the PFC algorithm is presented only for this process type.

The changes of y can be predicted also using the process model equation:

$$y_m(k)=-a_m\,y_m(k-1)+K_m(1+a_m)u(k-1) \quad (17)$$

with the process model output y_m and process model gain

K_m, time constant T_m and a_m= -exp(-Δt/T_m).

Supposing the actual input signal u is kept constant during the prediction horizon, the predicted model output becomes after n_p steps:

$$\Delta\hat{y}_m(k+n_p\,|\,k)=[1-(-a_m)^{n_p}]\cdot[K_m u(k)-y_m(k)] \quad (18)$$

Simple comparison between the predicted changes of the reference trajectory in Eq. (16) and the predicted changes of y_m in Eq. (18) leads to the manipulated variable equation (Richalet and O'Donovan, 2009, 2012):

$$u(k)=k_0[y_r-y(k)]+k_1 y_m(k) \quad (19a)$$

where:

$$k_0=\frac{1-\lambda^{n_e}}{K_m[1-(-a_m)^{n_p}]}, \qquad k_1=\frac{1}{K_m} \quad (19b)$$

6. SIMULATION

Simulations are important in order to get the best control structure and controller parameters. Following some simulations using the mathematical software Matlab are shown. The ambient (fresh air) temperature was simulated from collected real weather data (11 days in August 2010, Cologne, Germany). All dynamic processes were approximated by first-order lag terms without dead time. During the simulations it was assumed that the process and the controller models are the same. From the real process data a dynamic function is feed into the process to simulate the internal heat load, calculated out of the difference between the inlet air temperature and the outlet air temperature of the room.

Simulation 1: *100% fresh air and set-point 21°C*

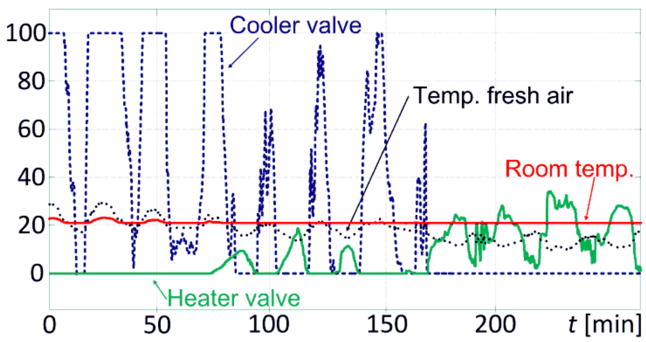

Fig. 14: Simulation 1: 100% fresh air, set-point 21°C

The simulated PFC control in Fig 14 represents the control of the initial parameters controlling on a set-point of 21°C and a flap position of 100% fresh air. At beginning of this control characteristic the cooler works at limit and the controlled variable cannot be forced to the set-point. This is due to the fact that the fresh air temperature is very high on a hot summer day in August and the cooler is at constructional limit. This limited cooling capacity on a very hot summer day was noticed before when looking on a previous real-time data recording.

Simulation 2: *100% fresh air and dead zone 19°C to 25°C*

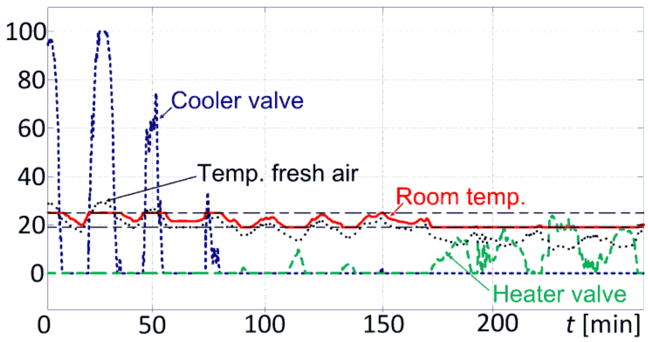

Fig. 15: Simulation 2: 100% fresh air, dead zone 19°C to 25°C

In Fig. 15 the floating temperature strategy is applied through the split into a heating set-point of 19°C and a cooling set-point of 25°C. The positive effect on the less activity of the valves results in the reduction of the deviation between the set-points and the fresh air temperature. The effect works stronger on the cooling performance while increasing the set-point from 21°C to 25 °C and still with a reduction of the valve activity, but less stronger on the heating performance decreasing from 21 °C to 19°C. The characteristic shows the floating of the temperature between the heating and the cooling set-points when no treatment by the heat exchangers is necessary. This cost free modus depends mainly on the interactions between the fresh air temperature, the internal heat load and the due of fresh air.

Simulation 3: *30% fresh air and set-point 21°C*

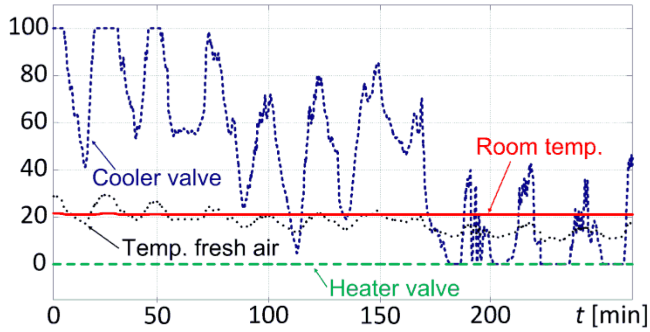

Fig. 16: Simulation 3: 30% fresh air, set-point 21°C

In Fig. 16 the fresh air due is simulated with the minimum of 30%. This results in a maximum due of recirculated air of 70%. The flap control is static with fixed position. The set-point for the room temperature is 21°C. When reducing the due of fresh air from 100% to 30% compared to Fig. 14 the heating performance will be reduced to zero, but the cooling performance is shifted to a permanent activity, even though when the fresh air temperature is below the set-point. This is due to the fact that the internal heat load shifts the temperature difference which needs to overcome by the cooler to higher values. This effect is stronger when a bigger amount of recirculated air is feed back to the process. The total amount of energy will be reduced, but notice here that cooling fluid is more expensive then the heating fluid.

Simulation 4: *30% fresh air and dead zone 19°C to 25°C*

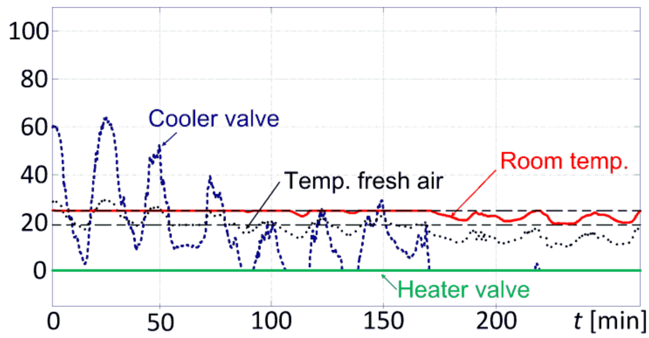

Fig. 17: Simulation 4: 30% fresh air, dead zone 19°C- 25°C

The floating temperature strategy is applied through the split into a heating set-point of 19°C and a cooling set-point of 25°C, see Fig. 17. This causes no change in the heating performance compared to Fig. 16, but in the cooling performance when increasing the upper set-point limit. When comparing these characteristic with Fig. 15 changing the fresh air due from 100 % to 30 % the big benefit is clearly seen on the heating performance, but the cooler still works more active.

The relative benefits on the control performance are presented for the simulations 1 to 4 in section 9.

7. REAL-TIME CONTROL

After experimental modeling and successfully simulating of the air handling unit of the Sanofi site A. Nattermann & Cie GmbH the simulated code of the PFC with floating temperature strategy was adapted to the PLC (Programmable Logic Control). One reason of the choice of the air handling system was a S7 based automation system. The code was programmed with SCL (Structured Control Language) and implemented as a functional block to the existing control program. The programming was done by internal teams; the implementation of the block was done by the supplier. Touching a running system is a critical point. The air handling units should guarantee permanently the GMP status; therefore all quality parameters are recorded and checked regularly. With qualified people of the supplier the modification was done quickly with the possibility to switch back to the standard control at any time. After placing the PFC block into the control program the air handling unit was still controlled by the PI controller until the final modification have been done during the running modus. After bumpless switching to the PFC controller, without any interference, the first control activity by the PFC was recorded in August 2011. Fig 18 shows a cut out of a control record of the PFC, when controlling the floating temperature strategy with set-point range of 19°C and 24°C. The control is very accurately and the set-points are not violated. With this kind of a sporadic data recording, which matches very well to the simulation results, the estimated benefit was calculated with around 25% of the reduction of the heating and cooling energy over the year for this specific air handling unit, mainly achieved through the applied floating temperature strategy, compared to the previous control on a fixed set-point of 21°C.

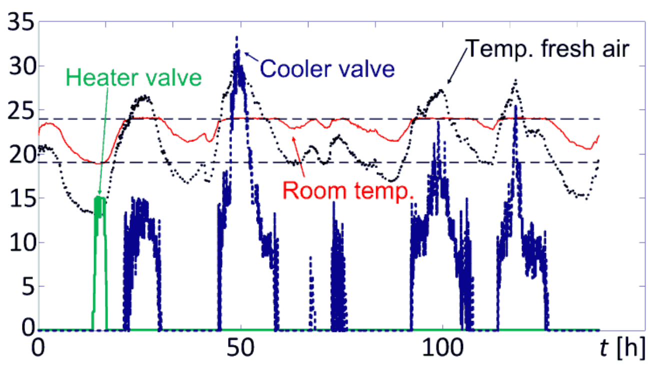

Fig. 18: Real-time control: PFC with floating set-point strategy

8. FLAP CONTROL

The fresh air temperature acts like a disturbance on the process. If no recirculation is possible and 100 % fresh air should be delivered to the room the treatment over the heat exchangers (heating- and the cooling-coils) needs to overcome the temperature difference between the incoming fresh air temperature and the desired set-point (e.g. 21°C). If a part of the outlet air of the room is recirculated to the process, the temperature difference between the desired set-point and the inlet temperature to the heat exchangers will be reduced. The mixed air temperature becomes the inlet temperature to the heat exchangers. The mixing process is a kind of heat recovery, because the good conditioned air with the desired temperature is fed back. Therefore - considering ideal case (no heat losses and no heat load) - the temperature difference which needs to overcome by the heat exchangers becomes smaller. The recirculation can be realized by using a kind of static recirculation, which results in a fixed due of the fresh and the recirculated air or by a dynamic flap control which acts between defined limits of air flows. The recirculation over dynamic flap control can reduce the energy consumption in a significant way.

So far the influence of air recirculating was shown but only with constant flap position. Better control performance can be achieved by setting the ratio fresh/recirculated air flow depending on the ambient air temperature. Dynamic flap control is the extension of the cascade control (Fig. 11) by an inner flap control loop, see Fig. 19. The inner loop is the control of the mixed air temperature (y_{air_mix}) with the flap (u_{flap}). This part is without consuming any kind of energy, excepting for the motors of the flaps, but no cooling or heating fluid. The middle loop is the control of the inlet temperature (y_{room_in}) using the heat exchangers (u_{HE}) for cooling and heating, but following the outer loop which gives the set-point for the room-in temperature. The treatment part by the heat exchangers is the cost intensive part and should be reduced by mixing the flap in a dynamic modus within the defined limits (e.g. minimum fresh air due of 30% as the lower limit of the fresh air flap).

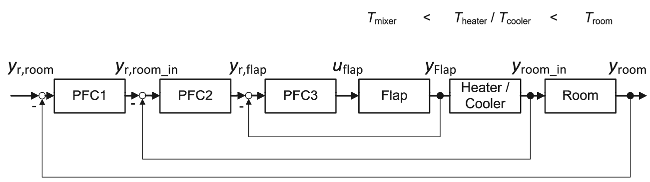

Fig. 19: Cascade control with dynamic flap control

Simulation 5: *Flap control with set-point 21°C*

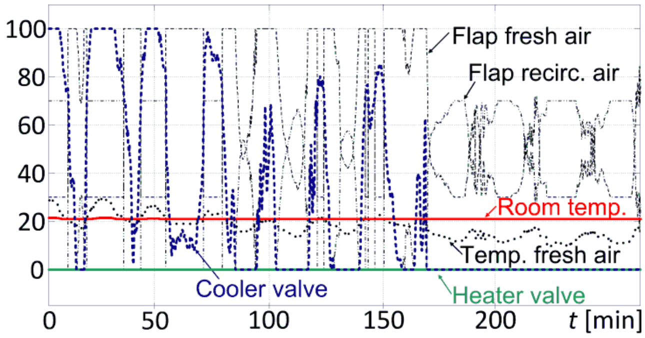

Fig. 20: Simulation 5: flap control, set-point 21°C

Fig. 20 shows the adaptation of a dynamic flap control. The actions of the fresh air flap are limited between 30 and 100%. The control of the recirculated air flap, inverse to the fresh air flap, results in limits between 70 and 0%. The room temperature is controlled on a set-point of 21°C. This characteristic shows the greatest benefit reducing the cooling and heating performance by dynamic flap control compared to the simulations 1 (Fig 14) and 3 (Fig 16) with fixed flap positions.

Simulation 6: *Flap control with dead zone 19°C to 25°C*

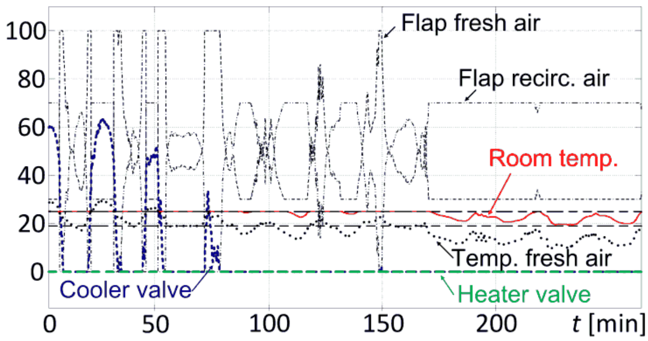

Fig. 21: Simulation 6: flap control, dead zone 19°C to 25°C

The simulation of Fig, 20 was extended by the floating set-point strategy in Fig. 21. This last simulated scenario achieves the best control performance and reduces the heating and cooling performance in a significant amount. The relative benefits on the control performance for the simulations 5 to 6 are presented in section 9.

9. PEFORMANCE CALCULATIONS

In this section the performance of the different control scenarios, showed in section 6 by the simulations 1 to 4 and in section 8 by the simulations 5 to 6, is evaluated by the calculation of the cumulative actions of the heating and cooling with the equations (20) and (21)

$$\textit{Cumulative actions heating valve} = \sum_{k=1}^{N} u_{\mathrm{HE\_heating}} \tag{20}$$

$$\textit{Cumulative actions cooling valve} = \sum_{k=1}^{N} u_{\mathrm{HE\_cooling}} \tag{21}$$

where N is the number of simulations. The calculations are done for each scenario and presented in Fig. 22, which shows the percentage savings related to the maximum consumer of the simulation. The calculated relative savings differ between the heating and cooling consumption and the total amount of both types of energy.

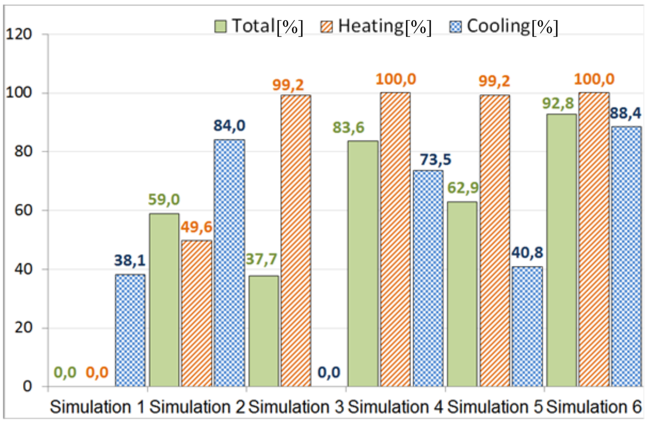

Fig. 22: Performance results with different control strategies based on the simulation in section 6 and 8

Analogues to Fig. 23 the same simulations have been repeated on real weather data based on a full year (Frankfurt, Germany).

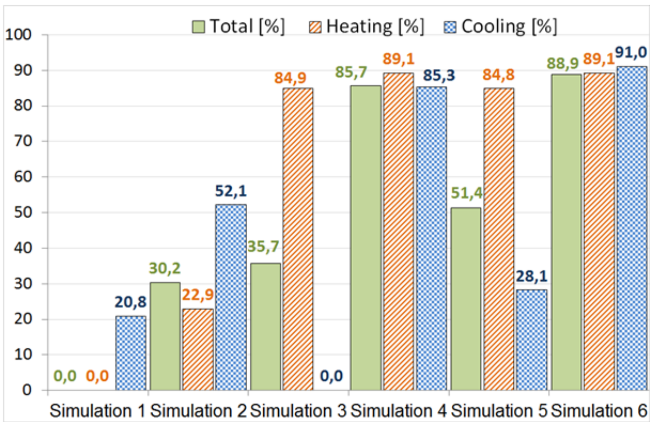

Fig. 23: Performance results with different control strategies with real weather data of a full year

The cumulative actions of the heat exchanger valve, giving the mass flow of the heat exchanger fluid, are directly proportional to the absolute energy consumption shown by (22); see also (1)

$$W\,[\mathrm{kWh}] \sim \dot{m}_{\mathrm{fluid}} \sim \sum_{k=1}^{N} u_{\mathrm{HE}} \tag{22}$$

10. CONCLUSIONS

A large pharmaceutical production site consumes every day a lot of energy. Beside the permanent inflations of the energy prices, energy consumption results in high energy cost. On the economical point of view the target is to reduce such costs and on the environmental point of view to reduce the CO_2 emissions. Both perceptions result in the fact that the energy consumption needs to be reduced. For a permanent responsible usage of our natural resources it is important to identify and stop wasteful processes. Therefore it is important to manage the usage of all kinds of energy and to be aware of the main consumers at the site, looking for potential to reduce the energy consumption. When tackling the HVAC system of the Sanofi site A. Nattermann & Cie GmbH the biggest energy consumer should be addressed, when running full speed over the year. The optimal mode of operation should be found in applying the best practices known as floating set-point, reduced mode at night and heat recovery. These control strategies for HVAC system can be realized with many standard solutions of different suppliers.

After the standard optimization the advanced control technologies follow, mostly to handle complex control tasks. The introduction of PFC for the HVAC process of this plant was proofed to be satisfactory, due to the well adapted methodology. Controlling a HVAC process is acting on a compound of different parts of heat exchange, with the cost intensive part of the classical heat exchanger treatment by the heating and cooling coils. The advantage of the PFC controller is the physical interpretability of the controller parameters using here the model of the thermal convexity of a heat exchange process; therefore the nonlinear function will be eliminated with this method. Another advantage is the simple realization of the manipulated signal constraints important for the limitation of the temperature floating between the set-points.

With the temperature controller the first step on HVAC process control with PFC was done, resulting in good savings using the floating set-point strategy. Although the temperature model is global, temperature characteristics vary over the world. Therefore the potential savings on the performance will be also shifted with different temperature characteristics to the applied strategies.

The next step will be the development of the simulated dynamic flap control to a real process, by integration of a further PFC control element to the control loop. HVAC process control with PFC should be completed by humidity control with a full integrated model to handle the coupled effect of temperature and relative humidity.

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