Clima p 216

Demand Guided Control of the Indoor Air Quality in Domestic Buildings

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Abstract

Intelligent ventilation techniques are able to maintain the air exchange rate in domestic buildings necessary to fulfil hygiene, health and building physics requirements. Controlled air ventilation is essential for minimising heating loads of low energy buildings. This paper deals with the development of a new indoor air condition control strategy for domestic buildings with a conventional heating system, non-central ventilation and outdoor blinds. The control uses information about the indoor and outdoor climate, user specific behaviour and solar energy gains. Firstly, a Matlab/Simulink model of the thermo-, hygrodynamical and air ventilation process behaviour is introduced. The model calculates the state variables of the indoor air of a room (temperature, humidity, air pressures, mass flows) depending on the actuators, existing interactions with adjacent rooms and internal as well as external disturbances. The model is used for simulation of the rather complex heating and ventilation system, the operation conditions and the closed loop system performance.

The demand guided control for the indoor air condition is achieved by using a hierarchical control structure. The supervisor co-ordinates the heating and air ventilation in such a way that the indoor air condition is kept within given limits. Depending on the indoor and outdoor air condition, the storage capacity of the building envelope, weather conditions and user behaviour, an appropriate operating regime (e.g. night setback, temperature setback) as wells as suitable setpoints for corresponding basic controllers are chosen by the supervisor control. A knowledge based strategy using experience is developed.

Furthermore, the setpoints are influenced by the presence of inhabitants. Since the indoor air condition is characterised by strong interactions among state variables, and exhibits strong non-linear behaviour depending on setpoint and ventilation , classical controllers for temperature and humidity are inefficient. For this reason a multivariable model reference adaptive control for air temperature and relative humidity is introduced as the basic control loop. For hygiene and health conservation reasons a certain air infiltration rate control is required. The demand guided control has been tested with the simulation model. Results while using the demand guided control for a test reference room are presented as well.

1 Introduction

Modern building design tries to meet the demand for minimal energy consumption and capital costs. Thermically high-insulated building envelopes, large insulating glazed areas, windows and doors with almost air-tight joints and a building-design which is suitable for utilising solar energy have led to a substantially reduced natural air circulation within the building. But due to inappropriate ventilation techniques a total amount of energy of up to 30% is wasted. Only intelligent ventilation techniques are able to maintain the necessary air infiltration rate in view of hygiene, health and building physics considerations. Controlled air ventilation is essential for minimising heating loads of low energy buildings. The development of efficient installation bus systems for domestic usage creates the conditions for a superior new quality in functionality and operation comfort of domestic technical equipment and appliances.

The presentation shows the development of a new indoor air condition control strategy for domestic buildings with a conventional heating system, non-central ventilation and outdoor blinds. This leads to a rather complex heating and air ventilation process where an appropriate control strategy must consider the indoor and outdoor air condition, user specific behaviour and solar energy use.

First, a process model for the thermal and hygrodynamical plant operation of the indoor air condition of domestic buildings is introduced in Section 2. The model programmed using *Matlab/Simulink* [Mat93] calculates the state variables of the indoor air of a room (temperature, humidity, air pressures, mass flows) depending on the actuators, existing interactions with adjacent rooms and internal as well as external disturbances. The model will be used for simulation of the rather complex heating and ventilation system behaviour, the operation conditions and the closed loop system performance.

The demand guided control of the indoor air condition is achieved by using a hierarchical control structure as shown in Section 3. The supervisor co-ordinates the heating and air ventilation in such a way that the indoor air condition is kept within given limits. Depending on the indoor and outdoor air condition, the storage capacity of the building envelope, weather conditions and user behaviour both an appropriate operating regime (e.g. night setback, temperature setback) and suitable setpoints for corresponding basic controllers are chosen by the supervisor control. A knowledge based strategy using experience is developed in Section 3.4.

Since the indoor air condition is characterised by strong interactions among state variables, and exhibits non-linear behaviour depending on the setpoint and ventilation, classical controllers for temperature and humidity are inefficient. For this reason a multivariable model reference adaptive control is introduced as a basic control loop as explained in Section 3.3.1. The controlled variables are the air temperature and humidity, i.e. control ensures that a given upper limit of relative humidity is not exceeded. If humidity is in given limits or especially since loose control of humidity is required, a certain air infiltration rate control is required for hygiene and health conservation reasons as explained in Section 3.3.2. The demand guided control has been tested with the simulation model as shown in Section 4. Results while using the demand guided control for a test reference room are presented as well.

2 Model of the Indoor Air Condition

2.1 Description of the System to be Controlled

The modelled room is characterised by a central heating system, where one radiator per room is equipped with an electrically operated valve, and a non-central ventilation system with a speed controlled fan. Additionally a supply opening in the envelope and relief opening in the door are implemented (Fig. 2.1). The window is equipped with outdoor blinds that can be positioned with regard to opened distance and fin angle.

2.2 Characterisation of the System

The system to be controlled (Fig. 2.2) contains the inputs radiator valve lift h, related fan speed n and position of the outdoor blinds l. Outputs are the air temperature ϑ_L and the indoor relative humidity φ_L. The disturbances are flow temperature ϑ_{VL}, outdoor air temperature ϑ_a; radiation i, internal heat gains $\dot{Q}_i$, outdoor relative humidity φ_a and internal moisture gains $\dot{m}_{D,i}$.

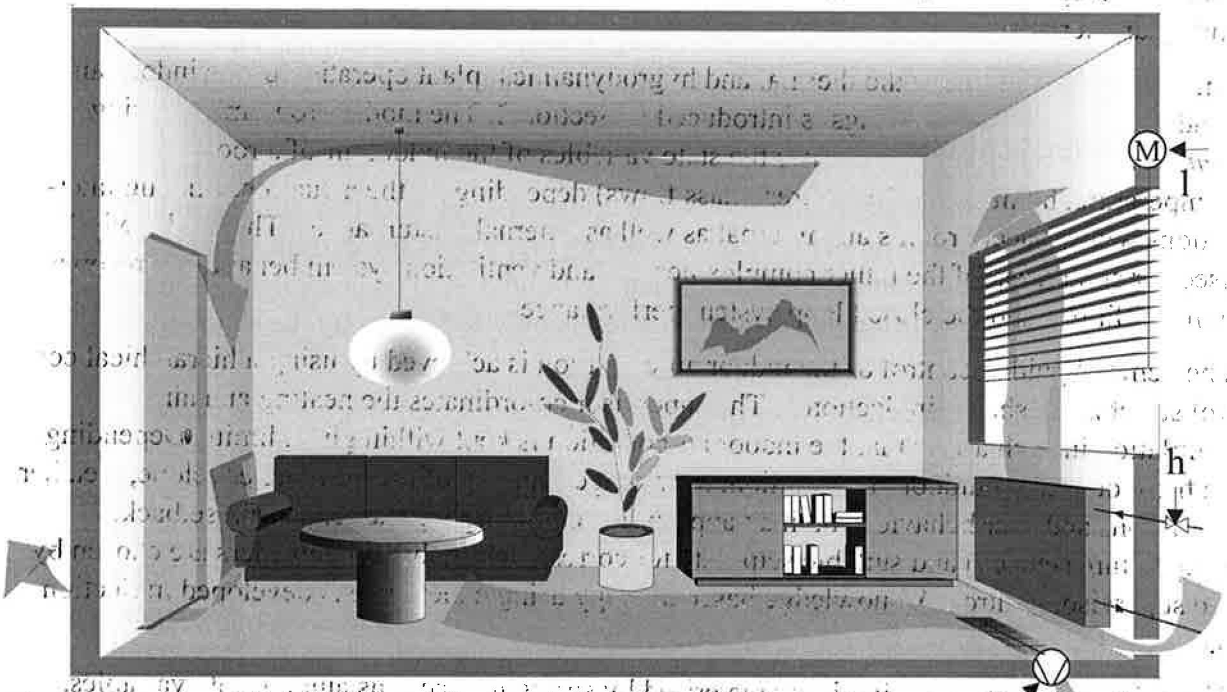


Fig. 2.1: Room with radiator, fan, and outdoor blinds

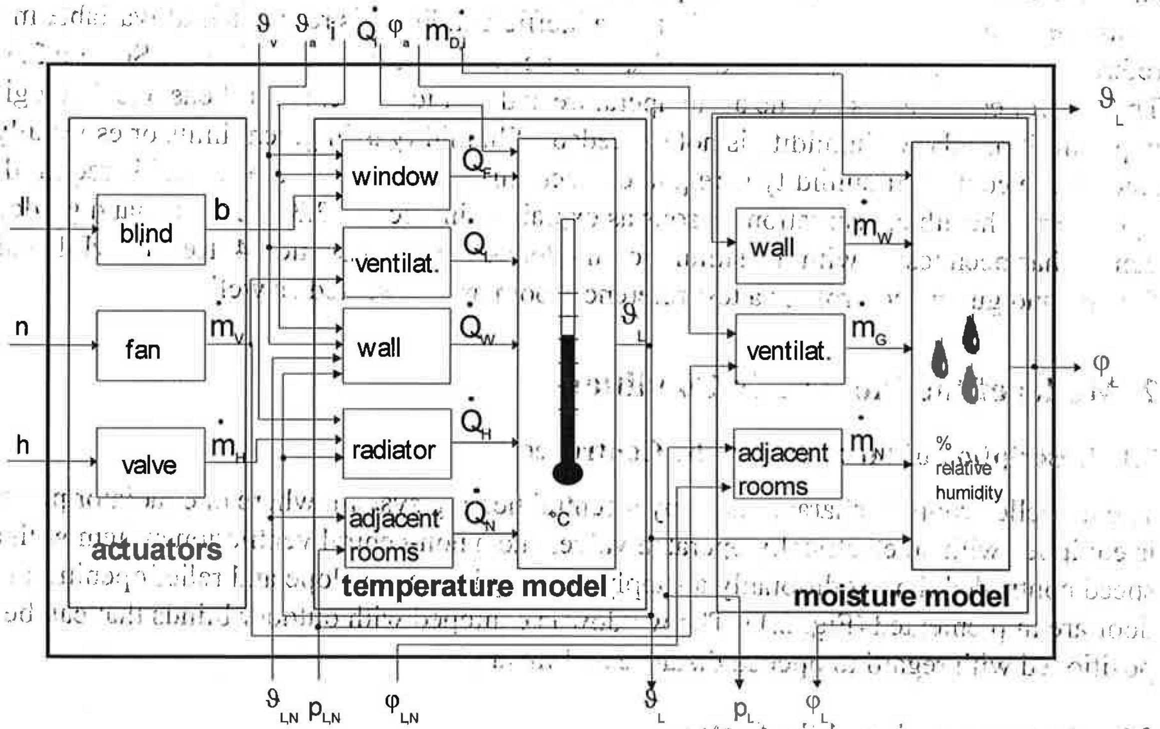


Fig. 2.2: Block structure of the room model

If a larger model of more than one room is planned with more room models, coupling variables have to be defined. These terms are the air temperature $\vartheta_{L,N}$, indoor relative humidity $\varphi_{L,N}$ and the indoor air pressure $p_{L,N}$.

A room model was developed to analyse the static and dynamic transfer behaviour. According to its physical effect the partial models of the various transfer systems are put together. Since the program *Matlab/Simulink* was used, the Laplace transform representation was adopted. In the s-plane the handling is more efficient and the outfit is much better than operating in the time domain. The couplings of various transfer functions like e.g. series and parallel connections are much easier to see. Another advantage is that it's possible to test all transfer functions for stability when cutting the coupling and using a defined input at this point.

2.3 Ventilation Model

Fig. 2.2 shows the influence of the ventilation on the relative humidity and the indoor air temperature. The ventilation model is characterised by Fig. 2.3. In the current version the supply and relief resistance are defined to be constant. In future it will be possible to influence the resistance. A controlled change of the ventilation is only supported by the speed controlled fan.

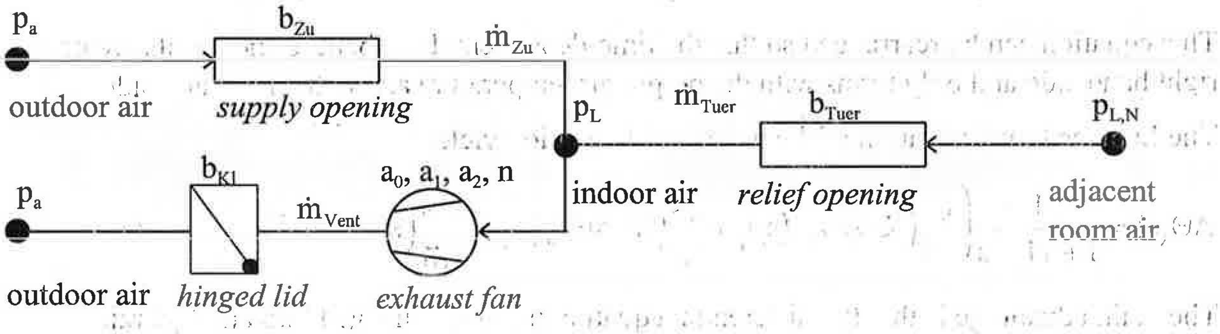


Fig. 2.3: Block diagram of the ventilation calculation

The room ventilation is calculated by one node and two mesh equations.

$$\dot{m}_{Zu} + \dot{m}_{Tuer} - \dot{m}_{Vent} = 0 \qquad (2.1)$$

$$(p_a - p_L) - (p_{L,N} - p_L) - (p_a - p_{L,N}) =$$
$$b_{Zu} \cdot \dot{m}_{Zu} \cdot |\dot{m}_{Zu}| - b_{Tuer} \cdot \dot{m}_{Tuer} \cdot |\dot{m}_{Tuer}| - (p_a - p_{L,N}) = 0 \qquad (2.2)$$

$$(p_a - p_L) - (p_a - p_L) =$$
$$b_{Zu} \cdot \dot{m}_{Zu} \cdot |\dot{m}_{Zu}| - \left((a_2 - b_{Kl}) \cdot \dot{m}_{Vent}^2 + a_1 \cdot \dot{m}_{Vent} \cdot n + a_0 \cdot n\right) = 0 \qquad (2.3)$$

where $b_{Kl} \to \infty$ if $\dot{m}_{Vent} < 0$.

The resistance coefficients b_{Zu}, b_{Kl} and b_{Tuer} depend on the local resistance coefficients ζ

$$b = \zeta \cdot \frac{1}{2 \cdot \rho \cdot A^2} \qquad (2.4)$$

and are assumed to be known.

If the fan is out of operation, the equations (2.1) to (2.3) are valid for the natural ventilation computation as a result of the pressure differences between the inside and outside of the building. The thermal buoyancy is neglected.

The calculated air mass flows are treated in both temperature and moisture models. They have a decisive influence on the dynamics and static of the room transfer behaviour.

2.4 Temperature Model

2.4.1 Partial model of the room air

Extensive investigations have been done with the simulation program ResCUE [Han94], computing the room air flow. The room has been divided in consideration of its flow behaviour into several zones (multi-zone model). At the current time no simplified description of the room air flow exists. This is caused by a strong dependence of the room air flow on the boundary conditions like e.g. wall surface temperatures, solar gains, positions of internal gains, supply speed and supply direction, position of internal flow drags.

Presently the air is considered as an ideal mixed zone. The partial differential equation (2.5) is the basis of the temperature model

$$m_L \cdot c_p \cdot \frac{\partial \vartheta_L}{\partial t} = \sum_{i=1}^{\infty}\left(\alpha \cdot A \cdot [\vartheta_{Ob} - \vartheta_L]\right)_i + \sum_{j=1}^{\infty}\left(\dot{m}_L \cdot c_p \cdot [\vartheta_{Zu} - \vartheta_L]\right)_j + \sum_{k=1}^{\infty}\dot{Q}_{i,k} \quad (2.5)$$

This equation can be rearranged so that the time-dependent disturbances and inputs are on the right hand side and only terms with the output air temperature are on the left hand side.

The Laplace transformation of the differential equation yields

$$\Delta\Theta(s) = \frac{1}{1 + T1_L \cdot s}\left(\sum_{i=1}^{\infty}\left(K \cdot \Delta\Theta_{Ob}(s)\right)_i + \sum_{j=1}^{\infty}\left(K \cdot \Delta\Theta_{Zu}(s)\right)_j + \sum_{k=1}^{\infty}\left(K \cdot \Delta\dot{Q}_i(s)\right)\right). \quad (2.6)$$

The main advantage is that the differential equation has been changed into an algebraic equation. The surface temperatures cannot be controlled. They essentially depend on the location and that is why they are not suitable for a measurable variable. These temperatures can be modelled and attributed to measurable homogeneous variables.

2.4.2 Partial model of a wall

As there is a great variety of wall structures it is rather difficult to model easily the exact wall transfer behaviour. The differential equations of a sandwich wall in the s-plane are solved in [Kna92]. Output is the wall surface temperature, inputs are the ambient temperatures. The transfer functions are transcendental.

A simplified approach for modelling the wall surface temperature is given by a first-order system with delay.

In the s-plane one obtains

$$\Delta\Theta_W(s) = \frac{1}{1 + T1_W \cdot s} \cdot \left(K_L \cdot \Delta\Theta_L(s) + K'_{F,s} \cdot \Delta i_{dir+diff}(s)\right) + \ldots$$

$$\ldots + \frac{e^{-T_{LW} \cdot s}}{1 + T1_W \cdot s} \cdot \left(K_a \cdot \Delta\Theta_a(s) + K'_{W,s} \cdot \Delta i_{dir+diff}(s)\right). \quad (2.7)$$

The parameter can be approximated via frequency response. According to equation (2.7) a simple transfer function was found and it's possible to describe an ordinary wall.

2.4.3 Partial model of a radiator

To begin with, a radiator can be described as an ideal mixed chamber

$$m_{HW} \cdot c_p \cdot \frac{\partial \vartheta_{HW}}{\partial t} = \dot{m}_{HW} \cdot c_p \cdot \left(\vartheta_{Z,HW} - \vartheta_{HW}\right) - \alpha_i \cdot A_i \cdot \left(\vartheta_{HW} - \vartheta_{HK}\right). \qquad (2.8)$$

The radiator temperature can be modelled in different ways. A first approximation is given by an easy static model and a second by a first-order system for more detailed computations like e.g. a cast iron radiator. An example of the first named model is characterised by

$$\vartheta_{HK} = \frac{\alpha_i \cdot A_i}{\alpha_i \cdot A_i + \kappa_a \cdot A_a} \cdot \vartheta_{HW} + \frac{\kappa_a \cdot A_a}{\alpha_i \cdot A_i + \kappa_a \cdot A_a} \cdot \vartheta_L. \qquad (2.9)$$

As a good approximation the heat conduction resistance can be neglected. This leads to a simple relation of the coupling factor

$$\kappa_a = \alpha_a. \qquad (2.10)$$

In consideration of $\alpha_i >> \alpha_a$ one yields in the s-plane

$$\Delta\Theta_{HW} = \frac{\frac{1}{\dot{m}_{HW} \cdot c_p + \alpha_a \cdot A_a}}{1 + T1_{HK} \cdot s} \cdot \left(\dot{m}_{HW} \cdot c_p \cdot \Delta\Theta_{Z,HW} + \alpha_a \cdot A_a \cdot \Delta\Theta_L\right). \qquad (2.11)$$

The return temperature is a system variable depending on the water mass flow, the flow and room air temperature. In verifications it was estimated that this model is to coarse. The radiator was divided into n-parts where all parts are in a series connection. For that purpose one can transform equation (2.11) into the representation

$$\Delta\Theta_{HW,n} = G(s)^n \cdot \left(\dot{m}_{HW} \cdot c_p\right)^n \cdot \Delta\Theta_{Z,HW} + \alpha_a \cdot \frac{A_a}{n} \cdot \sum_{z=1}^{n} \left(\left(\dot{m}_{HW} \cdot c_p\right)^{z-1} \cdot G(s)^z\right) \cdot \Delta\Theta_L$$

where

$$G(s) = \frac{\frac{1}{\dot{m}_{HW} \cdot c_p + \alpha_a \cdot \frac{A_a}{n}}}{1 + \frac{m_{HW} \cdot c_p}{n \cdot \dot{m}_{HW} \cdot c_p + \alpha_a \cdot A_a} \cdot s}.$$

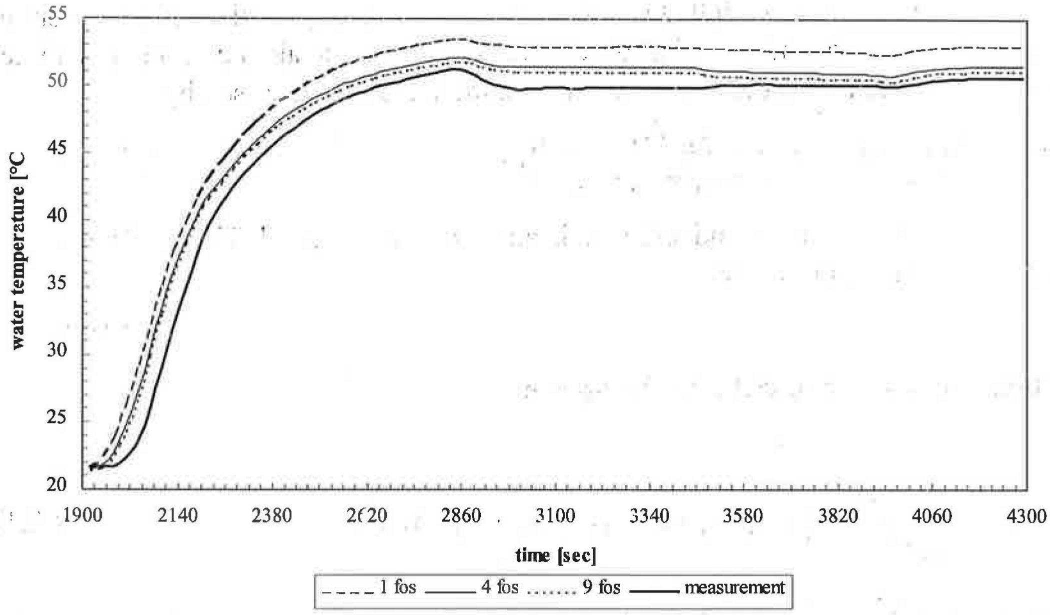


Fig. 2.4: Comparison of modelled and measured temperatures. The radiator was modelled as a series connection of n=1, n=4 and n=9 first-order systems (fos), respectively.

Measurements has been done in a radiator laboratory of the Institute for Thermodynamics and Technical Installation of Buildings. This laboratory is equipped with five boxes supplied by a heating system. Temperature sensors are arranged at all meaningful points, in the air and in the heating system.

A good agreement between measured and modelled values was reached when the radiator was assumed to be a series connection of nine identical first-order systems. Both the time constant and the gain must adapt in this case. It could be estimated that a series connection of four first-order systems is sufficient (Fig. 2.4).

2.5 Moisture Model

2.5.1 Partial model of the room air

The partial differential equation of the moisture the model based on is as follows

$$\frac{\partial m_{D,RL}}{\partial t} = \sum \dot{m}_{D,zu} - \sum \dot{m}_{D,ab} \,. \qquad (2.12)$$

The supplied moisture mass flows consist of the two components internal moisture gains and entered moisture connected with ventilation processes (2.13). The exhausted moisture mass flows consist of the moisture absorption of the wall and the moisture of the exhaust air (2.14)

$$\sum \dot{m}_{D,zu} = \sum_{l=0}^{q} \left(\dot{m}_{D,i}\right)_l + \sum_{l=0}^{m} \left(\dot{m}_{D,L,zu}\right)_l \qquad (2.13)$$

$$\sum \dot{m}_{D,ab} = \sum_{l=0}^{n} \left(\dot{m}_{D,W}\right)_l + \sum_{l=0}^{k} \left(\dot{m}_{D,L,ab}\right)_l \,. \qquad (2.14)$$

A closer inspection of $\dot{m}_{D,L}$ yields the connection of both supplied and exhausted moisture mass flows and the air mass flow as a result of the ventilation calculation

$$\dot{m}_{D,L} = x_D \cdot \dot{m}_L = x_D \cdot \lambda \cdot m_L . \qquad (2.15)$$

2.5.2 Partial model of the wall

The moisture absorption of walls depends on the water vapour partial pressure. Many calculations demonstrate that the vapour diffusion through a solid external wall is of less meaning in the time-independent case. That's the reason why the wall can be assumed as a first-order system that does not transmit the moisture to the outside surface. These assumptions are valid for hygrodynamical conditions in the wall.

Measurements by Setzer/Hohmann [Set94] show that an implementation of a moisture buffer function is useful

$$F(t) = \frac{\Delta g_{W,W}(t)}{\Delta \varphi_L(t)} = a \cdot e^{-\left(\frac{t}{T}\right)^m} . \qquad (2.16)$$

As a good approximation it can be assumed that $m \approx 1$. Given that $x_L << 0.6222$ and $x_S << 0.6222$ the following transformation is permissible

$$\varphi_L = \frac{p_D(x_L)}{p_S(\vartheta_L)} = \frac{x_L \cdot p_L}{(x_L + 0{,}6222)} \cdot \frac{(x_S + 0{,}6222)}{x_S \cdot p_L} \approx \frac{x_L}{x_S} . \qquad (2.17)$$

Furthermore, the following is valid [Kue95]

$$g_{W,W} = \int (\dot{m}_{D,W} \cdot A_W) dt . \qquad (2.18)$$

With the above mentioned assumption one obtains a first-order system for the wall. The input is the relative humidity and the output is the absorbed moisture mass. This yields

$$T \cdot \dot{g}_{W,W} + g_{W,W} = K \cdot \varphi_L \approx \frac{K}{x_S} \cdot x_L . \qquad (2.19)$$

The moisture mass flow transfer between the wall and the air is now given by

$$\Delta \dot{m}_{D,W}(s) = \frac{\frac{K}{x_S} \cdot s}{1 + T \cdot s} \cdot \Delta x_L(s) . \qquad (2.20)$$

3 Design of the Demand Guided Control

3.1 Control Objectives

A principal requirement controlling indoor air condition is thermal comfort with the components air temperature, humidity, air flow, and heat radiation. Thermal comfort depends on various physical variables and is an individual perception. Nevertheless, there are recommended values for these variables [Reck95]:

rate for outdoor air infiltration rate : $\lambda \geq 0.5\ h^{-1}$

Demand for fresh air depends on room usage, number of persons present, size of the residence and other variables. Because of mould growth and mites the mean value of the rate for outdoor air change should not fall below 0.5 h^{-1} even when the room is empty [Sch95]. Control of the mean value of λ with a user given setpoint $\lambda_{soll} \geq 0.5\ h^{-1}$ makes sense.

relative humidity : $35\% \leq \varphi_L \leq 70\%$

Air humidity influences evaporation on the skin. A humidity ratio of 12 g/kg should not be exceeded according to the sultriness function and because a fall below dew point could occur. If the relative humidity is too low, then clothes, carpets etc. dry out and dust development increases. Given high indoor humidity gains, automatic control ensures that a given max. value is not exceeded.

temperature : $20°C \leq \vartheta_L \leq 23°C$

Thermal comfort depends strongly on the sensed temperature ϑ_O, calculated roughly by using the radiation temperature ϑ_S of the surrounding surfaces including radiators and the local air temperature ϑ_L. For easy physical actions and normal clothes the equation $\vartheta_O = 0.5\ (\vartheta_S+\vartheta_L)$ can be used. For simplification it is assumed from now on, that sensed temperature in a well heated low energy building equals room air temperature. For the operating mode winter this temperature should be stabilised matching ϑ_{soll}. The simplification is not suitable for special cases like night cooling and transition season, though.

control system design criteria

Single room controllers in well-known building automation systems enable frequent setpoint changes for temperature in dependence on presence of inhabitants and e.g. night setback. It should be stressed that although temperature control systems are sometimes required to respond to step reference inputs when operating, this is not very common. Step inputs are used for test purposes, not because they represent the typical input, but because the response to them is informative and easily obtained. Overshoot beyond the final value, oscillations and position errors which are not negligible give rise to additional energy consumption. Consequently the design aims are nearly aperiodic step response without overshoot and oscillations at all desired temperature levels.

3.2 Properties of the Controlled System and Control Possibilities

The system room (plant) has certain properties, complicating the controller design:

structure with interactions and non-linear coupled parameters, time variant

The plant has a non-linear and time variant characteristic due to certain parameters (time constants and gains depend on temperature and air flow). The time variant air flow effects aerodynamic as well as thermal conditions for the time constants of the room. By increasing the fan speed step-wise an increase or decrease of the temperature and the relative humidity might occur, depending on the state of the air indoors and outdoors. Given this, the plant has non-linear behaviour. Certain conditions cause a sign reversal within the transfer function of the plant. Therefore, linearisations are difficult to apply. The control system design mostly neglects these properties which may result in oscillations, peak overshoot and poor performance.

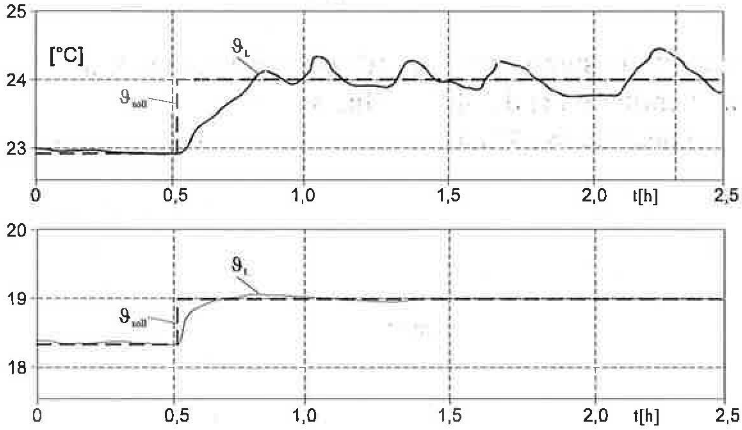


Fig. 3.1:

PI-control on different setpoints

The simulation results of conventional PI-control with the described room model are shown in Fig. 3.1. The step response in the second diagram shows an satisfactory behaviour, because the controller is adjusted to setpoint ϑ_{soll}=19°C. On higher values of temperature closed loop performance is unacceptable (first diagram).

great time constant differences, non minimum phase humidity controlled system

The transient process of the temperature when operating the radiator valve step-wise is characterised by small time constants of the indoor air warm up process and large time constants of the surrounding masonry warm up process. During the warm up, considering the humidity storage capacity of the walls, the relative humidity φ_L and the vapour pressure p_D will fall on a short term basis. Finally, however, the relative humidity will increase, since the lower vapour pressure will cause steam to be released out of the walls. Such a non-minimum phase behaviour complicates the automatic control design for the humidity.

achieving control objectives with restricted controllability

In addition to the mentioned difficulties, the indoor air condition controllability is rather restricted. First neglecting wall humidity storage capacity, the controlled system indoor air for the operating mode winter can be interpreted as a mixing chamber with pre-heater. According to the h-x-diagram the transition between the indoor air states $z_i(t_1)$ and $z_i(t_2)$, $t_1<t_2$, follow a solid line as shown in figure 3.2. Note that the indoor air condition $z_i(t_1)$ will not be in the target or comfort zone e.g. at the end of the night setback. The outdoor air (state z_a) passes the pre-heater and the supply condition is characterised by z_{Zu}. In winter it should be at a higher temperature than the desired indoor air state and at lower moisture content. The air leaving the

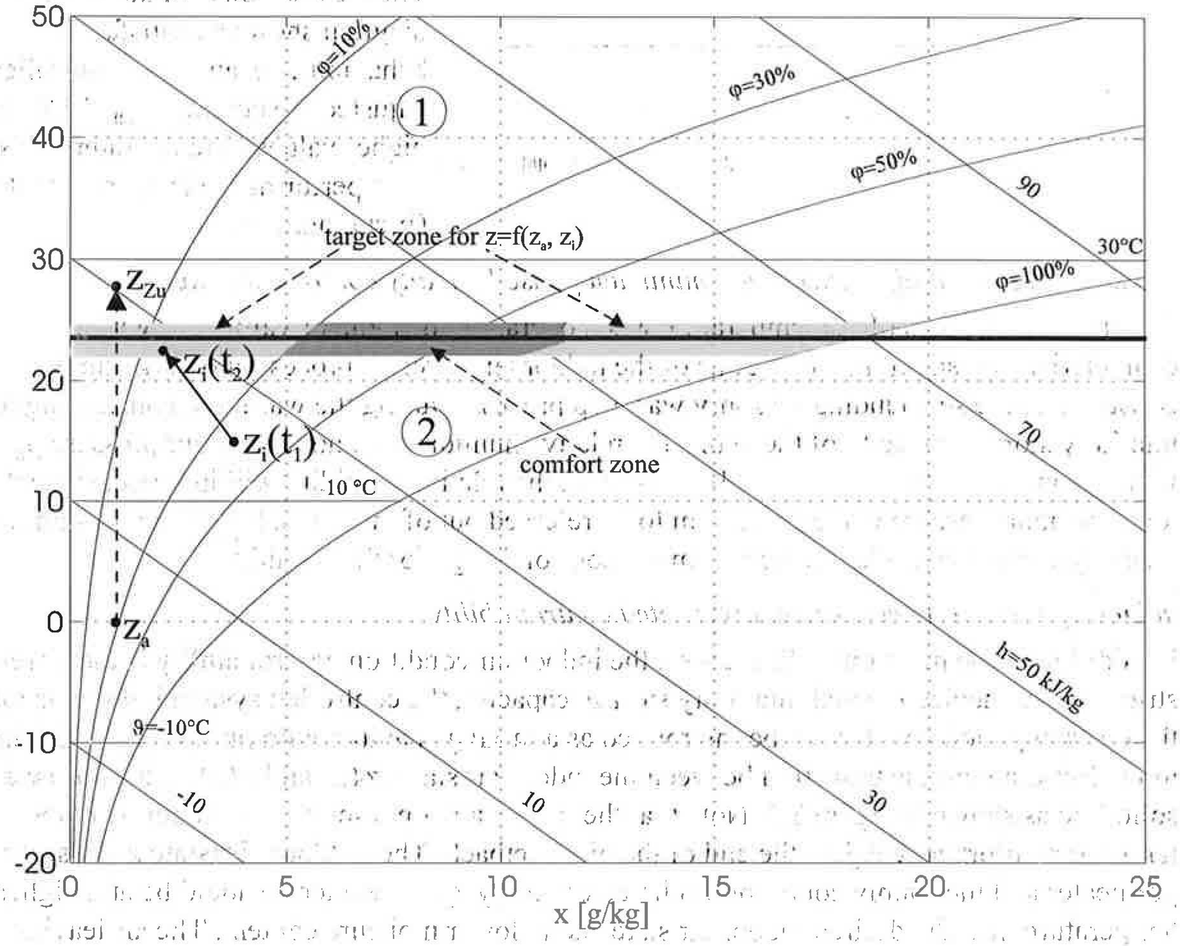
Fig. 3.2: h-x diagram

pre-heater forces $z_i(t_1)$ to $z_i(t_2)$. The desired comfort zone is not reachable with the states as shown in the figure. It is only possible, to reach the grey target zone. When wall humidity storage capacity is taken into consideration, the solid line in the h-x-diagram becomes distorted, the behaviour is non-minimum phase. Related to the indoor and outdoor air condition ($z_i(t_1)$, z_a), the reachability of the target zone in Fig. 3.2 is shown roughly by the placement of $z_i(t_1)$ and z_a in ① and ② respectively.

summer season

Strong limitations apply to the controllability in summer time (both $z_i(t_1)$ and z_a in ①). In order to avoid over heating of the room, ventilation and outdoor blinds have to be activated with view to outdoor air conditions in the near future. The comfort zone is rarely reachable.

winter season



Concerning temperature, this case (both $z_i(t_1)$ and z_a in ②) is easy to handle. The target zone is always reachable. However, the comfort zone is rarely reached due to low air moisture content. The desired air infiltration rate is ensured on a time average level by non-continuous ventilation.

transitional season

This case ($z_i(t_1)$ and z_a in ① and ② respectively or vice versa) might occur in summer or winter time. For cold weather conditions ($z_i(t_1)$ in ① e. g. due to solar gains and z_a in ②) the desired room air temperature is ensured by ventilating. Under warm weather conditions (z_a in ① and $z_i(t_1)$ in ②) the desired room air temperature is ensured by ventilating as well. The near future outdoor temperature tendency must be considered in both situations. The target zone is always reachable, though the comfort zone is reached only under certain suitable indoor and outdoor air conditions.

3.3 Basic Control Components

Basic controllers for stabilisation of room air temperature, relative humidity and air infiltration rate in dependency on supervisory control commands are arranged in a process-near information level. As detailed below in paragraph 3.3.1 a multivariable model reference controller is proposed to control the coupled process variables room air temperature and relative humidity. Disconnecting the sub-controller for humidity yields a single-input/single-output-control system for the room air temperature. If humidity is in given limits a controller for the air infiltration rate is shown in paragraph 3.3.2.

3.3.1 Multivariable model reference adaptive control of room air temperature and relative humidity

Single room control systems do not take into account the coupling of the process variables which often results in a poor performance of regulation. Moreover, setpoint dependent transfer characteristics of temperature and humidity and different time characteristics of the reaction of the room air and the surrounding walls recommend adaptive control strategies. These methods are capable of adapting to changing conditions of thermal, hygrodynamical and ventilating processes also in the case of first commissioning of the control systems. Due to the strong coupling relation between the process variables room air temperature and relative humidity they cannot be considered separately. These structural connections must be taken into account in the controller design. Modern control systems on the basis of process fieldbuses with distributed sensors and actors and powerful processors are suitable for implementation of advanced control strategies such as adaptive control systems.

In the following a time-discrete multivariable adaptive controller for temperature and relative humidity is described [Bier96]. The structure of a discrete multivariable model reference adaptive control system (MRAC) is shown in Fig. 3.3. The basic control loop consists of a linear controller and the multivariable process. A pre-specified multivariable reference model calculates the desired closed-loop behaviour for any bounded reference signal $\underline{r}(k)$. The speciality of these systems is that the output of the process follows the output of the reference model. Therefore, the reference model is connected in parallel to the basic control loop. The unknown controller parameters $\underline{\gamma}(k)$ can be adapted in such a way that the error signal $\underline{\varepsilon}(k)$ between reference model and process will be minimised. A suitable choice of the reference model results in the decoupling of the process variables, that is both process variables temperature and humidity can changed independently of each other.

The following applies for the reference model according to Fig. 3.4:

$$\underline{\tilde{y}}_k = A_M \underline{\psi}_{k-1} + B_M \underline{r}_{k-1} . \tag{3.1}$$

The model output

$$\underline{\tilde{y}}_k = \begin{pmatrix} \tilde{y}_{1,k} \\ \tilde{y}_{2,k} \end{pmatrix} \tag{3.2}$$

represents the desired process output at each time k

$$\underline{y}_k = \begin{pmatrix} y_{1,k} \\ y_{2,k} \end{pmatrix} = \begin{pmatrix} \vartheta_{L,k} \\ \varphi_{L,k} \end{pmatrix} . \tag{3.3}$$

$\underline{\psi}_{k-1}$ denotes a column vector with retarded values of the model outputs and the column vector $\underline{r}_{k-1}$ includes the retarded reference values. $\underline{\psi}_{k-1}$ denotes a column vector with retarded values of the model outputs and the column vector $\underline{r}_{k-1}$ includes the retarded reference values.

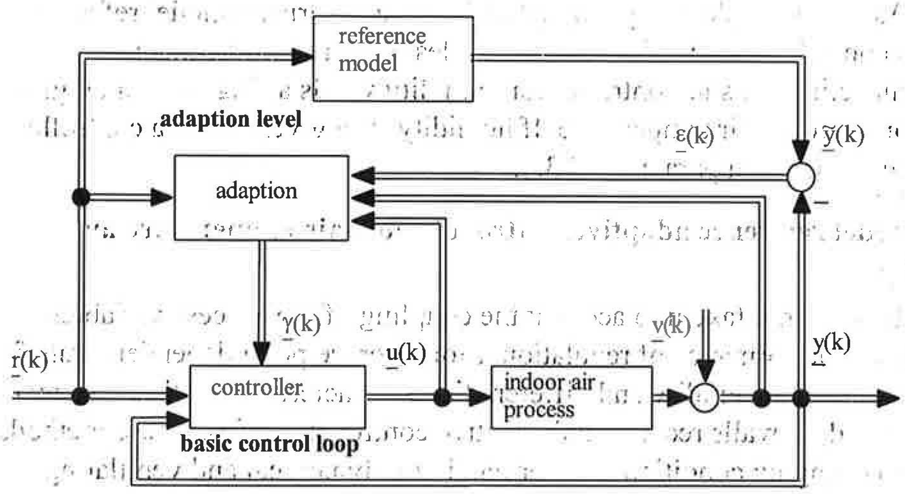


Fig. 3.3:

Basic structure of a discrete multivariable model reference adaptive control system

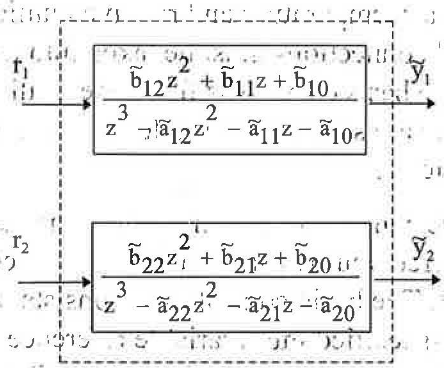


Fig. 3.4:

Discrete two-variable reference model

The control algorithm is derived from a simplified design model of the process according to Fig. 3.5. Such a model is used in [Bil84] to design an adaptive controller for temperature and humidity of a simulation chamber for environmental processes. This model describes the main effects sufficiently and can be easily handled with regard to its number of parameters.

In [Bil84] the influence of the surrounding walls is not included; here this effect is taken into account as slow changes of non-measurable disturbances $\underline{v}(k)$.

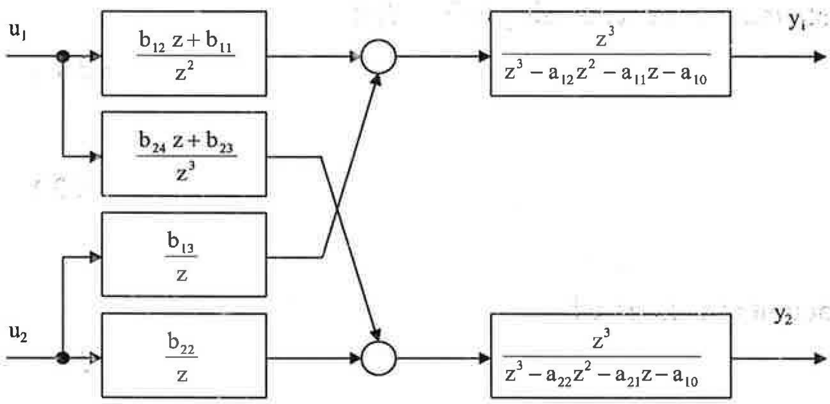


Fig. 3.5:

Discrete design model of the process

The simplified process model is denoted by

$$\underline{y}_k = A_p \underline{\Phi}_{k-1} + B_p \underline{u}_{k-1} \tag{3.4}$$

with the matrices

$$A_p = \begin{pmatrix} a_{12} & a_{11} & a_{10} & 0 & 0 & 0 & b_{11} & 0 & b_{13} \\ 0 & 0 & 0 & a_{22} & a_{21} & a_{20} & b_{24} & b_{23} & 0 \end{pmatrix} \text{ and } \quad B_p = \begin{pmatrix} b_{12} & 0 \\ 0 & b_{22} \end{pmatrix} \tag{3.5}$$

and the vectors

$$\underline{\Phi}_k = (y_{1,k} \; y_{1,k-1} \; y_{1,k-2} \; y_{2,k} \; y_{2,k-1} \; y_{2,k-2} \; u_{1,k-1} \; u_{1,k-2} \; u_{2,k})^T$$

and

$$\underline{u}_k = \begin{pmatrix} u_{1,k} \\ u_{2,k} \end{pmatrix} = \begin{pmatrix} h_k \\ n_k \end{pmatrix} \tag{3.6}$$

as the process input at each time k. The parameters of A_p and B_p are assumed to be unknown but constant, B_p is positive definite.

The discrete process outputs air temperature and relative humidity $\underline{y}_k$ and the model outputs $\underline{y}_k$ form the error signal

$$\underline{\varepsilon}_k = \underline{\tilde{y}}_k - \underline{y}_k = \begin{pmatrix} \varepsilon_{1,k} \\ \varepsilon_{2,k} \end{pmatrix}. \tag{3.7}$$

The control variable $u_{1,k}$ for actuating the radiator valve is

$$u_{1,k} = \underline{\gamma}_{1,k}^T \underline{x}_{1,k} \tag{3.8}$$

and the second component of the control vector $u_{2,k}$ for actuating the fan is

$$u_{2,k} = \underline{\gamma}_{2,k}^T \underline{x}_{2,k} \tag{3.9}$$

whereby $\underline{\gamma}_{1,k}$ and $\underline{\gamma}_{2,k}$ are time-varying controller parameters. According to [Bil84] the variation of the parameters of the sub-controller 1 for room air temperature

$$\underline{\gamma}_{1,k} = \underline{\gamma}_{1,k-1} + \frac{Q_1 \underline{x}_{1,k-1} \tilde{\varepsilon}_{1,k}}{1 + b_{1max} \underline{x}^T_{1,k-1} Q_1 \underline{x}_{1,k-1}} \tag{3.10}$$

and of the sub-controller 2 (relative humidity controller)

$$\underline{\gamma}_{2,k} = \underline{\gamma}_{2,k-1} + \frac{Q_2 \underline{x}_{2,k-1} \tilde{\varepsilon}_{2,k}}{1 + b_{2max} \underline{x}^T_{2,k-1} Q_2 \underline{x}_{2,k-1}} \tag{3.11}$$

form an asymptotic stable adaptive control system. In these expressions $\tilde{\varepsilon}_{i,k}$ are generalised error signals which are generated from the output errors by dynamic compensators. The specification of these compensators follows from conditions of stability theory. Q_1 and Q_2 are symmetric and positive definite weighting matrices. From a practical point of view they have a diagonal structure whereas the parameters b_{1max} and b_{2max} are maximum values of the process step responses of the first sampling interval.

The vectors $\underline{x}_{1,k}$ and $\underline{x}_{2,k}$ consist of values from the filtered reference signals, the sampled and retarded process signals and the control variables. Due to the coupled structure of the process both control signals $u_{1,k}$ and $u_{2,k}$ contain the other control signal respectively. The calculation of the control signals is carried out in a successive manner.

The output of the control variables $\underline{u}_k$ is limited by technical restrictions. Therefore, a suitable interpretation is necessary, because neither active cooling nor dehumidifying is possible. The controller parameters are „frozen" if the control signals are limited. Furthermore, a bumpless changeover of manual and automatic operation is realised.

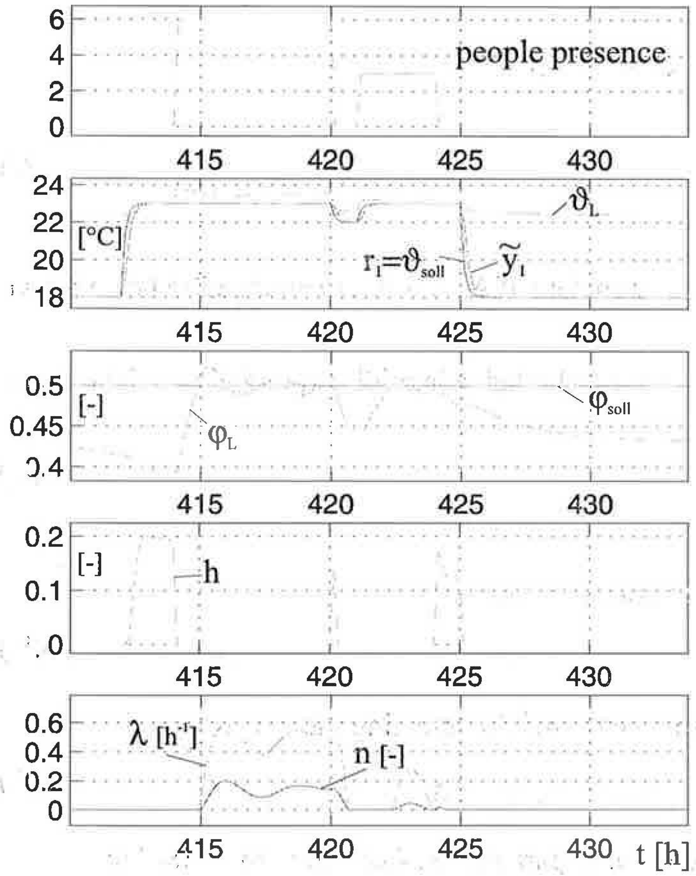


Fig. 3.6:

Simulation results of the adaptive control system

Fig. 3.6 depicts some simulation results of different situations and the presence of 6 or 3 persons with internal heat and moisture gains (first diagram). The second diagram shows the room air temperature ϑ_L and the setpoint ϑ_{soll} very close together with the model output y_1. Unfortunately, these two curves are not exactly to distinguish in the diagram. The third diagram depicts the relative humidity φ_L and its setpoint φ_{soll}. The two last diagrams show the control signals h and n as well as the air infiltration rate λ. The sampling period for all process signals is T = 20 s. ϑ_{soll} = 23 °C is valid for 413 h $\le t \le$ 420 h and 421 h $\le t \le$ 425h. φ_{soll} = 0.5 is valid

The temperature ϑ_L does not fall below the setpoint ϑ_{soll}, the increase of ϑ_L above ϑ_{soll} is due to internal heat gains. The increase of the relative humidity φ_L above φ_{soll} could be prevented only by dry air supply. The decrease of φ_L below 0.5 is caused by the loss of moisture gains. The control loop for the room air temperature shows a high performance whereas the control loop for the humidity tends to overshoot. This is due to the non minimum phase behaviour of the process which cannot be prevented in any way. According to paragraph 3.1 a greater tolerance is acceptable for this process variable. Therefore, the obtained performance is sufficient for this application.

3.3.2 Air infiltration rate control

For hygiene, health and building envelope conservation reasons a certain air infiltration rate has to be ensured. The exact value depends on room usage, number of persons present, residence size, and other variables. But even when the room is empty, the average air infiltration rate should not fall below 0.5 h^{-1} to avoid growth of micro-organisms. When the humidity satisfies the requirements, the corresponding subcontroller is disconnected from the system, and the input fan speed n is used to stabilise the user specified air infiltration rate on an average basis. The average air infiltration rate is imposed by alternating sudden short ventilation regardless whether the room is empty or not. If the room is occupied and ventilation is necessary the radiator valve must be closed in order to save energy. This is achieved by decreasing the room air temperature setpoint. This applies for operation mode winter only.

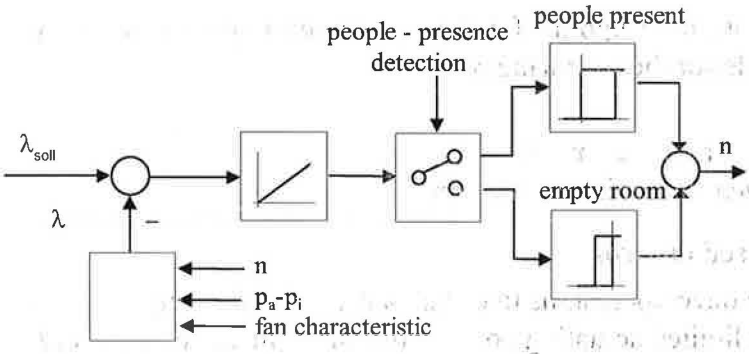


Fig. 3.7 :

Air infiltration rate control

The air infiltration rate λ is determined by the variables fan speed, pressure difference p_a-p_i and the characteristic of the fan [Häd96]. Fig. 3.7 shows the realisation of alternating sudden short ventilating.

The error signal λ_{soll} - λ is integrated for the last 5 hours and determines the percentage of air volume

$$V = \int_{t} (\lambda_{soll} - \lambda) dt \tag{3.12}$$

which has to be replaced. The result V is feed to a two-position controller with differential gap. For an absolute actuating error signal greater than the differential gap the controller either starts ventilation immediately when the room is empty or applies an extended differential gap when the room is occupied. In this latter case a longer time interval without ventilation occurs before the controller reacts to the error signal. The integration time range and the magnitude of the differential gap can be changed easily. The differential gap has been chosen for an empty room as

n=100 % for $\lambda_{soll} - \lambda \geq 1\ h^{-1}$, n=0 % for $\lambda_{soll} - \lambda \leq 0\ h^{-1}$

or when people are in the room as

n=100 % for $\lambda_{soll} - \lambda \geq 0\ h^{-1}$, n=0 % for $\lambda_{soll} - \lambda \leq -0.5\ h^{-1}$.

3.4 Hierarchical Control Strategy

3.4.1 Setpoint placement for comfort indoor air condition

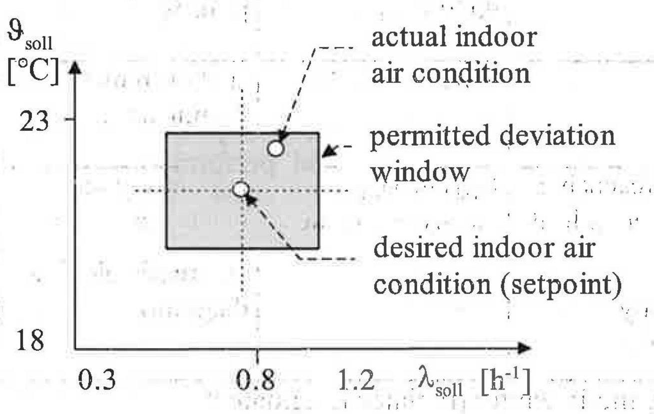


Fig. 3.8:

Temperature - air infiltration rate - diagram with setpoints for comfort indoor air condition

Situation depending setpoints for temperature, humidity and basic controllers have to be placed [Häd96]. The top level of the hierarchical control structure does not establish a setpoint for the relative humidity φ since a large deviation is acceptable for this value and it is not always possible to reach a given setpoint φ_{soll} at all. The permitted temperature deviation depends on the temperature itself (e.g. high temperature setpoint means small permitted deviation).

The setpoint for comfort indoor air condition is chosen by the user within a temperature - air infiltration rate - diagram (Fig. 3.8). The permitted deviations $\vartheta_{soll\ min} \leq \vartheta_{soll} \leq \vartheta_{soll\ max}$ and $\lambda_{soll\ min} \leq \lambda_{soll} \leq \lambda_{soll\ max}$ are automatically determined and indicated as a window. Linear functions are used to derive basic controller setpoints from the discussed setpoints. Setpoints are changed in normal operation mode for the following reasons:

- night setback,
- temperature setback when the room is empty,
- temperature setback when ventilation is switched on.

3.4.2 Case sensitive knowledge based control

The automatic control which is to be developed, aims to ensure a comfortable indoor air condition during the whole year. Due to limited actuating possibilities this aim can be achieved only by distinction between various cases of indoor and outdoor air conditions according to the h-x diagram and user behaviour considerations. Subject to the operating mode, basic controllers must be enabled/disabled and reversion actions have to be performed. This approach corresponds to a knowledge processing method using human experience for controlling indoor air condition. This paper distinguishes two fundamental operating modes by temperature conditions in summer and winter time, since temperature conditions influence most strongly the human well-being and heating energy consumption [Häd96]. For the first operating mode winter (Table 3.1), the heating system works as an actuator within the control system. In the second operating mode summer, however, the heating system can not be used since it is switched off. The fan is an actuator for both operating modes.

	operating mode winter			
	$\vartheta_a \leq \vartheta_L$		$\vartheta_a > \vartheta_L$	
	$\vartheta_L \leq \vartheta_{soll\,max}$	$\vartheta_L > \vartheta_{soll\,max}$	$\vartheta_L \leq \vartheta_{soll\,max}$	$\vartheta_L > \vartheta_{soll\,max}$
controlled variables	ϑ_L and λ or ϑ_L and φ_L when substantial internal moisture gains	ϑ-control off, φ- control off, ventilation: decrease ϑ_L	ϑ- control off, φ- control off, ventilation: increase ϑ_L up to $\vartheta_{soll\,min}$, if $\dot{\vartheta}_a \leq 0$ up to $\vartheta_{soll\,max}$	ϑ- control off λ, φ_L possibly, avoiding further increase of ϑ_L has priority
outdoor blinds	shut at night	none	none	shut or minimum illumination
objectives	ϑ_{soll} reachable, φ_{soll} mostly to high	ϑ_{soll} not reachable when internal heat gains, φ_{soll} mostly reachable	ϑ_{soll} and φ_{soll} mostly reachable	ϑ_{soll} mostly not reachable, φ_{soll} reachable (h-x diagram)

Table 3.1: Control strategy for the operating mode winter (radiator is actuator)

Some destinctions in Table 3.1 and 3.2 may seem similar or even identical. Every situation represents however different weather situations and may require different actions. A detailed discussion isn't possible within this paper.

	operating mode summer			
	$\vartheta_a \leq \vartheta_L$		$\vartheta_a > \vartheta_L$	
	$\vartheta_L \leq \vartheta_{soll\ min}$	$\vartheta_L > \vartheta_{soll\ min}$	$\vartheta_L \leq \vartheta_{soll\ min}$	$\vartheta_L > \vartheta_{soll\ min}$
controlled variables	λ or φ_L when internal humidity gains, avoiding a decrease of ϑ_L has priority	ventilation : decreasing ϑ_L	ϑ_L increase up to $\vartheta_{soll\ min}$, if $\dot{\vartheta}_a \leq 0$ up to $\vartheta_{soll\ max}$	λ , φ_L possibly, avoiding further increase of ϑ_L has priority
outdoor blinds	shut at night	shut at night	none	shut or minimum illumination
objectives	ϑ_{soll} not reachable, φ mostly within comfort zone	ϑ_{soll} mostly not reachable, φ_{soll} mostly reachable	ϑ_{soll} and φ_{soll} mostly exceeded, ϑ_L has priority	ϑ_{soll} mostly not reachable, φ_{soll} reachable (h-x diagram)

Table 3.2: Control strategy for the operating mode summer (heating system is switched off)

4 Simulation Results

The described control strategy has been tested on a *Matlab/Simulink* simulation model and gives the results shown in Fig. 4.1 (one winter and one summer day). The figure shows the person-present signal (present people cause moisture and heat gains) and the actuator values of the radiator valve (winter-operating mode) or of the outdoor blinds (summer-operating mode) and the resulting transition effects of the room air temperature.

In winter time the indoor air condition is easier to control. The setpoints of the controlled variables ϑ_L and λ are matched all the time. The average air infiltration rate of 0.5 h^{-1} is ensured by alternating sudden short ventilation. The small deviation of the temperature is due to assumed solid walls and the good heat insulation of the building. The sequence of decreased temperature setpoints because of special situations as described earlier (empty room, night setback, fan on) is clearly to be seen.

However, controlling the indoor air condition in summer time is more complicated. The only available actuators are fan and outdoor blinds. For the time interval t $\leq$ 211.4d and t $\geq$ 211.85d the ϑ_a-tendency dictates the ϑ_L-tendency. Internal heat gains are caused by present persons at t=211.3 d and t=211.75 d. The time period 211.4 d < t < 211.85 d represents the case minimum ventilation (λ = 0.5 h^{-1}) and demand guided shading. An indoor temperature of less than 24.5 °C can be achieved for a given outdoor temperature of up to 30 °C by the employed control strategy.

Simulation results demonstrate the capability of the applied strategy to control the indoor air condition throughout the year. During transitional season, however, different cases (see Table 3.1 and 3.2) may occur in a rather fast alternating manner. So it is possible to use solar heat gains effectively and reduce heat energy consumption.

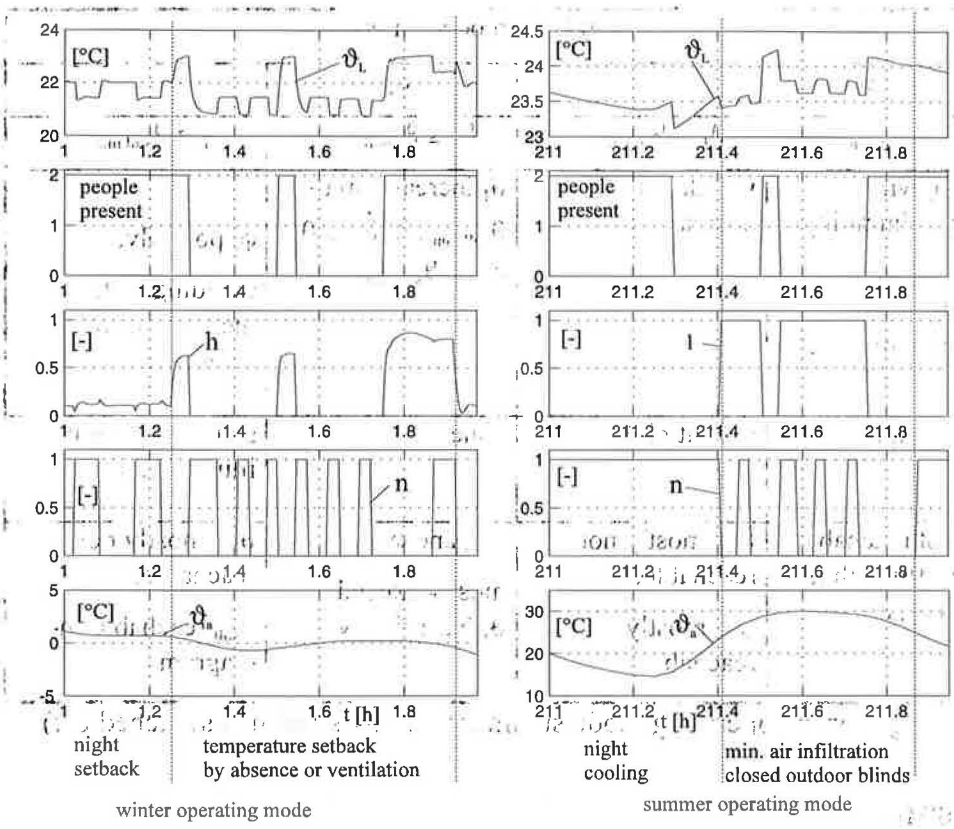


Fig. 4.1: Simulation results of the control strategy

5 Summary

The paper introduces a strategy for indoor air condition control for domestic buildings equipped with a heating system, non-central ventilation and outdoor blinds. The *Matlab/Simulink* simulation model, developed for the indoor air condition in a single room, which is based on theoretical process analysis and was up till now unknown in the literature with such a complexity, is an essential prerequisite for testing the developed control system. The necessary model accuracy was chosen according to the requirements of control design. The achieved simulation results show a satisfactory match with measured data. A model of an entire domestic building with user configured characteristics can be created by grouping room models.

To achieve demand guided control of the indoor air condition, a hierarchical control structure employing case-sensitive knowledge processing is proposed since the process characteristics are rather complex (e.g. reduced controllability). The developed control system considers indoor and outdoor air conditions and predicts room usage (empty/people present). The set-points and permitted deviations for the basic controllers are chosen depending on operating modes (winter/summer) and time or events (e.g. night cooling, empty room etc.).

The basic controller for temperature and humidity is a model adaptive controller because set-point dependent variations and ventilation dependent parameters do not suggest the use of classical controllers. The applied concept of adaptive control appeals to the user, since it offers simple commissioning, automatic adjustment to user behaviour and a simple operation interface.

According to the simulation results, the chosen strategy to control the indoor air condition works very well throughout the year. The developed control system is intended to be used for modern installation bus systems for domestic buildings. By introducing such powerful control

systems an increased domestic comfort and less energy consumption for heating can be achieved. The project is supported by the German Bundesministerium für Bildung, Wissenschaft, Forschung und Technologie, grant number 0329 706A.

6 Nomenclature

A	-	area
a, T, m	-	material specific parameter
c_p	-	specific heat capacity
g	-	specific time-dependent mass of the wall due to its water content
K	-	gain
m	-	mass
$\dot{m}$	-	mass flow
p	-	pressure
$\dot{Q}$	-	heat gain, heat flow
s	-	complex variable
t	-	time
T1	-	time constant
x	-	humidity ratio
α	-	heat convection number
φ	-	relative humidity
κ	-	coupling factor
λ	-	air infiltration rate
ϑ	-	temperature
h	-	radiator valve lift
n	-	fan speed
l	-	outdoor blind position
Δi	-	Laplace transform of $i(t)$
$\Delta\dot{m}(s)$	-	Laplace transform of $\dot{m}(t)$
$\Delta x(s)$	-	Laplace transform of $x(t)$
$\Delta\Theta(s)$	-	Laplace transform of $\vartheta(t)$

Subscripts

a	-	external, outdoor
Ab	-	exhaust
D	-	moisture, vapour
HK	-	radiator
HW	-	heating water
i	-	internal, inside
L	-	room air, ventilation
Ob	-	surface
S	-	saturation
W	-	wall
Zu	-	supply
N	-	adjacent room

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