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Control of thermally-activated building systems (TABS)

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Abstract

Integrating the building structure to act as an energy-storage, thermally-activated building system (TABS) has proved to be energy efficient and economically viable for cooling and heating of buildings. However control has remained an issue to be improved. In this paper, a method is outlined allowing both for dimensioning and for automated control of TABS, with automatic switching between cooling/heating modes for variable comfort criteria. The method integrally considers both HVAC and building automation design aspects, as well as the fact that during design and operation heat-gains are unknown, but that bounds of them normally can be specified. This integral method is termed the Unknown-But-Bounded or UBB method. Applying the method guarantees that comfort can be maintained, as long as the actual heat-gains stay within the predefined range between the lower and upper bounds. The UBB method can also handle non-predictable day-to-day variations as well as room-to-room variations of the heat gains. The paper outlines the underlying thermal models and assumptions, and gives the procedure and an example for the application of the method.

Keywords: Thermally-activated building systems, Concrete core conditioning systems; Building control; Heating; Cooling; Unknown-but-bounded approach

1. Introduction

Thermally-activated building systems (TABSs) have emerged as energy efficient and economical ways for cooling and heating of buildings. They integrate the building structure into the overall energy strategy of the building as energy storage. The dynamic thermal behaviour of building elements, such as structural floors and slabs, is exploited to provide either cooling by radiant and convective energy absorption or space heating by the release of stored energy. By contrast to radiant cooling by suspended ceiling panels, peaks in energy demand are flattened and the actual cooling is shifted to the colder night time [1,2].

The design and the dimensioning of a TABS are based on guidelines [3]. So far, control is implemented downstream in the design process. The specification of control algorithms is difficult because of thermal inertia

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Nomenclature

A,B,C,F,G	matrices of the state space model, various
C	capacitance, J/K
d	distance, thickness, m
LB,UB	lower/upper equivalent heat gain bound, W/m^2
$\dot{q}$	heat flux, W/m^2
R	thermal resistance, m^{2}K/W
$\widetilde{R}$	thermal resistance between core and room temperature, m^{2}K/W
$R_{l,f}$	thermal resistance façade, in relation to floor area, m^{2}K/W
R_t	thermal resistance piping system TABS, m^{2}K/W
t	time, s or h
$\underline{u}(k)$	disturbance vector, discrete time notation, various
$\underline{u}(t)$	disturbance vector, continuous time notation, various
$\underline{x}_k$	state vector, discrete time notation, °C
$\underline{x}(t)$	state vector, continuous time notation, °C

Greek symbols

Δ	difference
δ	outside diameter pipe, m
ϑ	temperature, °C

Subscripts

0,1,2, ...	side, index 0, 1, 2, ...
b	bound
c	core
e	equivalent
elb	equivalent lower-bound
eub	equivalent upper-bound
g	gain
i,k	indices
iW	internal wall
lb	lower bound
l,f	loss, façade
LmC,LmH	Limit cooling/heating
oa	outside air
p	pipe
r	room
s	slab
SpC,SpH	set-point cooling/heating
sw	supply water, inlet
t	total
ub	upper bound
w	water
x,y,z	coordinates, directions

of the system and because of the challenge to comply with comfort requirements in different rooms with different gains and these rooms being connected to the same supply of hot water. Various control approaches are implemented, but they often have disadvantages due to different approaches for cooling and heating, too frequent switching between the heating and cooling, the need for manual switching between the heating and cooling mode as well as the need for manual adjustment of parameters.

Therefore the objective was to develop control algorithms for year-round automated operation of TABS, which are simple to commission and which lead to an energy-efficient operation fulfilling the comfort requirements.

This paper presents a newly-developed, comprehensive strategy for the temperature control of rooms with TABS, considering the prediction uncertainty of heat gains during operation by specifying respective bounds. The strategy has been developed in the frame of a research project [4].

2. Control concepts for TABS

2.1. Conventional control-concepts for TABS

In the literature, only few have reported on the control of TABS: for example, Meierhans as one of the TABS pioneers [5], Olesen [6], Antonopoulos [7] and Weitzmann [8], the latter giving a short overview of so far proposed control-concepts. Tödtli et al. [4] evaluated existing control solutions and stated that these mostly have the following properties. (a) They are based on an outside temperature compensated water-flow temperature-control, where the set point of the flow temperature is shifted with varying outside temperature according to the heating curve (HC) without considering heat gains. The cooling curve (CC) typically is a constant-flow temperature set-point based on the maximum load situation. (b) No feed-back variable from the zones (return temperature, concrete core temperature or room temperature) is used for the control. Self-regulation of the concrete core conditioning system is assumed to be sufficient. (c) The heat and cold generation (heating, cooling or neutral) are enabled or activated dependent on the season and/or the outside temperature. (d) Free cooling, for instance by a wet cooling tower, is accounted for in a heuristic way.

2.2. New concept based on the integral approach

HVAC design is considered with questions like: (a) Is the TABS system (with ventilation according to indoor-air quality requirements) sufficient to cover heating and cooling loads? (b) Are auxiliary systems needed (for heating, for cooling, for both)? (c) Is TABS the most promising system? Building automation (BA) design is considered with questions like: (a) What is the range of the self-regulating effect? (b) What is the application range of outside temperature dependent on/off control? (c) In which cases is a room-temperature control necessary with sensors in reference rooms? (d) In which cases are individual room-temperature controllers needed?

For the control of TABS, the HVAC and the BA aspects cannot be treated separately. Therefore, an integral approach is proposed, considering, as the first element, both the HVAC and the BA design aspects. The second important element of the new approach is how external (solar) and internal heat-gains are considered. The gains are unknown, but bounds normally can be specified. These two elements lead to an integral method, termed the Unknown-But-Bounded or UBB method.

3. TABS and room models used in the UBB method

Within the UBB method, a dynamic thermal room model is used to determine the influence of internal and solar gains on the heating and cooling load. Therefore, this room model is described first, followed by the detailed description of the UBB approach.

3.1. Modelling of TABS (piping system)

For design purposes and performance simulations of TABS, a model depicting the heat transfer in the slab and to adjacent rooms was developed [3]. This TABS model – under certain restrictions – allows reducing the 3-dimension heat-transfer in the slab to a 1-dimension approach by establishing a correlation between flow temperature, core temperature (mean slab temperature in the plane of the piping system) and room temperature (i.e. operative temperature). The main parameter to model the piping system is the equivalent resistance R_t, called TABS-resistance, in which the geometrical characteristic, material parameters and the influence of the fluid mass flow are summarized (Fig. 1). The dynamic thermal behaviour of the upper and lower parts of the slab can be modelled in an arbitrarily complex manner ranging from a simple 1-node-model to detailed multi-layer models (cf. Section 3.2).

Integrated in building and system simulation codes, such as TRNSYS [9], the TABS model can be used to calculate the dynamic behaviour of whole buildings equipped with TABS.

3.2. Dynamic thermal room-model

Thermal inertia plays a pivotal role in the behaviour of TABS. Therefore in the UBB approach, a dynamic thermal room model must be used. Fig. 2 shows the structure of such a model. The slab's core temperature node ϑ_c is linked to the supply-water temperature ϑ_{sw} by the TABS-resistance R_t and to the (operative) room temperature ϑ_r by a resistance/capacitor element per slab layer (for modelling purposes the slab is subdivided into several layers). The heat gains (internal, solar) affect the room node, which is linked to the outside air temperature ϑ_{oa} by a façade-resistance $R_{l,f}$. Investigations have shown that it is essential to incorporate, besides the slabs, additional thermal mass and surface area in the room model to account for heat exchange

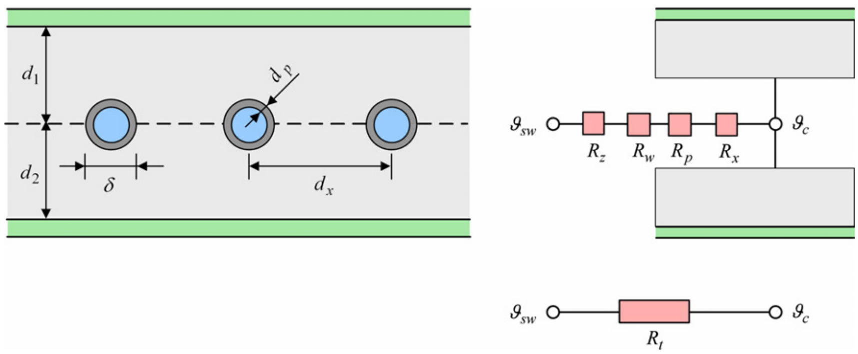

Fig. 1. Piping system embedded in a slab and corresponding representation as TABS model.

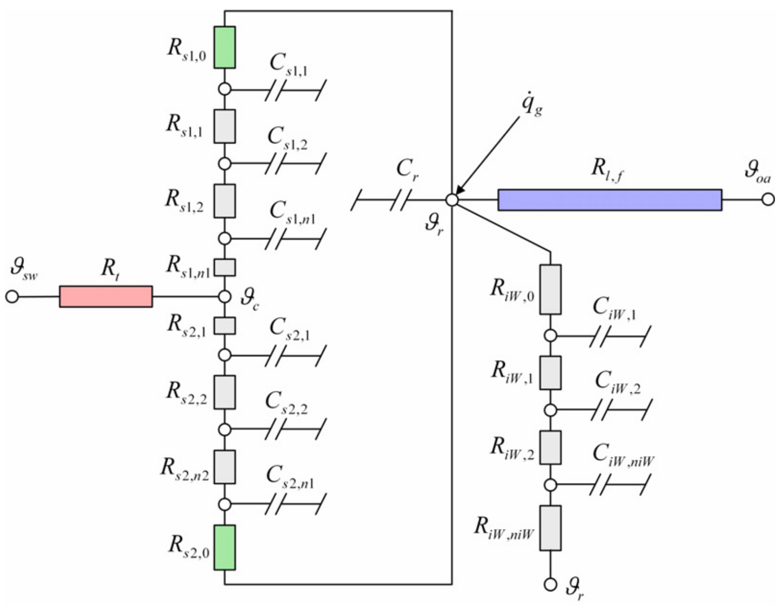

Fig. 2. Resistance-capacitor model of the room.

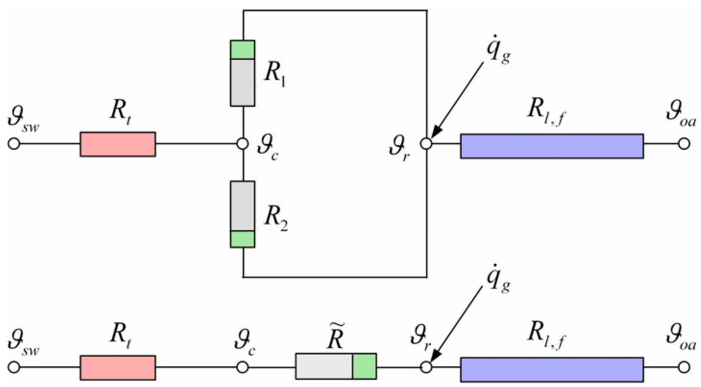

Fig. 3. Room model reduced to a steady-state resistance network.

from/to inside walls and furniture. This is realized by an internal wall link, which consists again of a resistance/capacitor element per wall layer.

3.3. Steady-state situation

As will be explained in Section 4.1, the heating and cooling curves can be determined on a steady-state basis. In this case, the influence of thermal capacities vanishes and the thermal room model reduces to a pure resistance network (Fig. 3). Thereby, the resistances R_1 and R_2 of the upper and lower parts of the slab (including floor and ceiling coverings as well as heat-transfer coefficients to the adjacent rooms) can be combined in the resistance $\widetilde{R}$.

4. The Unknown-But-Bounded (UBB) method

4.1. Basic principle of the UBB method

The first part of the UBB method comprises the choice of a base control concept. Several possibilities exist for this choice. In this paper, only the outside temperature compensated supply-water temperature control is considered as base control concept. There the supply-water temperature is shifted according to the heating/cooling curves to compensate for heat losses and gains through the façade, thus meeting the cooling/heating demand.

The second part is the consideration of internal and solar heat gains in the heating/cooling curves to account for their influence on room temperature and thus on thermal comfort. Heat gains vary with time, dependent on the presence of persons, the usage of equipment and the actual solar input. Therefore they are difficult to measure or predict and, as a consequence their actual value, are not available for control purposes. However, it is assumed that the designer of the control scheme can specify the lower-bound profile of the heat gains $\dot{q}_{\mathrm{g,lb}}$ and the upper-bound profile of the heat gains $\dot{q}_{\mathrm{g,ub}}$ which most likely will never be undercut and exceeded respectively (Fig. 4a), considering the uncertainty of prediction and the possible (time- and room-to-room-dependent) variability of the heat gains.

As a consequence, to cope with the possible span of the heat gains during the design phase, the so called Unknown-But-Bounded method has been developed. The underlying principles can now be summarized.

Due to the high thermal-inertia of TABS, the adaptation of the supply-water temperature within the day to react on load changes (gains) on the room side is ineffective. This in turn means that, if large gains are present only during a few hours a day, they must not be handled immediately but can be buffered in the slabs. Therefore the heating and cooling curves (under consideration of the heat gains) can be determined on a quasi steady-state basis. For this steady-state calculation, the profiles of the lower and upper heat-gain bounds have to be converted into the equivalent, single constant values $\dot{q}_{\mathrm{g,elb}}$ and $\dot{q}_{\mathrm{g,eub}}$, in the following and are termed Lower Bound (LB) and Upper Bound (UB), respectively (cf. Fig. 4a).

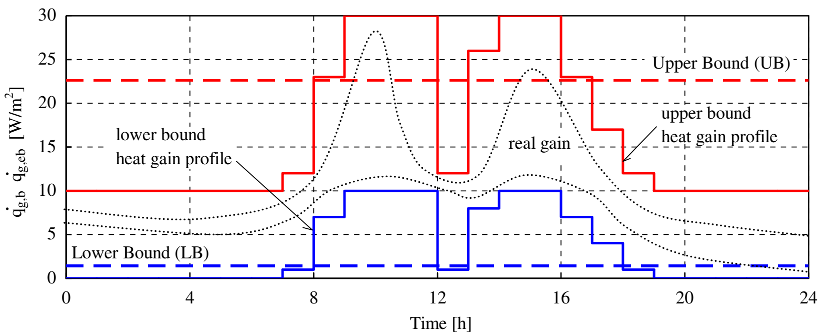

Fig. 4a. Typical profiles of upper and lower heat-gain bounds with respective equivalent upper and lower heat-gain bounds (which are constant values, termed Upper Bound and Lower Bound). The specific shape of the profiles results from people arriving at the office in the morning, turning on lights and equipment, the sun rising, people leaving the office at noon, people leaving at the end of the work day and the sun setting in the evening.

This conversion is done in such a way that the minimum/maximum room temperatures reached under exposure to the LB/UB (steady-state) correspond to the temperatures reached under exposure to the real profiles of the lower/upper heat-gain bounds (i.e. dynamic behaviour of the room).

For an accurate determination of the maximum/minimum room temperatures, a calculation procedure is needed which makes allowance for the dynamic behaviour of a room equipped with TABS. The dynamic thermal room model, described in Section 3.2, fulfils these needs and therefore is used to calculate the room temperature profiles under permanent (repeated) exposure to the profiles of lower/upper heat-gain bounds.

The maximum/minimum room-temperatures identified by the calculated room-temperature profiles are then used to determine the lower/upper equivalent heat-gain bounds (LB/UB), using the steady-state thermal room model (Section 3.3, Fig. 3).

Thus, applying the UBB method described above guarantees that room temperatures can be kept within the thermal-comfort range, as long as the actual heat gains stay within the predefined range between the lower and upper bounds.

4.2. Calculation procedure for the UBB method

The design and the dimensioning of the control of a TABS according to the UBB method follows the principal steps listed hereafter. These steps are described in detail in the following sections: (a) specify building components and TABS parameters; (b) specify heat-gain types and determine the resulting profiles of upper and lower heat-gain bounds; (c) calculate room temperature profiles under lower/upper heat-gain bound profiles; (d) calculate the lower/upper equivalent heat-gain bounds; (e) specify room-temperature set points heating/cooling (comfort range); (f) calculate heating/cooling curves and additional characteristic parameters; and (g) make optimization loops if requirements are not met.

4.3. Specification of building and TABS parameters

In the first step, the characteristic of the building under consideration is analyzed. Based on the properties of slabs, piping system and façade, the main parameters (R_t, $\widetilde{R}$, $R_{1,f}$) for the thermal model according to Fig. 3 are calculated. The corresponding definitions can be found in [3] and in Appendix 1.

4.4. Specification of heat-gain types and determination of resulting profiles for upper and lower heat-gain bounds

As a second step, the main components of the heat gains and the respective schedules are identified. In office buildings, gains from persons, equipment, lighting and transmitted solar-radiation are considered. From the individual gains, the resulting profiles of lower/upper heat-gain bounds are determined. Fig. 4a shows typical profiles of upper and lower heat-gain bounds (solid lines). If periods with clearly distinguishable gain-patterns exist (e.g. workdays – weekends, winter period – summer period), the respective profiles of lower and upper heat-gain bounds can be specified separately for each type of period. This leads to improved overall control behaviour of the system.

4.5. Model-based calculation of room-temperature profiles under LB/UB boundary-conditions

The calculation of the room-temperature profiles under lower/upper heat-gain bound profiles is performed using the dynamic room model described in Section 3.2. Here only a shortened version of the procedure is given. A more detailed description can be found in Appendix 2.

The differential equation system (DEQS, state space notation) for room, slab and wall temperatures with the input variables supply-water temperature, outside air temperature and heat gain is given as

$$\frac{\mathrm{d}\underline{x}(t)}{\mathrm{d}t} = F \cdot \underline{x}(t) + G \cdot \underline{u}(t); \quad \underline{u}(t) = \begin{bmatrix} \vartheta_{\mathrm{sw}}(t) \\ \vartheta_{\mathrm{oa}}(t) \\ \dot{q}_{\mathrm{g}}(t) \end{bmatrix}$$
$$\begin{bmatrix} \vartheta_{\mathrm{r}}(t) \\ \vartheta_{\mathrm{c}}(t) \end{bmatrix} = C \cdot \underline{x}(t) \tag{1}$$

If boundary conditions are given in discrete form (e.g. hourly values from simulation), the DEQS is transformed to discrete state space notation:

$$\underline{x}_{k+1} = A \cdot \underline{x}_k + B \cdot \underline{u}(k); \quad \underline{u}(k) = \begin{bmatrix} \vartheta_{\mathrm{sw}}(k) \\ \vartheta_{\mathrm{oa}}(k) \\ \dot{q}_{\mathrm{g}}(k) \end{bmatrix}$$
$$\begin{bmatrix} \vartheta_{\mathrm{r}}(k) \\ \vartheta_{\mathrm{c}}(k) \end{bmatrix} = C \cdot \underline{x}_k \tag{2}$$

Under the assumption that the profile of the lower heat-gain bound is perpetually repeated, the quasi-stationary profiles of all state-vectors (slab and wall temperatures) can be calculated. From these also the room $\vartheta_{\mathrm{r,lb}}(k)$ and core $\vartheta_{\mathrm{c,lb}}(k)$ temperature profiles are obtained. Because the system is linear, it is possible to calculate the room temperature as a superposition of the room temperature for the situation without gains ($\dot{q}_{\mathrm{g}} = 0$) and a room temperature increase caused by the gains for the situation where $\vartheta_{\mathrm{sw}} = \vartheta_{\mathrm{oa}} = 0$ °C. In the following, only the temperature increase caused by the gains is considered.

The same procedure is repeated for the upper heat-gain bound profile as input variable. Fig. 4b shows the room-temperature increases caused by the lower and upper bound heat-gain profiles defined in Fig. 4a.

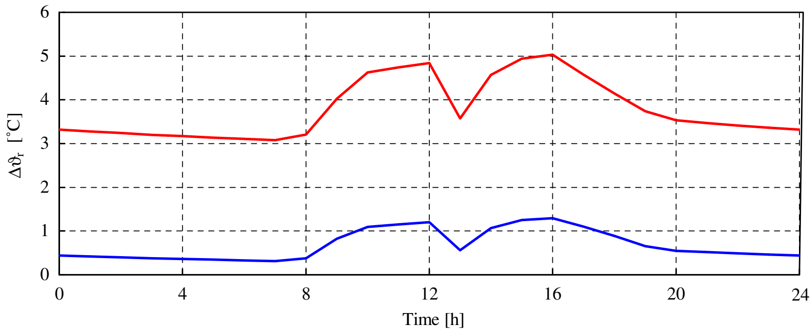

Fig. 4b. Resulting room-temperature increase under repeated exposure to the profiles of lower and upper heat-gain bounds.

4.6. Calculation of lower and upper equivalent heat-gain bounds (LB/UB)

Applying the minimum and maximum of the above calculated room-temperature increases in a steady-state energy balance for the room temperature of the resistance network given in Fig. 3, the Lower and Upper Bounds can be identified as

$$\dot{q}_{g,elb} = \frac{R_t + \widetilde{R} + R_{l,f}}{R_{l,f}(R_t + \widetilde{R})} \cdot \min_i\{\vartheta_{r,lb}(t_i)\} \tag{3}$$

$$\dot{q}_{g,eub} = \frac{R_t + \widetilde{R} + R_{l,f}}{R_{l,f}(R_t + \widetilde{R})} \cdot \max_i\{\vartheta_{r,ub}(t_i)\} \tag{4}$$

In Fig. 4a, the Lower and Upper Bounds, corresponding to the lower and upper heat-gain bound profiles specified, are printed in a dashed style.

4.7. Specification of applicable room-temperature set-point range (i.e. comfort range)

For the room temperature, a set-point range $[\vartheta_{r,SpH},\vartheta_{r,SpC}]$ has to be specified. The set-point range depends on the type of application (office, commercial, domestic building) and on the thermal-comfort requirements. Appropriate definitions can be found in respective standards, as e.g. [10]. The larger the set-point range, the more energy efficient the building can be conditioned and the more appropriate TABS are to cover the total thermal-loads of the building (heating and cooling). If the specified comfort range is too narrow, the room temperatures cannot float as much as needed for TABS for absorption and release of heat. Thereby the profitable use of the passive thermal-storage capacity of the TABS is reduced.

4.8. Determination of heating/cooling curves

In order to maintain the room temperature set-point $\vartheta_{r,SpH}$ at the Lower Bound, the TABS heating power which has to be transferred to the slab via the TABS water circuit is

$$\dot{q}_{w,SpH} = \frac{1}{R_{l,f}} \cdot (\vartheta_{r,SpH} - \vartheta_{oa}) - \dot{q}_{g,elb} \tag{5}$$

Similarly, in order to maintain the room-temperature set-point $\vartheta_{r,SpC}$ at the Upper Bound, the TABS cooling power is:

$$\dot{q}_{w,SpC} = \frac{1}{R_{l,f}} \cdot (\vartheta_{r,SpC} - \vartheta_{oa}) - \dot{q}_{g,eub} \tag{6}$$

In order to maintain $\vartheta_{r,SpH}$ at LB under steady-state conditions, or not to fall below $\vartheta_{r,SpH}$ at lower bound heat-gains $\dot{q}_{g,lb}$ under dynamic conditions: the supply-water temperature set-point for the room is

$$\vartheta_{sw,SpH} = \vartheta_{r,SpH} + \frac{R_t + \widetilde{R}}{R_{l,f}} \cdot (\vartheta_{r,SpH} - \vartheta_{oa}) - (R_t + \widetilde{R}) \cdot \dot{q}_{g,elb} \tag{7}$$

Similarly, in order to maintain $\vartheta_{r,SpC}$ at UB under steady-state conditions, or not to exceed $\vartheta_{r,SpC}$ at upper bound heat-gains $\dot{q}_{g,ub}$ under dynamic conditions, the supply-water temperature set-point for the room is

$$\vartheta_{sw,SpC} = \vartheta_{r,SpC} + \frac{R_t + \widetilde{R}}{R_{l,f}} \cdot (\vartheta_{r,SpC} - \vartheta_{oa}) - (R_t + \widetilde{R}) \cdot \dot{q}_{g,eub} \tag{8}$$

Eqs. (5) and (6) are valid for steady-state conditions only (i.e. constant heat gains between the LB and UB) while Eqs. (7) and (8) also hold for the dynamic case with dynamic heat-gains staying within the specified range between the lower and upper bound heat-gain profile. To simplify matters, the following analysis is done for the steady-state case.

Depending on the magnitude of the heat gain range, three cases can be distinguished, for which different requirements can be identified (see Fig. 5). Case (a): if the heat gain range is small ($\dot{q}_{g,eub} - \dot{q}_{g,elb}$ small), there

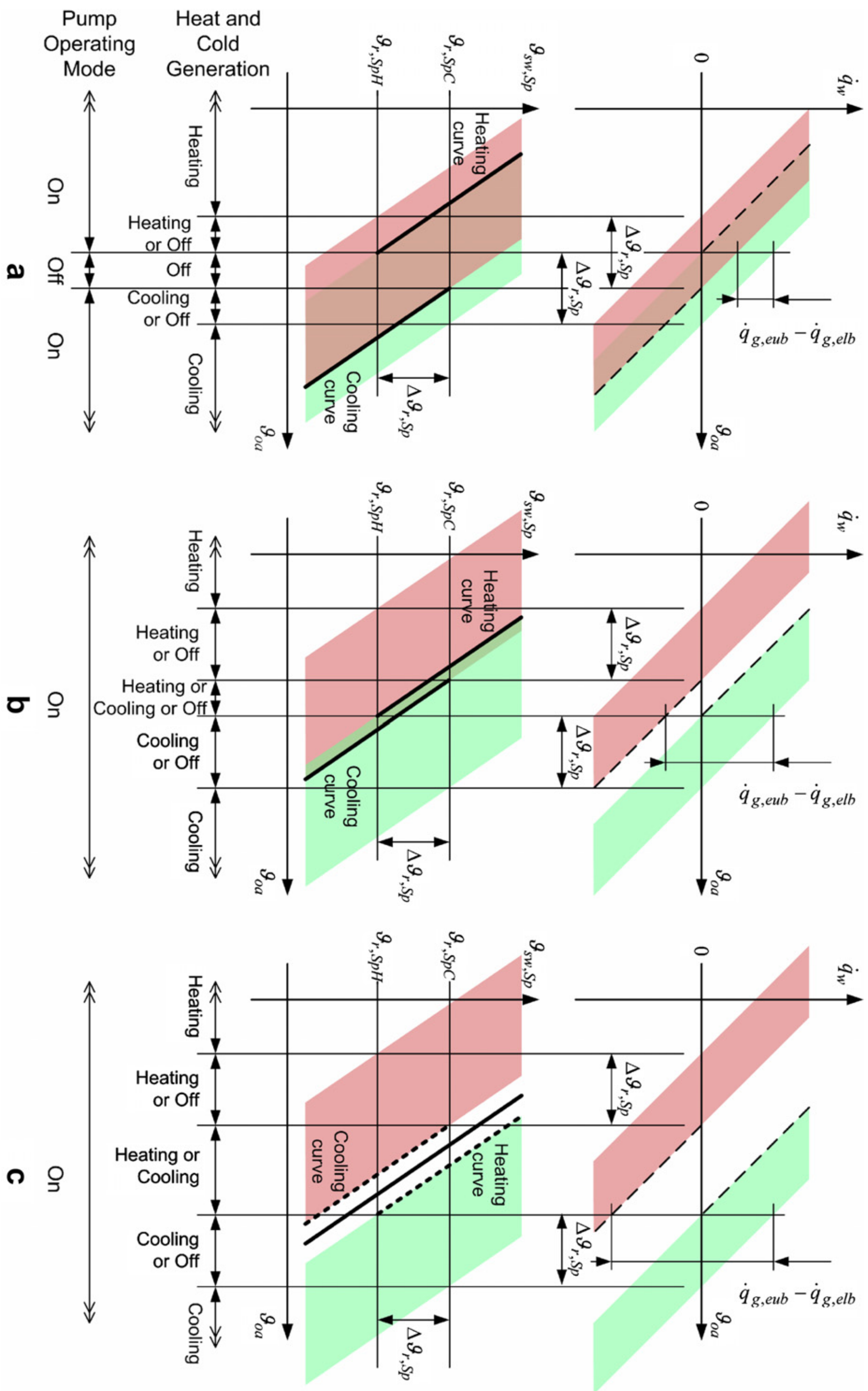

Fig. 5. Heating and cooling demands as functions of outside-air temperature and respective heating and cooling curves for the three cases (a) small, (b) medium and (c) large heat-gain ranges.

exists an outdoor temperature interval in which certainly neither TABS cooling nor TABS heating power is required. Case (b): for medium heat-gain range ($\dot{q}_{g,eub} - \dot{q}_{g,elb}$ medium), there exists an outdoor-temperature interval in which – without knowing the actual heat gains – it is not possible to determine whether there is heating or cooling demand or no power demand at all. Nevertheless, as long as the supply-water temperature can be kept between the heating and the cooling curves, thermal comfort can be maintained. Knowing only the actual supply-water temperature and the valve position, the supply-water temperature controller is able to automatically perform the corrective action (lowering ϑ_{sw} if it lies above the cooling curve, raising ϑ_{sw} if it lies below the heating curve or water circulation only if ϑ_{sw} lies between heating and cooling curve). Case (c): for large heat-gain ranges ($\dot{q}_{g,eub} - \dot{q}_{g,elb}$ large) it is not possible anymore to maintain comfort for all heat-gain situations as the heating curve lies above the cooling curve! In certain cases, the integration of a second control-loop (with room-temperature sensors in addition to the supply-water temperature loop) may solve the problem. But in most cases, this situation can only be handled by an extended UBB method, which accounts for auxiliary heating and cooling systems and/or additional sensors (not outlined here).

For all three cases (a) – (c), there are heating/cooling limits which demarcate outside-air temperature ranges, where only heating or cooling is required.

Office buildings typically fall into category (b) with a – in terms of outside temperature – large overlap of heating and cooling regions. Thereby the heating limit lies considerably higher than the cooling limit. Not being able to handle such an overlap in the outside-air temperature range, between heating and cooling limits, is one reason for the weakness of many conventional control-concepts (cf. Section 2).

4.9. Determination of heating/cooling limits

The heating and cooling limits represent outside-temperatures where the TABS heating or cooling demand reaches zero for any value of $\dot{q}_g$ within the bounds. For steady-state conditions at LB and UB, these limits can be calculated using (5) and (6) by setting the TABS heating/cooling power equal to zero. For the dynamic case with variable heat-gain bounds, $\dot{q}_{g,lb}$ and $\dot{q}_{g,ub}$, the heating and cooling limits can be determined by means of a model-based calculation of the room-temperature profiles $\vartheta_{r0}(t_i)$ similar to Section 4.5. Thereby a model slightly different to the one in Fig. 2 is used: the supply-water node is eliminated which is equivalent to a TABS water-circuit that is turned off or in idle operation. The heating and cooling limits then are determined as

$$\vartheta_{oa,LmH} = \vartheta_{r,SpH} - \min_i\{\vartheta_{r0,lb}(t_i)\} \tag{9}$$

$$\vartheta_{oa,LmC} = \vartheta_{r,SpC} - \max_i\{\vartheta_{r0,ub}(t_i)\} \tag{10}$$

If the difference between heating and cooling limits is positive, then case (b) or (c) according to Fig. 5 applies: if the difference is negative, then case (a) applies.

4.10. Maximum allowable equivalent heat-gain range

For a specified range of the room-temperature set-point, a maximum allowable equivalent heat-gain range $\Delta\dot{q}_{g,eMax}$ can be determined, which still can be controlled. This is the case if the set point of the supply-water temperature for heating ($\vartheta_{sw,SpH}$) equals that for cooling ($\vartheta_{sw,SpC}$). Thus

$$\Delta\dot{q}_{g,eMax} = \left\{\frac{1}{R_t + \widetilde{R}} + \frac{1}{R_{l,f}}\right\}\Delta\vartheta_{r,Sp} \tag{11}$$

Eq. (11) shows that the maximum allowable range of equivalent heat-gain (and thus steady-state heat-gain) is proportional to the range of the room-temperature set-points. The smaller the values of the resistances R_t, $\widetilde{R}$ and $R_{l,f}$ are, the larger becomes the allowable equivalent heat-gain range. For energy-efficient façades, $R_{l,f}$ has only a marginal influence, thus the thermal coupling between the room and the TABS water-supply is the decisive element.

5. Application of the UBB method: simulation example

The applicability and performance of the presented UBB method was assessed by detailed building simulations, using the building and systems simulation code TRNSYS [9]. The detailed building model and the comprehensive system-capabilities of TRNSYS guarantee for a realistic assessment of the control method.

5.1. Reference building

As a reference case, a typical office building was considered. The characteristics of the chosen space module are summarized in Table 1.

5.2. Heat-gain profiles

The definition of heat-gain profiles was based on the Swiss standard SIA (Swiss Association of Engineers and Architects) 2024 guideline [11]. An open-plan office was assumed. The solar gains were drawn from a simulation considering local meteorological data for Zurich/Switzerland. In the case where no detailed simulation is possible (angle-dependent solar transmission and secondary heat-transmission due to absorption), solar gains may be estimated using just the solar irradiation and a constant solar-heat gain coefficient of the glazing. The individual gains from persons, equipment, lighting and solar gains were summed to give the total heat gain profile. In Fig. 6, an evaluation of the heat gains for workdays, (Monday – Friday) during the summer period is shown. Because the sum of lighting and solar gains does not vary too much between sunny and overcast days, the difference between the lower and upper bound profiles of the heat gains is relatively small. The

Table 1
Characteristics of considered space and thermally-activated building system

Building	
Space length, width, height	6 m × 6 m × 3 m
Façade orientation	West
Façade area	18.0 m^2
Overall U-value façade	0.65 W/m^2K
Glazing fraction façade	42%
Internal-wall area (light-weight construction)	36 m^2
Air change infiltration	0.1 h^{-1}
Ventilation	according to indoor-air quality requirements (no cooling/heating by ventilation assumed)
TABS configuration	
TABS covering fraction (floor area)	80%
Thickness concrete slab	250 mm
Pipe spacing	200 mm
External/internal pipe diameter	20/15 mm
Specific mass flow rate[a]	15 kg/(h m^2)
Fictitious heat-transfer Coefficient TABS (U_t)[a]	12.5 W/(m^2 K)
Thermal resistance of flooring	carpet, 0.125 (m^2 K)/W

[a] Interms of floor area covered by TABS.

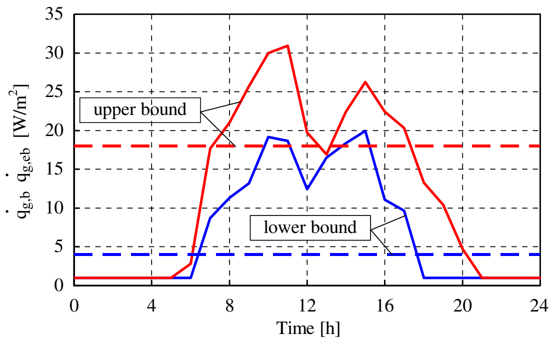

Fig. 6. Example of upper and lower bound heat-gain profiles for workdays during summer period with corresponding upper and lower bounds.

corresponding lower and upper equivalent heat-gain bounds (Lower and Upper Bound) for this case are 4.0 and 18.0 W/m^2, respectively. For the year-round control of the system, a distinction between the four major gain situations is made: workdays – weekends each for winter and summer periods.

5.3. Room-temperature set-points (comfort range)

The room-temperature set-points for heating and cooling were defined in accordance with the Swiss national standard SIA 382/1 [12]. The applicable room-temperature set-points depend on the maximum (expected) value of the outside-air temperature of each day (Fig. 7). The thermal-comfort ranges defined in this standard stem from the comfort definitions given in fundamental thermal-comfort standards (e.g. [10]), applying usual clothing habits and comfort requirements.

5.4. Characteristic data for the simulation example

The characteristic data for the simulation example were calculated from building and TABS parameters and the gain situation. These parameters are summarized in Table 2.

For a room-temperature set-point range of 21.0–24.5 °C, the maximum allowable equivalent heat-gain range $\Delta\dot{q}_{g,eMax}$ according to Eq. (11) is 16.8 W/m^2. The maximum effective equivalent heat-gain range reached is 18.0–4.0=14.0 W/m^2 (Table 2, Mo-Fr, summer). This means that the simulation example is classified as case (b) "medium heat-gain range". Because the smallest comfort range is critical concerning controllability, the conclusion is that comfort can be guaranteed year-round without constraint.

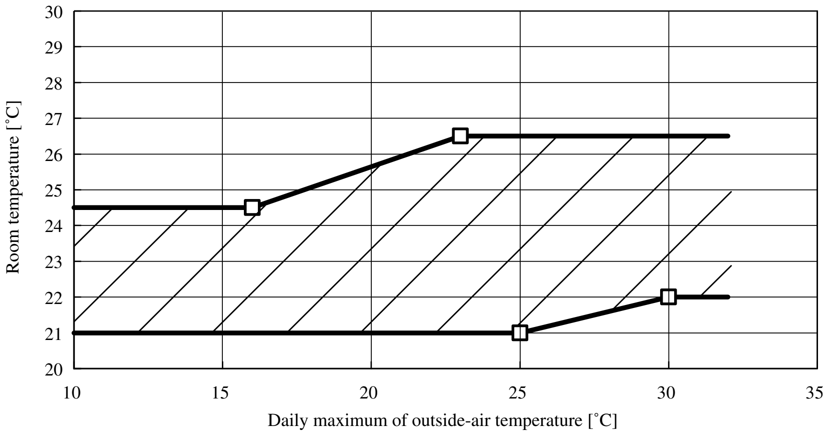

Fig. 7. Room temperature comfort-range (hatched area) according to Swiss national standard SIA 382/1 [12]. The comfort range is bordered by the outside-air temperature dependent room-temperature set-points for cooling (upper limit) and heating (lower limit), respectively.

Table 2
Characteristic parameters and data for the simulation example

Thermal resistances	m^{2}K/W
R_t	0.102
$\widetilde{R}$	0.127
$R_{l,f}$	2.356

Lower and upper bound heat-gains		Winter (W/m^2)	Summer (W/m^2)
Mo – Fr	$\dot{q}_{g,eub}$	17.3	18.0
	$\dot{q}_{g,elb}$	3.9	4.0
Sa – Su	$\dot{q}_{g,eub}$	7.2	7.8
	$\dot{q}_{g,elb}$	0.1	0.3

As an example, the heating and cooling curves for workdays in winter are plotted in Fig. 8. It can be seen that there exists a broad region between heating and cooling limits where – depending on the actual heat-gains – either heating or cooling may be required.

5.5. Simulation results

By applying the parameters determined in Section 5.4 in the controller of the TABS, a simulation for a whole year was performed. Thereby the following basic assumptions concerning control were considered: (1) Because fast changes of the supply temperature are mostly ineffective on the room side, and to prevent the system from frequent switching between heating and cooling mode, instead of the actual, the floating mean value of the outside air temperature over the last 24 h is used to determine the supply-water temperature set-point from the heating/cooling curves. (2) As the maximum daily outside air temperature is not available for control and therefore also the applicable room temperature set-points are unknown, an extrapolation method is used to determine the maximum daily outside-air temperature.

The results of the simulation are shown in Fig. 9. From the hourly values of the whole year, it can be seen that the room temperatures stay to a very great extent in the predefined comfort range. The few deviations can be explained by an inaccurate prediction of $\vartheta_{oa,max}$ and the incompatibility of the quasi steady-state approach for the cooling/heating curves with dynamic changes of the outside-air temperature (i.e. fast temperature drops/rises).

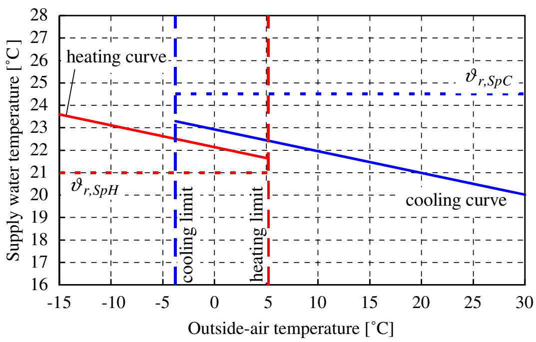

Fig. 8. Heating and cooling curves for the simulation example (smallest room-temperature set-point range, gain situation: workdays, winter).

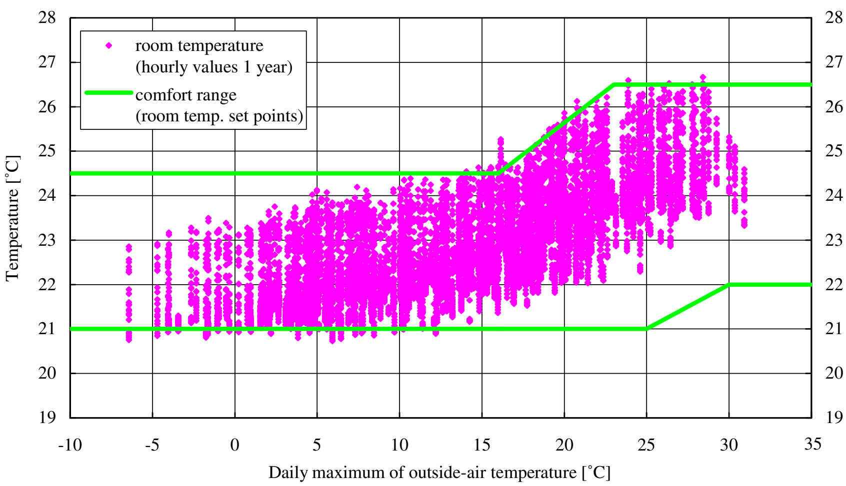

Fig. 9. Simulated room-temperatures for the reference building (hourly values).

6. Conclusions and outlook

With the Unknown-But-Bounded (UBB) method, a tool is available for both the dimensioning of TABS in the design process as well as for the control of TABS during operation. The method allows for automated control of TABS and incorporates automatic switching between cooling/heating modes for variable comfort criteria.

By applying the UBB method, (1) the design process including verification of the system's applicability for a given situation is facilitated, (2) the maximum cooling/heating capacity can be determined considering the uncertainty in predicting the heat-gains and their variability, (3) an indication is given whether an additional heating or cooling system or enhanced control is required, (4) parameter values for control units are delivered as a result, and (5) thermal comfort can be maintained, as long as the actual heat-gains stay within the predefined range between lower and upper bounds.

Using the UBB method, room to room heat gain variations also can be handled, in cases where different rooms are aggregated into one single TABS zone and which cannot be controlled individually.

Further developments will include the consideration of intermittent operation (on–off control step-function) with different on-and-off periods, the integration of additional control loops (with individual room-temperature sensors) and the extension of the method for additional control strategies as well as for auxiliary heating-and-cooling systems. Further details of the method can be found in the design and commissioning handbook, which will be published.

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Appendix 1

Calculation of thermal resistances for TABS and room models: definition of variables

A_{fl} total floor area, m^2
A_l total loss area façade, m^2
A_{ta} floor area covered by TABS piping system, m^2
c_a heat capacity air (room), J/kgK
c_w heat capacity water (fluid), J/kgK
d_1,d_2 thickness of upper/lower part of slab, m
d_p pipe's wall-thickness, m
d_x pipe spacing, m
h_{t1},h_{t2} total heat-transfer coefficient from slab to upper/lower room, W/m^2K
l pipe length of one pipe loop, m
$\dot{m}_{sp}$ specific mass flow-rate, relating to TABS area A_{ta}, kg/hm^2
n_a (air) infiltration coefficient, h^{-1}
R_{cov1},R_{cov2} resistance of floor/ceiling covering, m^2K/W
U_l mean U-value of façade (loss area), W/m^2K
V_a air volume (room), m^3

Greek symbols
δ outside diameter of pipe, m
λ_p heat-conductivity pipe-wall material, W/mK
λ_s heat-conductivity slab material, W/mK
ρ_a density of room air, kg/m^3

The basic approach for the modelling of TABS and the mathematical derivation of formulae for their calculation can be found in [1 and 3]. Based on fundamental relations for the transient heat-transfer in a threedimensional slab with integrated piping system, a one-dimensional correlation is derived. With faint restrictions due to the necessary approximations, this correlation allows the calculation for piping systems embedded in the core of a slab, based on the equations for transient heat-transfer in a plane wall. The 1-dimensional room and TABS model is represented by the thermal resistances according to Figs. 1 and 3. In the following, the formulae for these resistances are listed: these only apply if the two limiting conditions $\frac{d_i}{d_x} > 0.3$ and $\frac{\delta}{d_x} < 0.2$ (i = 1,2) are met.

$$R_w = \frac{\left(d_x \frac{A_{fl}}{A_{ta}}\right)^{0.13} \left(\frac{\delta - 2d_p}{\left(\dot{m}_{sp}\frac{A_{ta}}{A_{fl}}\right) l}\right)^{0.87}}{8.0\pi} \tag{A-1}$$

$$R_p = \frac{\left(d_x \frac{A_{fl}}{A_{ta}}\right) \cdot \ln\left(\frac{\delta}{\delta - 2d_p}\right)}{2\pi\lambda_p} \tag{A-2}$$

$$R_x = \frac{d_x \cdot \ln\left(\frac{d_x}{\pi\delta}\frac{A_{fl}}{A_{ta}}\right)}{2\pi\lambda_s} \tag{A-3}$$

$$R_z = \frac{1}{2\left(\dot{m}_{sp}\frac{A_{ta}}{A_{fl}}\right) c_w} \tag{A-4}$$

$$R_t = R_w + R_p + R_x + R_z \tag{A-5}$$

$$\widetilde{R} = \frac{1}{\frac{1}{R_1} + \frac{1}{R_2}} = \frac{1}{\frac{1}{\frac{1}{h_{t1}} + \frac{d_1}{\lambda_s} + R_{cov1}} + \frac{1}{\frac{1}{h_{t2}} + \frac{d_2}{\lambda_s} + R_{cov2}}} \tag{A-6}$$

$$R_{l,f} = \frac{1}{\frac{A_l \cdot U_l}{A_{fl}} + \frac{n_a \cdot V_a \cdot \rho_a \cdot c_a}{A_{fl}}} \tag{A-7}$$

Appendix 2

Calculation of room and mass temperatures using the dynamic thermal room-model.

The differential-equation system (DEQS) for room, slab and wall temperatures of the dynamic thermal room model according to Fig. 2 is given as:

$$C_r \frac{d\vartheta_r}{dt} = \frac{1}{R_{l,f}}(\vartheta_{oa} - \vartheta_r) + \frac{1}{R_{iW,0}}(\vartheta_{iW1} - \vartheta_r) + \frac{1}{R_{s1,0}}(\vartheta_{s1,1} - \vartheta_r) + \frac{1}{R_{s2,0}}(\vartheta_{s2,n2} - \vartheta_r) + \dot{q}_g$$

$$C_{s1,1} \frac{d\vartheta_{s1,1}}{dt} = \frac{1}{R_{s1,0}}(\vartheta_r - \vartheta_{s1,1}) + \frac{1}{R_{s1,1}}(\vartheta_{s1,2} - \vartheta_{s1,1})$$

$$C_{s1,2} \frac{d\vartheta_{s1,2}}{dt} = \frac{1}{R_{s1,1}}(\vartheta_{s1,1} - \vartheta_{s1,2}) + \frac{1}{R_{s1,2}}(\vartheta_{s1,n1} - \vartheta_{s1,2})$$

$$C_{s1,n1} \frac{d\vartheta_{s1,n1}}{dt} = \frac{1}{R_{s1,2}}(\vartheta_{s1,2} - \vartheta_{s1,n1}) + \frac{1}{R_{s1,n1}}(\vartheta_c - \vartheta_{s1,n1})$$

$$C_{s2,1} \frac{d\vartheta_{s2,1}}{dt} = \frac{1}{R_{s2,1}}(\vartheta_c - \vartheta_{s2,1}) + \frac{1}{R_{s2,2}}(\vartheta_{s2,2} - \vartheta_{s2,1})$$

$$C_{s2,2} \frac{d\vartheta_{s2,2}}{dt} = \frac{1}{R_{s2,2}}(\vartheta_{s2,1} - \vartheta_{s2,2}) + \frac{1}{R_{s2,n2}}(\vartheta_{s2,n2} - \vartheta_{s2,2})$$

$$C_{s2,n2} \frac{d\vartheta_{s2,n2}}{dt} = \frac{1}{R_{s2,n2}}(\vartheta_{s2,2} - \vartheta_{s2,n2}) + \frac{1}{R_{s2,0}}(\vartheta_r - \vartheta_{s2,n2})$$

$$C_{iW,1} \frac{d\vartheta_{iW,1}}{dt} = \frac{1}{R_{iW,0}}(\vartheta_r - \vartheta_{iW,1}) + \frac{1}{R_{iW,1}}(\vartheta_{iW,2} - \vartheta_{iW,1})$$

$$C_{\mathrm{iW},2}\frac{\mathrm{d}\vartheta_{\mathrm{iW},2}}{\mathrm{d}t}=\frac{1}{R_{\mathrm{iW},1}}(\vartheta_{\mathrm{iW},1}-\vartheta_{\mathrm{iW},2})+\frac{1}{R_{\mathrm{iW},2}}(\vartheta_{\mathrm{iW},n\mathrm{iW}}-\vartheta_{\mathrm{iW},2})$$
$$C_{\mathrm{iW},n\mathrm{iW}}\frac{\mathrm{d}\vartheta_{\mathrm{iW},n\mathrm{iW}}}{\mathrm{d}t}=\frac{1}{R_{\mathrm{iW},2}}(\vartheta_{\mathrm{iW},2}-\vartheta_{\mathrm{iW},n\mathrm{iW}})+\frac{1}{R_{\mathrm{iW},n\mathrm{iW}}}(\vartheta_{\mathrm{r}}-\vartheta_{\mathrm{iW},n\mathrm{iW}}) \tag{A-8}$$

As the core temperature node is assumed massless, the correlation for ϑ_{c} can be formulated as an algebraic, equation:

$$0=\frac{1}{R_{\mathrm{s1},n1}}(\vartheta_{\mathrm{c}}-\vartheta_{\mathrm{s1},n1})+\frac{1}{R_{\mathrm{s2},1}}(\vartheta_{\mathrm{c}}-\vartheta_{\mathrm{s2},1})+\frac{1}{R_{\mathrm{t}}}(\vartheta_{\mathrm{c}}-\vartheta_{\mathrm{sw}}) \tag{A-9}$$

Since the DEQS (A-8) is linear, it can be formulated in state space notation as

$$\frac{\mathrm{d}\underline{x}(t)}{\mathrm{d}t}=F\cdot\underline{x}(t)+G\cdot\underline{u}(t);\quad \underline{u}(t)=\begin{bmatrix}\vartheta_{\mathrm{sw}}(t)\\ \vartheta_{\mathrm{oa}}(t)\\ \dot{q}_{\mathrm{g}}(t)\end{bmatrix}$$
$$\begin{bmatrix}\vartheta_{\mathrm{r}}(t)\\ \vartheta_{\mathrm{c}}(t)\end{bmatrix}=C\cdot\underline{x}(t) \tag{A-10}$$

By (zero-order hold) discretization of the matrices of the continuous state space model (sampling time T_{s})

$$A=\mathrm{e}^{F\cdot T_{\mathrm{S}}}=I+F\cdot T_{\mathrm{S}}+F^2\cdot\frac{T_{\mathrm{S}}}{2!}+F^3\cdot\frac{T_{\mathrm{S}}}{3!}+F^4\cdot\frac{T_{\mathrm{S}}}{4!}+\dots$$
$$B=\int_0^{T_{\mathrm{S}}}\mathrm{e}^{F(T_{\mathrm{S}}-\sigma)}\mathrm{d}\sigma=F^{-1}(A-I)G \tag{A-11}$$

the discrete state space model is obtained:

$$\underline{x}_{k+1}=A\cdot\underline{x}_k+B\cdot\underline{u}(k);\quad \underline{u}(k)=\begin{bmatrix}\vartheta_{\mathrm{sw}}(k)\\ \vartheta_{\mathrm{oa}}(k)\\ \dot{q}_{\mathrm{g}}(k)\end{bmatrix}$$
$$\begin{bmatrix}\vartheta_{\mathrm{r}}(k)\\ \vartheta_{\mathrm{c}}(k)\end{bmatrix}=C\cdot\underline{x}_k \tag{A-12}$$

Applying the profile of the lower heat-gain bound $\dot{q}_{\mathrm{g,lb}}$ (assuming $\vartheta_{\mathrm{sw}}=\vartheta_{\mathrm{oa}}=0$), the values of the state vector for each time step can be formulated according to:

$$\begin{aligned}
&\underline{x}_{\mathrm{lb}}(0)=\underline{x}_{\mathrm{lb}}(0)\\
&\underline{x}_{\mathrm{lb}}(1)=A\cdot\underline{x}_{\mathrm{lb}}(0)+B\cdot\underline{u}_{\mathrm{lb}}(0)\\
&\underline{x}_{\mathrm{lb}}(2)=A\cdot\underline{x}_{\mathrm{lb}}(1)+B\cdot\underline{u}_{\mathrm{lb}}(1)=A^2\cdot\underline{x}(0)+AB\cdot\underline{u}_{\mathrm{lb}}(0)+B\cdot\underline{u}_{\mathrm{lb}}(1)\\
&\vdots\\
&\underline{x}_{\mathrm{lb}}(n)=A^n\cdot\underline{x}(0)+A^{n-1}B\cdot\underline{u}_{\mathrm{lb}}(0)+A^{n-2}B\cdot\underline{u}_{\mathrm{lb}}(1)+\dots+AB\cdot\underline{u}_{\mathrm{lb}}(n-2)+B\cdot\underline{u}_{\mathrm{lb}}(n-1)\overset{!}{=}\underline{x}_{\mathrm{lb}}(0)
\end{aligned} \tag{A-13}$$

For quasi-steady-state conditions, $\underline{x}_{\mathrm{lb}}(n)$ equals $\underline{x}_{\mathrm{lb}}(0)$. In this case, (A-13) can be solved to obtain the state vector for time step 0 as

$$\underline{x}_{\mathrm{lb}}(0)=(I-A^n)^{-1}\left\{\sum_{i=0}^{n-1}(A^{(n-i-1)}B\cdot\underline{u}_{\mathrm{lb}}(i))\right\} \tag{A-14}$$

Combining (A-14) and (A-13), the values of the state vector for the remaining time steps can be calculated. From these results, the requested room temperature profile can be obtained.

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