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Sizing Thermally Activated Building Systems (TABS): A Brief Literature Review and Model Evaluation

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SUMMARY

While Thermally Activated Building Systems (TABS) is a recognized low-energy HVAC candidate system for net-zero-energy buildings, sizing of these systems is complex due to their slow thermal response. In this paper, seven design and control models have been reviewed and characterized systematically with an aim to investigate their applicability in various design scenarios and at different design stages. The design scenarios include variable space heat gain, different building thermal mass and varying pump operating hours. Three design stages were considered, including feasibility study, early design decisions and detailed sizing. The applicability to different design stages was evaluated based on a compromise between accuracy and ease of implementation of the design methods. Five of the models were shortlisted for a future simulation-based evaluation and recommendation.

INTRODUCTION

TABS are radiant cooling systems with fluid carrying pipes embedded in the building structure. They have potential to save energy and facilitate renewables integration, but their design/sizing is complex. TABS have different peak cooling load, 24-hour total cooling energy and hydronic cooling load than all air systems. For a detailed description, please refer to (Feng, Schiavon, and Bauman 2012). Three simple control models of TABS have been proposed in the literature, most of which are single to bivariate linear models (Olesen and Dossi 2004). Recently, more complex integrated design and control models (Lehmann 2007, Gwerder et al. 2008, 2009) have been proposed. These models require an iterative solving approach and include multiple input variables.

Design of TABS primarily encompasses *sizing of the design parameters* on the TABS side. TABS design parameters include pipe spacing, water flow rate, pipe position in the slab, circuit temperature rise, circuit pressure drop, water supply temperature, slab thermal mass, heat transfer coefficient and surface temperature. Associated with sizing of TABS are selection and sizing of other ancillary components, for example cooling plant, circulation pump, dehumidifier and sometimes concurrent or alternative cooling systems. TABS cannot be applied in all climate types and internal heat gain conditions.

The aim of this study was to identify and evaluate design and control models of TABS.

METHODS

Literature review

A literature search was performed in Google scholar using key terms “Radiant cooling” + “design”, “Thermally active building systems” + “design”, “TABS” + “design”, “TABS” + “control”, “Low temperature heating and high temperature cooling”. In addition to peer reviewed papers and dissertations, several conference papers were also screened. One international standard, one guidebook and nine peer-reviewed papers were found, exclusively dealing with the design and control of TABS. Sixteen additional references directly or indirectly related to control of TABS were reviewed. Three rule-based, one hybrid, two physically based and one building energy simulation-based methods were identified and characterized by system types, active surface, design approach, design parameters and validation. This classification is presented in Table 1 in the results section. Although, this study is primarily oriented towards identifying design models for TABS, in reality control and design of TABS is a highly integrated topic (Gwerder et al. 2008). Hence, models proposed for both design and control have been included in this study.

Most of the TABS design literature including design standard (ISO 2012) recommends for sizing of water supply temperature alone since this parameter determines the choice and size of the condensing unit (e.g. cooling tower) and pumps. Hence in this study, too, models of water supply temperature of TABS are investigated. Models which require slab thermal resistance information take pipe spacing, water flow rate, pipe position in the slab as inputs. The pipe spacing is based on constructability, desired surface temperature homogeneity and rigidity of the pipe, while mass flow rate is based on desired temperature rise and pressure drop and Reynolds number in the hydronic circuit.

Model classification

Table 1 displays the design and the control equations that have been selected for further comparison and classification in Table 2. Table 2 has seven headers. The system type refers to the sub-category of TABS classified by active surface (like wall, ceiling, floor) and thermal mass (slab thickness, active area, pipe depth). Under design approach, these TABS models, with their underlying assumptions and implementation methods, are presented as:

- i) *Model* - rule based or physically based, like numerical and analytical models, etc.
- ii) *Model purpose* – design, control, or a combined design and control approach.
- iii) *Model assumptions* – whether the model accounts for difference in heat transfer mechanism of a radiant system and convective system, or if it assumes a steady state or dynamic boundary conditions
- iv) *Implementation methods* – non-iterative, iterative, or using transient simulation.

Henceforth, in this paper, the models will be referred to by their respective numbers in Table 2. The rules in rule-based models are either derived empirically or from simulations. Model assumptions may affect the complexity of the model and its applicability in certain design phase. In addition to model accuracy, ease of implementation can promote early adoption of a design method. For example, a non-iterative method like excel spreadsheet calculation has low computation cost. The design parameters are calculated from the quasi-steady-state conditions that the system is designed to meet on a design day without involving multiple transient simulation runs or solving 1st order differential equations. The iterative methods can range from sensitivity analysis at the basic level to more complex optimization and reliability

based methods. Building energy simulation tools like TRNSYS and IDA ICE have been used by researchers to vary the design parameters of TABS and measure the effect on the desired thermal comfort performance. Simulations have been used to derive design models as a function of one or many variables and may not be extrapolated to scenarios beyond those used in the sensitivity study. In optimization, the design parameters are varied by optimization algorithm until the design objective is met within a desired error or tolerance range. One major limitation of optimization in design phase of building services is the amount of uncertainty in the input data. Reliability based design, on the other hand, accounts for uncertainties in inputs and associates a probability to the predicted design performance. This is widely used in machine design, product design, quality control and systems engineering, but is still at a nascent stage for building systems (Hopfe and Hensen 2011; Chen and Yu 2009) at least in practice. The *design/control parameters* column in Table 2 provides the list of design and control inputs for the models. The sixth column, *validation procedure*, enumerates the number of cases simulated as examples of application or for validation of the models. Under the *comments* heading the applicability of the design methods under various design scenarios have been discussed. The selected models have been classified by varying internal gains (VIG), varying pump operation hours (VPO) and thermal mass (TM). VPO is similar to pulse width modulation (PWM) pump operation, like, 8 hours precooling is a special case of VPO when the PWM period is 24 hours. VPO may be desirable for two reasons, i) precooling that can take advantage of alternative cooling sources like night ambient outdoor temperature or off-peak electricity tariff and ii) increased energy efficiency with PWM (Lehmann et al. 2011). Inclusion of VIG in the design model will lead to more robust design. The effect of zone thermal mass is primarily to shave and shift the peak cooling load. The classification is based on the hypothesis that the models that account for TM or VIG will be sensitive to changes in these parameters and therefore display more consistent comfort performance under different design scenarios, while the simpler models will not be responsive to these changes.

RESULTS AND DISCUSSION

The TABS models are shown in Table 1 followed by classification of existing TABS design and control models in Table 2. Out of seven models, three are control models, two design models and the rest combined design and control models. Equation 1 is for open loop control of outdoor air temperature compensated water supply temperature and is not dependent on any other design parameter like internal or solar heat gains. Equation 2 represents a zone operative temperature feedback control of water supply temperature in addition to outdoor air temperature compensated control. It is very similar in structure to equation 6, except for the heat gain part in equation 6, which is replaced by a function of zone temperature in equation 2. Equations 3 and 4 constitute the ‘simplified sizing by diagrams’ method of the latest design standard for TABS, ISO 11855 (ISO 2012). Both are component equations of models 3 and 4, that have similar structure but different coefficients, 3 being for continuous pump operation and 4 for precooling. This method allows sizing of water supply temperature on a design day, the aim of the standard being to guide adoption of renewable energy sources. These are steady state models of supply water temperature that assume a constant average surface temperature of the slab during the operation. The coefficients of equation 4 are given for south, east and west zones based on cooling load profiles, with south zone having the strongest correlation with the 24 hours cooling load due to solar load. The coefficients have higher values for 8 hours pump operation as opposed to continuous pump operation for the same cooling load, which can be explained from energy balance of the supply and the demand side aggregated over a 24 hours period. Equation 5 is a steady state model which relates the supply water

temperature under PWM, given the supply water temperature under continuous pump operation is known. It is based on the principle that energy extracted by the slab during switched-on period of PWM is equal to that under continuous pump operation. This equation therefore reduces the 24 hours water supply temperature by a factor which is a function of the duty cycle and resistance of the tubing. Model 5 can be implemented in conjunction with any of the other three models for non-continuous pump operation.

Table 1. Model equations for water supply temperature

No.	Water supply temperature models	Sources
1	$T_{ws} = 0.35(18 - T_{oa}) + 18$ (°C)	(Olesen and Dossi 2004)
2	$T_{ws} = 0.52(20 - T_{oa}) + 20 - 1.6(T_{op} - 22)$ (°C)	(Olesen and Dossi 2004)
3	$T_{ws} = T_s - \frac{Q}{h} \cdot 1000(\tilde{R} + R_t)$	(ISO 2012))
4	$T_s = T_{rsp} + coeff \cdot Q$	
5	$T_{wspm} = T_{ws} - \left[\frac{R_t}{\tilde{R} + R_t} (T_{rsp} - T_{ws}) \right] \left(\frac{\Delta t_c}{\Delta t_{c1}} - 1 \right)$	(Gwerder et al. 2009)
6	$T_{ws} = T_{rsp} + \frac{R_t + \hat{R}}{R_f} (T_{oa} - T_{rsp}) + (R_t + \hat{R})q_{ub}$	(Gwerder et al. 2008)

Nomenclature

T_{oa}	24 hours running mean outdoor dry bulb temperature, °C
T_{op}	Zone operative temperature, °C
Q	Specific heat; 24 hours sum of internal and 10% solar gain with two internal gain profiles, continuous and with 2 hours recess, Wh/m ²
\tilde{R}	Resistance between tubing and component surface, K-m ² /W
R_f	Thermal resistance of the building envelope, K-m ² /W
R_t	Tubing thermal resistance for constant mass flow rate, K-m ² /W
T_{rsp}	Room operative set point temperature, °C
$\Delta t_{c1}/\Delta t_c$	Total pump running hours as percentage of cooling hours, <i>unitless</i> .
T_{ws}	Water supply temperature for 24 h operation, °C
T_{wspm}	Water supply temperature for precooling, °C
q_{ub}	Upper bound steady state internal and solar heat gain that would produce the same maximum zone temperature as the dynamic cooling load profile in a given space, W
$coeff$	Coefficient of equation for calculating the design active surface temperature from 24 hours accumulated cooling energy demand. This coefficient is varies with zone orientation, precooling and continuous pump operation and internal load profile, <i>unitless</i>
h	Number of hours of pump operation
T_s	Temperature of active surface

Table 2. Classification of existing design methods of TABS by system type, design approach, design/control parameters and validation procedure.

Model no.	Paper/author	System type		Design approach				Design and control parameters	Validation procedure	Comments
		Active surface	Thermal mass	Model	Purpose	Model assumptions	Implementation methods			
1	(Olesen and Dossi 2004)	Ceiling + Floor	CC, 180 mm thick slab, 48 m ² active area, pipe in the middle	Rule based single variable	Control	Linear relation between cooling load and outdoor air dry bulb temperature, equation source not reported.	Non-iterative	Outdoor air dry bulb temperature	30 cases simulated in total for the climates of Wurzburg and Venice in TRANSYS	No VIG No VPO No TM Variations of the model must be tried for feasibility study
2	(Olesen and Dossi 2004)	Ceiling + Floor	CC, 180 mm thick slab, 48 m ² active area, pipe in the middle	Rule based bivariate	Control	Linear relation between cooling load, outdoor air dry bulb temperature and zone operative temperature, equation source not reported	Iterative	Outdoor air dry bulb temperature and zone operative temperature	8 cases simulated in total for Wurzburg, Venice (a) in TRANSYS, 12 cases simulated for Phoenix, Miami and San Francisco (b) in IDA ICE	VIG and TM using zone temperature feedback. No VPO Condensing unit selection, must be implemented with building energy simulation
3 and 4	EN 15377-3:2006	Ceiling/ceiling+ floor both	Similar system configuration as reported by 6	Hybrid (combination of linear regression and physically based steady state model)	Design	Linear relation between 24-hour cooling load and radiant surface temperature; coefficient derived for different zones, internal load profile and pump operation hours, by dynamic simulation. The slab temperature constant 24 hours.	Iterative	Pump operation, slab and tubing resistance, room temperature setpoint, number of thermally active surfaces, zone orientation, internal load profile Model 3 refers to continuous operation and 4 for precooling	No validation reported in the standard, presumably derived by method reported in paper no. 6.	VIG using empirical relation. Two specific VPO. No TM except for the case for which the coefficients were derived. Zone level modulation of water supply temperature. Condensing unit selection and detailed design with further validation

5	(Gwerder et al. 2009, 1606-1616)	Floor	CC, 250 mm thick slab, 25 m ² active area, pipe depth not reported	Physically based and quasi-steady state/design and control	Control	Requires preliminary water supply temperature calculation for continuous pump operation	Non-iterative. This paper also mentions a first order equation model of pulse width modulation. The simpler model showed comparable results and was thus chosen	Water supply temperature for variable pump operation hours, slab and tubing thermal resistance, internal heat gain, room set point temperature	Laboratory tests	VIG <i>same as above</i> . VPO TM Requires further validation
6	(Gwerder et al. 2008, 565-581)	Floor	CC, 250 mm thick slab, 29 m ² active area, pipe depth not reported	Physically based and quasi-steady state	Design and control	24 hours constant cooling load, continuous pump operation, considers difference in heat exchange mechanism of a TABS and overhead system	Iterative/reliability based	Water supply temperature, envelope, slab and tubing thermal resistance, internal heat gain, room set point temperature	One design scenario as an example, simulated in TRANSYS	VIG <i>using steady state formulation of limiting internal heat gain</i> . No VPO TM Complex and not practically feasible without an existing mathematical model.
7	(Lehman n, Dorer, and Koschen z 2007, 593-598)	Ceiling	CC, 300 mm thick slab, 30 m ² active area, pipe depth not reported	Building energy simulation (physically based)	Design	Considers difference in heat exchange mechanism of TABS and overhead system	Sensitivity study based	Water supply temperature, pump operation hours	6 simulations purportedly conducted in TRANSYS	VIG <i>by empirical relations</i> . VPO TM Detailed design for non-standard zones

VIG – Varying internal gains

VPO – Varying pump operation hours

TM – Thermal mass of the zone

CC – Concrete core

Gwerder et al. (Gwerder et al. 2009) also reported a transient model of water supply temperature for PWM that uses time constants of the zone as well as the slab. They compared the transient method with the former single variable steady state method and reported negligible difference in performance of the two models by laboratory tests. The zone set point temperature and heat gain are assumed to be constant for the entire operation cycle, for example 24 hours. Equation 6 is a steady state model of water supply temperature as function of envelope resistance, slab resistance, water supply mass flow rate and tubing characteristics, and zone set point and outdoor air temperatures. Implementation of Equation 6 is a two-step process, calculating the maximum temperature rise in space under transient heat gain profile and then calculating the steady state solar and internal heat gain, q_{ub} that can produce this temperature rise. This method therefore requires disaggregating the effect of internal and solar gains from that of conductive heat gain. Implementing such a method at early design stage is inconvenient unless a mathematical model of the zone already exists. Model 7 is a simulation based method of iteratively finding water supply temperature for given pump operation hours. The authors of this model did not report any equation, but derived a graphical example of the working principle of TABS, similar to those in Annex A of ISO 11855-4. Therefore no representative equation for this model could be included in Table 1. This method was presumably used to derive the coefficients of the regression equation for model 3 in standard (ISO 2012). A nomogram of this model is given in Figure 11 in ISO 11855-4.

All of the design methods except model 1 require dynamic simulations at least in calculation of cooling load. Depending on the type of the building model and treatment of T_{op} , model 2 may or may not account for the presence of thermal mass. Babiak and Kolarik et al. (Babiak 2007; Kolarik et al. 2011) reported that thermal mass of the building can have considerable influence on daily comfort performance of TABS. Models 5-6 all include the effect of building thermal mass in the cooling load calculation in some form, discussed in the comments section of Table 2. Equations 3 and 4 also account for thermal mass but specific to the conditions for which the coefficients were derived. These conditions have not been reported in the source. Models 1 and 5 are suitable candidates for feasibility study and early design phase spreadsheet calculations. Model 3 can also be implemented on spreadsheet if the 24 hours quasi-steady state cooling load is used. ISO 11855 however recommends use of building energy simulation to calculate the cooling load from convective system for use in this model. But the 24 hours hydronic cooling energy extracted by the waterside of TABS has been reported to be higher than that of all air system by Feng et al. (Feng, Schiavon, and Bauman 2012). This cooling energy demand cannot be predicted without modeling and simulation of TABS surface. This entails that more accurate calculation of design water supply temperature using model 3 and 4, should be iterative. By far it can be stated that model 3 accounts for maximum number of influencing factors in design of TABS and hence should display the most consistent performance under varying design conditions. It could be the best candidate design model for detailed design and sizing of cooling plant with proper cooling load calculation. The discrepancies between the comfort performance of this model with convective cooling energy demand and iteratively calculated TABS hydronic cooling energy should be assessed by dynamic simulation. The design models should also be tested for the time window of input variables like outdoor air dry bulb temperature and internal and solar heat gain, influencing thermal performance of TABS, ranging from an hourly average to 24 hours to even a 3-4 days running mean. This depends on thermal response time of TABS, a property that varies by density, thickness and area of the slab from 4 to 5 hours at the lower end up to 13 hours for ceiling and even higher for floor cooling (Babiak 2007). A seasonal simulation must be performed to determine the influence of zone dew point temperature on the water supply temperature for sizing the dehumidification system and the condensing unit. Thermal mass of the slab may govern the deployment of additional chilled water storage and

influencing TABS deployment in certain design scenarios. Future tests should be performed i) to validate coefficients of models 3 and 4 for more complex building geometries, varied construction types and internal heat gain profile, ii) to size thermal mass for more systematic zoning of TABS affecting design water supply temperature, iii) to assess the consistency of comfort performance of the studied models.

CONCLUSIONS

Several design and control models of TABS were reviewed from the literature and seven of them were classified. Most of these models allow sizing of water supply temperature, since it affects the choice and size of the cooling plant. The simplest model is a single variable one and does not account for varying thermal mass, internal heat gains and pump operation modes, but can be easily implemented in spreadsheet or in optimization loop. Simplified method in ISO 11855 uses a mix of physically based and correlation models. It can be implemented on spreadsheet at early design phase with quasi-steady-state cooling load, after validating its coefficients for wider range of design scenarios. Since hydronic loop cooling load is different from all air system cooling load, modeling and simulation of TABS may, however, be desirable for more accurate results. Different design model should be used for precooling and continuous operation as can be explained from energy balance standpoint.

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