Critical Simulation Based Evaluation of Thermally Activated Building Systems (TABS) Design Models

By

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Abstract

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Thermally Activated Building Systems (TABS) is a recognized low-energy HVAC system. Sizing of these systems is complex due to their slow thermal response. Limited cooling capacity of these systems and inadequacy of conventional sizing method, that assumes high factor of safety, is preventing early adoption of these systems. TABS, however, is proven to be energy-efficient and capable of preserving comfort in several commercial buildings of Europe. There is, however no comprehensive case study report on comfort performance of TABS in the US. With this being the background, my dissertation aims to identify and recommend a design method for TABS that balances between accuracy of multivariable complex design models, high computational cost of models requiring an iterative approach and computational ease of simple single to bivariate linear design models. The dissertation work involved: 1) a systematic qualitative review of seven TABS design models from the literature, and 2) a simulation based quantitative comfort performance assessment of three shortlisted design models. I reviewed seven design and control models and characterized them systematically with an aim to investigate their applicability in various design scenarios and at different design stages. All of these models size water supply temperature (WST) as this parameter will be used for selection and sizing of the cooling plant or the condenser unit. The design scenarios include variable internal heat gain, different building thermal mass, varying pump operating hours and varying solar gain due to orientation. Other parameters affecting cooling load and thermal performance of TABS that were held constant in this study included window-to-wall area ratio, zone volume, construction insulation, supply air temperature and volume flow rate of the ventilation system, external shading, location, TABS mass flow rate, pipe layout, active surface configuration and TABS thermal properties. I considered three design stages: feasibility study, early design decisions, and detailed design sizing and the selection criteria are reliability and ease of implementation. Results of the qualitative analysis indicated that based on the above-mentioned criteria, a hybrid model recommended by ISO 11855 is the best candidate for detailed design and sizing of the cooling plant. An outdoor temperature (T_{oa}) compensated model, a zone operative

temperature (OT) feedback based model and the hybrid model from ISO 11855 were isolated for transient simulation based quantitative evaluation in terms of a novel comfort exceedance metric. This metric accounts for both duration and severity of discomfort and is weighted by instantaneous occupancy. For comfort analysis in terms of zone OT, zone RH was maintained using humidistats. TABS was the only cooling system in the building. Twelve simulations were carried out in a standard 5 zone small office building for CZ03 in EnergyPlus v7.0 under 2 different heat gains and 2 construction types. Results of the simulation study indicated that both the T_{oa} compensated model and zone operative temperature feedback based model provided equally good comfort in 14 out of 20 design scenarios including zone orientation. However, the zone OT feedback model responded better to the heat gain and thermal mass conditions as expected, and is therefore recommended as a more robust model for early and detailed design phase implementation. The hybrid model recommended by ISO 11855 resulted in comfort exceedance of 10% to 48%, while the recommended threshold exceedance for this study was 3-5%. This model also resulted in significantly reduced discomfort using 24 hours hydronic cooling energy of TABS instead of the design day 24 hours cooling energy of convective system.

Dedication

To my father and mother

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List of Acronyms

CC	Concrete core conditioning
CD	Room peak cooling load on design day
CE	24 hours cooling energy on TABS waterside
DBT	Dry bulb temperature
HVAC	Heating Ventilation and Air conditioning
PWM	Pulse Width Modulation
RH	Relative humidity
TABS	Thermally Activated Building Systems
TM	Thermal mass
VAV	Variable Air Volume
VPO	Variable pump operation
VIG	Variable internal heat gain
WST	TABS water supply temperature
ZEB	Zero energy buildings

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Chapter 1

1. Introduction

1.1 Motivation

Commercial buildings have a significant impact on energy use and the environment. They account for approximately 19% (18.2 quads) of the total primary energy consumption in the United States (U.S. Energy Information Administration). The number of buildings is increasing steadily because new buildings are added to the national stock faster than old buildings are retired. Energy efficiency measures and load management on the other hand have resulted in considerable increase in peak power reduction in 2010 compared to 1999 (U.S. Energy Information Administration January 2012), out of which 65 % reduction was contributed from energy efficiency measures. The U.S. Department of Energy's (DOE) Building Technologies Program has established a goal to create the technology and knowledge base for marketable zero-energy commercial buildings (ZEBs) by 2025 calling for more aggressive energy efficiency measures across the country. Based on 4 years of measurement of six low energy buildings in different parts of the US, Torcellini et al. (2006) came to the conclusions that today's technology can substantially change the energy performance of buildings provided they are designed, deployed and operated as integrated systems. They also pointed out that low energy buildings may under-perform compared to target design performance i.e. they consume higher energy than predicted. Under such circumstances of uncertain performance in the operational phase, adoption of the latest off-the-shelf technologies suffers as designers fail to convince the owners about the potential energy savings achievable with a certain technology. There is a need for proper design guidelines and more accurate performance prediction tools for these new building technologies that will accommodate the operational uncertainties in the design. Hence one of the key goals in securing net zero energy commercial buildings is development of a knowledge base of effective combinations of building technologies for better deployment and integration of these technologies on a mass scale. Thermally Activated Building Systems (TABS) is one such off-the-shelf technology that offers potential for energy savings, as learned from case studies in Europe (Kalz et al. 2006; Kalz et al. 2010; Schmidt and Kaiser 2007). Unfortunately there is not enough literature on simulated/measured energy savings of TABS in the US. Most of the energy studies are concerning chilled ceiling. Using EnergyPlus simulations Tian and Love (2009) that water side free cooling with TABS and VAV, in addition to well-designed control of the systems could provide as much as 42% energy savings compared to the base case of TABS with campus chilled water supply, for the climate of Calgary. Another IDA Indoor Climate and Energy simulation based study by Kolarik et al. (2011) showed that with improved control of TABS cooling (supplied by ground heat exchanger) could save 15% to 49% annual primary energy in an office building in the three US climates, depending on the location. From TRANSYS annual energy simulations of ventilation assisted TABS in the climate of Omaha, Nebraska, Henze et al. (2007) reported a 20% energy penalty of pure VAV systems compared to the former system in a typical compact office building. In spite of the projected energy savings, this system has not seen enough market penetration owing to limited cooling capacity and lack of reliable design tool.

1.2 Background on TABS

System description

TABS is a form of radiant zone thermal conditioning system that exchanges more than 50% of the heat within the space by thermal radiation (Babiak et al. 2007). Hydronic radiant systems use water as the heat carrier and take advantage of the high thermal capacity of water to minimize the temperature rise of the cooling medium required for conditioning a space. Radiant systems can be of different kinds based on the heat transfer mechanism between the active surface and the pipe and whether the system is thermally coupled with the building structure or thermally insulated. There are primarily three categories of hydronic radiant systems:

1. Panels: heat carrier close to the surface, not structurally integrated,
2. Embedded systems: Pipe embedded in structure physically but insulated from of the structure as much as possible,
3. Thermally active building systems: The heat carrier is embedded in the building structure and thermally integrated with the structure.

TABS differs in thermal performance from the rest of the radiant system due to the integration of the thermal mass, that results in slower system thermal response compared to rest of the radiant systems. This can lead to possible asynchrony of occurrence of thermal load in space and plant operation, taking advantage of temporal efficiency of natural cooling sources like night cooling. In REHVA (Babiak et al. 2007), a further classification of embedded heating/ cooling systems into 3 sub-groups based on pipe configurations has been given:

1. Type E - Pipes of different diameters embedded in structure, mostly floor and ceiling, at different depths.
2. Type F - Capillary pipes embedded in a layer at the inner ceiling or as a separate layer in gypsum
3. Advanced systems with capillary pipes embedded in gypsum board or plaster with Phase change material.

The Type E TABS is most widely used in Europe and has the lowest installation and operation cost. To the author’s knowledge only a very few commercial buildings in the US have radiant panels and type E TABS. While type F has been rarely deployed worldwide, the 3rd category of

TABS is still in research phase, mostly implemented in a few experimental buildings. In Type E typically pipes 15 mm to 20 mm in diameter and 150 – 300 mm apart are embedded near the neutral axis of the slab. The pipe configurations can be meander, double meander or spiral. The pipe layout and the center to center pipe spacing influence the surface temperature distribution of TABS. In this thesis TABS will refer to type E. The rest of this section will describe Type E TABS.

Thermal response of TABS

The main difference between TABS and other radiant systems, like panels and embedded systems is that the thermal response time of TABS is much higher. Thermal response time is defined as the rise-time characterizing the response to a time-varying input of a first-order of a linear time-invariant system. It is measured as the time required by a system's step response to reach 63.2% of the final value due to a step input. TABS slab thickness may vary from 75 mm to 300 mm depending on the structural design. Slab response time may vary considerably by density, thickness, area of the slab and depth of tubing in the slab, from 4 to 5 hours at the lower end up to 13 hours (Kalz 2009; Sakellariou 2011; Sourbron et al. 2009).

Cooling capacity

One of the shortcomings of TABS is limited cooling capacity as the active surface temperature is limited by room dew point temperature and discomfort from radiant asymmetry. Hence, they cannot be implemented in all climates, unless heat gain is adequately controlled in space. Moreover the conventional approach of designing HVAC systems with high factor of safety leaves the designers with little confidence in systems with such limited capacity. As per the latest standard on TABS, ISO 11855 (ISO 2012), the cooling power of TABS in W/m^2 can be obtained from combined heat transfer coefficient and the mean differential surface temperature, raised to some power. The coefficient of this equation varies with surface configurations (floor/ceiling/wall). For example, a ceiling surface with a minimum allowable mean surface temperature of 19°C, a surface heat transfer coefficient of 8.92 W/m^2 K (ISO 2012) and a maximum allowable zone operative temperature of 26°C has the maximum cooling capacity of 75.8 W/m^2. For more detail refer to section 7 of part 2 of this standard for a list of possible mean surface temperature for different boundary conditions. Due to slow thermal response of TABS, the cooling capacity of TABS should be calculated as 24 hours sum of total energy extracted rather than a peak value. Lehmann et al. (2007) reported cooling capacity of TABS in terms of 24 hours energy, while the cooling power was deduced from the same and pump operation hours, water supply temperature and allowable temperature rise in space. The 24 hours summation is however just an assumption that has not been investigated or justified. From first principles the ideal summing period is expected to vary by thermal response time of the system. Further, waterside dynamics of TABS is decoupled from room side dynamics by its thermal mass. Using dynamic simulation Feng et al. (2012) found that the cooling energy removed by the water side of TABS is higher than that removed at its active surface. Figure 1.1 excerpted from this paper displays the difference between the cooling energy removed by the water side of TABS and cooling energy of a convective system for the same set of zone operative setpoint temperature. The figure shows that the peak cooling load of TABS is 15.5% higher than that of all-air system for the same proportion of radiant and convective instantaneous load. TABS, however, meets a higher fraction of radiative heat gain than a convective system, resulting in higher overall cooling load.

Self-regulating effect

Figure 1.2 taken from (Lehmann et al. 2007) shows the convective and the radiative heat transfer mechanisms in space. The time scale in which the air heats up directly from the convective fractions of cooling load is much lower than that of temperature change of the active surface. This temperature constancy is also attributed to the high thermal mass of other surfaces that participate in radiative heat exchange. The surface temperature constancy relative to that of the zone air is called a "self-regulating effect", which means if the air is warmer than the active surface, the surface behaves as a cooler, while removal of the convective load can quickly bring down the air temperature below surface temperature, in which case the surface behaves as a heater. The switching effect can be enhanced if the surface temperature is kept as close to the comfort temperature as possible (Weitzmann 2004).

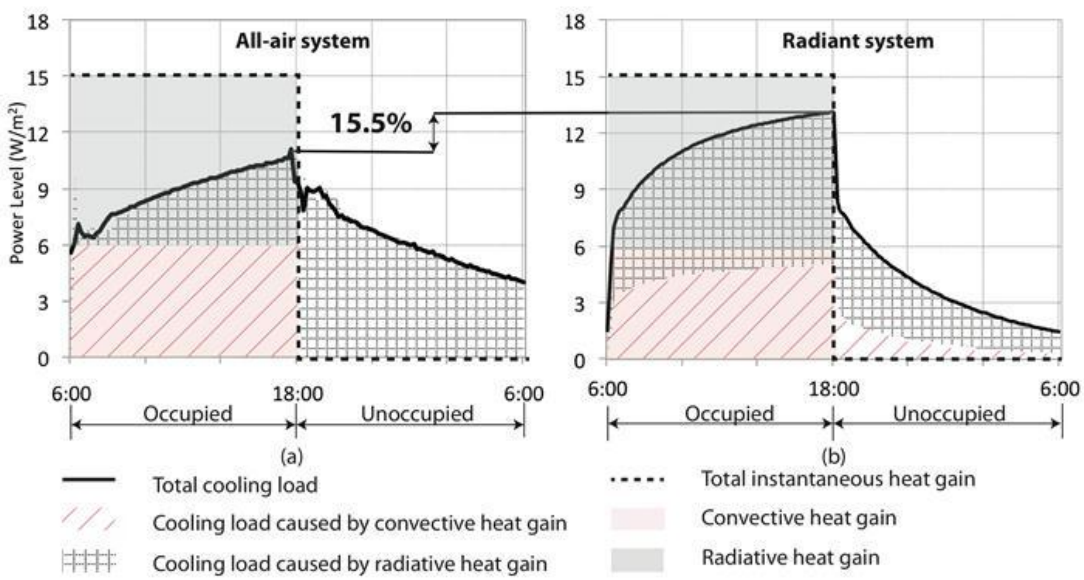


Figure 1.1 Comparison of cooling load and heat transfer breakdown for a simulated case: (a) all-air system and (b) radiant system (surface cooling) (Feng et al. 2012)

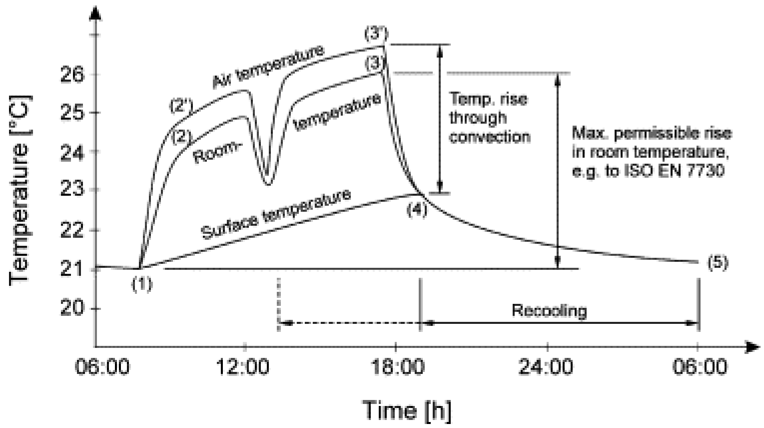


Figure 1.2 Typical diurnal temperature cycle for TABS application (Lehmann, Dorer, Koschenz 2007)

Comfort surveys in buildings with radiant systems

While a significant fraction of the commercial floor area in Germany approximately amounting to 49,000 m^2 of floor area have some form of radiant cooling, very few buildings in the US are equipped with radiant cooling (Kalz 2009). However, while the energy efficient performance of TABS makes it a potential candidate for future NZEBs, limited cooling capacity of TABS as compared to conventional VAV systems can pose a challenge in terms of thermal comfort, especially in hot climates. Thermal comfort surveys show that buildings equipped with radiant systems, in addition to natural ventilation and other mixed mode strategies can provide comfort in the moderate US climate. This result was obtained from analysis of CBE thermal comfort surveys conducted in 7 mixed mode buildings constructed post 2003. They are described in Table A.1 of Appendix A. The buildings are all located in moderate climate having ASHRAE design day summer temperature range of 27.7°C to 33.6°C and corresponding R.H. from 40% to 20.8%.

The results of the survey are displayed in Figure 1.3. I analyzed these results as part of my dissertation. The occupants' general satisfaction was found to be high in mixed mode buildings with positive response in thermal comfort category, similar to findings reported by Brager and Baker (2009). The results of the survey reflect an apparent positive potential of application of TABS in certain climates in the US. However, presence of mixed mode systems in addition to radiant cooling and possibility of correlation between satisfaction levels with building features subjects these results to further scrutiny outside the scope of this work. Therefore, the results of this survey have been included in the introduction chapter only for supporting my dissertation and not discussed in detail further. Most of these projects were initiated by innovation minded design engineers, whom Hopfe et al. (2009) would classify as early adopters. Another key aspect of these projects was an integrated design process where the architects and the mechanical engineers received significant support from environmentally concerned clients.

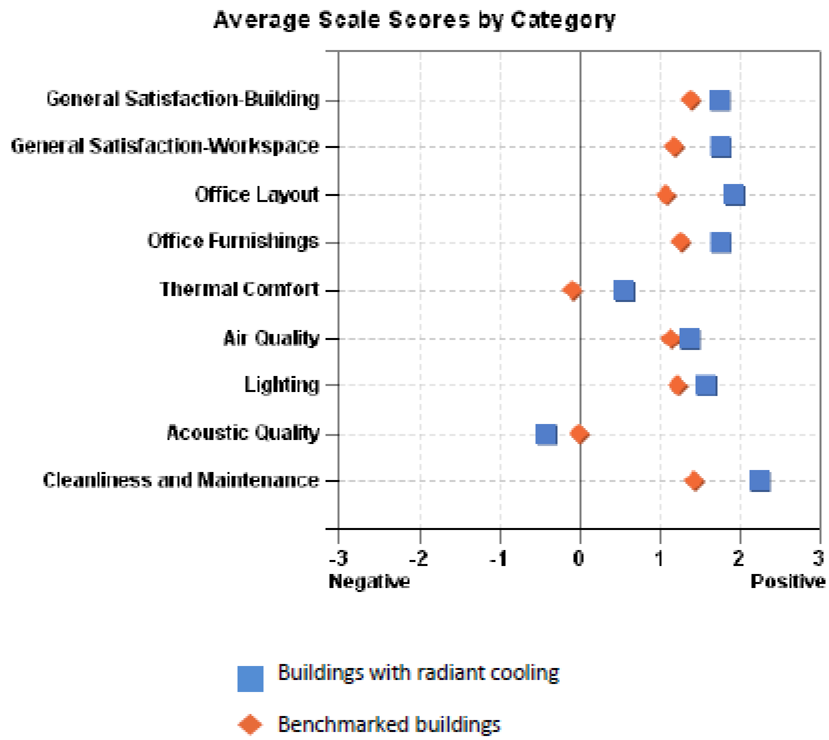


Figure 1.3 Comparison of general comfort performance of 7 buildings equipped with radiant cooling and mixed mode cooling with those in CBE database constructed post 2004 in the US. For more details on the CBE survey methods please refer to (Zagreus et al. 2004)

1.3 Objectives

The objective of this work was to develop a framework for classifying and characterizing, both qualitatively and quantitatively, the design and control models of TABS found in the literature, as a step towards development of future recommendations for sizing TABS in different design scenarios. The quantitative assessment is conducted in terms of comfort performance of the design models.

Most of the TABS design literature including design standard ISO 11855 (ISO 2012) provides recommendations for sizing of water supply temperature alone since this parameter determines the choice and size of the condensing unit (e.g. cooling tower), and sizing of water circulation pump. The cooling generation source will be designed to supply the lowest required water supply temperature for a given period of time to meet thermal comfort requirement on a design day. Therefore, in this study models pertaining to sizing water supply temperature of TABS were only investigated. Other design parameters like pipe spacing, water flow rate, pipe position in the slab were considered more implicitly as part of slab thermal resistance, but no effort was made to actively size them for system performance. The different design scenarios that would require different sized TABS were selected from literature review.

I attempted to answer the following research questions through this study,

1. How design and control models affect TABS comfort performance for the peak summer design day?
2. Which of the studied TABS design model gives the most consistent comfort performance under different design scenarios?

Design scenarios in this case refer to outdoor weather parameters and building construction features and thermal characteristics that affect the cooling load of TABS. Assuming that there is no complementary cooling system, the purpose of sizing of TABS would be to stipulate system parameters necessary to allocate enough cooling capacity for meeting the design (peak) cooling load, and hence preserve comfort temperature in the conditioned space. Consistency of comfort performance, therefore, means that the same design and simulation model, by the virtue of its inputs and adaptive parameters, should be robust enough to provide equal comfort under design scenarios that affect the TABS cooling load differently. TABS cooling load in this case refers to 24 hours room cooling load, which may be different from the system cooling load. The parameters of the empirically derived models should adapt to the changing boundary conditions. Literature review suggested that cooling load or heating load (waterside) of TABS is higher (in some cases as high as 33%) than that of convective systems, the difference being attributed to difference in heat transfer mechanisms between the two systems. However it is not clear what fraction of this difference is propagated to design water supply temperature or pump running hours or both and its resultant influence on comfort or energy. Answer to the first question is expected to partially address this issue. The second question is seeking to find which model should be recommended such that one model fits most design scenarios, in other words which of the studied models is the most robust one. This research addresses design scenarios where TABS is the main cooling system in space.

1.4 Approach

The research approach included both qualitative and quantitative analysis of TABS design models from literature.

Qualitative analysis involved carefully classifying the models by their design inputs, model type (like physically based or regression model), model purpose, model assumptions, implementation method and methods of validation or examples of application.

In quantitative evaluation I performed building energy simulations on a set of shortlisted models, using consistency of comfort performance under different scenarios as the evaluation metric. The underlying assumption here is that TABS water supply temperature should be so sized that it is able to maintain similar indoor comfort temperature irrespective of difference in building and weather characteristics that might have influenced the cooling load of TABS. The building energy simulations were performed in EnergyPlus v.7.0. Comfort performance is measured by a new comfort exceedance metric that accounts for both duration and severity of discomfort. Full-factorial design, defined later in the research methods chapter, is deployed for designing simulation cases. Since water supply temperature is the only varying parameter in these simulations for a given design scenario, a simulation based comparison is expected to generate reasonable results based on which a model can be chosen for recommendation.

Despite the differences in the heat exchange mechanisms between TABS and overhead (OH) convective systems, the design cooling load is calculated according to EN 15255 (BSI 2007) as the ‘basic room cooling load’ under constant comfort conditions for a convective system of unlimited maximum cooling power. In order to answer the two questions, I proposed a new comfort exceedance metric that accounts for both severity and duration of discomfort resulting from inadequate design. Statistical significance of the difference between the comfort performances of these models was tested using non-parametric method. The threshold for acceptable comfort exceedance was chosen to be 3-5% (DIN EN 2007-08; EN 2007). A model was expected to be robust if it could maintain the daily discomfort within this threshold for the maximum number of the design scenarios simulated.

Control of TABS is inseparable from its design. Sizing of auxiliary equipment (like pumps), selection and sizing of cooling sources and their associated energy, and cost implications are contingent upon the lowest water supply temperature and the corresponding maximum operating hours (including start time and end time) required for maintaining comfort in a given design scenario. However, water supply temperature and pump operation hours are the only two flexible design parameters that can be controlled in operation phase as well. Some of the investigated TABS models are control models, while others are hybrid (design and control) models. Therefore design day simulation results would reflect the operational efficiency of the system in addition to sizes of plant equipment.

The energy implications are best understood if the entire operation phase is simulated. Since in building energy simulation typical representative weather data is used, simulating seasonal performance can capture an average energy performance of an HVAC system ignoring system degradation. However, in this study energy equipment are not simulated directly, as my intention in this work was only to compare the comfort performances of the selected design models.

Very often the design water-supply-temperature predicted by the models studied may not be practically achievable because of zone dew point considerations. This could hinder selecting models based solely on comfort performance analysis, since the temperature actually used in the field may not be the ones recommended by the models. In this analysis, the zone dew point temperature is carefully controlled using unitary cooling system with reheat for enhanced dehumidification.

1.5 Scope and Limitation

Simulation based evaluation has its pluses and minuses. The advantage of a building energy simulation based evaluation framework is its scalability to other design scenarios, ease of replication and convenience of performing controlled experiments. Building energy simulations are good for performance comparison purpose, but not for model validation unless a simulation model calibrated to a real case is used. Even then too, extrapolation beyond calibration conditions may not be feasible. The minus point is that simulation models and methods include approximations and have their underlying assumptions, some of which are known and others are unknown. This is why simulation models never match reality and need calibration (Maile et al. 2010), (Augenbroe 2002)

This thesis does not aim to validate the shortlisted TABS control and design models individually under different design scenarios. The models investigated as part of this thesis include physically based (or analytical models), regression models and hybrid models. None of these models are complete, but some are more accurate than others in terms of input parameters and model equations and in their ability to capture transient behaviour of the systems. Comparison with first principle or analytical models can help in validating the regression-based models. Therefore first principle models need not be validated. Regression based models or hybrid models should be validated either by comparison with analytical relations, or by practical implementation in controlled environment, or simulation using calibrated model of any space with TABS, as these models have been derived for a given set of boundary conditions which cannot be extrapolated to other conditions. Such validation is essential in the final design stage when the construction and geometry of the building is frozen. Since the main purpose of this work is comparison of model performances using comfort as the metric, such validation may not even be necessary. However, some models may be meant for a given set of boundary conditions that are grossly different from the scenario under consideration. In such cases preliminary investigation of the models, using just spreadsheet calculation may be sufficient to disclose any infeasibility of application. Similar situations have been encountered in this study that required modification of coefficients/inputs of one of the models tested and will be reported later in this thesis.

The hybrid models or rule-based models included in this study have been derived mostly from simulations and iterations for a different set of climates and boundary conditions. Therefore, the most consistent comfort performance of one such model under the design scenarios studied does not guarantee it to be the most robust. Its good performance could be attributed to the explicit or implicit equation parameters or its coefficients. We could still assume the higher the number of inputs (especially in hybrid models), the higher is its accuracy, that could be further reinforced by adapting the model coefficients to a wider range of design scenarios covering more climate types, building geometry and construction peculiarities, heat sink type and size, beyond the scenarios for which the models were derived.

Chapter 2

2 Literature review

2.1 Introduction

I reviewed seven design and control models/methods of TABS and characterized them systematically for assessing their applicability in various design scenarios and at different design stages. The term method will be used for the seventh design model reviewed, since this method refers to a sensitivity study based graphical design method, recommended for design scenarios not covered by design standards, rather than any specific model equation. The design scenarios will be discussed later in this chapter. The design stages include: i) feasibility study, ii) early design decisions, and iii) detailed design sizing. The applicability in different design stages is evaluated based on ease of implementation of the design methods. TABS have different peak cooling load, 24-hour total cooling energy and hydronic cooling load than all air systems, for a detailed description please look at (Feng et al. 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 (Gwerder et al. 2008; Gwerder et al. 2009; Lehmann et al. 2007) 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 chapter is to qualitatively identify and evaluate design and control models of TABS for further simulation based evaluation and final recommendation for use in design.

2.2 Method

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, I also screened several conference papers. I found one international standard, one guidebook and nine peer-reviewed papers, 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 2.2.

Most of the TABS design literature including design standard (ISO 2012) provides recommendations for sizing of water supply temperature alone since this parameter determines the choice and size of the condensing unit (e.g. cooling tower). Hence in this study, too, I only investigated the models of water supply temperature of TABS. 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 desirable temperature and pressure drop and turbulent flow (Reynolds number) in the hydronic circuit. While design and control of water supply temperature has been extensively studied, slab thermal mass has never been considered as an active design parameter.

2.3 Model classification criteria

In this section I will first describe the criteria for classifying the models and then assess the models in light of those criteria. Table 2.1 Model equations for water supply temperature displays the design and the control equations that have been selected for further comparison and classification presented in Table 2.2. This table has seven headers, among which “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 the heading “design approach”, the design and control models of TABS, with their underlying assumptions and implementation methods, are presented under three subheadings:

1. *Model* - rule based or physically based, like numerical and analytical models, etc.
2. *Model purpose* – design, control, or a combined design and control approach.
3. *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
4. *Implementation methods* – non-iterative, iterative, or using transient simulation.

Henceforth, in this thesis, the models will be referred to by their respective numbers in Table 2.2.

Table 2.1 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	$T_{ws} = T_s - \frac{Q}{h}.1000(\tilde{R} + R_t)$	
4	$T_s = T_{rsp} + coeff.Q$	(ISO 2012)
5	$T_{wspm} = T_{ws} - [\frac{R_t}{\tilde{R} + R_t}(T_{rsp} - T_{ws})](\frac{\Delta t_c}{\Delta t_{c1}} - 1)$	(Gwerder and others 2009)
6	$T_{ws} = T_{rsp} + \frac{R_t + \hat{R}}{R_f}(T_{oa} - T_{rsp}) + (R_t + \hat{R})q_{ub}$	(Gwerder and others 2008)

$coeff$	Coefficient of linear equation for calculating the design active surface temperature as function of 24 hours accumulated cooling energy demand. This coefficient is different for different zone orientation, precooling and continuous pump operation and internal load profile, *unitless*
h	Number of hours of pump operation
Q	Specific daily energy gains in room during design day, consists in the sum of heat gain over 24 hours period divided by room area ISO 11855-4, kWh/m^2
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
$\tilde{R}$	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
$\Delta t_{c1}/\Delta t_c$	Total pump running hours as percentage of cooling hours, *unitless*.
T_{oa}	24 hours running mean outdoor dry bulb temperature, °C
T_{op}	Zone operative temperature, °C
T_{rsp}	Room operative set point temperature, °C
T_s	Active surface temperature of TABS
T_{ws}	Water supply temperature for 24 h operation, °C
T_{wspm}	Water supply temperature for precooling, °C

The rules in rule-based models are either derived empirically or from simulations. 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. *Model assumptions* may affect the complexity of the model and its applicability in a certain design phase. In addition to model accuracy, ease of implementation can go a long way to promote early adoption of a design model or method by the design community. 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 designer's experience can be enhanced by developing an intuitive design user interface. The iterative methods can range from sensitivity analysis 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 iteratively in a fixed design scenario to measure the effect of the change on the desired thermal comfort performance. This is equivalent to an empirical model of a design parameter as a function of one or many environmental and building design 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 with constraints until the design objective is met within a desired error or tolerance range. One major limitation in applying optimization in the 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. Reliability based optimization is widely used in machine design, product design, quality control and systems engineering, but is still at a nascent stage for building systems (Chen et al. 2007; Hopfe2009).

Table 2.2 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*	*Implementa-tion 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 or	Ceiling/ ceiling+ floor	Similar system configuratio n as reported	Hybrid (combination of linear regression and physically	Design	Linear relation between 24-hour cooling load and radiant surface	Iterative	Pump operation, slab and tubing	Presumably based on paper no. 6. No validation	VIG *using empirical relation.*

	ISO 11855	both	by 6	based steady state model)		temperature; coefficient derived for different zones, internal load profile and pump operation hours, by dynamic simulation. The slab temperature constant 24 hours.		resistance, room temperature setpoint, number of thermally active surfaces, zone orientation, internal load profile	reported in the standard	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)	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 *same as above.* VPO TM Requires further validation

6	(Gwerder et al. 2008)	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 *using steady state formulation of limiting internal heat gain.* No VPO TM Complex and not practically feasible without an existing mathematical model.
7	(Lehmann et al. 2007)	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 *by empirical relations.* VPO TM Detailed design for non-standard zones

The design/control parameters column provides the list of design and control inputs for the models. The sixth column, validation procedure, enumerates the number of cases simulated or tested 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 sizing and control models could take in to account varying internal gains (VIG), varying pump operation hours (VPO) and zone thermal mass (TM). VPO refers to any form of pulse width modulation (PWM) pump operation. In this study I considered precooling as a special case of VPO with 24 hours period of PWM. 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 selected models from the literature are classified by VIG, VPO and TM, 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.

2.4 Model classification

Out of seven models/methods, three are control models, two design models/methods 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. Note that the authors did not provide any specific definition of the zone operative temperature i.e. whether it is instantaneous or peak or average of a certain time window. In simulation based study that I have conducted later I have used average zone operative temperature of 24 hours. Equations 3 and 4 constitute the 'simplified sizing by diagrams' method in 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. While, these results are qualitatively intuitive from knowledge of building physics, ISO 11855 has no mention of the boundary conditions under which the coefficients of these equations were derived. Moreover, the total cooling energy for the equations is calculated as "the calculation of the heat gains has to be carried out by means of the value of the total cooling energy to be provided in the day in order to ensure comfort conditions at the average operative temperature", without clarifying if this the room side cooling energy or system side cooling energy. The good information is the associated error range of 15-20% of these models. Equation 5 is a simplified 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. 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 from laboratory tests. The zone set point temperature and heat gain are assumed to be constant for the entire pump operation cycle.

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. By far model 6 is the most sophisticated model of TABS intended for both design and control. This model is also closest to reliability based design, in that it accounts for extreme cases of space heat gain, thereby covering the entire possible range of heat gain in spaces supplied by a single cooling source, especially in face of uncertainty in heat gain data. 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 sensitivity study of WST and 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 this study. 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 shown in figure 11 in (ISO 2012), also presented in this section as Figure 2.1. All of the design methods except model 1 require dynamic simulations and hence must be iteratively implemented. Depending on the type of the building model and simulation used, model 2 may or may not account for the presence of thermal mass.

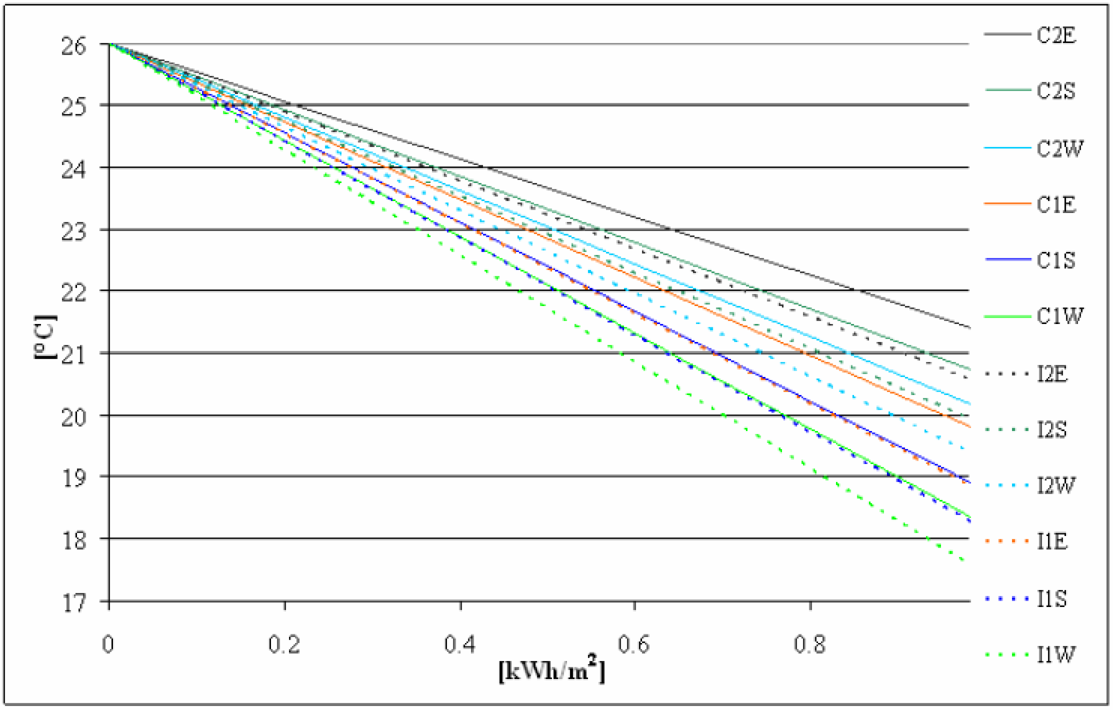


Figure 2.1 Linear diagram of active surface temperature as function of 24 hours sum of heat gain (Q), coeff and zone maximum allowable operative temperature, C=continuous pump operation, I =intermittent pump operation, E=East zone, W=West zone, S=South zone, 1 =continuous internal heat gain, 2=internal heat gain with 2 hours recess from 12:00 noon to 2:00 p.m.

Babiak (2007) and Kolarik et al. (Babiak 2007; Kolarik and others 2011) reported that thermal mass of the building can have considerable influence on daily comfort performance of TABS. On the other hand Feng et al. did not find any significant difference between cooling load of TABS in lightweight and heavyweight building construction. 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.

The purpose of this qualitative classification was to shortlist TABS design and control models for further dynamic simulation based evaluation of the consistency of comfort performance of the models under different design scenarios. In this paragraph the design scenarios will be carefully derived from model inputs and building characteristics, that are expected to influence the cooling load of TABS. For instance, solar load increases the radiant component of the cooling load, in addition to direct absorption by the active surface, resulting in higher peak cooling load of TABS than that of all-air system. Effect of solar load on TABS sizing can be assessed either by varying fenestration size or changing zone orientation. The latter approach is adopted in this study. In addition to solar load, effect of varying internal loads on comfort performance of TABS models is tested. On the other hand, thermal mass of the construction has negligible effect on cooling energy requirement of TABS (Feng et al. 2012). But thermal mass of the construction influences the cooling load profile and hence hourly comfort temperature distribution. Thus, varying construction thermal mass is considered as another treatment. Heat transfer analysis of a room equipped with TABS using EnergyPlus simulations, performed by Feng et al. showed that conduction heat gain is the most important component of TABS cooling load that distinguishes it from that of all-air system. The simulated case in the above study is rather unrealistic with four walls exposed to the exterior. This could have exaggerated the results. Therefore envelope resistance is not varied in this study. From energy conservation standpoint the total energy absorbed by the cold slab surface of 24 hours must be removed by cold water in TABS. The energy removed is a function of both WST and the corresponding operating hours. The different pump operation hours are not included in this study, as the models accounting for PWM could be compared independently from those of continuous operation using the same framework later.

2.5 Model shortlisting

The primary criterion for shortlisting of TABS design and control models, in this study, is the balance between ease of implementation and model accuracy. Ease of implementation really refers to usability of any service. In software industries usability assessment is an internal process where representative users are asked to test the usefulness of software from time to time throughout its design phase and compare it with alternative means of achieving the same task or competing software (Design@ IBM 2012). If designing software for aiding HVAC designers in TABS design is assumed to be the final goal of a project, then the current study would fall within the first phase of usability assessment, i.e. identifying a suitable design model. While, such usability analysis involving real users is outside the scope of this project, it is rather intuitive that the number of design inputs, the time taken to assess performance of a design alternative including interpreting software result and the transparency of analysis are some of the factors that can affect the ease of implementation of a model. I chose those models for simulation study that accounted for the majority of factors affecting TABS cooling load, while having lesser input fields than the most sophisticated model and the results of which were easy to interpret. The idea is that if a certain design model or method can give fast and accurate results, comparable with those from a complex model with several inputs, then why not use the simple model.

I grouped the sub-headings discussed in Table 2.2 into three broad conceptual and more tractable categories like reliability, design inputs/ number of inputs and ease of implementation. I selected these broad categories such that if I ask engineers or designers they would be able to easily assign a weight to each of these categories according to their priority. I observed that some components of the design approach (the type of model and model assumptions), the validation procedure, and also partly, the design inputs (classified as design and control parameters in Table 2.2), constitute how reliable a model will be. The number of design inputs could also mean a balance between robustness and accuracy on one hand, and information required and availability on the other. For example the average value or peak value of a boundary condition, for instance, outdoor DBT, may be more easily available (and even less uncertain) than hourly outdoor DBT. Factors such as having a prerequisite model for design calculation, use of complex building energy simulation, iterative approach or simple spreadsheet calculations determine how easily a design method can be implemented. Following are the results of this classification:

Model 6 (Gwerder et al. 2008): Most reliable model, accounts for uncertainty: design for entire range of cooling load in a building, complex implementation

Model 3 (ISO 11855-4): More reliable than rule based model, coefficients must be validated for complex building geometry

Model 2 (Olesen and Dossi 2004): Low reliability, better address design conditions with room temperature feedback, good initial model for optimization based design

Model 1 (Olesen and Dossi 2004): Easiest implementation, least reliable

2.6 Summary

In this chapter I reviewed several design and control models of TABS from the literature and classified seven of them. 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. ISO 11855 “Design by diagram” method uses a mix of physically based and correlation models. From this qualitative evaluation I could draw the following inferences,

Model 1 and model 2 (Olesen and Dossi 2004) are suitable for feasibility study.
Model 3 and 4 (ISO 11855) is ideal for:

- Early design decision with quasi-steady state cooling load
- Detailed design phase
- Precooling scenario

Model 6 (Gwerder et al. 2008) can be used for:

- Single water supply temperature for different zones in a building with different cooling demand
- Detailed design phase

Model 5 (Gwerder et al. 2009) is specifically suitable for precooling or pulse width modulation scenario, but in general recommended for all design stages.

Further, 24 hours cooling load of TABS must be calculated using transient simulation for detailed design, only if the difference between the 24 hours room cooling energy and the former propagates down in comfort performance well. However, this qualitative method used in this chapter is not enough for making concrete design recommendations. It is primarily meant for screening of models and methods for further simulation based evaluation; design, methods and results of which will be reported in the next few chapters.

Chapter 3

3 Research methods

3.1 Introduction

After a systematic classification and comparison of seven of the design models/methods from literature I shortlisted three of them for further validation under varying internal gains, zone orientation and building constructions using EnergyPlus v7.0. These parameters were selected for two reasons, firstly they were expected to have considerable influence on the cooling load of any HVAC system, and secondly these parameters vary from building to building. A simulation of prototypical small commercial office buildings stock in the US performed by Huang and Broderick (2000) found that 40% of the cooling load is attributed to internal gains and 42% to the solar gains. Of these the building constructions are responsible for differences in the occurrence of the peak cooling load, and distribution of the cooling load over the larger time span. Other parameters affecting cooling load and thermal performance of TABS that were held constant in this study included window-to-wall area ratio, zone volume, construction insulation, supply air temperature and volume flow rate of the ventilation system, external shading, location, TABS mass flow rate, pipe layout, active surface configuration and TABS thermal properties.These are equations 1-3 and 4 in Table 2.1 of Chapter 2. The final goal of this simulation based study is to recommend a new TABS design method that balances between the accuracy of complex hybrid and/or transient design models and ease of implementation of single disturbance empirical models, catering to most of the design scenarios. The method of shortlisting of the models for simulation and identification of design scenarios have been discussed in Chapter 2. The results of continuous (24 hours) pump operation scenarios will only be reported in this thesis, leaving the PWM cases for future work.

3.2 Research method

The purpose of this simulation based study is to test how the shortlisted models differ from each other in terms of comfort performance, and which of the models gives the most consistent comfort performance under different design scenarios. I constructed a full factorial experiment with 3 factors, viz. varying internal heat gain (VIG), varying thermal mass (TM) and varying zone orientation, the first two factors have two levels each in statistics jargon and the last factor has five levels. The derivation of these design scenarios has been described in Chapter 2. Effect of WST and associated operating hours on zone thermal comfort is tested using building energy simulation tool EnergyPlus.

The potential difference in cooling load of TABS and convective systems has triggered this new research interest, the intent being to establish the correct cooling load for TABS design. This difference can be best understood by simulating the transient behavior of TABS and its thermal interaction with the conditioned space. Such an approach is more time consuming and resource intensive than a simple spreadsheet calculation. Therefore, investing effort in doing this elaborate calculation during the early design phase will make sense only if this difference in the cooling loads makes a significant difference in the TABS design parameters necessary to provide the desirable thermal comfort. WST is just one such design parameter of TABS. Therefore, in addition to testing the comfort performance of the as-recommended design models of WST, I also performed a sensitivity analysis varying the 24 hours cooling energy of TABS for model 3, the objective function being comfort performance of the TABS design model.

Full-factorial design

The design scenarios forming the simulation cases were formulated into a full factorial design. In statistics, a full factorial experiment is an experiment whose design consists of two or more factors, each with discrete possible values or "levels", and whose experimental units take on all possible combinations of these levels across all such factors. A full factorial design may also be called a fully crossed design. Such an experiment allows studying the effect of each factor on the response variable, as well as the effects of interactions between factors on the response variable. A factorial experiment can be analyzed using box plots and parametric or non-parametric Analysis of Variance (ANOVA). The distribution of TABS 24 hours cooling energy (CE), WST and daily comfort exceedance as function of different design scenarios will be represented as box plots in this study. The box plots are implemented using Python library. A box-plot is a way of graphically summarizing a data distribution. In a boxplot the thick horizontal line in the box shows the median. The bottom and top of the box show the 25th and 75th percentiles, respectively. The horizontal line joined to the box by the dashed line shows either the maximum or 1.5 times the interquartile range of the data, whichever is smaller. Points beyond those lines may be considered as outliers and they are plotted as circles in the boxplot graphs. The interquartile range is the difference between the 25th and 75th percentiles. Statistical significance of the differences between model to model thermal comfort performances are reported using non-parametric statistical significance test, Kruskal Wallis ANOVA, performed using SPSS.

Performance metric

Assuming that there is no complementary cooling system, the purpose of sizing of TABS would be to stipulate system parameters necessary to allocate enough cooling capacity for meeting the design cooling load, usually the peak cooling load and hence preserve comfort temperature in the conditioned space. Comfort was therefore chosen as the performance criterion over energy, for comparing of simulation results of the design models. Comfort will be measured as comfort exceedance.

Comfort exceedance is a metric that is evolving these days and attracting investigation due to several factors namely, how to accommodate uncertainty of on-site renewable energy supply (assuming sufficient energy storage is not provided) and of utility conducted demand response, and finally application of low energy or energy efficient HVAC systems that have limited capacity and complex and unpredictable cooling performance. A few examples of these types of HVAC systems or rather methods are natural ventilation, hybrid ventilation and radiant cooling systems with high thermal mass like TABS. It is the thermal mass of TABS interacting with the water-side on one hand and space thermal mass of the other hand that makes exact performance prediction of these systems difficult. Ascertaining and adopting a more concrete and pragmatic exceedance policy can therefore minimize the uncertainties associated with adoption of the above systems and corresponding occupants' thermal comfort expectations. Further this metric could be used to select complementary or alternative HVAC systems for mixed-mode buildings.

Comfort exceedance metric is a measure of percentage of occupied time that the operative temperature inside a building falls outside the expected comfort range. ASHRAE 55 (2010) does not recommend any metric for comfort exceedance evaluation. But European Standard EN 15251:2007 (EN 2007) recommends three different metrics: percentage of occupied hours outside the range, degree-hours criteria and PPD weighted criteria. . Percentage outside range refers to per cent of occupied hours (hours during which the building is occupied) when the PMV or the operative temperature is outside a specified range, degree-hours criteria is the time during which the operative temperature exceeds the specified comfort range during occupied hours weighted by some function of the number of degrees beyond the range and PPD weighted criteria suggests the accumulated time indoor temperatures are outside the expected comfort range weighted by some function of PPD. While the first of the metrics does not account for intensity or severity of excursion, the second and last metrics are excursion durations weighted by factors that account for severity of deviation from comfort zone. In practice the recommended comfort exceedance is 3-5% total, which can only be applied to the first metric. Borgeson and Brager (2011) found that the difference between exceedance performance of different comfort models itself is higher than the recommended acceptable excursion. The authors also came up with a new exceedance metric defined as:

$$Exceedance_M = \sum_{i=0}^{all\ hours} \begin{cases} n_i \ if\ PPD_M > 20 \\ 0\ if\ PPD_M \leq 20 \end{cases} / \sum_{i=0}^{all\ hours} n_i \qquad (7)$$

Where n_i is the number of occupants in a given hour i, M is the comfort model, discomfort is the percentage occupant dissatisfied according to model. Its unit is percentage of occupant-hours. One advantage of this metric is that it can be expressed in percentage and hence compared with the current recommendation of 3-5%. Secondly it also accommodates the uncertainties in comfort models by weighing any level of discomfort beyond 20% equally. Thus with this model it does not matter how many people are really dissatisfied beyond 20% PPD. This also means a higher sensitivity to values that just happen to be on one side or other of the 20% threshold. Such a unit could be less tangible than PPD weighted hours or degree-hours criteria, which accounts for severity as well as duration. Degree-hours and PPD weighted criteria, however, can overestimate the severity of exceedance, as they are not weighted by current occupancy. Therefore, in this study a new comfort exceedance metric was introduced that accounts for duration as well as severity of discomfort, the severity being weighted by instantaneous occupancy. The operative temperature range used in this study is 23.7-26.7°C. This corresponds to comfort class B of ASHRAE 55-2010, i.e. -0.5<PMV<0.5±0.5 and PPD=10% for a met value of 1.2, a clo value of 0.5, relative humidity 50%, an air speed of 0.15 m/s (Schiavon and Hyot 2012). The new comfort exceedance metric will be called "Occupancy-weighted PPD hours" and computed as per *Equation 8.*

$$Exc = \sum_{i=0}^{all\ hours} \begin{cases} n_i \times PPD_i \ if \ PPD_i > 10 \\ 0 \ if \ PPD_i \leq 10 \end{cases} / \sum_{i=0}^{all\ hours} n_i \times 100\% \qquad (8)$$

Where n_i and PPD_i are the number of occupants and the corresponding PPD>10 in the i^{th} hour of a day. Hourly discomfort distribution addresses the differences in pump operation hours and difference in zone orientations, while cumulative discomfort can be used more for technical feasibility assessment and identifying alternate strategies. Note a constant clo and met value are assumed and zone relative humidity is maintained to ensure that the difference in comfort performance depends only on zone operative temperature (a function of cooling provided by TABS). But obviously it is not free from the inherent uncertainty in the PMV-PPD model, even though it addresses the severity of discomfort better than just *PPD-weighted hours* or just *occupancy weighted hours* with high PPD.

Energy Simulation software

EnergyPlus v7.0 was used for the simulation study because it performs a fundamental heat balance on all surfaces in the space and has been extensively validated (US Department of Energy and USA.gov). Also since it is able to integrate the heat transfer calculation in the water-based embedded cooling systems with the changing space conditions, it is able to capture the transient behaviour of the systems. Furthermore, the energy management module of this software allows for easy implementation model equations using virtual objects like "sensor", "actuator", and the actual control "program". Windowing on the time series, for instance, calculating a moving average over different time blocks can be performed conveniently using the "trendvariable" that tracks any variable over the length of the averaging window.

Building and climate

A five zone (four cardinal directions and a core zone) building model is used for testing the TABS WST design models. This model was chosen as it is a standard five zone model of a small single storey office building in EnergyPlus, close to the reference size of a small office building in US Department of Energy Commercial Reference Building Models of the National Building database (US DOE 2012). This would ensure ease of replication of results elsewhere without having to model the building geometry again. It also facilitated the investigation of the different zone orientation on design of WST of TABS. Each zone in this building is 2.5 m high, with a floor area given in

Table 3.2. This height was selected keeping scalability in mind for future comparison of HVAC systems that require plenums. This, however, may affect the comfort performance result for individual cases, which is not important in the current study. The orientations accounted for differences in solar gain, one of the major influencing factors for zone cooling load. The constructions are similar to descriptions in ASHRAE Handbook of Fundamentals for lightweight and heavyweight treatments. Both construction types have similar U values. I investigated the California climate zones CZ03, CZ04, CZ05, CZ06, CZ07 and CZ08 as potential climates for the simulations. The climates were shortlisted by the fraction of the temperate hours, the relative fraction of too hot and too humid hours, the summer design day temperature and the coincident dew point temperature (descriptions of 16 climate zones of California, ASHRAE weather data). The classification of moderate, too hot and cold is based on an outdoor DBT value of 30°C, determined through iteration. Based on these characteristics CZ03 was chosen for simulations.

As all the zones have similar window-to-wall ratio and near similar exposed wall/floor area, the effect of orientation on cooling demand and TABS model performance can be studied with this building geometry. Figure 3.1 shows the building model simulated.

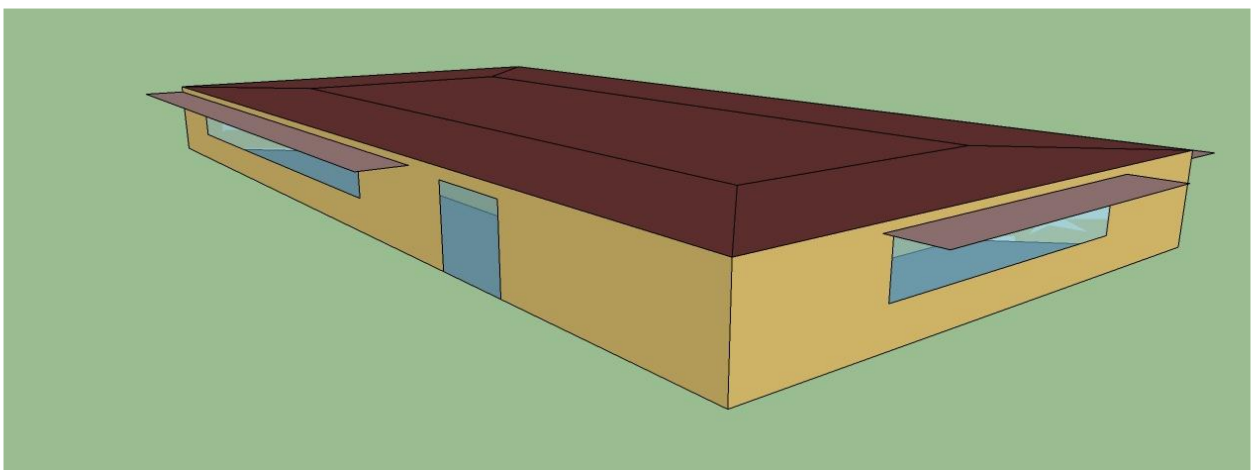


Figure 3.1 Sketchup model of the simulated office building

Table 3.1 Climate analysis for simulation

CA Climate zone	0.4% cooling design day max DBT	WBT at max DBT	Daily DBT range	Fraction of annual moderate hours	Fraction of too hot hours	Fraction of too humid hours
03	27.7	18.3	7.4	0.37	0.03	0.23
04	33.5	19.4	12.1	0.4	0.2	0.2
05	29	17.1	10.2	0.55	0.03	0.2
06	32.9	20	9.5	0.5	0.04	0.3
07	28.9	19.9	4.9	0.6	0.75	0.26
08	33.4	20.1	10.4	0.55	0.16	0.16

Table 3.2 Simulation boundary conditions

Building description	Single story, 4 perimeter zones and one interior zone, intermediate floor level				
Climate	San Francisco, California Climate zone 3				
Zone data	**S**	**E**	**N**	**W**	**C**
Area (m^2)	99.2	42.7	99.2	42.7	182.5
Glazing fraction	0.23	0.25	0.23	0.25	0
Radiant slab data	**Pipe internal diameter (m)**	**Slab thickness (m)**	**Effective thermal conductivity (W/m-K)**	**Tube spacing (m)**	**Depth of pipe in slab (m)**
Ceiling slab with exposed concrete	0.013	0.25	0.8	0.2	0.125
Ventilation rate during occupied hours (m^3/s) for high occupant density (Standard 62.1-2010)	0.106	0.045	0.106	0.045	0.195
Ventilation rate during occupied hours (m^3/s) for low occupant density (Standard 62.1-2010)	0.04	0.017	0.04	0.017	0.073
Solar shading	External screen activated when the vertical irradiation on the window exceeds 80 W/m^2. The screen has a transmittance of 0.05. This is an approximation of real shading since only on/off state is considered.				
Occupant density (m^2/person) 1. low 2. high	 22.3 (Schiavon et al. 2011) 8.4				
Lighting density (W/m^2) 3. low 4. high	 6 13				
Equipment density (W/m^2) 5. low 6. high	 6 34				
Pump running hours	100 % (24 h availability, zone temperature setpoint based operation).				

Simulation cases

The simulation cases are identified as follows, thermal mass-internal gain, for example $HWHG_1$ would mean a case of heavy construction with high internal gain, where 1 indicates the model number and HW – Heavy construction, LW – Light construction, HG – High internal gain, LG – Low internal gain. Therefore a total of 12 cases were simulated for models 1, 2 and 3, as $HWHG_1$, $HWLG_1$, $LWHG_1$, $LWLG_1$, $HWHG_2$, $HWLG_2$, $LWHG_2$, $LWLG_2$, $HWHG_3$, $HWLG_3$, $LWHG_3$ and $LWLG_3$. Results will be analyzed for these combinations for each of the south, east, north, west and core zones. The three simulated construction types and associated material properties are given in Table 3.2 and 3.3 respectively. 3 mm clear glass double glazing with 13 mm air gap with U value 2.72 W/m^2-K is used. The values in the brackets in table 3.2 are U-value and thermal mass per unit surface area for each building component. The simulation cases automatically include different solar loads in different zone orientations due to building geometry. As for internal heat gains, the light power density, equipment power density and occupant density were informed by works on US and international surveys of office buildings, best practices adopted to achieve 30% energy efficiency compared on ASHRAE 90.1-2010 standard, other papers reporting full factorial designs, even though technologies in research phase are not included (Borgeson and Brager 2011; Dubois and Blomsterberg 2011; Dunn and Knight 2005; Ryckaert et al. 2010; Schiavon et al. 2010; Schiavon et al 2011). Readers are directed to Figure 3.2 Occupancy and lighting schedule typical weekday occupancy and lighting schedule.

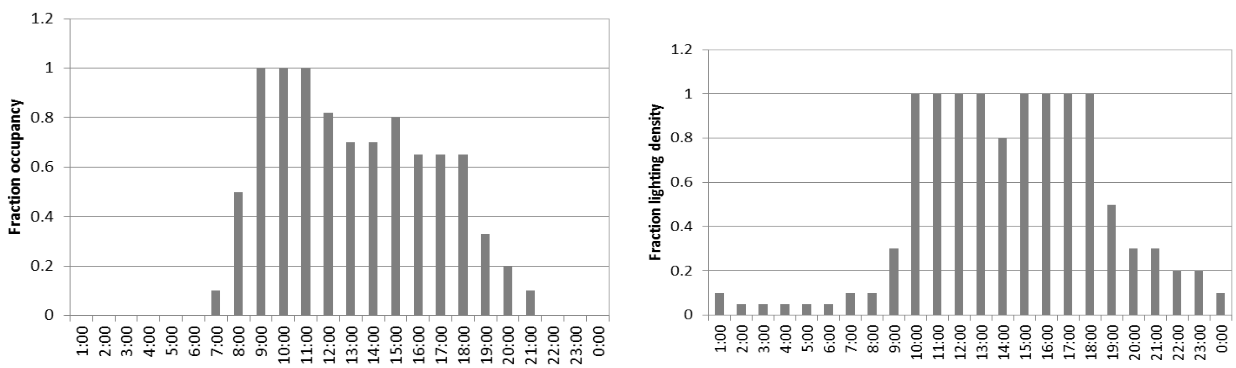


Figure 3.2 Occupancy and lighting schedule. Occupancy and lighting schedule are taken from European standard, EN 15232 (EN 2012).

Table 3.3 Construction materials (The values in the brackets in table 3.2 are the U-values and the thermal mass per unit surface area of each building component)

Construction type	External wall	Floor	Ceiling	Internal wall
HW	100 mm brick, 200 mm heavy weight concrete, 50 mm insulation board, 50 mm air space, 19 mm gypsum (0.5, 612.4)	200 mm heavy weight concrete, 100 mm heavy insulation (0.23, 464.2)	125 mm heavy weight concrete, tubing, 125 mm heavy weight concrete	19 mm plasterboard, 200 mm heavy weight concrete, 19 mm plasterboard (1.68, 490.32)
LW	0.8 mm steel siding, 50 mm insulation board, 50 mm air space, 19mm gypsum (0.51, 19.4)	100 mm lightweight concrete, 100 mm heavy insulation (0.23, 64.2)	125 mm heavy weight concrete, tubing, 125 mm heavy weight concrete	19 mm plasterboard, 50 mm air space, 19 mm plasterboard (1.58, 30.32)

HVAC system, solar shading and control

The HVAC system in each zone is a thermally active radiant slab (EnergyPlus name: LowTemperatureRadiant:VariableFlow). The radiant cooling system set point is controlled at 25°C operative zone with a deadband of 2K, i.e. the water supply volume flow rate varies linearly from 0 to maximum between 24° and 26°C. Note that this range is narrower than that of the comfort zone used in this study, because high thermal mass of TABS could lead to overshooting of zone operative temperature beyond the boundaries of the comfort zone. The main purpose of the air system was to maintain minimum required ventilation in space while not adding any heating or cooling to the space, and to control the zone dew point temperature to the point that all the models could be tested without condensation. The zone RH set point is 40% using a humidistat object in EnergyPlus. I found that that I could use two alternative HVAC systems for meeting the ventilation and the zone RH set point; the first method would be to create a dummy adiabatic zone and control supply air through that zone from a constant volume reheat system, that added no extra heating or cooling to the zone, and the second method is to use a zonal heating and cooling unit with enhanced dehumidification. I used the second option with a very narrow range of supply air temperature, using a combination of several EnergyPlus components, viz. AirloopHVAC:Unitarycoolonly, Coil:Cooling:DX: TwoStageWithHumidity ControlMode and a heating coil. The minimum and maximum supply air temperature of the reheat was controlled within zone comfort range used in this study. . Minimum required ventilation for acceptable air quality as per ASHRAE 62.1-2010 was maintained during occupied hours. The total air volume supplied required to be supplied for high and low occupancy are 0.497m^3/s and 0.2m^3/s respectively, according to the minimum ventilation requirement of 0.0085m^3/s in case of unknown occupancy (≥20m^2/person) as per Table 6-1 in ANSI/ASHRAE

Standard 62.1-2010. The ventilation rates were maintained slightly higher than the above values for maintaining zone humidity without adding cooling or heating. This study was performed without considering infiltration, since minimizing uncontrolled infiltration heat and moisture gain is a recommended design and construction goal for future buildings, for ensuring better indoor environment control. One implication of this assumption was that the majority of the latent heat gain was from occupants and therefore the maximum relative humidity occurred during the occupied period, even though controlled. This assumption could, however, be a limitation in the context of existing buildings with diverse levels of infiltration. The ultimate goal of this study is comparative evaluation of the models under consideration; hence as long as all the models are compared for similar design scenarios, the purpose of the investigation is served.

Overhang is implemented as a practical solution to glare, while external screen was activated when the vertical irradiation on the window exceeded 80 W/m^2 during the occupied hours. The screen has a total solar transmittance of 0.05 and operates as completely retracted or activated. In addition to this, all the windows have a 1.3 m deep overhang. Even though the existence of a screen attenuates the effect of different solar gains on different orientations, nonetheless it is a practical to have a solar screen for glare protection.

Control of TABS in practice does not use zone temperature set point. But, in EnergyPlus it is mandatory to use a zone temperature set point based control, at least to my knowledge.

Implementation of model equations

The model equations 1, 2 and 3 were implemented in the EnergyManagementSystem module of EnergyPlus.

Table 3.4 Parameters of model equations

Parameters of the model equations	
T_{op}	24 hours moving average zone operative temperature of previous day
T_{oa}	24 hours moving average of outdoor air dry bulb temperature
Δt_c	24 hours
T_{rsp}	26°C
$\tilde{R}$	0.04 m^2-K/W (calculated as per Appendix B.2 of ISO 11855-2)
R_t	0.03 m^2-K/W (calculated as per Appendix B.2 of ISO 11855-2)
$coeff - \frac{1000}{h}\left(\tilde{R} + R_t\right)$	-10.4 (continuous operation, south zone) - 8.3 (continuous operation, east zone) -11.3 (continuous operation, north zone) -11 (continuous operation, west zone) -8.3 (continuous operation, core zone)
Q	24 hours specific cooling energy as per ISO 11855, kWh/m^2.

The T_{op} in equation 2, has not been clearly defined by the authors, i.e. whether it is average of 24 hours or peak or instantaneous. Instantaneous zone temperature feedback would lead to rapid changes in the WST setpoint, which may not benefit due to thermal decoupling of waterside and room-side of TABS by its own thermal mass (Koschenz et al. 2007, Gwerder et al. 2008). Therefore other alternatives would be to use zone peak temperature of 24 hours or average temperature. I tested for both of the above. Peak zone temperature led to very low supply water temperature below 12°C and was found incompatible for this equation. Therefore I used 24 hours moving average zone temperature as T_{op}. The limitation of this method is that in climates with high diurnal range the resultant T_{op} may not reflect the high zone temperature during the occupied period. Q in ISO 11855 is calculated as “the calculation of the heat gains has to be carried out by means of the value of the total cooling energy to be provided in the day in order to ensure comfort conditions at the average operative temperature”, without clarifying if it is water-side or room side cooling energy. I used the 24 hours design cooling energy calculated from an all air system for model 3, as per ISO 11855. Coefficients derived for TABS energy could be different from those derived using convective systems. For example, for the south zone with open windows the coefficient can be expected to be higher than that of the current equation under consideration. Further the “radiant cooling energy” is a function of water supply temperature input and hence will be different from model to model. To derive this value by iteration, TABS parameters must be changed several times until the closest to the desired comfort performance is reached probably with a chosen tolerance. This process is cumbersome and not pragmatic. Another approach would be to use a percentage increase on standard cooling energy, similar to (Schiavon et al. 2011). While having standard values of these percentages would be useful, such values don’t exist yet. However, Feng et al. (2012) found a range from 3.9-6.5% for a set of boundary conditions closest to reality. Even though the boundary conditions of the current model are different, nonetheless 6.5% could be considered as an extreme scenario since this value corresponds to a case of low insulation and all four walls exposed. Since, practically, the office is never assumed to be completely empty during the lunch hours, the coefficient from model 3 for constant internal gains during the occupied hours is used. Note that as per ISO-185511, the differences in the coefficients between cases with constant internal gains with/without 2 hours break varies from 11.5% to 25% (based on orientation), being higher for gains with break. The coefficients of model 3 pertaining to only east, south and west orientations of the zones are given. The coefficients for north and core zones were derived from first principle assuming steady state condition and 24 hours constant room set point, cooling surface temperature and 24 hours operation. The corresponding formula is given in the following equation.

$$h_c = \frac{1000}{24 \times coeff} \tag{9}$$

Where h_c is the combined heat transfer coefficient between the cooling surface and the zone air using convective heat transfer coefficient calculated from EnergyPlus and assuming radiation heat transfer coefficient as 5.5 $W/m^2.K$ (ISO 11855-2), the h_c used in this equation is 8.14$W/m^2.K$. Rt and $\hat{R}$ have been calculated from the thermal properties of the pipe and the slab concrete used as per as per Appendix B.2 of ISO 11855-2.

All the models simulated required prior 24 hours design day data, which means at least one day before the period of interest for condition initialization. I modified the EnergyPlus weather (EPW) file for simulations with two consecutive days of design day conditions for this purpose. I performed the analysis only for the second day, and used the first day for condition initialization. Results of the simulation study are reported in the following chapter.

Chapter 4

4 Results

Results of EnergyPlus simulations of WST models, for each of south, east, north, west and core zones of a standard small size office building, with 12 different boundary conditions of heat gain and construction types will be presented in this chapter. The model predicted WST, TABS 24 hours cooling energy extracted on the hydronic side (CE) and occupancy-weighted PPD of the three design models are the main simulation outcomes compared in this section. Distribution of these variables are represented using box plots, while statistical significance of the differences between model to model comfort performances are reported using non-parametric statistical significance test.

4.1 Cooling demand and cooling energy

The "cooling energy demand" (CD) refers to 24 hours cooling energy of an ideal convective system for 25°C zone setpoint operative temperature, while CP is the peak cooling demand in the zone for a convective system. "%dif_n" refers to (CE_n-CD)/CD*100 %, where n is the model number. From Table 4.1, note that the hydronic cooling energy of TABS ($CE_{1,2,3}$) is always higher than the convective cooling demand (CD), calculated for 25°C zone operative temperature set point, while TABS cannot always meet the set point temperature. In fact for the HWHG case in the south zone, for the same hourly zone operative temperature distribution as with TABS, the corresponding cooling load of the convective system is 400 Wh/m^2, which is 68% lower than the 24 hours cooling energy of TABS. The difference is more pronounced for LG cases, with zones that are expected to have higher solar load, like S, W and E. The S zone operative temperature is reported in

Figure 4.3.

One possible reason could be the lower internal gain leading to lower surface temperatures and hence higher conduction gains when TABS is operating. In order to test this I compared one of the wall surface temperatures and the total conduction heat gain of the south zone, for a HG and LG scenario. Note that in these cases the models 2 and 3 would provide different water supply temperature. Therefore model 1 was chosen for this investigation. From Figure 4.1it can be seen that the peak temperature on the inside surface of the south wall in LG condition is 25°C, while the same in the HG scenario is 27.5°C. The claim is further confirmed by net positive heat gain from the surface in LG case and net heat loss to the surface in HG over 24 hours. The difference could also be because of a higher ratio of solar to internal gain and its resultant effect on the heat exchange processes of TABS with the rest of the zone. Feng et al. (2012) also reported that the difference between CE and CD in the absence of internal gains and in presence of solar loads is high. The authors, however, did not study how conductive gains from building envelope change in presence and absence of internal gains.

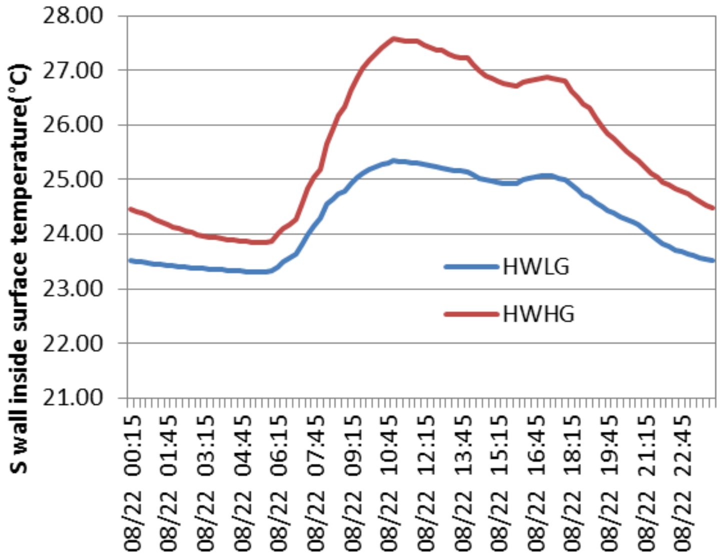


Figure 4.1 South wall inside surface temperature for identical TABS WST during the design day

CP varies with construction type ranging from 3% for C zone to 10% in E zone, which is because high thermal mass basically distributes the total cooling energy over the entire conditioning period, in other words acting as a thermal storage system. However, from just the visual inspection of the data we can see that CD is independent of construction type, but CC varies. For instance, in the south zone CC calculated using model 1 for HW is 11% higher than that for LW construction when internal heat gain is high. This difference also varies with the model. The same difference for model 2 is 3%. When I compare the CC for LG, the corresponding differences are 4% and 3%, for models 1 and 2 respectively, which may be ignored being in the range of variations due to simulation approximations and simplifications. Zone-wise CC varies as W>E>S>C>N. This pattern is noticeable irrespective of the WST model. From the results of CD, design of TABS based on model 3 (ISO11855), therefore, would mean equal treatment of the two construction types.

Table 4.1. 24 hours convective cooling demand, 24 hours TABS cooling energy, percentage difference, simulated for different models for a single day in the design period.

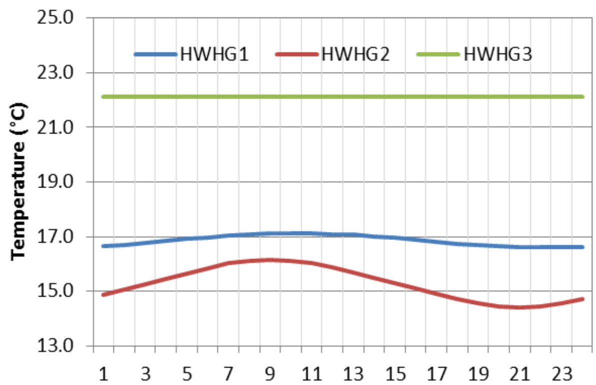
	S				**E**			
	HWHG	**HWLG**	**LWHG**	**LWLG**	**HWHG**	**HWLG**	**LWHG**	**LWLG**
CD	437.7	173.2	437.5	179.2	461.2	193	462.5	198.7
CP	46.7	19	49	20	47	19	52	23
CE_1	674.5	406	609.5	391	683	374.7	630.5	334
$\%dif_1$	54%	134%	39%	118%	48%	94%	36%	68%
CE_2	673	400	654	387	695	371	654.5	347
$\%dif_2$	54%	131%	50%	116%	50%	92%	41%	74%
CE_3	474	167	469	171.4	443.5	176	440	160
$\%dif_3$	8%	-3%	7%	-4.5%	-4%	-9%	-5%	-19.5%
	N				**W**			
CD	409.5	152.7	409.8	159	455.5	186	455.7	194.7
CP	44	17	46	18	46.8	19	48	20.5
CE_1	597	375.8	542.7	317	691	418	605.7	394
$\%dif_1$	46%	133%	32%	99%	52%	126%	32.6%	100%
CE_2	610	330	571	326	694	412.5	637	407.5
$\%dif_2$	49%	116%	39%	105%	52.5%	121%	40%	108%
CE_3	357	120	353.5	110	482	198	473	197
$\%dif_3$	-13%	-21%	-14%	-37%	6%	6.4%	4%	1
	C							
CD	440.7	171	440.7	169.7				
CP	44.6	17	46	18				
CE_1	599	321.6	591	289				
$\%dif_1$	36%	87%	34%	71%				
CE_2	626	310	596	306				
$\%dif_2$	42%	81%	35%	81%				
CE_3	379	145	378	148				
$\%dif_3$	-14%	-15%	-14%	-13%				

The difference in the CEs resulting from model 1 and model 2 are minimal. Later on it will be seen that this difference is also negligible in comfort performance using statistical analysis. Even then, a noteworthy pattern is that CE of model 2 that uses Top feedback is higher for HG cases and lower for LG cases than those of model 1. In other words zone temperature feedback seems to better accommodate for differences in internal gains, as expected. The CE of TABS as per model 3 is 48% to 58% lower than that of model 1 and 2 depending on the other boundary conditions, on the other hand, much closer to corresponding CDs. Therefore, overall, model 3 (ISO 11855) provides inadequate cooling and will need further investigation in terms of thermal comfort.

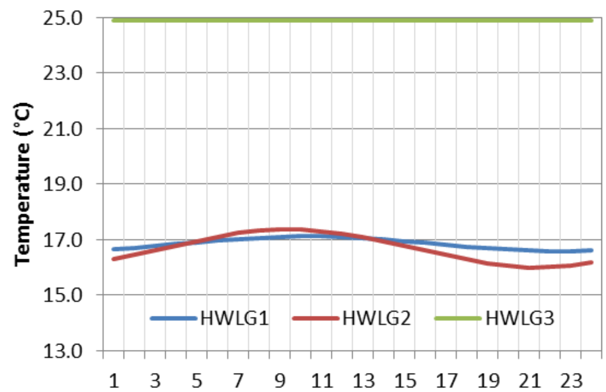
4.2 Water supply temperature of TABS

Model 1 is meant to provide cooling when the outdoor DBT (T_{oa})is above 18°C, and the calculated water supply temperature will always be lower than 18°C depending on the difference between the T_{oa} and 18°C. Model 2 is meant to provide cooling whenever the T_{oa} is above 20°C and/or zone operative temperature is above 22°C, the model constant in this case being 20°C. For a building of given geometry and construction type, the WST of model 3 depends on how one calculates the Q, the *coeff* and assigns a zone set point temperature. Figure 4.2 and

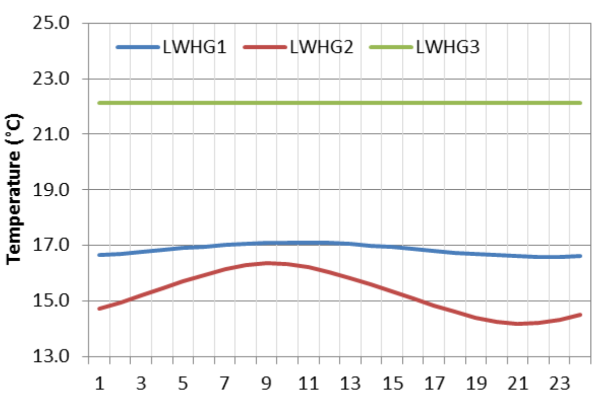
Figure 4.4 show the water supply temperature distribution over the course of a day during the design period for the south zone and the west zone respectively.



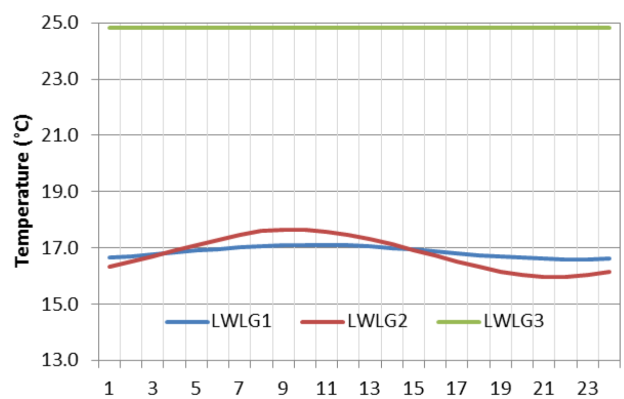
(a) Heavyweight construction-high heat gain



(b) Heavyweight construction-low heat gain

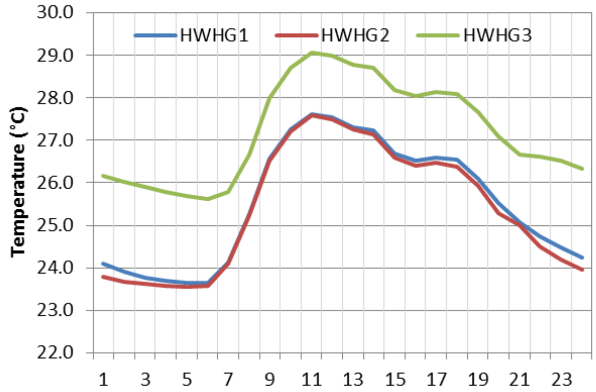


(c) Lightweight construction-high heat gain

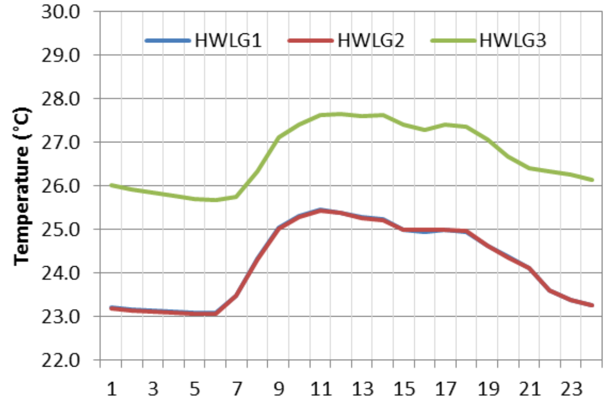


(d) Lightweight construction-low heat gain

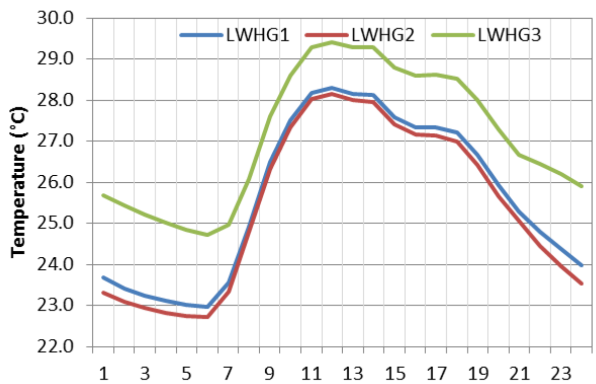
Figure 4.2. Distribution of supply water temperature over the course of the design day for the south zone. Model 1 is independent of zone orientation, thermal mass, heat gain.



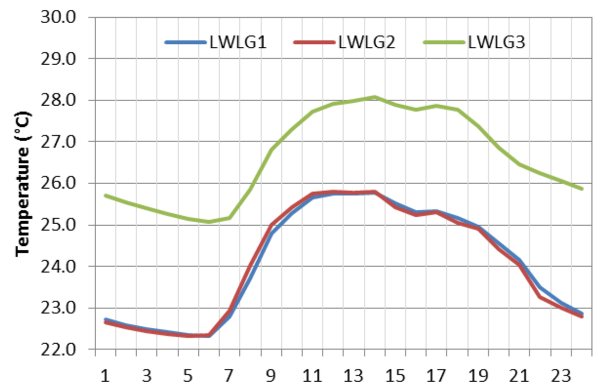
(a) Heavyweight construction-high heat gain



(b) Heavyweight construction-low heat gain

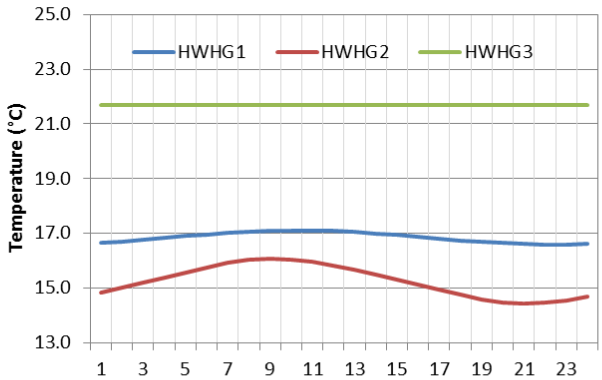


(c) Lightweight construction-high heat gain

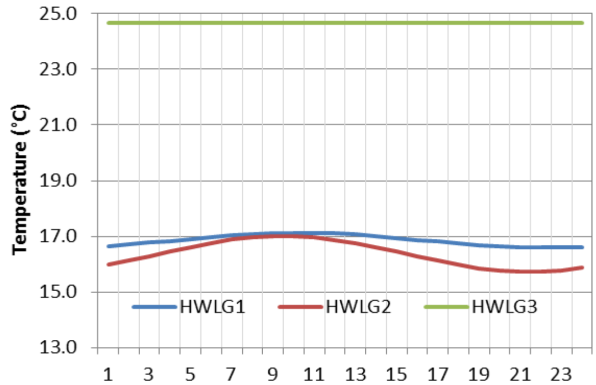


(d) Lightweight construction-low heat gain

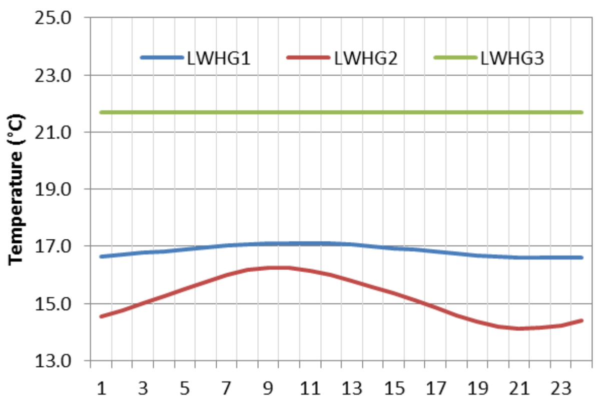
Figure 4.3. Distribution of south zone operative temperature on the design day following models 1,2 and 3



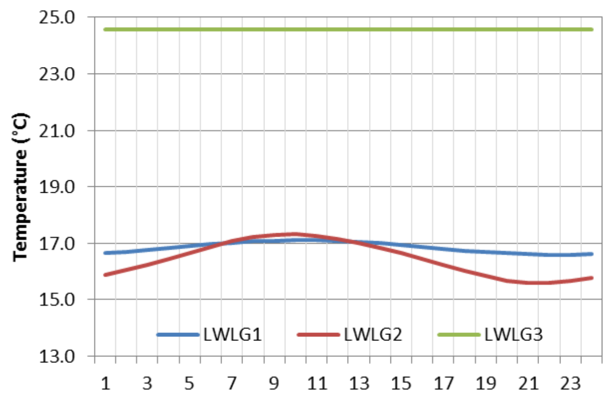
(a) Heavyweight construction-high heat gain



(b) Heavyweight construction-low heat gain

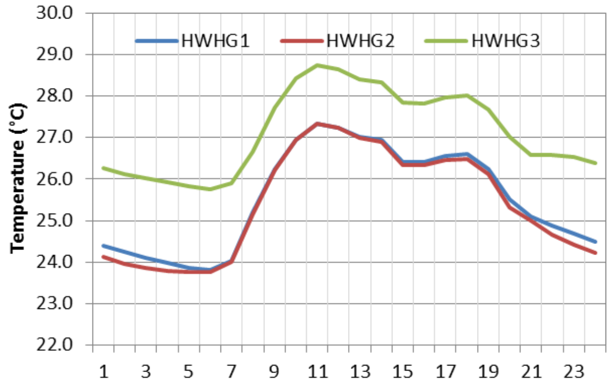


(c) Lightweight construction-high heat gain

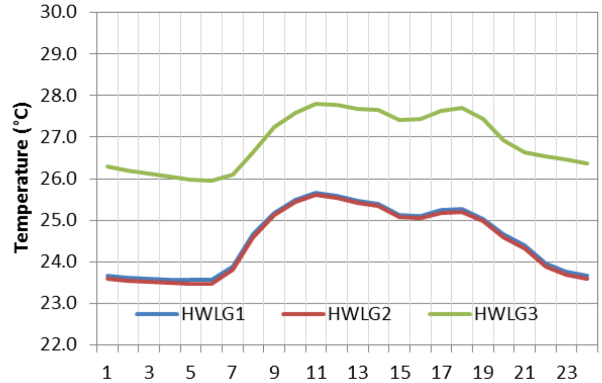


(d) Lightweight construction-low heat gain

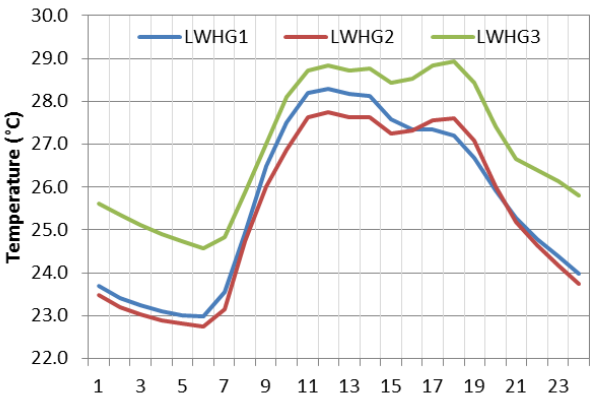
Figure 4.4. Distribution of supply water temperature on the design day for the west zone. Model 1 is independent of zone orientation, thermal mass, heat gain.



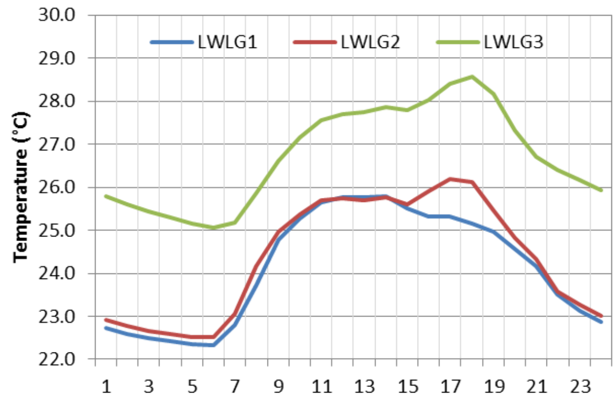
(a) Heavyweight construction-high heat gain



(b) Heavyweight construction-low heat gain



(c) Lightweight construction-high heat gain



(d) Lightweight construction-low heat gain

Figure 4.5. Distribution of west zone operative temperature on the design day following models 1,2 and 3

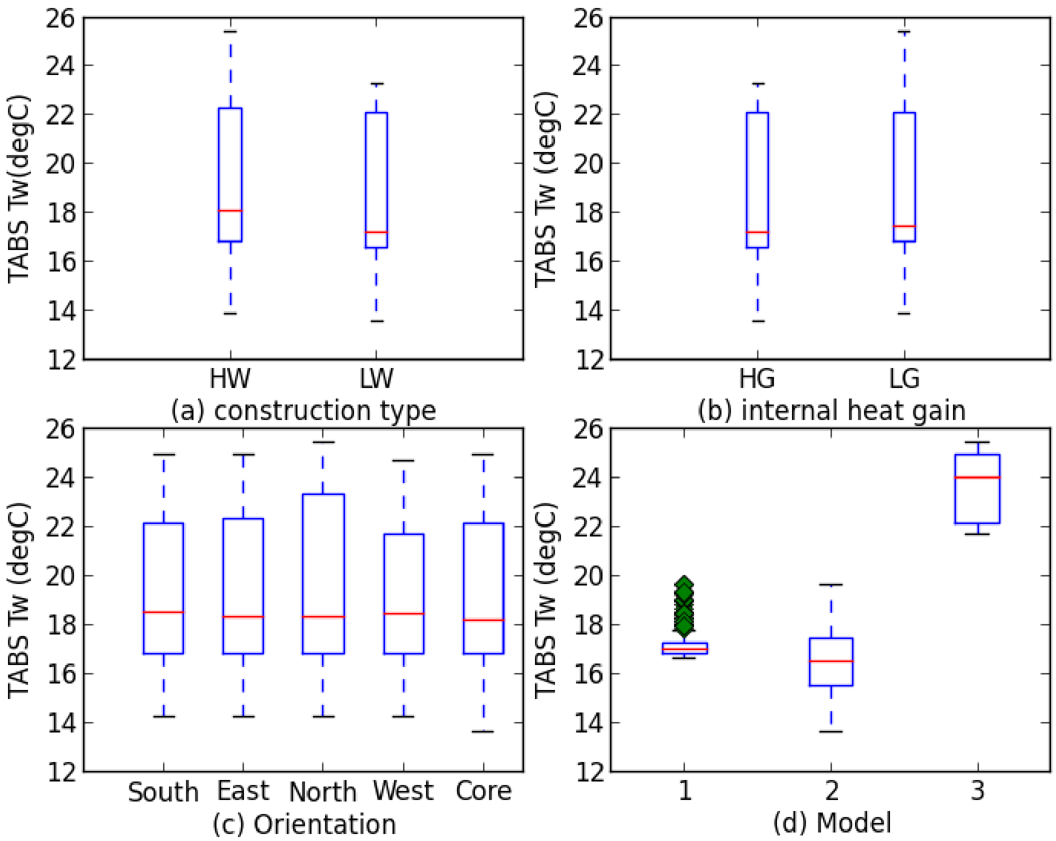


Figure 4.6. Boxplot distributions of water supply temperature by construction type, internal heat gain, orientation and models.

Figure 4.3 and

Figure **4.5** show the corresponding zone operative temperature distribution over the course of the day resulting from following models 1, 2 and 3. Model 1 being function of 24 hours running average outdoor dry bulb temperature simply acts a smoothing filter in this case, following the T_{oa} curve and allocates values lower than 18°C. Model 2 is computed as function of both T_{oa} and average zone operative temperature, therefore showing variations across construction types and heat gains, unlike model 1. The resulting curve of zone operative temperature after the averaging treatment has a slope opposite to that of water supply temperature. Following the peak shaving effect of thermal mass of a zone, the range of water supply temperature is smaller for heavy construction (2K) than for lightweight construction (2.5K). Heat gain intensity moves the curve up or down the scale by 1 K. Model 3 on the other hand displays highest variation (3.5K as per Figure 4.5) in WST by heat gain, but independent of construction type, as it uses CD of convective system in the equation. Similar figures representing water supply temperature distribution for east, north and core zones are given in Appendix B.

4.3 Comfort performance

Comfort exceedance is reported as occupancy weighted PPD hours. The cumulative occupancy weighted PPD hours should be interpreted as equal to 100% if all the occupants at every occupied hour are dissatisfied, the performance threshold being 3-5%. Table 4.2 shows the total Occupancy weighted PPD hours in all the zones on a design day under different design scenarios and different water supply temperature models. Some general conclusions from this table are that: i) model 2 is able to provide the best and the most consistent comfort performance across all the design scenarios (14 out of 20 design scenarios), ii) as was interpreted from the CCs of model 1 and 2, the difference between the comfort performance of model 1 and 2 are negligible, except for the case LWHG, where model 2 is able to provide more comfort than model 1 of the order of 40%-50% better.

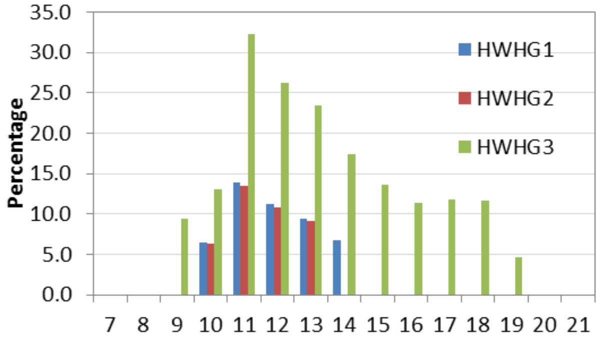
Figure 4.7 and

Figure 4.8 are the diagrammatic representations of the distribution of the same occupancy weighted PPD across the day in the south zone, such that all the bars taken together for a given model and a given design scenario sum up to the cumulative values presented in Table 4.2. Note that over the course of the day, also, the occupancy weighted PPD values match well between model 1 and model 2, the only difference being in the case of LWHG. A similar conclusion can be drawn from

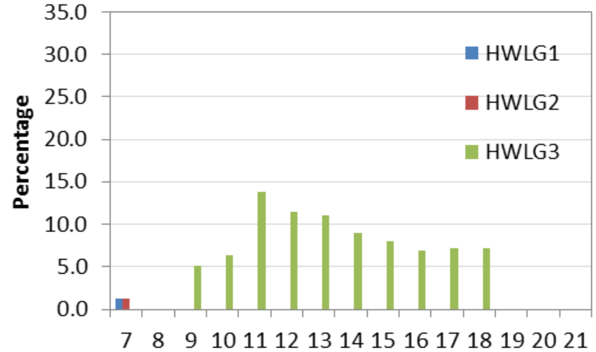
Figure 4.8 for the west zone. The similarity in comfort performance is more apparent from the zone operative temperature profile of the south zone during the design day resulting from model 1 and 2. In LWHG case of west zone, the zone operative temperature is generally lower for model 2 than model 1, only exceeding the latter in the late afternoon, a trend noticeable in case of lightweight construction. This difference is, however, not captured in the comfort exceedance figures as the PPD during the occupied hours is mostly less than 10%.

Table 4.2. Occupancy weighted PPD hours for PPD>10% resulting from models 1, 2 and 3 in all the zones expressed in percentage.

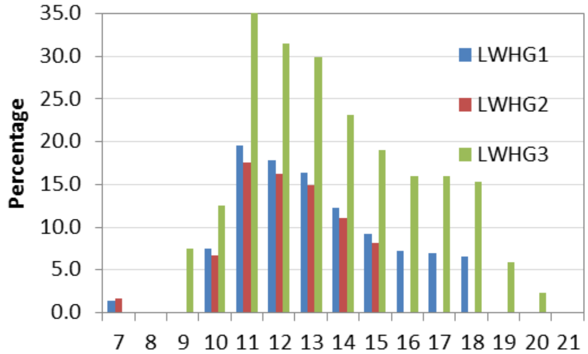
	S (1	2	3)	**E** (1	2	3)	**N** (1	2	3)	**W** (1	2	3)	**C** (1	2	3)
HWHG	6	5	21	2.5	2.4	20	5.6	4.6	26	2.6	2.6	18	10	8.5	44
HWLG	0.1	0.1	10	0.5	0.5	12	0.1	0.8	12	0.1	0.1	12	0	0	17
LWHG	13	9	26	9.4	8.7	26	12	8.5	30	11	9.7	21	16	15	48
LWLG	1	1	13	1.3	1.2	11	0.9	1	11	1	1	14	0.2	0.2	13



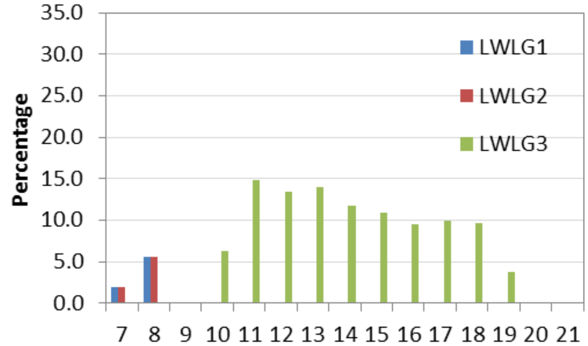
(a) Heavyweight construction-high heat gain



(b) Heavyweight construction-low heat gain



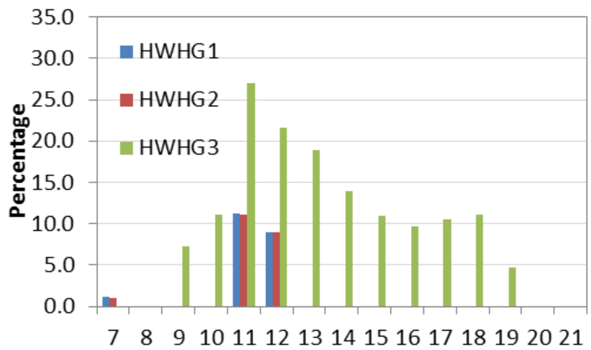
(c) Lightweight construction-high heat gain



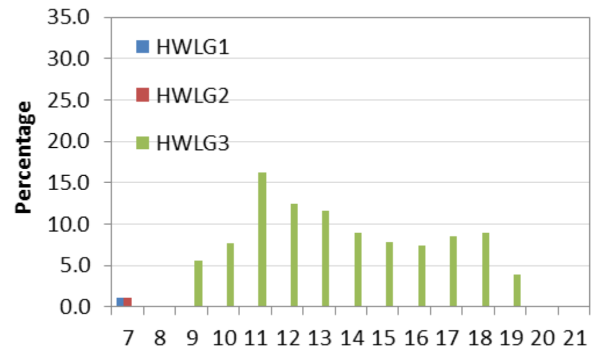
(d) Lightweight construction-low heat gain

Figure 4.7. Discomfort distribution on a design day as Occupancy weighted PPD for PPD>10% in the south zone, resulting from different water supply temperature models.

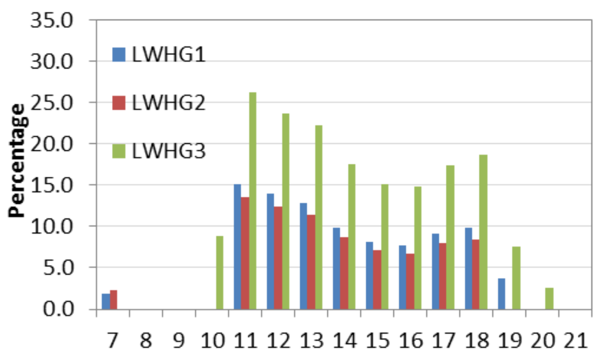
Model 3 has far exceeded the discomfort threshold of 3-5% and needs correction. Several factors may influence this outcome including the calculation of Q, *coeff* and room set point, which will be investigated in the following chapter.



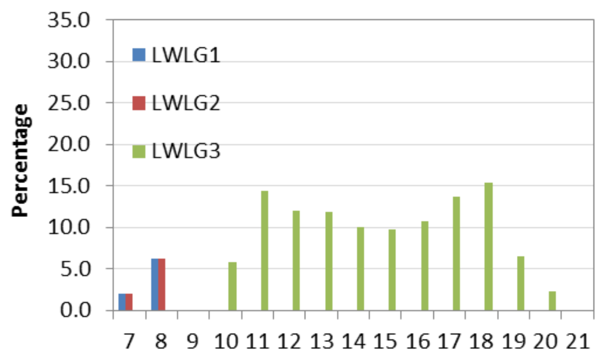
(a) Heavyweight construction-high heat gain



(b) Heavyweight construction-low heat gain



(c) Lightweight construction-high heat gain



(d) Lightweight construction-low heat gain

Figure 4.8. Discomfort distribution on a design day as Occupancy weighted PPD for PPD>10% in the west zone, resulting from different water supply temperature models.

As expected, the most comfortable design scenario is heavyweight construction and low internal gain followed by lightweight construction and low internal gain. The distributions carry more information, in that the peak discomfort almost coincides with the hours of highest cooling load.

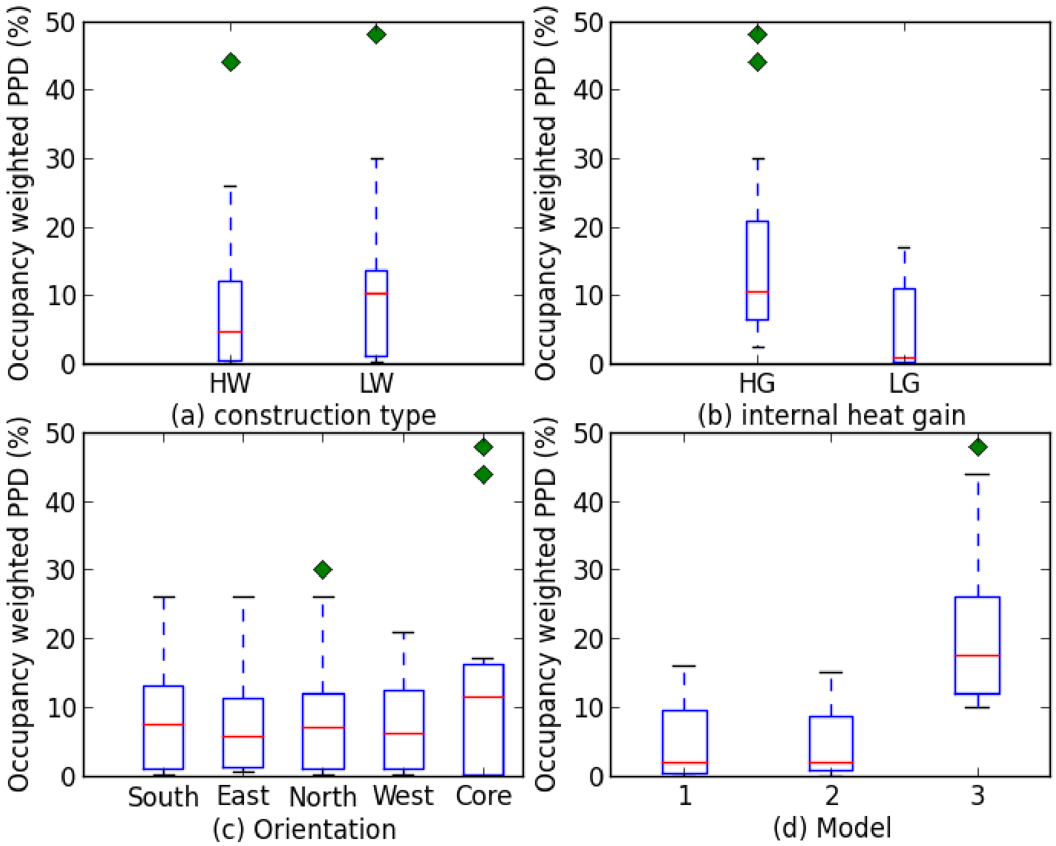


Figure 4.9. Boxplot distributions of Occupancy weighted PPD by construction type, internal heat gain, orientation and models.

In

Figure 4.8, again, we observe that the peak hours of discomfort are concurrent with the peak cooling load in the zone; for example, the west zone has a dual peak for initial heat gain due to occupancy and solar load in the late afternoon.

Box plots were constructed to investigate data distribution of comfort exceedance by median, 25th and 75th Quartiles, shown in Figure 4.9. This will help single out and visualize the contribution of each of these parameters on resultant spread of water supply temperature and comfort performance of the models. From Figure 4.9 it can be observed that occupancy weighted PPD has smaller spread and lower median value <5% in a massive building, compared to >10% in light weight construction. Obviously impact of internal heat gain on comfort performance is pronounced, followed by construction type and then zone orientation, probably due to use of shading devices. Again, the difference between the comfort performances of model 1 and model 2 are negligible. There is however, a slight difference in the spread of the discomfort, in that model 2 has narrower range of occupancy weighted PPD than model 1.

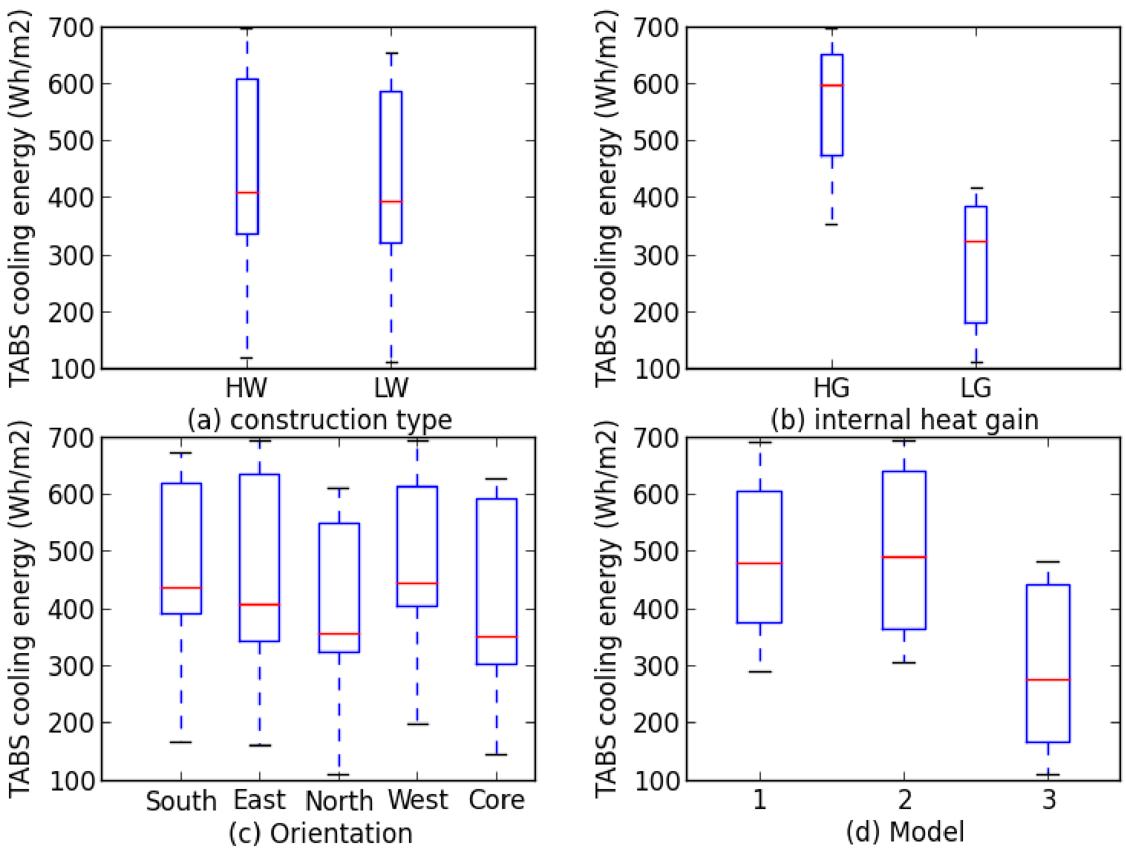


Figure 4.10. Boxplot distributions of TABS CE by construction type, internal heat gain, orientation and models.

The CE of TABS as shown in Figure 4.10 shows a significant difference between LG and HG scenarios with the median ranging from 350 Wh/m^2 to 600 Wh/m^2. Note that the CE has the lowest median for the north and core zone corresponding to highest discomfort in these zones. The WST in these zones could be biased by coefficient chosen for model 3.

Summarizing the results from this Chapter, model 1 and model 2 have minimal difference in the allocated WST and hence cooling capacity of TABS and comfort performance. However, model 2 seems to accommodate the variations in construction type better than model 1 resulting more consistent comfort performance across construction types. Both the models also provided the highest comfort performance, restricting the design day occupancy weighted PPD hours to 5% in 14 out of 20 design scenarios investigated. Model 3 shows the worst comfort performance with a median occupancy weighted PPD of 15%, therefore requiring further investigation. In the following chapter the statistical significance of difference between comfort performance of model 1 and model 2 will be reported, along with further investigation of model 3.

Chapter 5

5 Discussion

In the previous chapter several results of water supply temperature, cooling capacity and comfort performance of three TABS models have been presented, and analyzed as functions of construction type (heavy vs. lightweight), internal heat gain, zone orientation and model number. In Chapter 4, I reported that model 1 and model 2 have similar comfort performance. However, model 1 considers only an outdoor weather parameter, i.e. 24 hours average outdoor DBT ($T_{oa)}$, while model 2 includes a feedback of zone operative temperature, the coefficient being similar to the gains factor of standard proportional control. The former approach can allow a spreadsheet based calculation and the latter will require a detailed simulation of building model equipped with TABS. If it can be proved that model 1 will suffice for the initial design phase of TABS, it will save the time and the resources required for transient simulation. In this chapter, therefore, I used non-parametric ANOVA to test the statistical significance of the differences in the comfort performances of model 1 and model 2, measured as occupancy weighted PPD hours.

5.1 Statistical comparison of model 1 and model 2

In the simulation based study conducted here, WST of TABS calculated by various models ranged from 14°C to 25°C, which is a large spread for a constrained parameter like this, in particular reference to choice of environmental cooling water source. Comfort performance of these models expressed in terms of daily occupancy weighted PPD also shows a spread from 0 to 20% a part of which could be attributed to differences in internal heat gain and construction type. On the other hand the spread of comfort performance varies from model to model, (0-16%, standard deviation=5%) for model 1 and for model 2 (0-15%, standard deviation=4%) and the highest in the case of model 3 (10%-48%, standard deviation=10%). Table 5.1 gives the median, mean and standard deviation of comfort performance of TABS models by construction type, internal heat gain, zone orientation and model. The difference from model 1 to model 2 is negligible In order to test the statistical significance of this difference I performed a non-parametric ANOVA, also called Kruskal Wallis ANOVA. Non-parametric test was used, as the occupancy weighted PPD for the different design models showed a discontinuous non normal distribution due to small sample size. The distributions are shown in Figure 5.1.

Table 5.1. Median of PPD-weighted discomfort (row1), mean and standard deviation of PPD-weighted discomfort by model (row 2 and row 3 respectively)

Parameter	M1	M2	M3
Median	1.8	1.9	17
25^{th} percentile	0.1	0.2	10.3
75^{th} percentile	10	8	26

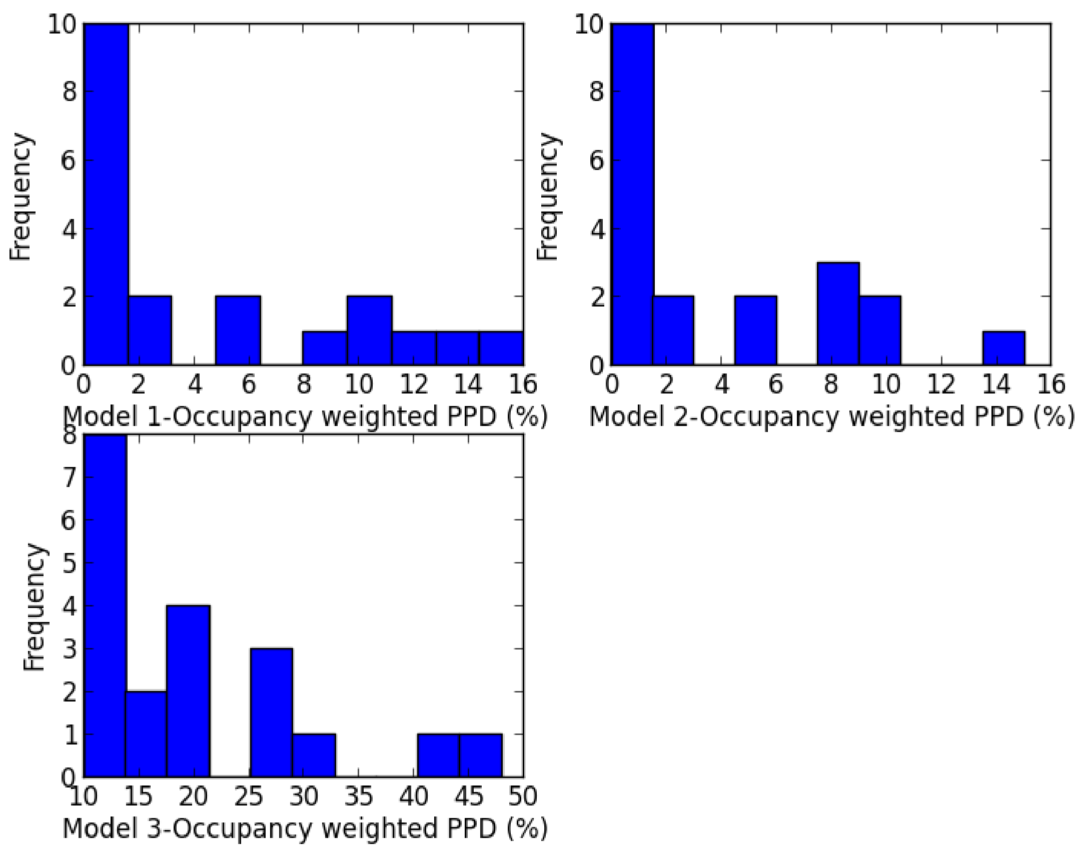


Figure 5.1. Frequency distribution of Occupancy weighted PPD hours for models 1, 2 and 3.

The results of the Kruskal Wallis ANOVA showed that while comparing model 1, 2 and 3, the resultant p is 0, comparing models 1 and 2, p is >0.7. Therefore comparing the design models of TABS WST, based on occupancy weighted PPD, I can infer that the differences between the comfort performances of these models across all the design scenarios studied, is not statistically significant. Hence, a designer can use a simple univariate design model of WST for CZ03 for design scenarios reported in this thesis. However, considering higher cooling energy extraction capability and probably better comfort performance of model 2 for the design scenario of lightweight construction with high internal gains, as can be seen from Figure 4.8, I would still recommend model 2 for early design phase of TABS. Further, from the results of WST predicted by these two models it was evident that model 2 accounted for difference in heat gain (1K difference), while model 1 did not. However, this difference was compensated by the lower pump operation hours for model 2. Such opportunity may not exist in warmer climates where 24 hours pump operation is necessary. Therefore before assigning equal weightage to both model 1 and 2 as recommendation, studies must be conducted in warmer climates and higher heat gain scenarios.

5.2 How could model 3 be improved?

Model 3 was excerpted from the international TABS design standard ISO11855. However, from the results of this study's WST simulations, it seemed obvious that this model is not suitable for providing comfort in CZ03 for the design scenarios studied. While model 1 and 2 predicted a WST of 14°C to 17°C, WST predicted by model 3 ranged from 21.5°C to 25°C. Such high values could be attributed to several contributing factors, viz. 24 hours cooling energy, Q, the coefficient of the model, *coeff*, and the zone maximum comfort limit T_{rsp}.

I selected the scenario of south zone HWHG and modified the above mentioned model inputs iteratively. The first step was to change Q. In place of "standard cooling load" I used the CE of TABS calculated from simple univariate model 1. The resultant Q is 674 Wh/m^2 in place of 473 Wh/m^2 previously used, which corresponds to a 42.5% difference. The calculated WST is 19.5°C. The resultant 24 hours TABS CE was found to be 603 Wh/m^2, which was still lower the actual Q used in the formula. The comfort performance was drastically improved from 21% to 9.75% occupancy weighted PPD hours. A similar approach with HWLG scenario resulted in 10% change in comfort performance. The Q used was 406 Wh/m^2, while the corresponding TABS CE obtained from simulation was 273 Wh/m^2. Model parameters like the *coeff* can also be changed iteratively in EnergyPlus using EMS module until the desired comfort performance is reached. However, this adjustment should be done logically rather than arbitrarily. I found that for the design scenario of HWHG in the south zone, model 1 and 2 were able to provide maintain the discomfort threshold of 5%. The corresponding WST is 17°C for 20 hours of pump operation. With this WST I calculated the *coeff* of the model and found it to be 14.4. In future a more first principles approach could be used to adjust the values of the coefficients for different zones, under different design scenarios. However, evaluation of comfort performance of the model in terms of occupancy weighted PPD could be complex to implement in every simulation time step. A more tractable metric would be difference between the zone setpoint temperature and the peak temperature of the day.

5.3 Which of the TABS design model from literature gives the most consistent comfort performance under different design scenarios?

One of the major goals of this study was to understand that of the chosen models with different levels of complexity, which model is able to maintain comfort under majority of design scenarios. The obvious expectation was that a model that accounts for inputs that vary under these design scenarios, for instance, heat gain, building thermal mass and zone orientation amongst several others, will be the most robust one. From this perspective model 3 scored the most based on a qualitative analysis presented in Chapter 2. This is because this model has the maximum number of inputs including the direct design cooling load calculated using transient simulation and it is partly physically based.

From simulation study of a single story small office building in CZ03, it was found that both model 1 and model 2 could provide equal comfort performance across all the design scenarios considered. Both model 1 and model 2 were able to maintain the discomfort threshold of 3-5% for 14 out of 20 scenarios simulated. Model 2 was found to have lesser spread of occupancy weighted PPD and better comfort performance in the above mentioned design scenarios than model 1. This could probably be attributed to the fact that lightweight construction benefitted better from zone temperature feedback than heavy construction. Also outdoor DBT compensated model did not account for heat gain in space. Kolarik et al. (2012) have reported a comfort exceedance of 17.5 % working hours/year using model 2 in heavy weight construction for CZ03 based on comfort class B of EN15251. However, this scenario included activation of solar shading (internal and external shades) controlled by solar radiation at the internal surface of the window and supply air temperature at 19°C which would have provided additional cooling. Further, from the results of WST predicted by these two models it was evident that model 2 accounted for difference in heat gain (1K difference), while model 1 did not. However, this difference was compensated by the lower pump operation hours for model 2. In case of warmer climate and higher internal gains, when the pump is required to operate for 24 hours, model 2 can be expected to provide more consistent comfort performance. However, such a conclusion cannot be justified without further simulation based studies with model 1 and model 2 for more severe design conditions.

5.4 Practical feasibility of design models

In order to test if the design models could be practically implemented design scenarios close to real cases were chosen, for instance, the model chosen was close to the standard reference small office building in terms of geometry and thermal properties. Shading was provided to avoid glare and ventilation maintained at minimum required for air quality purposes, with TABS as the only cooling system in the building. The model predicted values of water supply temperature range from 14°C to 25°C, which can be provided by a wide of range plant equipment from chiller at the lower end to cooling tower at the higher end (Kalz 2009). For instance, for an average design outdoor wet bulb temperature during night hours of 16-19°C, the WST provided by a wet cooling tower of efficiency 75% would be 20±2°C. Furthermore average annual ground water temperature in CZ03 is higher than 19°C. A 14°C water supply temperature predicted by model 2, as in the case of lightweight construction with high internal heat gain in the west zone, would mean that only a chiller could meet this design WST setpoint, the energy consumption of which will depend on the COP. But greater limit could be imposed by dehumidification need and discomfort from radiant asymmetry in space. Typically the water supply temperature is restricted by the dewpoint temperature of the zone. Maintaining a zone dew point temperature below 14°C would mean high dehumidification energy cost at the lower end of the design values. Therefore further evaluation of the tested design models should be performed under similar design constraints, imposed by energy and choice of cooling source.

Chapter 6

6 Conclusion

The study was driven by two motivating factors:

i) Any HVAC system should be designed to provide thermal comfort irrespective of climate, construction type and other parameters that may affect the cooling load in space.

ii) Like design of any other engineering system, the chosen design method should be a trade-off between the accuracy of iterative multivariate models and simplicity of univariate non-iterative design models.

The objective of this study was to develop a framework for classifying and characterizing, both qualitatively and quantitatively, the design and control models of TABS from the literature, as a step towards development of recommendations for sizing TABS for different design scenarios. Comfort exceedance was chosen as the quantitative assessment criterion.

I conducted a detailed literature review and a qualitative classification of seven models of water supply temperature (WST) of TABS from the literature comparing their suitability for different design phases, varying design conditions such as internal gains, building thermal mass and pump operation hours and ease of implementation in terms of number of inputs, iterative or non-iterative approach. The results of this qualitative study are presented in Chapter 2. Based on reliability, design inputs and ease of implementation, five design models were shortlisted and three of those models were evaluated using transient building energy simulations, the results of which are reported in chapters 4 and 5. For model definition and description, the readers should refer to Chapter 2.

6.1 Qualitative and quantitative analysis

From the qualitative analysis I found that model 1 and model 2 (Olesen and Dossi 2004) are suitable for a feasibility study of TABS. Model 3 and 4 (ISO 11855) are ideal for early design decision with quasi-steady state cooling load as well as detailed design phase and in precooling scenario. Model 6 (Gwerder and others 2008) can be used for single cooling generator for different zones in a building with different cooling demand and is more suitable for detailed design phase. Model 5 (Gwerder and others 2009) is specifically suitable for precooling or pulse width modulation scenario, but in general recommended for all design stages.

Three shortlisted models were simulated for four sets of design conditions including two levels of internal heat gain and two construction types. Simulation studies were performed using the Energyplus 5 zone small office building model in California CZ03 climate zone. The design models were compared for general comfort performance and comfort performance consistency under different design scenarios using a new comfort exceedance metric, occupancy weighted PPD. The discomfort threshold was assigned to be 3-5%. Results of the simulation indicated that both model 1 and model 2 produced similar comfort performance for the design scenarios studied, maintaining the discomfort threshold in 14 out of 20 cases. The only scenarios where these models failed to maintain the threshold was lightweight construction with high heat gain and core zone. Model 2 was found to have lesser spread of occupancy weighted PPD and better comfort performance in the above mentioned design scenarios than model 1.

Model 3 which uses 24 hours cooling energy of convective system in the design equation displayed the worst comfort performance with highest spread. Using the TABS 24 hours hydronic energy obtained from simulation of model 1, for the case of HWHG construction in the south zone resulted in significant reduction of daily discomfort, but still worse than both model 1 and model 2. For a desired WST of 17°C, the higher model coefficient was found more suitable for the above mentioned design scenario. Based on this study model 2 seemed to provide the most consistent comfort performance across all the design scenarios.

Further these models should be evaluated in face more rigid constraints on choice of cooling source and energy requirements.

6.2 Future work

TABS is an advanced zone conditioning system, highly energy efficient using water as the heat carrier. It is an active research topic and needs detailed investigations and more tractable as well as accurate design models for field deployment. The current study is subject to various limitations due to the simulation environment, design scenarios and climatic data considered. Sufficient effort was put in to control the boundary conditions for best results expected from this study. There could be multiple factors responsible for the comfort performances of the models considered including coefficients of linear models and zone and base water supply set point temperatures.

Parameters and inputs of model 3 should be adjusted to match the design boundary conditions like climate, window-to-wall ratio, exposed surface are to floor area ratio etc. and comfort expectations of people in a given climatic region.

Even though there was no statistically significant difference between comfort performance of model 1 and model 2, model 1 should still be tested for more severe heat gain conditions. It could, however, serve as initial value for more complex design models.

Further parametric studies should be performed for different solar heat gain levels and different zone set point temperature. The effect of hourly modulation of water supply temperature was not clear from this study but can govern deployment of night-time cooling sources like cooling tower. Hence this effect should be investigated better.

A new metric for comfort exceedance, "Occupancy weighted PPD hours" was developed as a part of this study that encompasses both severity and duration of discomfort. In future usability or affordance of this metric should be verified with the designers and it should be compared with other metrics. Other metrics should be developed that could be easily measured in operating buildings for comparing the design goal with the design performance.

Results of the simulation show that higher comfort could be maintained in most of the simulated cases, if TABS cooling energy is used in model 3 than 24 hours convective system's cooling load. TABS energy extraction is function of supply water temperature. However such a process would be iterative. Methods simpler than multiple iterations to accommodate difference in energy extraction of TABS and convective systems should be implemented as standalone programs and also in building energy simulation software. Maintaining identical boundary conditions in the simulation studies, as much as possible, it was found that under circumstances of low internal heat gain, the cooling energy extracted on the waterside of TABS could be greater than convective system's cooling load by even 100%. The reason behind this difference between scenarios of low and high internal heat gains was found to be lower internal surface temperature leading to higher conduction heat gain, at least in part. However, prior to drawing conclusion, the results of the TABS energy extraction should be validated with a 3 D transient heat conduction model of the slab with a time series of surface temperature applied on its surface. This may require appropriate modelling of the water-side of TABS, since EnergyPlus uses a heat exchanger model.

Models of heat gain range from detailed standard values used in building energy simulation programs to single variable semi-parametric regression models. However, none of them have been implemented in practical and complex buildings with high solar gains to compare and validate their performance. For wider adoption of TABS in more challenging climates better solar heat gain control algorithms (possibly model predictive) must be designed.

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Appendix A

Table A.1. List of buildings from CBE database with mixed mode and radiant cooling

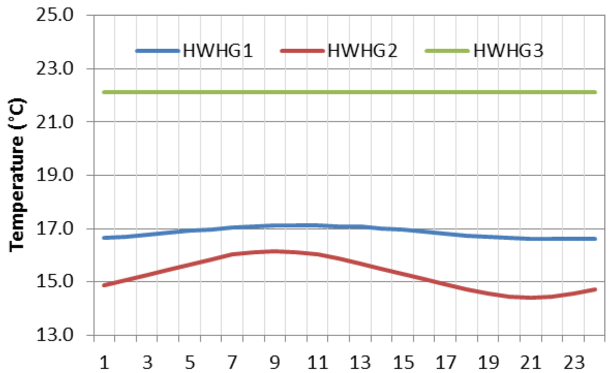
Building index	YOC	Climate zone	Use and loads	Envelope	Ventilation system	Radiant cooling system	Respo-nse rate
1	2007	CA climate zone 3	offices and classrooms	Arbor shaded door, window control automated for natural ventilation, daylighting design	Natural ventilation and under floor air distribution system	Thermo-active floor	0.06
2	2009	CA climate zone 3	offices	Daylighting design with fixed external shading devices on south and west	Mixed mode, demand controlled ventilation, under floor air distribution	Thermo-active ceiling	0.37
3	2007	CA climate zone 4	design studio	Daylighting design with large windows, fixed external shade, photo-sensor and occupancy sensor based artificial lighting control	Natural ventilation	Thermo-active floor	0.64
4	2008	WS	offices	Adequate solar shading, photo sensor and occupancy sensor controlled lighting	Natural ventilation	Ceiling panel	0.58
5	2004	CA climate zone 4	offices, labs and conference room	Narrow floor plate and long north and south faces, horizontal external shading devices on the south, operable vertical louvers in the north, light selves with clearstory windows all around	Seasonal mixed mode system in office, mechanical ventilation in research labs and conference room	Thermo-active floor	0.66
6	2006	OR	jury assembly area, common corridor and lobby	Recycled metal panel cladding, each courtroom shaped like a droplet of water day-lit using clerestory window, dimmable and day light controlled artificial lights, solar shading and high performance glazing, in the atrium in particular.	Displacement ventilation	Thermo-active floor	0.38
7	2005	CA climate zone 4	classrooms and laboratories	Large window with spectrally selective low e glazing, external shading devices, glare control louvers and interior light shelves on the south side, PV covers 25 % of west roof, cool roof	Natural ventilation and ceiling fans in classrooms and laboratories	Thermo-active floor	1

Table A.2. Thermal properties of building materials

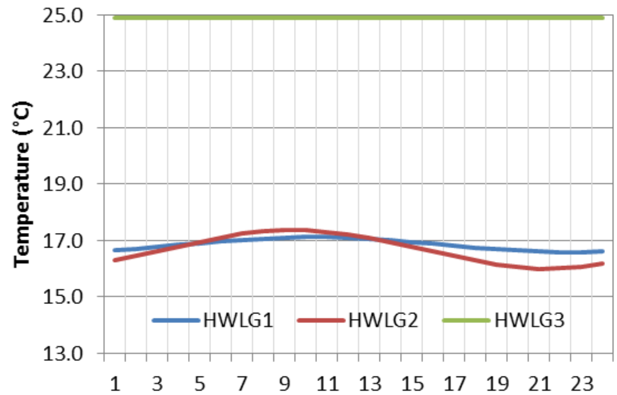
Material	Thermal conductivity (W/m-K)	Density (kg/m^3)	Specific heat (kJ/K)
Brick	0.84	1700	0.800
Heavyweight Concrete	1.4	2300	1.0
Lightweight concrete	1.28	600	1.0
Insulation	0.04	25	1.0
Heavy insulation	0.025	30	1.40
Air space	x	x	x
Plasterboard	0.14	950	0.840
Steel siding	50	7800	0.480

Appendix B

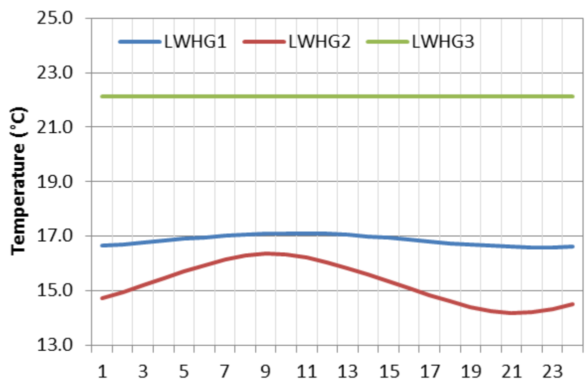
Figures



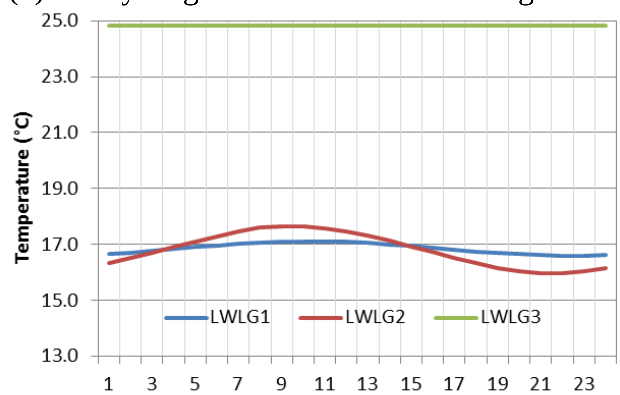
(a) Heavyweight construction-high heat gain



(b) Heavyweight construction-low heat gain

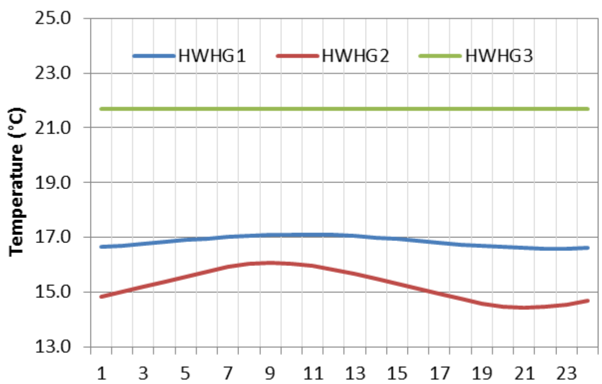


(c) Lightweight construction-high heat gain

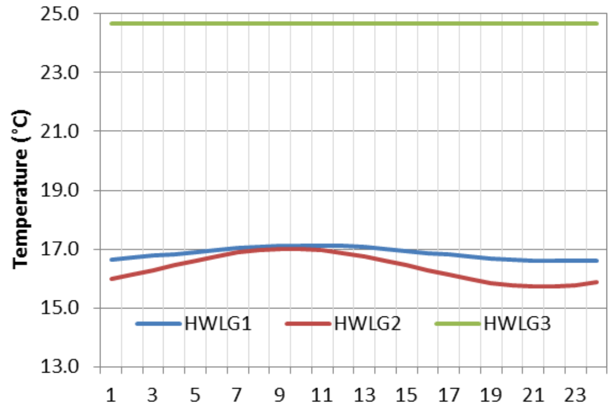


(d) Lightweight construction-low heat gain

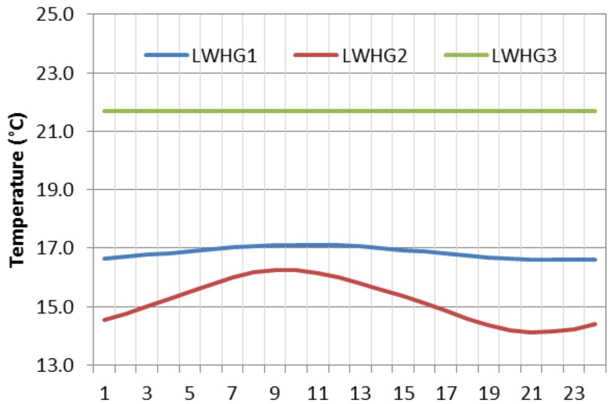
Figure B.1. Distribution of supply water temperature on the design day for the east zone. Model 1 is independent of zone orientation, thermal mass, heat gain.



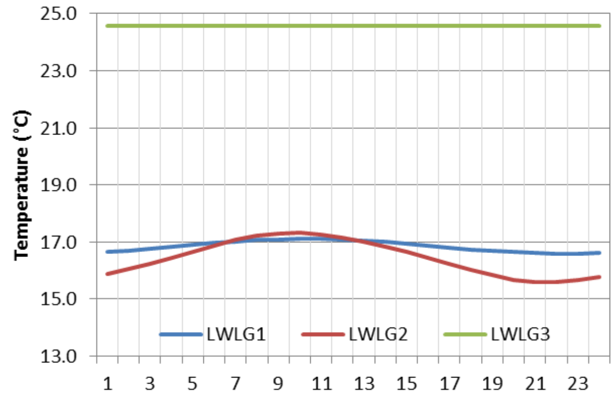
(a) Heavyweight construction-high heat gain



(b) Heavyweight construction-low heat gain

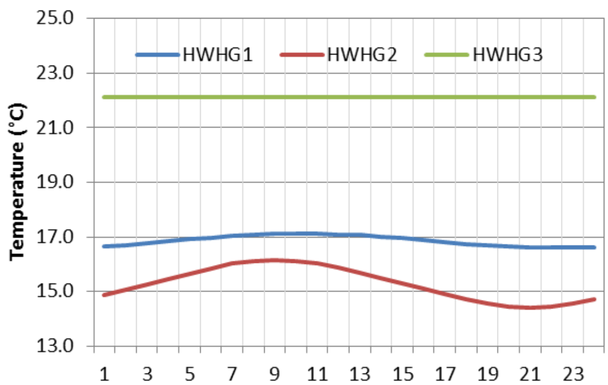


(c) Lightweight construction-high heat gain

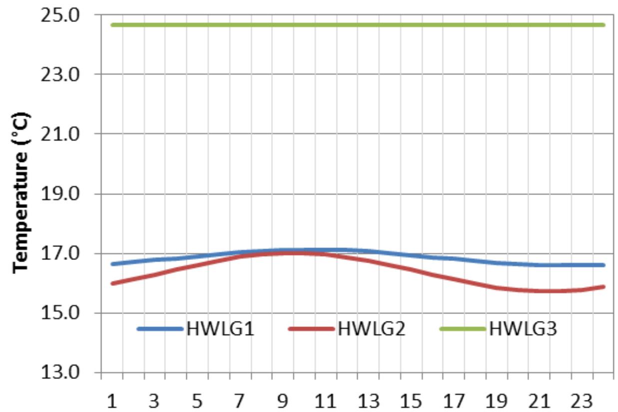


(d) Lightweight construction-low heat gain

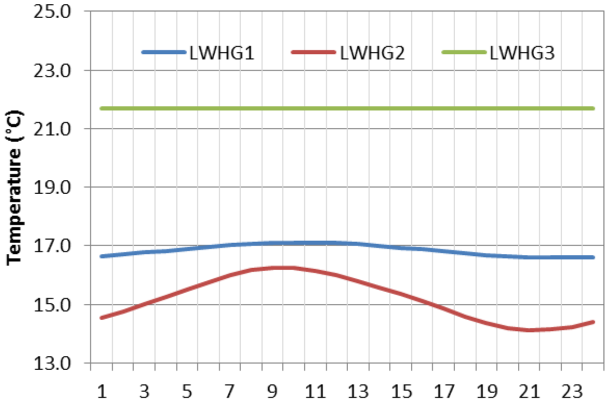
Figure B.2. Distribution of supply water temperature on the design day for the north zone. Model 1 is independent of zone orientation, thermal mass, heat gain.



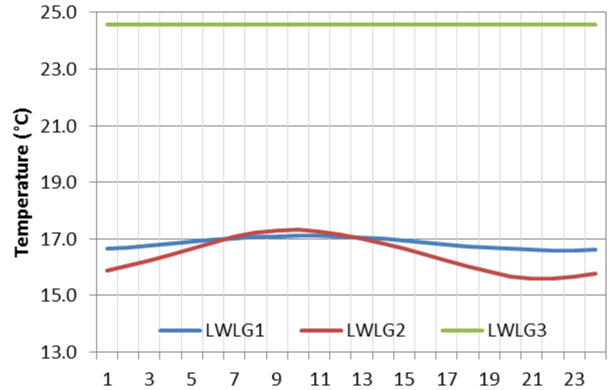
(a) Heavyweight construction-high heat gain



(b) Heavyweight construction-low heat gain



(c) Lightweight construction-high heat gain



(d) Lightweight construction-low heat gain

Figure B.3. Distribution of supply water temperature on the design day for the core zone. Model 1 is independent of zone orientation, thermal mass, heat gain.