A new control strategy for high thermal mass radiant systems

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Abstract

This paper presents a new controller for high thermal mass radiant systems that can be implemented within a typical Building Automation System. We illustrate its performance using an EnergyPlus model representing a single zone, middle floor of an office building in Sacramento, California. The results of a small sensitivity analysis show that when compared to common practice in the US this approach reduces electricity cost and energy consumption by up to 40% and 35%, respectively, while maintaining comparable comfort conditions in the zone. Furthermore, this design & control approach could eliminate the need for a chiller in most California climate zones for typical office design loads.

Introduction

We define high thermal mass radiant systems as thermally activated building systems (TABS) or embedded surface systems (ESS) (ISO, 2012) that have a response time calculated at 63% (i.e., time constant) (Ning et al., 2017) of several hours or more. It is a challenge to control these systems to maintain comfort conditions due to their thermal inertia and the associated time constant (typically 4 to 8 hours) relative to the 24-hour period of typical building loads.

Model Predictive Control (MPC) is the current state-of-the art as it can find a near optimal control solution when a suitable model and reliable load forecast is available (Oldewurtel et al., 2012). However, unlike other industries such as automotive and chemical engineering, which benefit from scaleable and repeatable deployments, in the buildings domain each building is it's own unique protoype and poses unique challenges. For the HVAC industry, MPC is complex to implement and maintain, often computationally intensive, and outside the realm of feasibility for controls contractors who work under time constraints. Moreover, current Building Automation System (BAS) software does not have the programming capability to implement MPC. Though we have used MPC in the past, both in whole building simulation and over short timeframes in actual buildings (Feng et al., 2015), the technology is still in the research domain rather than industry best practice within the HVAC field.

On-off or proportional-only closed loop controllers with either the zone air or the slab temperature as the control variable are the most common approaches in practice today. The former leads to large zone air temperature overshoots and frequent inefficient heating/cooling operation within the same 24-hour period. The latter is more stable but often fails to meet comfort requirements without manual trial-and-error configuration. In many cases, particularly for TABS systems, the supply water temperature is the controlled value at the zone instead of the valve position, and the system operates in a constant flow manner.

In a recent survey of radiant buildings in the US (Higgins et al., 2015), 43% used the zone temp as the sole control variable, while 26% used the slab temp. In addition, we interviewed 20 mechanical designers with extensive expertise working with radiant systems and found wide variation in perceived best practice.

Additionally, in many cases, the temperature setpoints for the cooling and heating water plant are also considered a part of the control strategy at the zone level, as opposed to *disturbances* to the controller at the zone level. Examples of this approach include outdoor air temperature based resets for plant supply water temperatures.

A paper by Romaní et al. (2016) provides an up-to-date and comprehensive literature review of previously assessed control strategies for radiant systems. It focuses on TABS systems only, and does not address the potential of high mass ESS to provide many of the same benefits and challenges. The introduction of the paper by Schmelas et al., 2015 also provides a concise evaluation of control startegies for high thermal mass radiant systems, before presenting a novel adaptive model predictive controller using multiple linear regression.

Particularly notable papers that evaulate control strategies that do not use an underlying model are discussed below. A paper by Sourbron and Helsen, 2014, which found that controlling to ceiling surface temperature using setpoints with a fixed offset from comfort limits was the best solution of the controllers evaluated with TRNSYS. Gwerder et al., 2008 developed the Unknown But Bounded (UBB) method, which focuses on design and control approaches that define an upper and lower range of zone loads and control the zone using a constant flow, variable temperature approach. Olesen, 2007 used TRNSYS simulations to investigate a constant flow, variable temperature feedforward control (i.e., without a feedback loop), where the temperature is defined by the outside air temperature. Notably, the radiant system operates only during the nighttime hours (18.00 to 6.00) and was able to maintain comfortable conditions in the zone in two different climates.

We agree with the statement that "Since TABS react slowly, only day-to-day room temperature compensation is promising …instant correction can not be achieved with TABS." (Gwerder et al., 2009). This implies that for a control strategy to be effective, it must react to, and incorporate feedback from past conditions, and not simply respond to the condition at the particular moment a control decision is made.

Based on the literature, from a comfort perspective the most effective control strategies for high thermal mass radiant systems are those that operate at constant flow and variable temperature (based on the outside air temperature). These feed-forward, model-free approaches do not incorporate feedback and their performance depends on how closely the design matches reality. Additionally, in the majority of applications, heating and cooling can occur throughout the 24 hour period and thus the approach does not take full advantage of the thermal inertia of the system. None of the prior work presented a control strategy that explicitly limits the operation period of the radiant system, and controls using historical feedback from the zone, without using MPC approaches. Presenting and assessing such a control strategy is the aim of this paper. This new controller for radiant systems locks out the operation of the radiant system during predefined hours of the day, and then uses a zone air temperature feedback loop cascading onto a slab temperature feedback loop, to maintain the zone at comfortable conditions (more details below).

Method

Description of the models

We evaluate the controller using whole building energy simulation tools. We selected EnergyPlus as the simulation engine as it implements the full ASHRAE Heat Balance method while providing a detailed model at zone, system and plant levels (Crawley et al., 2008), and has a validated radiant system module (Chantrasrisalai et al., 2003; Strand and Pedersen, 1997). We built a single zone model using EnergyPlus using the Python eppy package (Philip, 2016), representing a single California Title 24 compliant zone in the middle floor of a large office building. Virtually every input parameter can be modified programmatically, which will allow us to perform a wide reaching sensitivity analysis across climates, geometry, constructions, loads, and schedules. However, for this paper we focus our assessment on the zone described below.

The single zone's dimensions are 25 x 5 x 3 m, with a total area of 125 m^2. We chose to simulate the model with climate data from Sacramento, CA corresponding to California climate zone 12. The cooling design day peak dry and wet bulb temperatures are 38 °C and 21.5 °C, respectively. The heating design day values for both temperatures is -0.3 °C. Table 1 describes the properties of the building zone and its heat gains. We defined non-regulated internal heat gains and schedules in our model with values found in the DOE large office prototype building model (US DOE, 2013), which is defined for a typical building compliant to ASHRAE 90.1-2013 standard. Non-regulated components include occupant density and plug loads. Title 24 non-regulated heat gains listed in Table 1 are from the Title 24 Nonresidential ACM Reference Manual (California Energy Commission, 2016). We defined the South façade, along with its window, as the only exposed surface to outside weather conditions. We defined this exterior wall as a medium thermal mass wall with three layers. The outside layer is normal weight concrete with thickness, thermal conductivity, specific heat, and density of 100 mm, 1.2 W/m·K, 800 J/kg·K, 2240 kg/m^3, respectively. The middle layer is an insulation layer with thickness, thermal conductivity, specific heat, and density of 59 mm, 0.03 W/m·K, 1,500 J/kg·K, 15 kg/m^3, respectively. The inside layer is plasterboard with thickness, thermal conductivity, specific heat, and density of 13 mm, 0.16 W/m·K, 1090 J/kg·K, 800 kg/m^3, respectively. The total U-value of the wall is 0.35 W/m^2·K. The other three sides of the zone have an adiabatic boundary condition. The floor and ceiling are thermally interconnected to represent a middle floor zone. The window-to-wall ratio is not the maximum allowed by Title 24 prescriptive approach; instead we defined it at 20%, a more reasonable design value for a TABS system. Larger window area requires additional steps in the building design in the form of shading, increased indoor air speeds to ensure occupant comfort, or increased capacity in the ventilation system. Infiltration is 0.537 l/s per area of exterior surface (0.64 ACH) and reduces to a quarter of the value during operation of the ventilation system, as in the case of the benchmark models (US DOE, 2013). We assume occupancy to be from 8.00 to 18.00. We set the operative temperature comfort bounds to 22.3 °C and 26 °C. This corresponds to -0.5 to +0.5 predicted mean vote (pmv) at an air speed of 0.1 m/s, relative humidiy of 35%, occupant metabolic rate of 1.2 met, and a clothing insulation of 0.7 clo. Any of these values can change and can be updated to have a dynamic comfort setpoint throughout the year but we chose to define constant setpoints for simplicity. The ventilation heating and cooling coils are available one hour prior to occupancy with fans available an additional 15 min earlier. We defined the ventilation system with dual setpoints at 15 °C and 21 °C. The ventilation airflow is oversized by 30% which is common practice for DOAS systems to receive credits under rating systems such as LEED. The design day peak instantaneous heat gains are 38.5 W/m^2. The design day peak and 24-hour average cooling load, as calculated using an all-air system controlling to the same operative temperature comfort bounds, are 27.5 W/m^2 and 11.3 W/m^2 respectively.

The TABS case has a pipe diameter, spacing, and depth of 15.9, 152.4, and 76.2 mm respectively. The ceiling and floor slab thickness is 203.3 mm and made of normal-weight concrete. We define the concrete with thermal conductivity, specific heat, and density of 1.8 W/m·K, 900 J/kg·K, and 2240 kg/m^3 respectively. The floor slab has commercial flooring with a thermal resitance of 0.217 m^2K/W. The total water flow rate was also constant at 0.759 L/s, or 0.06 L/s for each of the 12 tubing circuits (or 'loops') in the slab.

The above radiant system is only feasible in new construction. We also simulated two more radiant systems that can potentially be implemented in existing buildings with the addition of PEX tubing embedded in a topping layer above the structural slab. The topping layer and the structural slab may or may not be decoupled through the addition of an insulation layer and the decision may depend on local building codes. ISO 11855 defines a coupled topping layer as TABS while a decoupled layer is defined as ESS. A lack of insulation will activate more of the building's thermal mass, and is the preferred approach to increase the thermal mass activated by the radiant system. We defined a 76.2 mm concrete topping layer with the same thermal normal weight concrete properties described above. The tubing is located at the bottom of the topping with over 60 mm of screed between the top of the tubing and the floor surface. The diameter and the spacing of the pipe remain the same as in the TABS system, as well as the water flow rate, thicknesses and thermal properties of the structural slab, etc.

Table 1: Properties of office single zone and its internal loads. Zone complies with Title 24-2013 prescriptive limits. ASHRAE 90.1-2013 standard requirements are also shown for comparison.

South exterior wall	**Title 24-2013**	**ASHRAE 90.1-2013**
Materials		
U-value	0.35 $W/m^2\,K$	0.44 $W/m^2\,K$
South window		
WWR ratio	Maximum = 40% Model = 20%	Maximum = 40%
U-value	2.0 $W/m^2{\cdot}K$	2.8 $W/m^2{\cdot}K$
SHGC	0.25	0.25
Loads		
Lights	8.6 W/m^2	8.8 W/m^2
Plug loads	14.4 W/m^2	8.1 W/m^2
People	9.3 m^2/person	18.6 m^2/person
Ventilation	Maximum of 7.08 L/s per person or 0.07 L/s per m^2	Sum of 2.5 L/s per person and 0.3 L/s per m^2

As the focus of this paper is to evaluate control strategies at the zone level, we defined the mechanical plant with district heating and cooling to supply the water temperatures to the radiant system. We used a coeifficient of performance (COP) of 5.55 for cooling and 4.2 for heating to make approximate conversions to electricity consumption. We defined a two-stage direct exchange (DX) coil for the ventilation system to ensure that any dehumidification requirements can be met without requiring a lower water temperature for the entire cooling water plant, as would be the case if the ventilation system used a water coil. We set the COP of the DX coil to 3.58 for the first stage and 4.2 for the second. We assumed there is sufficient capacity to supply the requested water temperature at all times. These are the minimum efficiency values for a centrifugal chilled water plant (less than 300 tons) and for heating it is the minimum efficiency value for a water source heat pump required by code in California (California Energy Commission, 2013). The same applies for the DX coil efficiencies. Future research will evaluate different plant designs and how the effectiveness of the plant changes.

As EnergyPlus provides relatively limited control options for radiant systems, we chose to implement the control strategies in the C programming language to provide full flexibility. We then used the Functional Mockup Interface to control the radiant system in EnergyPlus in co-simulation time, using the EnergyPlus External Interface objects and the Qtronic FMU-SDK (Qtronic, 2016).

Note on load variation

The majority of whole building energy simulation work uses deterministic load schedules, though large collaborative efforts such as IEA Annex 66 indicate that the research field is gradually moving towards stochasticly defined loads. However, all simulation papers on radiant system control strategies to date have used deterministic internal load profiles. When combined with highly efficient envelopes and shading systems, this yields load profiles that often vary little from day to day, and in some cases throughout the year - the only variation is due to conduction through the (well-insulated) exterior envelope. However, real buildings have varying internal loads both from day-to-day and throughout the year. This will significantly affect the outcome of the feedforward control strategies that use pre-defined values (often based solely on the outside air temperature), and that do not incorporate a feedback loop that responds to the actual zone conditions. To the best of our knowledge this effect has not been captured in simulation studies of radiant systems to date.

Although well-designed shading is highly recommended for TABS systems, we purposefully did not include shading in this model in order to ensure that the model included significant intra-day and intra-month variation in loads. Though excluding solar shading does not provide truly 'realistic' load variation (e.g., modelling internal loads that vary stochastically) it does come significantly closer to doing so than using fixed internal load schedules and a highly efficient shading.

Sensitivity analysis

We performed a small full factorial sensitivity analysis to evaluate a range of control strategies for the model described above. We tested 36 cases for the new controller: three different cooling supply water temperatures (16, 18, and 20 °C) and at 12 different periods in which the radiant system is not available to operate ('shutoff' periods that are 12 hours long, moving forward in increments of two hours – e.g. midnight to noon, 2:00 to 14:00, etc.). We present those results against 3 cases of a 'typical' controller that is common in the USA, operating at those same three water temperatures. As Sacramento is cooling dominated, and heating performance is not the primary focus of this paper, we kept a constant hot water temperature of 30 °C.

Description of the new controller

We developed a controller that responds to both zone and slab temperature conditions, and allows a user to specify periods during the day in which the radiant system can not operate. Figure 1 shows a schematic diagram of the controller in cooling mode. The primary control loop is an on/off controller that controls the radiant system valve in response to the error between temperature sensor in the slab, placed close to the surface, and the slab setpoint. We initialize the model with heating and cooling setpoints of 19 °C and 23 °C, respectively.

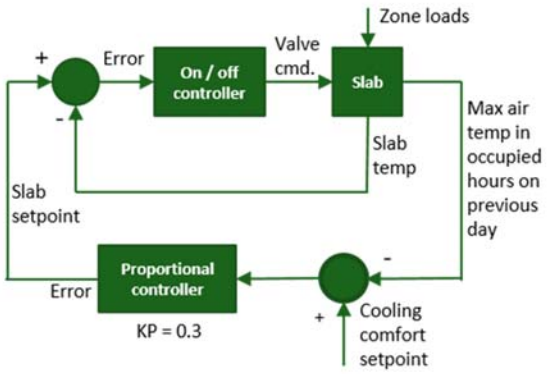

Figure 1: Schematic diagram of the controller in cooling mode. The same approach applies in heating mode, but using the minimum instead of maximum air temperature on the previous day, and heating instead of cooling comfort setpoint.

A secondary cascading control loop uses a proportional controller to control the primary loop setpoint. It operates using the error between the maximum/minimum zone air temperature during occupied hours on the previous day relative to the comfort setpoint for cooling/heating. This secondary controller activates once at the end of the occupied period each day, and the controller gradually makes the change to the slab setpoint over the next 12 hours. The comfort setpoint is 0.5 °C above and below the heating and cooling limits (respectively) of the comfort bounds defined for the zone. In this way, the controller gradually responds to changes in the zone loads over the course of several days.

In addition, the controller can only operate in one mode each day - either intermittent cooling, off for the entire day, or intermittent heating. This ensures one entire day between mode changes, avoiding wasted energy use from heating and cooling during the same day.

Lastly, the designer (or potentially even an operator in response to long term changes in electricity pricing periods) selects a period in which the radiant system does not operate - e.g., shutoff from 14.00 to 2.00. This allows the designer to design a cool and warm water plant that is optimally selected for the conditions that occur during the hours when the radiant system may operate. The most salient example here is to design the system to benefit from lower nighttime drybulb and wetbulb temperatures to generate cool water using an evaporative cooling tower instead of a chiller.

Figure 2 shows a week of data from Aug 27 to Sep 3 to illustrate how this controller operates for the single zone described in the previous section. The results from this entire annual simulation, as well as the results for the lowest comfort and lowest energy cost cases, are also available for readers to interactively explore in html files (here[1] or here[2]), on the University of California's open-source eScholarship platform. The particular case shown below uses a supply water temperature of 20 °C and a shutoff period from 14:00 to 2:00. The beginning of the selected period shows the radiant slab operating at almost peak capacity (e.g., water flowing through the tubing for the entire 12 hours of possible operation) and maintaining comfortable conditions. It also highlights that the controller responds well to load variation, as the slab cooling setpoint increases when zone loads (and zone temperatures) decrease during the subsequent week. Compared to existing control approaches for high thermal mass radiant systems, this removes the need for manual trial-and-error modification of setpoints, operation schedules, and other parameters, such as feedforward approaches that define the supply water temperature based on the outside air temperature. Additionally, this controller uses the supply water temperature directly from the plant instead of mixing locally at the zone as with controllers in which water temperature is the control variable. Thus, this control approach does not require mixing valves or small pumps at the individual zone level, providing significant initial and ongoing maintenance cost savings. Furthermore, this approach reduces pumping power (i.e., cumulative time that water flows through the PEX tubing) by always using the maximum possible temperature differential available between the room and the supply water temperature coming directly from the cool or warm water plant. Finally, though this controller does not find the optimal control solution by any means, it is far better than current practice, and it has the major advantage that it can be implemented within a BAS, by a typical controls contractor, without the complexity of MPC and associated practicality and robustness concerns.

Surface temperature versus in-slab temperature

The interactive online versions of Figure 2 (here[3] or here[4]) also show the temperature of the slab construction at the depth at which the tubing is located ('Slab core') for reference. As with all whole building energy simulation tools, EnergyPlus represents heat transfer through surfaces using the one dimensional heat transfer simplification. Thus, any property (temperature, heat flux, etc.) at a particular depth within the surface is an average throughout the entire surface at that depth. In most cases, this is not an issue as the properties are reasonably

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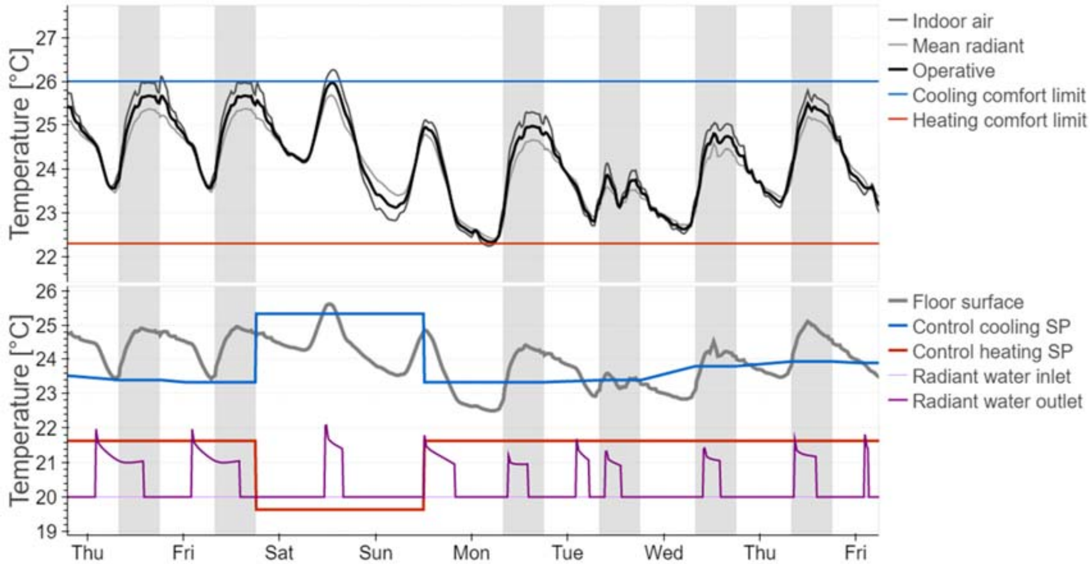

Figure 2:Time-series example showing peak system capacity operation with 20 °C chilled water supply temperature for several summer days for the TABS case. The radiant system is shut off from 14:00 to 2:00. (Top) indoor zone comfort conditions, (middle) radiant system response and setpoint adjustment, and (bottom) fluid flow rates and system operation. Shaded area shows the occupied hours, from 8:00 to 18:00.

uniform in reality. However, this is not the case with a high thermal mass radiant system – when water flows through the slab, the temperature of the water is different from the temperature of the surrounding concrete. This is what causes the 'Slab core' temperature to change almost as a step function when water is flowing through the slab, and when it is not. Thus, this output should not be used in the simulation control strategy as the output is not realistic when compared to a temperature sensor embedded in the slab at this depth. Instead, we use the floor surface temperature as the process variable in simulation. However, in real buildings, temperature sensors are typically installed at or near the same depth as the tubing, and certainly a few inches below the surface. This discrepancy makes direct comparison of control strategies in simulation and reality very difficult.

Description of the typical practice controller

For comparison, we compare the results of the new controller to those from a simple on/off controller. This controls the valve position using a fixed air temperature setpoint of 21.1 °C and 23.9 °C for heating and cooling, respectively. The radiant system is available to operate 24 hours per day. This represents a relatively common practice in the US for these types of systems, based on designer surveys (Higgins et al., 2015), as well as field studies of several radiant buildings performed by the authors. Some of the field studies used proportional valves instead of two position valves, and thus used a proportional controller instead of an on/off controller. The proportional band was typically 0.6 °C on either side of the setpoint. We did not include this controller in the results as we found that this approach led to heating and cooling operating during the same day, and poorer results overall (Bauman et al., 2015).

Results and discussion

Given the Sacramento climate, typical office internal heat gains, and the highly-insulated envelope, the zone spent most of its time in cooling mode, or off. Thus, most this discussion focuses on analysing the cooling results.

Figure 3 shows the annual summary results for all simulations. We calculated the energy cost using actual electricity tariff data available from a large utility provider in California (Pacific Gas & Electric). One important point to note here is that the energy costs do not include the electricity *demand* charges. The peak monthly demand level is calculated at the building/campus electricity incomer, and it is not reasonable to do that calculation at a single zone level. However, the electricity demand charge savings for the all nighttime operation cases would be significant compared to cases where the electricity usage related to the cooling/heating plant operation coincides with a higher electricity base load that typically occurs during occupied hours.

Figure 3 clearly shows that the new controller is an improvement over current industry practice. For both TABS cases, and for it maintains comparable or better comfort conditions while reducing energy consumption. It also significantly reduces cost of operation under time-of-use electricity tariffs because it allows the designer/operator to explicitly specify particular hours in which the radiant system is shut off, shifting the time at which heating and cooling related electricity use occurs, while responding automatically to maintain comfort. This is of particular relevance given the changing nature of net demand on the electric grid over the coming years, and the inherent ability of high thermal mass radiant systems to temporally shift these loads.

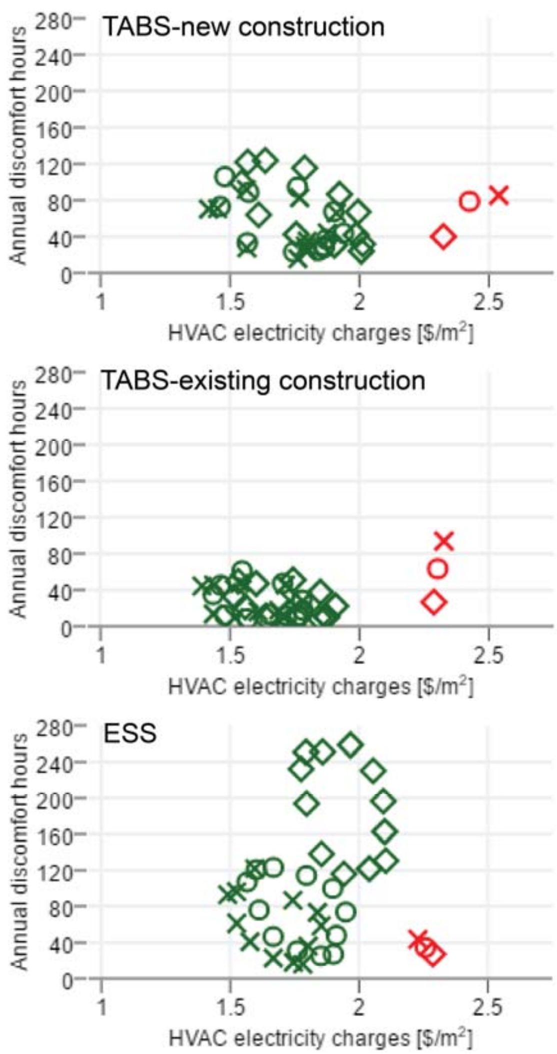

Figure 3: Annual summary results showing comfort, energy consumed, and electrical energy cost of the same radiant system using different controllers. The TABS-new construction, TABS-existing construction, and the ESS case are the upper, middle, and lower plots, respectively. The ×, ○, and ◇ shapes represent chilled water temperatures 16, 18, and 20°C, respectively. Green represents the new controller strategies while red represents typical radiant system control. The other variable for the new controller is the shutoff period – 12 cases for each water temperature in which the shutoff period moves forward in 2 h increments.

Interestingly, discomfort hours increase with decreasing cooling supply water temperature for the typical control strategies. As one would expect, though not shown in the figure, the cumulative pumping hours of the radiant slab also decreases with decreasing water temperature. Overall energy use increases with decreasing supply water temperature for these control strategies – the effect of cooling and heating occurring within the same day counteracts the reduced pumping power. In the case of the ESS system and the typical controller, the system is less directly coupled to the entire zone's thermal mass, and thus can more quickly change temperature in response to a change in operative temperature in the zone, with less on an energy penalty caused by heating and cooling within the same day.

In contrast to the typical controller, for the new controller, both energy use and discomfort hours decrease with decreasing water temperature as it can more effectively use the higher temperature differential provided by the lower water temperature, without exceeding the comfort bounds. The worst performing cases for the new controller are those that use the 20 °C cooling supply water temperature. This temperature is sufficient to maintain comfort for the typical controllers, as water can flow through the radiant slab 24 hours per day if necessary. However, for the new controller, where the radiant system is constrained to operate for a maximum of 12 hours per day, the operative temperature on the cooling design day for the TABS-new construction case typically causes more discomfort hours. This effect is less of an issue in the TABS-existing construction case as the zone has significantly more thermal mass. The effect is especially true in the ESS system in which a higher supply water temperature and the insulation between the tubing and the slab allows less thermal energy to be stored in the structural slab.

Figure 4 illustrates that the hours during which cooling occurs also clearly have an effect on the range of operative temperatures in the zone within each day. Water temperature also has an effect but it is far less significant and thus we omitted it from the figure for clarity. Again, the zone is predominantly in cooling mode throughout the year, and so these results are not relevant for heating mode. It is clear that the maximum range occurs when the radiant system significantly pre-cools the zone (e.g., radiant system shut off from 8:00 to 20:00). This strategy generally corresponds to a higher number of comfort exceedance hours, and lower energy costs, in the annual summary results for the new controller shown in Figure 3.

Interestingly, the *minimum* daily range of zone temperatures occurs when the radiant system shutoff period is from 14:00 to 2:00. Another way of stating this is that when the radiant system cooling operation period *leads* (i.e., occurs earlier than) the zone cooling load by approximately 4 hours - an 'afternoon shutoff' strategy - this provides the most uniform comfort conditions in the zone. For each water temperature, the simulations with the lowest annual discomfort hours (in Figure 3) were those in which the radiant system was shut off during the afternoon. While the energy cost savings are not as high as the full nighttime precooling strategy, this approach still benefits from avoiding cooling during most of the peak electricity price period (from 12:00 to 18:00 for the PG&E utility tariff), while providing the most uniform comfort conditions throughout the day in the zone.

Figure 5 illustrates the effect that different radiant system operation periods have on the range of operative temperature during occupied hours by focusing on the

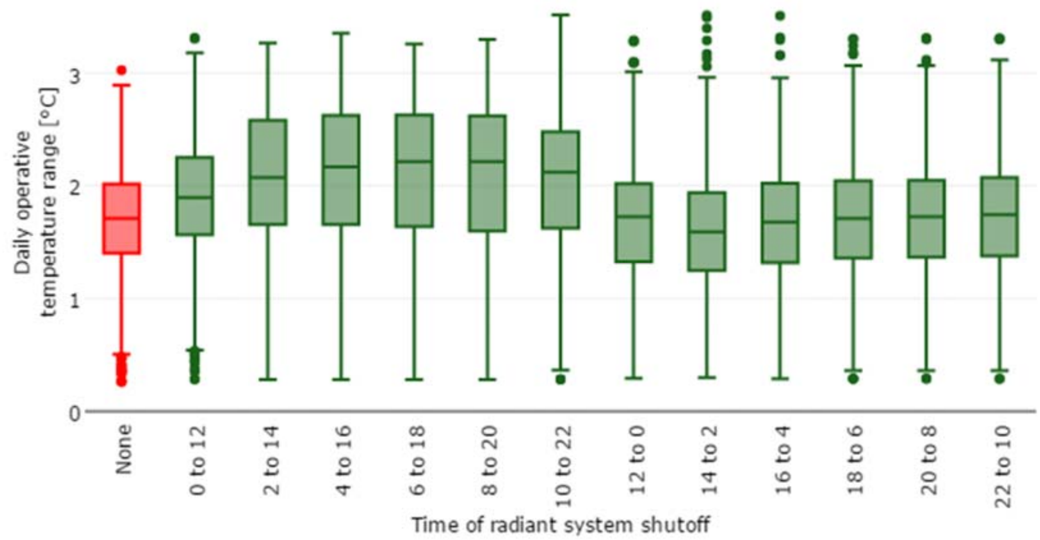

Figure 4: Boxplots showing the distribution of the daily range of zone operative temperature during occupied hours for each annual simulation, for each water temperature, for each shutoff period, for the TABS-new construction case.

cooling design day. To further illustrate this effect, in the digital files associated with this publication ([here](#)[5] or [here](#)[6]), we also include three separate design day animations (one for each cooling supply water temperature) which show the effect of moving a fixed radiant system operation period in increments of one hour.

Simplified method to estimate optimal lead time

The lead time that generates the most uniform comfort conditions is clearly a valuable piece of information for a designer. However, it may not be feasible to perform an an annual whole building energy simulation to discover what this lead time is.

One way to estimate the amount of lead time to provide the most uniform comfort conditions is to use two simplifications: that the zone and slab can be approximated by a first order lumped capacitance system, and that both control *disturbances* (i.e., zone cooling load) and control *inputs* (i.e., hydronic cooling load handled by the radiant system) are sinusoidal within a 24 hour period

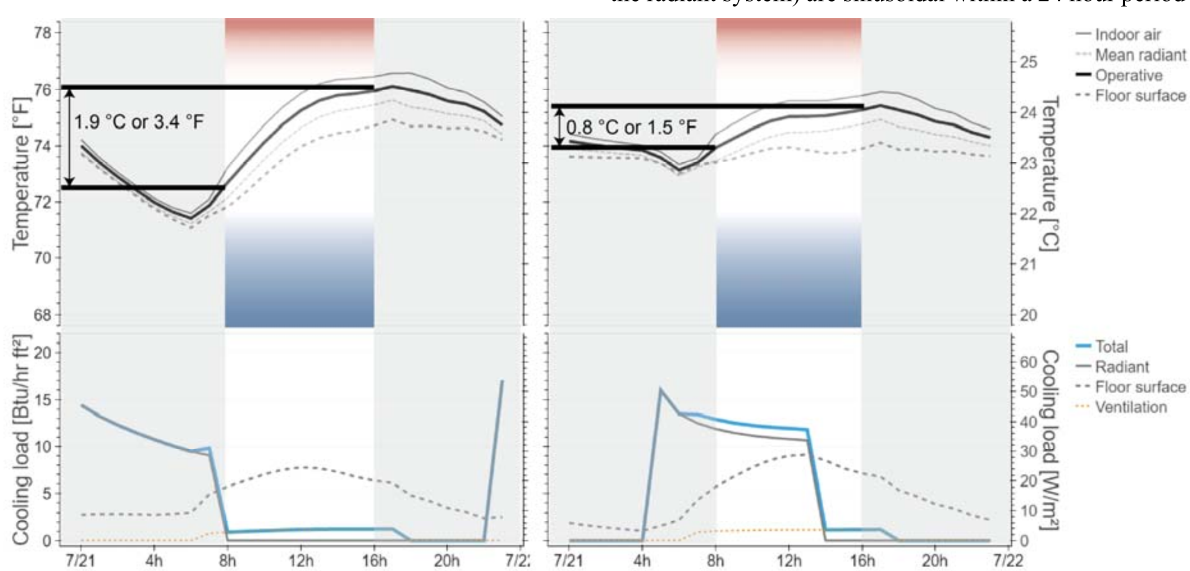

Figure 5: Design day simulation for the TABS case showing the effect of varying the time at which a fixed 9-hour period of radiant system operation occurs – (left) night time precooling and (right) early morning precooling (i.e. afternoon shutoff). The underlying model is identical in both cases, and identical to that presented earlier in the paper. The cooling supply water temperature is 18 °C for these two design day simulations.

[5] berkeley.box.com/s/fj4jkb477sh688n28stls1c7klxjekgq

[6] www.escholarship.org/uc/item/5tz4n92b

(T). Clearly these are major simplifications. The slab alone is at the very least more appropriately modeled as a second order system, though first order is a reasonably accurate approximation unless there is a very large tubing spacing or the tubing is close to the surface. However, the cooling loads are clearly not perfectly sinusoidal, nor is the hydronic cooling load.

Using the equation for frequency response of a first order system with time constant, or response time, τ_{63}, the lead time phase angle (ϕ) is:

$$\phi = -\text{atan}(2\cdot\pi\cdot\tau_{63}/T) \tag{1}$$

Or, we can also present this estimated lead time in hours, t_L:

$$t_L = \phi\cdot T/2\cdot\pi \tag{2}$$

Using a simplified first order lumped capacitance model:

$$\tau_{63} = \rho\cdot c\cdot d/h \tag{3}$$

where ρ is the conductivity of the concrete, d is the depth of slab, c is the specific heat capacity, and h is the sum of the combined heat transfer coefficient for both floor and ceiling. More accurate estimates require more complex approaches, such as those decribed in Ning et al., 2017. For the TABS system described in this paper, in which τ_{63} is 6.3 h, using this method estimates that the most uniform comfort conditions will occur when cooling operation leads the cooling loads by 3.9 h. This is in approximate agreement with the results of the EnergyPlus simulations (4 hours), though we cannot say how close the results truly are as the sensitivity analysis only evaluated the new controller with shutoff periods that vary in steps of 2 hours.

For a wide range of typical concrete floor constructions and cooling load profiles, the optimal lead time will be between 3 and 5 h - for less and more massive systems, respectively. Note that regardless of the mass of the system, the phase angle will never exceed a lead time of 6 h (i.e., a phase angle of 90°) for loads that vary with a typical 24-hour period.

This simplified approach could be improved upon by doing a frequency domain analysis of the cooling loads and using harmonics to better represent them than a single sinusoid. However, that adds a lot of complexity, requires *a prediction of the load profile in advance*, and is unlikely to be used by a design engineer. There's also a very real limit to what level of load variation a slab system can respond to, given the thermal inertia involved. The above simplified approach treads a fine line between a more complex implementation that takes a lot of time to define for each zone but yields an accurate result, and a simple approach that yields an approximate result quickly.

Retrofit applications

Note that we included three cases for the radiant slab design. In one, the radiant system is a typical TABS design for new constructions, with the PEX tubing embedded in the structural slab. In the other two, we modeled a design in which the PEX tubing is in a topping layer above an existing floor slab. In one approach, the topping layer is thermally connected to the structural slab and in the other it is decoupled through the use of insulation between both layers; the former called TABS-existing and the latter called ESS in Figure 3. Figure 3 shows that both types of radiant systems are feasible options in major retrofits of an existing office building from a HVAC perspective. Other design considerations that need to be taken into account include the structural design of the existing building. An additional concrete topping layer adds a significant amount of weight which may become critical, particularly in a multi-story high story building.

In the US, existing office buildings predominantly use Variable Air Volume systems and have high floor-to-floor heights to accommodate ductwork in the return plenum above the drop ceiling. By installing a radiant floor in a thin topping slab layer on the existing floor, and removing the drop ceiling, the concrete ceiling (of the floor above) is exposed to direct radiative exchange with the floor and the heat transfer dynamics are similar to those of a TABS system. Therefore, these buildings may also be candidates for this type of precooling control strategy in California.

Topping layer materials

Aside from the results presented above, which use a concrete topping layer, we also simulated a screed topping layer – also a common construction material for topping slabs. We defined the screed with thermal conductivity, specific heat, and density of 0.41 W/m·K, 840 J/kg·K, and 1200 kg/m^3, respectively. These results are not shown in this paper. However, they indicated that a screed topping layer has acceptable performance in the TABS-existing case but not for the ESS case. A 16 °C supply water temperature with typical control resulted in lowest discomfort hours (240 hours) for all simulated cases with ESS and a screed topping layer. This is still far higher than any of the other cases. The discomfort hours reached above 1,500 hours in cases that use the new controller since it is restricted to only operate 12 hours per day. The poor performance of the ESS case with a screed topping layer is because screed is not an effective heat transfer medium when compared to concrete – it's thermal conductivity is four times lower when compared to concrete. This is an important factor that affects surface temperature and ultimately heat transfer with the space. Interestingly, the TABS-existing case with a screed topping layer was still able to maintain acceptable comfort conditions with either controller. It benefits from the fact that the tubing is at the bottom of the topping layer. This allows more heat transfer into the structural slab from the PEX tubing, allowing it to be precooled and to engage in thermal storage.

In general, our conclusion regarding a screed topping layer is that if insulation is used to decouple the structural slab and the topping layer, then a topping layer that has better thermal heat transfer properties must be used in order to maintain occupant thermal comfort. An alternative is to supply lower water temperature to the radiant system, or to run the system for a longer period.

Discussion

It is worth highlighting that there are many advantages to night time cooling operation of the radiant system. Many are commonly noted in previous studies, such as that it is typically the most energy efficient time for generating cool water as outside drybulb and wetbulb temperatures are the lowest during the 24-hour period, and that this period also has the lowest electricity consumption prices (a $/kWh charge). However, other benefits are not commonly noted in the literature. For example, many utilities around the world also apply a demand charge - a $/kW charge on the peak building electricity use each month. This can be significant, typically 20-40% of the electricity bill in California. It is very unlikely that peak building electricity use will occur during the night and thus, operating the radiant system during the night will entirely avoid the electricity demand charges that are associated with cooling in a more traditional HVAC system.

One other major potential advantage is that in hot/dry climates it is often feasible to generate cool water at sufficiently low temperatures during the night using only evaporative cooling sources for the entire year. This avoids the significant initial and ongoing cost of a chiller. For example, Figure 6 shows that the 99th percentile of night-time wetbulb temperatures are below 18 °C in many California climates. Thus, it is possible to generate 20 °C cool water with a 2 °C cooling tower approach temperature all year round in these climates.

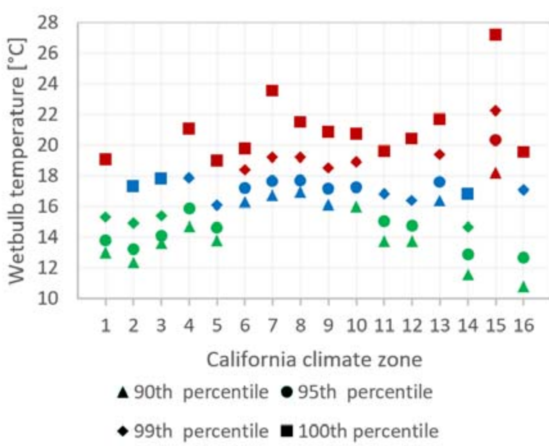

Figure 6: Upper percentiles of annual outside wet-bulb temperature distributions between the hours of 21:00 to 9:00 in all 16 California climate zones. Green and blue colors indicates the approximate feasibility of a design using only an evaporative cooling tower to supply 18 °C and 20 °C water respectively during this period, for each percentile.

This implies that a chiller-less building design is possible in these cases. If increased zone air movement (to expand the upper comfort bound) and/or lower approach temperature (or sub-wetbulb cooling technologies) are part of the design, it is feasible to remove the need for a chiller for office buildings in almost all California climates. Even where it is not possible to entirely remove the need for a chiller, it is still possible to generate water under 20 °C for over 90% of the nighttime pre-cooling hours of the year for all but one California climate (15, Brawley).

Future work

Further improving and evaluating the controller

We believe that using a running mean as the process variable for the primary control loop will provide more stable operation, and further improve the controllers' performance. We plan to further investigate this along with other improvements to the new controller, and evaluate them against other common control strategies. For example, the common practice in Europe: constant flow, variable temperature based on outside air conditions.

Designing without chillers in California

We will further investigate the feasibility of designing chiller-less office buildings in California. There are example buildings already constructed and successfully operating in milder climate zones, such as the David Brower Center in Berkeley (CZ 3) that prove that this is possible. Our aim is to provide resources for designers that allow them to apply this approach throughout the 16 California climate zones.

Load variation

Another key question that we have is how a more realistic representation of internal loads will affect the results. Though we approximated this effect by excluding shading in our model, future work will include stochastically defined load schedules that represent typical load variation in office zones and well-shaded fenestration.

Release EMS version of controller

Later this year we plan to publicly release a version of the new controller that uses EnergyPlus Energy Management System (EMS) objects instead of the co-simulation approach used in this paper, so that this controller can be more widely used by practitioners in simulation-assisted design.

Conclusions

The results show that the new controller has lower energy cost and equal or better thermal comfort compared to common control strategies used in the US. This applies for the TABS cases in new and existing buildings, and for the ESS cases at the lowest water temperature. The results also show that the controller is flexible and the designer can implement it with different shutoff hours to accomplish different performance goals. For example, the designer can maximize the use of the thermal comfort range that will minimize energy consumption, or maintain more uniform zone temperature conditions throughout the day. For the model used in this study, the lead time that provided the most uniform temperatures was 4 h, and a simplified analysis that this 'comfort optimum' lead time will range from 3-5 hours in other high thermal mass radiant systems with different construction properties. The appropriate lead time may also be used to increase the amount of heat gains that the radiant system can handle, though we did not investigate this phenomenon. The results of a sensitivity analysis show that when compared to common practice in the US, this approach reduces electricity cost and energy consumption by up to 40% and 35%, respectively, while maintaining comparable comfort conditions in the zone. However, these use a highly simplified cool and warm water plant model. In reality, this design & control approach could eliminate the need for a chiller in many California climate zones by using evaporative cooling during night time hours. Even in the most extreme climates, an office building could operate in free-cooling mode for over 90% of the year. This would yield enormous energy and cost savings compared to current approaches to HVAC systems for office buildings in California.

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