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A multi-method investigation into design and control of radiant cooling and heating systems

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A multi-method investigation into design and control of radiant cooling and heating systems

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Abstract

A multi-method investigation into design and control of
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Radiant cooling and heating offers compelling building energy performance opportunities compared to conventional cooling and heating systems. First, radiant systems operate with more moderate supply water temperatures, and – if designed and controlled strategically – can achieve better cooling and heating plant efficiency than conventional systems. In many scenarios, the supply water temperature needed for cooling is warm enough that an evaporative fluid cooler (water-side economizer) can be used in lieu of vapor-compression equipment. Second, whereas the timing and magnitude of cooling or heating plant loads for conventional systems are basically equivalent to the moment-to-moment needs for cooling or heating indoor spaces, high thermal mass radiant systems naturally decouple space heat transfer rates from plant heat transfer rates. As a result – if designed and controlled strategically – high thermal mass radiant systems can: utilize smaller plant equipment; operate more often at part capacity; and/or shift the timing of plant operation to periods with lower electricity prices or more favorable outdoor conditions.

However, these benefits are not unreservedly guaranteed. In fact, industry standard design procedures and the common tools for estimating cooling and heating loads obscure these opportunities and can lead designers to select systems that: operate with less favorable supply water temperatures, require larger equipment than necessary, operate more often at full load, and concentrate plant operation during periods with higher electricity prices and less favorable outdoor conditions.

For this dissertation we used several methods to investigate the design, control, and energy performance of radiant systems. First, we reviewed literature from simulations and case studies of radiant buildings which have built a strong case for the possible energy performance benefits of the technology. Then, by analyzing energy use intensity data from a statistically relevant sample of buildings we found that in practice, large office buildings with radiant cooling in mild climates are using 31% less energy on average than comparable existing building stock. This difference is almost certainly due to a variety of factors, but radiant cooling is very common among those buildings with the lowest energy use intensity.

Then, we conducted structured interviews with design professionals experienced with high thermal mass radiant cooling systems and documented the design and control strategies they

typically employ in practice. We discovered that although there are many similarities among the strategies used, there is not consensus on best practices, and there is a lack of standards, tools, and guidelines to support design and control of high thermal mass radiant systems. Our interviews also revealed many opportunities to improve design and control that could reduce cost and improve energy performance of high thermal mass radiant. systems.

Next, we performed experiments to compare the space cooling loads for radiant and all-air systems. We conducted a series of very accurate, realistically scaled, multi-day, side-by-side tests that measured the space cooling rates required for each system to maintain equal indoor comfort conditions. These experiments proved that the peak space cooling load for a radiant system is larger than that of an all-air system (2–21%), and that in some cases the cumulative cooling load can be much larger than that of an all-air system (2–40%). These differences occur because a portion of the heat that would be absorbed by non-active surfaces in a building with all-air cooling, is instead extracted by radiation heat transfer with the internally cooled surfaces. Then, since less heat is stored in non-active thermal masses, less heat can be released passively to the environment when there is an opportunity to do so. Consequently, the differences are largest in scenarios with an opportunity for passive cooling overnight – such as with natural ventilation night pre-cooling. Importantly, these findings prove that the magnitude and timing of cooling loads for a space depend on the type of system and control strategy that is used to provide cooling – factors that are conspicuously absent from standard cooling load calculations and the associated system design procedure.

In view of these findings, we developed a thorough critique of the standard definition of "space cooling or heating loads" and the associated system design procedure, then we developed a new definition and an improved system design procedure. We argue that in addition to omitting important heat transfer fundamentals, the standard design procedure fails to account for the impact of system controls, does not facilitate many important design objectives, and imposes simple constraints that overlook fundamentals about thermal comfort. To resolve these issues, our improved approach shifts the focused objective of the system design procedure away from satisfying a singular – "ideal" – space cooling or heating load, and instead orients the designer toward selecting and sizing components and their controls that best satisfy performance objectives such as thermal comfort, indoor air quality, or life cycle cost.

Finally, to demonstrate the practical consequences, we simulated a high thermal mass radiant system designed with the standard procedure and compared it to several example design alternatives developed with the revised procedure. This comparison showed that the improved design procedure: reduced the size of cooling plant equipment by as much as 50%, increased median cooling supply water temperature by as much as 5.2 °C (9.4 °F), reduced chilled water consumption during periods with high electricity prices by as much as 100%, and reduced annual occupied discomfort hours by as much as 55%.

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I could not have wished for more exceptional team members than those individuals I collaborated with most closely in conducting and co-authoring research as a doctoral student. Stefano Schiavon, Paul Raftery, and Fred Bauman from the Center for the Built Environment were incredible mentors and were also unpretentiously involved in the fine details of research planning, execution, and interpretation for all aspects of the research presented herein. I am happily indebted to many co-authors on all publications with which I was involved as a doctoral student, including: Carlos Duarte, Megan Dawe, Gwellen Paliaga, Farhad Farahmand, Marco Pritoni, Spencer Dutton, Haida Tang, Jingjuan (Dove) Feng, and Jovan Pantelic. Each of these individuals are admirably creative, productive, and cooperative research colleagues. I am grateful to Mark Modera and Gail Brager for their gracious encouragement, constructive feedback, and pragmatic leadership.

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INTRODUCTION

In consideration of pressing interests to develop a more responsible global energy system, improving the efficiency of heating, cooling, and ventilation in buildings is a critical objective. Buildings account for 32% of global final energy use, and 19% of global energy-related greenhouse gas emissions; 25% of this energy use is from commercial buildings and 75% is from residential buildings (IEA, 2013; Lucon et al., 2014). Overall, heating, cooling, and ventilation are responsible for 40% of the final energy use by commercial buildings (IEA, 2013; Lucon et al., 2014). However, the importance of heating and cooling varies throughout the world, accounting for between 18%–74% of final energy use by commercial buildings in different regions (Ürge-Vorsatz et al., 2015). In the United States, heating, cooling, and ventilation are responsible for 43% of site energy use by commercial buildings (DOE, 2012; EIA, 2012). In some consistently hot climates, cooling alone can account for more than 60% of the final energy consumption associated with commercial buildings (Chou, personal communication, 2020; Chou & Lee, 1988; Chua et al., 2013; Turiel et al., 1985).

Additionally, global energy use by buildings could triple by the middle of this century due largely to increased access to energy resources by billions of people in developing countries (Lucon et al., 2014). In part because of climate change, and in part because projected growth in energy access is concentrated in hotter climates, demand for cooling and ventilation is expected to account for the largest increase in global electricity consumption between 2020–2050 (IEA, 2018). Cooling and ventilation is currently responsible for 10% of global electricity consumption, but could account for 37% of the projected increase in global electricity consumption between 2020–2050 (IEA, 2018). The potential growth in energy use for cooling in metropolitan Mumbai alone is nearly one-quarter of the current energy use for cooling in the entire United States (Sivak, 2009) and the total potential demand for cooling in India is 12 times that of the United States (Davis & Gertler, 2015). In Europe, as communities adapt to climate change, annual energy use for cooling is expected to increase by 30-116% between 2005–2030 (Aebischer et al., 2007; Auffhammer & Mansur, 2014; Brunner et al., 2006; Hitchin et al., 2013).

Furthermore, heating, cooling and ventilation require large but intermittent electricity demands. In regions where these end uses are major components of the annual maximum electrical demands they can strain electricity systems, increase electricity system design capacity, increase regional capacity reserve requirements, and increase electricity costs. The role of either heating or cooling in electricity supply management is regionally heterogeneous, and depends largely on climate and the heating and cooling technologies commonly used (Auffhammer et al., 2017). In some regions, electrical demands associated with heating and cooling require dispatch of dirtier, less efficient, generators which increase marginal greenhouse gas emissions (Callaway et al., 2018; Graff Zivin et al., 2014; Hawkes, 2010; Shrader et al., 2019; Siler-Evans et al., 2012, 2013). Correspondingly, improving the efficiency and grid-responsiveness of heating, cooling and ventilation systems could offer unique benefits compared to other efficiency or electricity supply management strategies and should be an integral aspect of global efforts to develop sustainable energy systems.

There are many possibilities to improve performance of cooling, heating, and ventilation systems in buildings. This dissertation investigates several issues related to radiant cooling and heating. Radiant cooling and heating has been promoted on the basis of improved energy efficiency, demand response, comfort, indoor air quality, and architectural design. Many radiant buildings have demonstrated outstanding performance in these regards, and application of the technology in commercial buildings appears to be expanding – especially among high performance buildings and zero-net-energy buildings (Higgins, 2016; Higgins & Carbonnier, 2017; Maor & Snyder, 2016). Radiant is currently used in more than 40% of zero-net-energy commercial buildings (Higgins & Carbonnier, 2017). Furthermore, many researchers have conducted simulation studies and field evaluations which conclude that buildings with radiant cooling and heating can consume much less electricity than buildings with conventional all-air cooling and heating systems. Tian and Love (2009b) conducted parametric simulations which revealed that radiant cooling can use less electricity than all-air cooling in all climate zones, as long as climate-appropriate design strategies are used. Moore (2008a, 2008b) showed that in some climates a high thermal mass radiant system with an evaporative fluid cooler (water-side economizer) and indirect evaporative cooling for ventilation air could eliminate the need for vapor-compression equipment. Compared to a variable-air-volume system, such a system could reduce primary energy use for cooling and ventilation by 58–66%, and reduce peak electrical demand by 34–61% (Moore, 2008a, 2008b). Duarte et al. (2018) conducted parametric simulations with high thermal mass radiant cooling systems which showed that relatively warm cooling supply water temperatures – within the reach of evaporative fluid coolers – could satisfy comfort in a variety of climate zones and for a wide range of space heat gain rates.

Although radiant heating and cooling is common among high performance buildings and zero-net-energy buildings, the technology is not common in general, most professionals in the buildings industry are unfamiliar with radiant systems, and there are not well-established guidelines for design of radiant systems and their control sequences. This dissertation builds upon the existing foundation of knowledge related to radiant cooling and heating: it summarizes and clarifies several fundamental concepts; builds a more comprehensive understanding of how the technology is most commonly understood and implemented; questions and critiques some heretofore standard practices; proposes new methods and definitions to better support design and operation of radiant systems; substantiates and expands upon previously controversial research findings about fundamental thermodynamic requirements for radiant cooling; and advances new procedures for design of cooling systems that could facilitate improved performance.

Many aspects of this dissertation apply to radiant heating and cooling generally. However, where detailed investigation required focus on either heating or cooling, we focused on cooling. We chose this focus because of the critical global needs to advance energy efficient cooling strategies. amd because broader adoption of radiant systems has been restrained in part by challenges with design and control for operation in cooling mode. Still, most radiant systems used for cooling also provide heating, so even where our investigations focus particularly on cooling mode, we sometimes still refer to "radiant heating and cooling" systems in a general sense. Furthermore, there are also several types of radiant systems. Although many aspects of this dissertation are relevant to all types of radiant systems, where differentiation was necessary we focused on high thermal mass radiant systems – those radiant systems for which steady- state heat transfer calculations will not accurately represent dynamic performance in operation. We chose this focus because previous research has indicated that high thermal mass radiant systems unlock unique system efficiency opportunities, and because design and control of high thermal mass radiant systems has been challenging for designers in practice. Throughout the dissertation, we endeavor to clarify which observations are expected to be generalizable, which are limited to cooling only, and which are limited to high thermal mass radiant cooling. . Where possible, we explain how we expect findings to differ between radiant system types, however we did not specifically investigate the differences between system types.

Throughout this dissertation we use the term “high thermal mass radiant system”, which does not have a standard definition. Existing standards (ISO, 2012) and design guides (Babiak et al., 2009) have classified radiant heating and cooling systems according to construction and geometric characteristics, and grouped system types into three main categories: radiant ceiling panels, embedded surface systems, and thermally active building systems (TABS). However, Ning et al. (2017) showed that these categories do not account for differences in thermal behavior, and proposed an alternate classification scheme for design and control of radiant systems based on thermal response time. The researchers revealed that following a change in supply water temperature or flow rate, radiant ceiling panels approach steady-state in less than 10 minutes, while different types of embedded surface systems require 1–9 hours, and TABS systems require 9–19 hours (Ning et al., 2017). At the same time, as Olesen (2007) and Henze et al. (2008) have explained, the rate at which a radiant system cools or heats a space can change naturally and instantaneously in response to changes in space heat gains and indoor thermal conditions. The practical consequence of both phenomena is that steady-state heat transfer calculations cannot accurately represent dynamic performance of many radiant system types. Since both phenomena are related to the and thermal diffusivity and geometry of internally heated or cooled surfaces, and since the issues are pertinent to both TABS system and embedded surface system types, we chose to use the broad term “high thermal mass radiant system” to refer to any radiant system for which steady-state heat transfer calculations will not accurately represent dynamic performance in operation.

Chapter 1 investigates the deceptively simple question: *does radiant cooling save energy?* The chapter dissects this question critically and draws on several existing sources of information to articulate a judiciously skeptical assessment about the energy benefits of radiant cooling. We compile and review literature that has used building energy simulations to compare energy use of radiant cooling to that of conventional overhead-mixing all-air cooling. We aggregate and summarize the energy efficiency opportunities enabled by the technology; then, in light of these theoretical prospects, we outline system design and control decisions that would be critical to achieving potential energy savings in practice. Finally, we compare the measured energy use intensity for commercial buildings with radiant cooling to that of comparable standard commercial building stock – as recorded in the *US Department of Energy Building Performance Database*. Ultimately, the chapter argues that the energy performance for radiant cooling depends on many factors, which opens the door to questions about: (1) how radiant cooling is implemented in practice, and (2) whether or not current practice agrees with the theoretical representation of the technology in research literature.

Chapter 2 investigates how radiant cooling is implemented in practice. In particular, the chapter catalogs and compares the design and control strategies that are currently used for high thermal mass radiant buildings in the United States and Canada. We document the results from structured interviews with several leading design professionals who have considerable experience with design and control of high thermal mass radiant systems. Interviewees identified specific system configuration details and sequences of operations, then explored the practical reasons underlying design decisions and tradeoffs. We synthesize the results to describe several common design themes and to highlight areas where designers have considerably different approaches. The results reveal that there is: (1) not a consensus approach for the design and control of high thermal mass radiant cooling, (2) some disagreement and misunderstanding about certain fundamentals, (3) considerable opportunity to improve cost, control, and energy performance for high thermal mass radiant, and (4) a lack of standards, tools, and design guidelines to support more advanced system design and control.

The remainder of this dissertation builds on lessons learned from Chapters 1 and 2, and focuses on improving the fundamental understanding of certain aspects of radiant cooling, and improving the practical methods used to design, size, and simulate radiant cooling systems. Chapter 3 and Chapter 4 present experiments that compare the space heat extraction requirements[1] (space cooling load) for radiant systems and conventional all-air systems. Chapter 5 challenges – and proposes specific revision of – the industry standard definition of "space cooling (heating) load", and the associated design sizing procedures. A summary of these chapters, and the connections between them, follows. The industry standard method for sizing cooling systems and equipment assumes that space heat extraction requirements (space cooling loads) are independent from the type of system used. Although there is some previous evidence that this assumption is incorrect (Bauman et al., 2013; Feng et al., 2013, 2014b; Niu et al., 1995, 1997; Schiavon et al., 2011; Schiavon, Lee, et al., 2010), the observations presented in Chapter 2 – and an earlier study by Feng et al. (2014a) – reveal that the standard method is regularly used to design radiant systems. This may have important implications for the cost and performance of radiant systems, but the general understanding of this issue is limited. The previous evidence is somewhat fragmented, has been critiqued and doubted by researchers and professionals, and has not been presented in a comprehensive way that compellingly articulates the consequences of the differences.

To build upon previous evidence, and to support improving standard practice, Chapter 3 and Chapter 4 investigate the fundamental differences between space heat extraction requirements (space cooling load) for radiant cooling and conventional all-air cooling. In these chapters we present results from a series of multi-day side-by-side comparisons of the two system types in a pair of realistically scaled experimental testbed buildings, with equal internal and solar heat gains, and maintained at equivalent comfort conditions (operative temperature). In Chapter 3 we

[1]Much of this dissertation deals with the fact that the current standard definition of "cooling load" – and the associated system design sizing procedure – does not properly represent radiant cooling systems. Since we explicitly challenge the standard definition, we feel it is inappropriate to rely on the term "cooling load" throughout this dissertation. Instead, we use more fundamental terminology and careful explanations to convey our observations and arguments. In particular, we use the term "space heat extraction requirement" to describe the rate at which terminal heat transfer devices would need to extract thermal energy from a space to achieve particular objectives. This is admittedly awkward because most researchers and practitioners would use the term "cooling load" to describe the same concept, even though their use of the term would not comply with its standard definition. Moreover, the term "cooling load" is often used inappropriately – sometimes to describe heat gains, or intermediate heat transfer within a space – so when making precise distinctions between the different times, rates, or purposes of different heat transfer processes the term "cooling load" can be confusing. For further clarification about terminology, see Chapter 3, Chapter 5, and Appendix A. Chapter 3 includes concise summary definitions of terms that are needed to explain the experimental results in Chapter 3 and Chapter 4. Chapter 5 directly critiques the standard notion of "cooling load", and the associated system design sizing procedure – and proposes comprehensive revision of the explanatory sections and definitions in *ASHRAE Fundamentals 2017 Chapter 18: Nonresidential Cooling and Heating Load Calculations*". The comprehensive revision is included in Appendix A.

document methods that apply to the experiments discussed in both chapters, then we present results from two simple experiments that confirm – far beyond the bounds of measurement uncertainty – that to maintain equal operative temperature, the peak space heat extraction requirement (peak space cooling load) and cumulative thermal energy use for radiant cooling must be larger than for all-air cooling. Correspondingly, if a radiant system and an all-air system in equivalent spaces were to extract heat at the exact same rates, the two spaces would have different operative temperatures – the space with radiant cooling would be warmer. This is neither an advantage or deficiency, per se, it is just a fact. Additionally, we interpret the results to explain exactly why these differences must occur, and we discuss the implications in general. In Chapter 4 we pointedly explain where industry standards, design procedures, and building energy modeling tools fail to account for these fundamental differences. Then we present results from experiments which reveal that the differences are influenced by the characteristics of heat gains and by the availability of passive heat rejection. More specifically, we prove that the magnitude and timing of differences between cooling loads for radiant and all-air systems is influenced by the relative portion of heat gains that are convective and radiative, and we demonstrate that diurnal availability of a substantial passive heat rejection strategy – such as natural ventilation night pre-cooling – can have a large impact on the relative difference between space heat extraction requirements (space cooling load) for radiant and all-air cooling systems.

Chapter 5 directly addresses – and proposes practical resolutions to – several shortcomings with the current standard definition of "space cooling (heating) load" (as stipulated by *ASHRAE Fundamentals 2017 Chapter 18: Nonresidential Cooling and Heating Load Calculations* (2017a)) and with the current standard cooling (heating) system design sizing procedure for radiant systems (as stipulated by *ASHRAE Systems & Equipment 2016 Chapter 6: Radiant Heating and Cooling* (2016a)). We review the current standards and draw on various research results to explain the various ways that the standards are fundamentally flawed. We explain how these standards have influenced the modeling, sizing, and operation of radiant cooling and heating systems in practice, and we present simulation results to articulate the practical consequences of the flawed definition of "space cooling (heating) load" and associated design sizing procedure when used for design of high thermal mass radiant cooling systems. Importantly, Chapter 5 proposes a new definition for "space cooling (heating) load" that better suits a variety of systems and control strategies. The proposed definition is composed as a comprehensive revision to the explanatory sections of *ASHRAE Fundamentals Chapter 18: Nonresidential Cooling and Heating Load Calculations* (2017a). The comprehensive revision is necessary because the current narrow conception of "space cooling (heating) load" is pervasively entwined with every issue that is described by the standards.

CHAPTER 1

A critical review of energy savings claims for buildings with radiant cooling and heating

Chapter Abstract

Many building energy simulation studies have concluded that radiant cooling and heating can reduce energy consumption compared to all-air systems. However, there has been very little research published about whether or not buildings with radiant cooling and heating use less energy in practice. In this chapter, we review previous simulation studies that predict energy use for buildings with radiant systems in comparison to buildings with all-air systems, and we explain how and why the energy savings potential differs by climate and application. We summarize the specific mechanisms by which radiant cooling can reduce energy consumption compared to conventional all-air cooling, and we also highlight important choices about system design and control that could influence the energy performance ultimately achieved. To explore whether or not radiant cooling and heating reduces energy use in practice, we compared the measured site energy use intensity for a group of buildings with radiant cooling and heating (n=23) to the site energy use intensity for a subset of baseline buildings (n=2,592) listed in the *US Department of Energy Building Performance Database*. We conclude that the median Energy Use Intensity (EUI) for buildings with radiant cooling and heating is 31% lower than that of baseline buildings, in the same type category and climate zones (p = 0.006, Cohen's δ =0.84). Furthermore, we consider several potentially confounding factors that contribute to methodological uncertainty in this statistical assessment, we estimate the sensitivity of our results to those factors, and we recommend possibilities for future research.

1.1 INTRODUCTION

Radiant cooling and heating is promoted as a pathway to reduce energy use and peak electrical demand in buildings compared to conventional all-air systems. Using simulations and laboratory studies, researchers have demonstrated the ideal possibilities for radiant systems, but few researchers have documented the extent to which these advantages have actually translated to measurable energy benefits in practice. Confirming the tangible impact of radiant systems is a knotty prospect. Since there are currently relatively few radiant buildings overall, data on measured performance is sparse compared to the amount of data available for the conventional building stock. The technology is not common enough to appear as an equipment type category in the US Commercial Building Energy Consumption Survey (EIA, 2012). Moreover, notwithstanding the many fundamental reasons to expect that radiant is more advantageous than all-air systems, and despite the fact that many high performance buildings employ radiant, it is difficult – from a statistical basis – to disentangle a) the contribution of radiant cooling toward the apparent benefits demonstrated by individual case studies, from b) the contribution of other factors that would also improve performance.

In this chapter we review previous simulation studies that have estimated the potential energy savings for radiant systems compared to conventional all-air systems, we provide a consolidated inventory of the specific mechanisms by which radiant cooling can reduce energy use, and we consider counterposing factors that could erode the potential energy performance for buildings with radiant cooling if they are not avoided through proactive design and operation. Then, to assess whether or not buildings with radiant use less energy in practice, we present a comparison of measured site energy use intensity (site-EUI) for a group of buildings with radiant cooling to the site-EUI for a comparable subset of baseline buildings. Finally we consider the potentially confounding factors that make it difficult to attribute observed differences solely to the presence of radiant cooling.

1.2 SIMULATION STUDIES CONCLUDE RADIANT CAN REDUCE ENERGY USE

There has been substantial research to develop and validate numerical methods that properly estimate the fundamental heat transfer mechanisms involved with radiant cooling and heating systems(Chantrasrisalai et al., 2003; Niu et al., 1995, 1997; Strand et al., 1999; Strand & Baumgartner, 2005; Strand & Pedersen, 2002; Yu et al., 2014) . In the years since these methods were incorporated into publicly available building energy simulation engines, a multitude of researchers have conducted detailed whole building energy simulations to predict the energy benefits of radiant cooling. These studies have established strong evidence that radiant cooling offers the potential substantial energy savings and peak electrical demand reduction (Lim et al., 2014; Rijksen et al., 2010) compared to conventional all-air cooling systems. The following review synthesizes the findings from several simulated comparisons of radiant and all-air systems.

Feustel and Stetiu set the stage for simulated comparisons; the authors estimated that radiant cooling can reduce peak electricity use for fans and pumps by 75% compared to conventional variable-air-volume all-air systems (Feustel & Stetiu, 1995; Stetiu, 1998, 1999).

Niu et al. (2002), Jeong et al. (2003), and Hao et al. (2007) each simulated low thermal mass radiant cooling in hot-humid climates with various dehumidification strategies. The studies predicted annual primary energy savings between 8–53% depending on the dehumidification strategy employed. They predicted the largest savings for scenarios that used enthalpy recovery and chilled water in series to dehumidify ventilation air. Niu et al. (2002) also explained that if chilled water is used for dehumidification, radiant cooling will only reduce chiller electricity consumption if two chillers are used: one that operates at low temperature for dehumidification, and another that operates at a warmer temperature for the radiant system.

Niu et al. (1995) and Sodec (1999) both simulated low mass radiant cooling in temperate climates. The authors each concluded that radiant systems and variable-air-volume all-air systems consume a comparable amount of electricity when chilled water is generated by a conventional chiller. They attributed the smaller than expected overall savings to the fact that all-air systems benefit from air-side economizer cooling in temperate climates, while radiant systems usually do not. However, the authors also showed that radiant systems could save 10–20% compared to all-air systems if they employ water-side economizer cooling (evaporative fluid cooling). The studies did not consider air-side economizer cooling or natural ventilation cooling in radiant buildings.

Kim and Olesen (2015a, 2015b), Rijksin et al. (2010), Lehmann et al. (2007), and others have used simulations to demonstrate that high thermal mass radiant systems can be controlled in a way that the chiller plant operates overnight to pre-cool the building mass. These researchers demonstrated that the strategy can reduce peak electrical demand by 30–50% compared to variable-air-volume all-air systems, and can reduce the required size of the chiller plant. However, as documented in Chapter 2 this control strategy has been challenging for practitioners to institute in practice and is not widely adopted (Paliaga et al., 2017, 2018).

Tian and Love (2009b) conducted parametric simulations to compare performance of radiant cooling and conventional all-air cooling in an office building across a range of climate zones. The authors included ventilation air dehumidification where needed, and assessed the impact of both air-side economizer cooling for all-air systems, and water-side economizer cooling (evaporative fluid cooling) for radiant systems. This study demonstrated that radiant cooling can use less electricity than all-air cooling in all climate zones, as long as climate-appropriate design strategies are used. It should be noted that in all cases the radiant systems included ventilation air heat recovery, while the all-air systems did not – this may give an unfair advantage to the radiant system. Tian and Love also showed that in some climates and applications conventional all-air systems with air-side economizers would use less electricity than radiant cooling with a conventional chiller (Tian & Love, 2009a).

When all of the efficiency advantages for radiant cooling are employed together in a favorable climate, the overall savings can be especially large. Through energy simulation, Moore (2008a, 2008b) proved that for a high thermal mass radiant system in a cold semi-arid climate an evaporative fluid cooler (water-side economizer) can be used in lieu of vapor-compression equipment. The system was controlled to pre-cool building mass overnight and indirect evaporative cooling was used for ventilation air – no dehumidification was required. The simulations showed 58–66% primary energy savings and 34–61% peak electrical demand savings compared to a variable-volume all-air system (Moore, 2008a, 2008b).

1.3 LIMITED EVIDENCE ABOUT PERFORMANCE OF RADIANT IN PRACTICE

The evidence about energy performance for radiant cooling in practice is more limited.

A recent survey assessment of commercial building energy consumption in the United States indicated that the median energy use intensity for buildings with radiant cooling is 14–66% lower than standard buildings of comparable type and climate zone (Higgins & Carbonnier, 2017). Although radiant cooling is currently installed in a small portion of buildings overall, it is a common strategy among buildings with the lowest energy use intensity (Higgins, 2016; Maor & Snyder, 2016; Paliaga et al., 2016). In a study that reviewed 90 “high performance building” case studies, Maor and Snyder found that more than 25% included radiant cooling or heating.

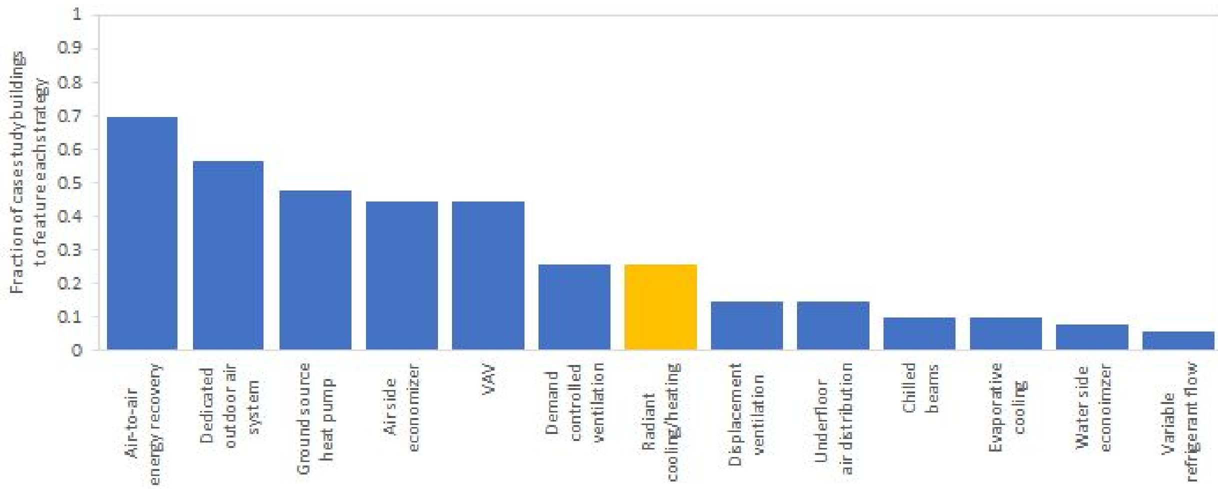
Moreover, the number of high performance buildings with zero net energy aspirations has increased rapidly in recent years (Higgins, 2016), and consequently application of radiant cooling appears to be expanding. In a study that reviewed design and energy performance for 26 zero-net-energy buildings Carbonier and Higgins found that more than 42% included radiant cooling or heating (Higgins & Carbonnier, 2017).

These studies all reveal an association between radiant cooling and lower energy use, but they do not consider whether or not confounding variables could be the cause for lower energy use. Each of these studies also explain that high performance buildings employ a variety of system types – often as multi-layered hybrid combinations. According to Maor and Snyder (2016) more than 80% of high performance buildings integrate three or more different mechanical system strategies to benefit energy efficiency. Figure 1.1 (top) summarizes the relative prevalence of different mechanical system strategies among the 90 “high performance building” case studies assessed by Maor and Snyder (2016). Figure 1.1 (bottom left) summarizes the relative prevalence of different heating and cooling distribution strategies among the 26 zero net energy buildings studied by Higgins and Carbonnier (2017). Figure 1.1 (bottom right) summarizes the relative prevalence of different heating and cooling plant strategies from the same buildings.

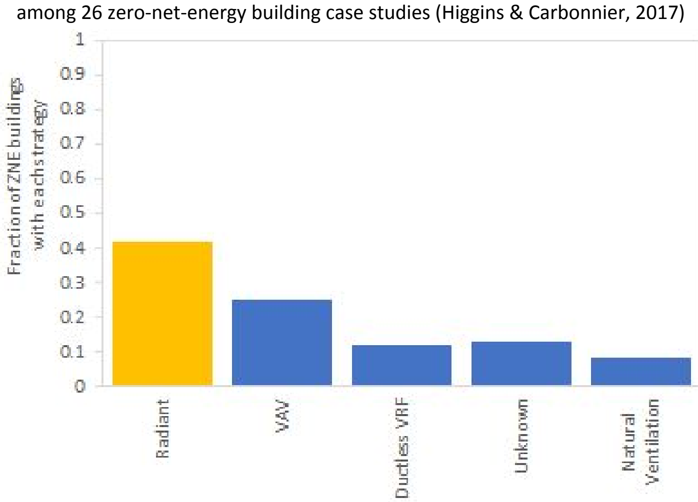
In the field of building science, it is very rare and difficult to conduct a full-scale paired assessment of measure-specific impacts. Even when two buildings of the similar type-category have the different elements that we wish to compare, a real effect may be difficult to detect because many uncontrolled exogenous influences can cause within type-category variation that obscures the real impact of the measure, and may in fact be larger than the effect of the measure. This challenge is even difficult to overcome in pre-post assessments of a single building because there are many variables that cannot be controlled, including: occupancy rates, activity, climate,, and other coincident system changes. *ASHRAE Guideline 14* (2014) offers several methods to reduce uncertainty associated with full-scale paired assessment of energy and demand savings in buildings. However, the guideline does not provide methods to assess measure-specific impacts by statistical analysis of the differences between large populations of buildings – the type of comparison presented by Higgins et al. (2017; 2016) and Maor and Snyder (2016).

In regard to field assessment of energy savings for radiant, there are two studies that come as close as possible to full-scale paired comparisons. First, Sastry and Rumsey (2014) compared energy use and comfort in opposing wings of a single office building in Hyderabad – one with radiant cooling and one with variable-air-volume all air systems. The researchers found that the radiant system used 34% less energy, cost slightly less to construct, and achieved better thermal comfort for occupants. However, to the contrary, Tian and Love (2009a) used measurements to calibrate building energy simulations for a radiant building in Calgary, and estimated that it used 43% more energy for cooling, heating and ventilation than an all-air system would have. The researchers attributed the deficiency to flawed control strategies in the real building, and to the fact that all-air systems can benefit from air-side economizer cooling while radiant systems do not. These opposing results underscore that the potential advantages of radiant cooling may depend on climate, and that its full benefits may only be realized when design and operation align proactively to effectuate the specific efficiency opportunities enabled by the technology.

Relative prevalence of mechanical system strategies among 90 "high performance building" case studies (Maor & Snyder, 2016)



Relative prevalence of cooling/heating distribution strategies among 26 zero-net-energy building case studies (Higgins & Carbonnier, 2017)



Relative prevalence of cooling/heating plant strategies among 26 zero-net-energy building case studies (Higgins & Carbonnier, 2017)

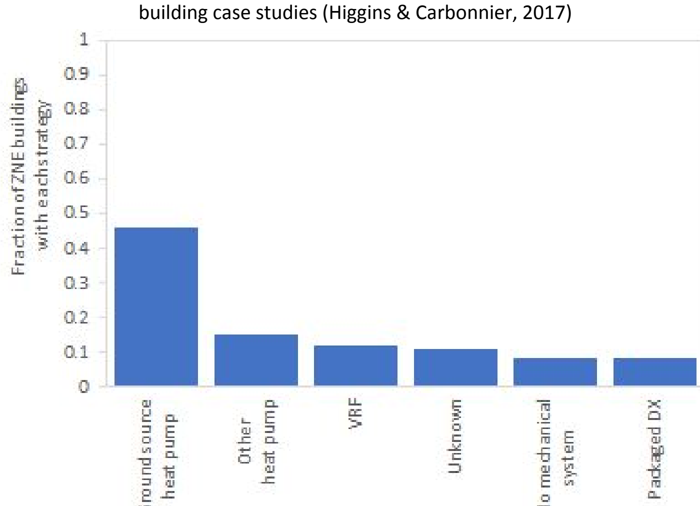


Figure 1.1: Relative prevalence of different mechanical system strategies among the 90 "high performance building" case studies assessed by Maor and Snyder (2016) (top) and among the 26 "zero net energy" buildings studied by Higgins and Carbonnier (2017) bottom. Higgins and Carbonnier (2017) accounted separately for distribution strategies and plant strategies.

1.3.1 The need for a comparative statistical analysis

Use patterns, design and installation quality, and energy performance for any heating, cooling, and ventilation technology will vary widely among a large population of buildings. Therefore, it is tenuous to rely on physical experiments, simulations, or case studies, to determine whether or not buildings with radiant cooling are actually achieving better energy performance than all-air buildings in practice. Physical laboratory experiments and case studies present particular examples, and the design characteristics associated with such studies are carefully controlled. Physical experiments reveal fundamental relationships with high resolution, and so they can be used to assess the effect of specific design decisions, but they cannot tell whether or not all the factors generally present in practice will result in radiant buildings that use less energy than all-air buildings. Similarly, simulations also compare specific examples with characteristics that are carefully selected by researchers. With enough knowledge about common practices and the critical design factors, building energy simulations may provide an effective opportunity to demonstrate what design decisions matter most – sensitivity analysis techniques are an excellent approach to study the importance of numerous design factors at once (Garcia Sanchez et al., 2014; Heiselberg et al., 2009; Henze et al., 2006; Le Dréau & Heiselberg, 2014). However, the limitation of such an analysis is in properly defining the parametric space to be analyzed. In other words, it is difficult to simulate the things that you do not know about, or which a model cannot represent. For example, even with knowledge about the different sequences of operation employed for radiant buildings, it would be difficult to simulate the range of options because the system control options available for radiant systems in building energy simulation engines is limited.

To determine if radiant buildings – in general – actually use less energy in the real world, we have to study buildings in the real world. Specifically, this determination requires careful statistical comparison of the measured energy performance from an appropriate number of buildings, with and without radiant cooling and heating. However, even with annual energy performance data from an adequate number of buildings, the question cannot be answered simply by comparing the median energy use for a population or radiant buildings and a population of baseline buildings. There are many potentially confounding factors that ought to be considered.

1.3.2 Potential confounding factors

If a population of buildings with radiant cooling uses less energy than a population of baseline buildings – as shown by Higgins and Carbonnier (2017) and Maor and Snyder (2016) – we can conclude that there is a correlation between building energy use and system type, but we cannot necessarily attribute causality in the relationship. It is possible that other confounding factors are more important determinants of building energy use and that radiant cooling just happens to be coincidentally associated. The following paragraphs explore a range of factors that could conceivably confound a statistical assessment of whether or not radiant cooling uses less energy than all-air cooling in practice. Some of these factors can be controlled for in an appropriately designed statistical comparison, but other potentially confounding factors are more difficult to avoid.

First, we can be certain that climate impacts energy consumption. In part because radiant systems have limited cooling capacity, and in part because it is more challenging to design radiant cooling systems for buildings in humid climates, it is probable that buildings with radiant cooling are predominantly located in milder climates. Therefore, a statistical comparison should be very careful to control for climate in the same way that a paired field assessment must control for or correct for differences in weather during the study periods (ASHRAE, 2014).

Since radiant systems have a smaller cooling capacity than all-air systems, it is also possible that radiant cooling is predominantly employed in buildings that have smaller heat gains, and so would naturally consume less energy. Controlling for heat gain should not be conflated with controlling for building type. While it is true that different building categories have different median energy consumption, the distribution of energy use intensity among buildings within a category is substantial. For example, a study that compares energy use intensity of laboratories with radiant cooling to energy use intensity for laboratory buildings in general, would incorrectly compare a group of laboratories with low internal heat gains – perhaps teaching laboratories are able to use radiant cooling – to the whole group of laboratories, which includes a much wider distribution of internal heat gains. To ensure that such a comparison is fair, we would either need to ensure that the distribution of heat gain rates for the sample of laboratories with radiant cooling is not significantly different from the distribution of heat gain rates for laboratories in general, or we would need to compare the radiant laboratory group against an appropriate subset of the conventional laboratory group that does have a similar distribution of heat gains.

A similar, but slightly different problem would occur if radiant tends to be used in particular building types, that for a separate reason happen to use less energy than commercial buildings in general. This could occur if, hypothetically, radiant cooling is used more often in high-value multi-story office buildings because it enables designs with more leasable floor area, and this building type happens to also use less energy than commercial buildings in general.

As discussed previously, radiant cooling is a relatively common strategy among 'high performance buildings' (Maor & Snyder, 2016) and 'zero net energy buildings' (Higgins & Carbonnier, 2017). Buildings in these categories are not random and representative samples of a general population; sustainability was certainly an intention from the onset, and proactive design with these goals in mind would have influenced many aspects that impact building energy use. For example, by some estimates a relatively small fraction of commercial buildings actually employ parametric building energy simulation in the design phase, yet likely more of the buildings with radiant cooling have benefited from parametric simulation.It may not be fair to compare these buildings against the general stock without acknowledging that many factors influence their excellent energy performance.

On this note, radiant cooling is also not the only mechanical strategy among the highest performing buildings. Figure 1.1 shows the distribution of different mechanical system types among high performance buildings studied by Maor and Snyder (2016) and zero net energy buildings studied by Higgins and Carbonier (2017). While radiant cooling provides sufficient performance to enable the degree of energy performance desired by these projects, it is clearly not a necessary condition. In this light a statistical assessment of the energy benefits of radiant cooling might only be appropriate where it draws a comparison among those buildings that were designed for very high performance.

Figure 1.2, adapted from Maor and Snyder (2016), plots the distribution of site-EUI for different mechanical system types among 90 case studies of high performance buildings. Although many of the buildings with the lowest energy use intensity employ radiant cooling, the best variable-air-volume all-air buildings have site-EUI that is equal to or lower than that of most buildings with radiant cooling.

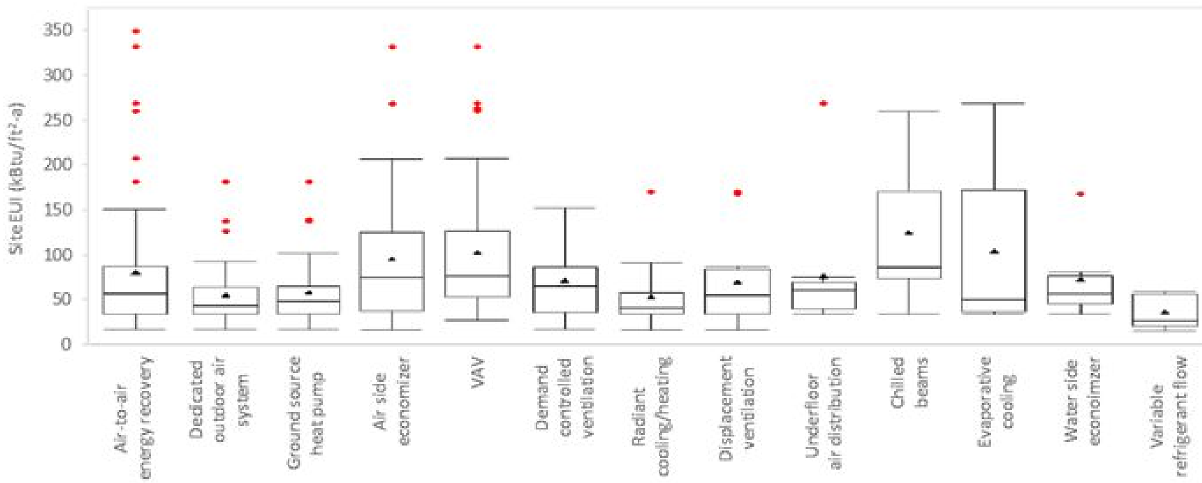


Figure 1.2: Distributions of site-EUI for "high performance buildings" that utilize various mechanical system strategies (Maor & Snyder, 2016).

The impact of some confounding factors can be expected to be attenuated by having a large enough pool of diverse data. For example, comparison of just a few radiant buildings and just a few baseline buildings could be biased if, by chance, the weather during the years when the radiant buildings were measured was different than the years when the baseline buildings were measured. This type of sampling error is random, and can be attenuated when there are enough randomly selected samples in each group. However the problem is not resolved if there is a fixed bias in the measurement. This could occur if, for example, all the data for energy use intensity in radiant buildings is collected by a research team that uses the gross floor area to calculate energy use intensity, and all of the data for other building types is collected by a research team that uses net floor area to calculate the metric.

Furthermore, confounding factors are not attenuated by more samples if there is a systematic relationship between the site-EUI in either sample group and any characteristics that are not symmetric between the sample groups. For example, if radiant buildings are predominantly used in buildings with low internal gains, then a comparison of many radiant buildings and many baseline buildings would naturally show that radiant buildings use less energy, but not because they use radiant cooling systems per se. At its worst, such an analysis would be tantamount to claiming that "buildings with smaller heat gains use less energy."

To clarify, it is acceptable to have systematic relationships between site-EUI and other group characteristics, as long as the distribution of values for that characteristic is similar in both groups, and as long as the influence of that characteristic is similar for both groups. For example, it is okay that there is a systematic relationship between the number of operating hours in a building and energy use intensity, but if there is such a relationship it is important that the two sample groups have similar distributions of operating hours and that operating hours influence energy use in radiant buildings and baseline buildings in similar ways. This issue is important because it means that we do not have to subset every sample group to a fixed value of each independent variable that could have influence.

1.4 THE SPECIFIC EFFICIENCY ADVANTAGES OF RADIANT COOLING

Several researchers have identified reasons that radiant cooling can reduce energy consumption and peak electrical demand compared to all-air systems (Kim & Olesen, 2015a, 2015b; Rhee et al., 2017; Tian & Love, 2009b). We summarize the variety of explanations as the following five specific energy advantages, ordered according to their relative importance:

1. Radiant cooling operates with relatively warm chilled water temperatures. Chiller efficiency can be improved if chillers are designed and controlled to operate at warmer temperatures. Further, radiant cooling can also allow use of very high efficiency primary cooling sources, such as evaporative fluid coolers (water-side economizers), and direct ground or water body heat exchange.
2. High thermal mass radiant systems can allow for cooling plant operation during periods when electricity prices are lower, and when primary cooling sources may operate more efficiently.
3. Electricity use for thermal distribution in radiant buildings can be lower than in all-air buildings. Although radiant buildings require more electricity for pumping, the fan electricity savings from reducing airflow rates to the minimum ventilation requirements can be larger than the increase in electricity use for pumping.[2]
4. By decoupling ventilation from space cooling, radiant systems can avoid the need for terminal reheat, and avoid energy consumed by incidental dehumidification that occurs when air is cooled with low temperature chilled water, or direct expansion.
5. The air temperature in buildings with radiant cooling is somewhat warmer than in buildings with all-air systems at equivalent comfort conditions. Consequently, heat gains from ventilation air are somewhat smaller in radiant buildings and there are more hours when outdoor air provides free cooling. The impact of this factor is minor.

The aforementioned advantages are all theoretical opportunities for energy savings compared to an ideal all-air system. However, in light of what researchers have shown about the inefficiencies of all-air systems in operation there are also several other reasons to expect that radiant cooling would use less energy than all-air systems in practice, including:.

1. Radiant cooling largely avoids thermal losses in distribution, whereas all-air systems are prone to substantial losses by duct leakage and by heat transfer from duct walls.
2. All-air systems are notorious for excess energy use due to simultaneous heating and cooling – an issue that should be easier to avoid with radiant systems.

[2] Feng and Cheng (2018) used energy simulations to compare a high thermal mass radiant system and a variable-air-volume system in San Francisco. The authors showed that the variable-air-volume system spent so much time operating at minimum ventilation rates that the DOAS for the radiant system used more fan energy.

3. Minimum airflow rates for terminal units in all-air systems can result in excess fan energy use compared to what might be expected. Since radiant decouples ventilation and space conditioning, fans in the dedicated outdoor air systems for radiant buildings can operate to provide only as much airflow as is required for indoor air quality.
4. The idealized performance for all-air systems usually incorporates economizer cooling, but various studies have revealed that economizers do not function properly in most cases (Cowan, 2004; Felts & Bailey, 2000; Hart et al., 2008; Jacobs et al., 2004; Jacobs & Higgins, 2003; Mowris et al., 2015). This failure is especially true for light commercial buildings which use rooftop packaged units.
5. Conventional all-air systems are prone to faults, especially related to supply airflow rates, and refrigerant charge issues. Katipamula and Brambley (Katipamula & Brambley, 2005a, 2005b) estimate that existing packaged vapor-compression systems use 15-30% excess energy due to faults and improper maintenance.

1.5 CRITICAL DESIGN AND CONTROL DECISIONS FOR RADIANT COOLING

Despite the many potential advantages, the full benefits of radiant cooling may only be realized when design and operation align proactively to effectuate the specific efficiency opportunities enabled by the technology. Decisions throughout the arc of building design and operation will influence the performance of radiant cooling systems in practice. It would be wise of practitioners to proceed with an awareness of the possible challenges and deficiencies, that they might strategically avoid them. The following summary outlines several critical issues that could erode the potential energy benefits of radiant cooling:

1. First and foremost, heat gains should be proactively reduced in radiant buildings because the magnitude of heat gain has a non-linear impact on energy use for cooling. The space cooling capacity for an internally cooled surface can be increased by reducing the surface temperature, but a lower surface temperature requires colder chilled water supply temperature which reduces cooling plant efficiency. Moreover, in many climates a lower surface temperature may also require additional energy use for dehumidification to avoid surface condensation.
2. To maintain equivalent comfort conditions as an all-air system, radiant cooling must extract more heat overall (Feng et al., 2013, 2014b; Woolley et al., 2018a, 2019).
    a. This phenomenon is partly attributed to the fact that interior surfaces of building envelope elements (outdoor exposed surfaces) are cooler, so total heat gains by conduction through the envelope increase. The effect would be small for very well insulated, internal gain dominated buildings, but could be more substantial for buildings with large envelope to floor area ratios, large window to wall ratios, or poorly insulated walls and windows. The effect would be most pronounced in scenarios where the internally heated or cooled surface surface is integrated into an envelope wall or ceiling.
    b. In an all-air building, a substantial portion of the heat gains each day are absorbed in masses, then released passively to the environment overnight. Radiant cooling extracts heat from all surfaces to which it is exposed, and so influences the way

that heat is absorbed, stored and released from non-active masses in a building. As a result, radiant cooling can reduce the opportunity for nighttime passive cooling by extracting a larger portion of the heat gains each day.

Chapter 3 and Chapter 4 present laboratory experiments that tested these effects. The experiments showed that to maintain equivalent comfort conditions, radiant cooling extracted 7% more heat than all-air cooling each day (Woolley et al., 2018a). In scenarios that increased the opportunity for passive cooling with night ventilation, the difference was as much as 40% (Woolley et al., 2019). At this scale, the extra thermal burden for a radiant system might not be easily overcome by improved efficiency elsewhere in the system. This issue might be mitigated if the radiant system can use a water-side economizer (evaporative fluid cooling), or if controls strategically coordinate radiant cooling and night flush cooling so that heat is absorbed and stored in building masses during the day instead of being extracted instantaneously by internally cooled surfaces.

3. Design and control of radiant systems should generally aim to keep chilled water supply temperature as warm as possible, and to operate the chilled water plant in a way that benefits from elevated temperature.
    a. Unfortunately, as documented in Chapter 2, research has revealed that in practice many buildings with radiant cooling operate chillers at conventionally low chilled water temperature – often because low temperature chilled water is used for dehumidification or other services within the building (Paliaga et al., 2017, 2018).
    b. If strategic design can increase the required chilled water temperature enough, radiant cooling can enable a step change in cooling plant efficiency by eliminating the need for vapor-compression equipment. In many scenarios, evaporative fluid coolers or direct ground or water body heat exchange could provide adequate chilled water temperature (Duarte et al., 2018).
    c. On the opposite end of the spectrum, warmer chilled water temperature may have diminishing benefits as the larger flow rates required could increase energy use for distribution. The optimal balance between these factors is not well understood and further investigation is warranted.
4. Similar to the effect of chilled water supply temperature, cooling plant efficiency is also influenced by the environmental conditions coincident with operation – every cooling plant (including evaporative fluid coolers) is more efficient when it is cooler outside. Uniquely, high mass radiant systems decouple the timing of cooling plant operation from the timing of space cooling, so they can allow plant operation overnight when ambient conditions are more favorable to efficiency and when wholesale electricity prices, and time of use retail electricity prices are lower. In Chapter 5, we present simulation results that demonstrate this demand shifting. Unfortunately, our research has revealed that very few radiant buildings are actively controlled to shift cooling plant operation to cooler periods (Paliaga et al., 2017, 2018). In Chapter 2 we present results from structured interviews with radiant designers that document the design and control strategies that are currently common among radiant cooling systems..This advantage will only be realized with strategic system design and operation.

5. Most radiant systems are not able to overcome all heat gains in all zones at all times. This is especially true for high thermal mass radiant systems, which – due to limited ability for rapid self-regulated response to dynamic heat gains – inevitably allow some drift in operative temperature. Accordingly, most buildings with radiant cooling also incorporate supplemental cooling from air systems. The design and control of these supplemental air systems – especially the way they are coordinated with radiant systems – could influence the amount of cooling provided by each system, occupant comfort, indoor air quality, and overall energy performance.

   As documented with greater detail in Chapter 2, and by Paliaga et al. (2017, 2018), our research indicates that practically all buildings with high mass radiant cooling utilize dedicated outdoor air systems for ventilation, and that these systems are also controlled to provide supplemental cooling. In most cases, chilled water flow to the internally cooled surface is controlled by either a slab temperature setpoint or an indoor air temperature setpoint, and supplemental cooling from air systems is controlled to an indoor air temperature setpoint that is somewhat higher than that which controls the radiant system. Building energy simulations by Chung et al. (2017) suggested that on a peak day, supplemental cooling with this control strategy may provide more than half of the sensible zone cooling required to maintain steady indoor air temperature. Furthermore, the choice of slab temperature setpoint affects the portion of cooling provided by each system. In this light, we have the following concerns about design and control of supplemental cooling systems:

   a. There are possibilities for conflict between radiant systems and supplemental air systems. When air systems are controlled to provide constant supply air temperature for ventilation, they might end up heating while radiant systems are cooling.

   b. In some conditions, as Chung et al. (2017) revealed, the slab might release heat to the zone at the same time that the air system operates to cool the zone.

   c. Often, the supply airflow rate from ventilation systems is increased to provide supplemental cooling. Although this strategy would provide the sensible zone cooling to maintain zone temperature setpoints, since supplemental cooling is most likely needed during hotter periods in the day, the strategy could unintentionally increase total system cooling requirements.

   d. If supplemental air systems are controlled to maintain constant indoor air temperature, they will naturally reduce the amount of sensible cooling delivered by high thermal mass radiant systems with constant slab temperature. This would occur because the zone cooling rate from high thermal mass radiant systems would naturally increase when zone air temperature increases so holding air temperature constant will deaden the self-regulating response. One solution to this challenge would be to allow zone temperature to drift somewhat throughout the day, and only operate supplemental cooling to avoid discomfort when the capabilities of a self regulated capacity response by the high mass radiant system have been fully exhausted.

6. Dedicated outdoor air systems in radiant buildings are often oversized for the needs of supplemental cooling and to achieve credit for green building certification programs.. In scenarios where it is possible, these ventilation systems include exhaust air heat recovery and are controlled by a demand controlled ventilation sequence to maintain an indoor CO_2 setpoint. However, according to our research (Paliaga et al., 2017, 2018), the controls for dedicated outdoor air systems are generally not designed to give preference to economizer cooling. Unless the control strategies are carefully specified, on mild days – when the radiant system can maintain comfort without any support from the supplemental air system – the air system might operate at minimum ventilation flow rates even though the building could benefit from economizer cooling.
7. The control point selected for radiant systems (the measurement that control sequences compare to a setpoint) can influence the sizing of, and energy use by, hydronic systems and the cooling plant. If flow to a high thermal mass radiant system is controlled in response to indoor air temperature, cooling plant operation may be initiated too late to maintain comfort reliably, the resulting supply water temperature would need to be lower than preferred, and the cooling plant capacity would need to be larger than necessary. In Chapter 5, we present simulation results that demonstrate these issues. If supplemental air systems are included in such a design, they would need to be larger than if cooling plant operation had been initiated earlier. Operative temperature control – which has been suggested by various researchers (Simone et al., 2007), but challenged by others (Dawe et al., 2020) – would not resolve this issue.
8. In general, we expect that energy use for fans and pumps associated with a radiant system can be much smaller than the fan energy use for an all-air system. However, this should not be taken for granted. In our side-by-side laboratory comparison of radiant and all-air cooling (see Chapter 3 and Chapter 4), the pump energy use for the radiant system was larger than the fan energy use for the all-air system. This occurred because the air handler was larger than needed for our experiments so the fan operated at part speed. This scenario is not necessarily limited to the unique arrangement of our laboratory apparatus. Feng and Cheng (2018) conducted simulations to compare energy performance of high thermal mass radiant cooling and a conventional variable-air-volume all-air system; they showed that in scenarios where a variable-air-volume system regularly operates at minimum airflow rates, fan energy for the variable-air-volume system can be smaller than fan energy for the DOAS associated with a radiant system. This gives weight to the importance of efficient hydronic system design. Tang et al. (2018) demonstrated a pulse width modulated flow strategy for radiant systems that simultaneously improves the thermal performance and reduces pumping energy use during part capacity operation.

In view of these design factors, it is clear that while there are many fundamental reasons that radiant buildings can be expected to use less energy than all-air buildings, there are many practical design and control decisions that can erode the potential energy performance of radiant systems. Further research is warranted to assess the sensitivity to these factors. For example, simulations could be performed to demonstrate many of the risks discussed, to determine whether or not non-ideal design and control decisions would impact energy performance in substantial ways compared to what could be expected from typical all-air buildings.

1.6 STATISTICAL COMPARISON OF ENERGY USE FOR BUILDINGS WITH AND WITHOUT RADIANT COOLING

1.6.1 Statistical methods

We conducted a statistical analysis to compare energy use intensity for commercial buildings with radiant cooling to that of standard commercial buildings. To accomplish this, we used building energy use intensity data reported by Higgins and Carbonnier (2017) and by the *US Department of Energy Building Performance Database* (DOE, 2017).

The *US Department of Energy Building Performance Database* is the largest collection of data about the energy-related characteristics of commercial and residential buildings in the United States. Aside from energy use intensity data, the *Building Performance Database* also includes information about building classification, location, age, floor area, occupant density, and operating hours. Although the database includes some information about systems in each building, it does not specifically identify buildings that use radiant cooling.

The data reported by Higgins et al is a collection of energy use intensity data for 23 commercial buildings with radiant cooling from 19 different cities in North America.

Before working with actual data, it is important for a comparison study to determine the number of samples that would be required from each group in order to have a reasonable expectation of detecting a difference between the groups with confidence. Therefore, a comparison study ought to begin with an a-priori assessment of statistical power. Statistical power is the probability that a statistical test will be able to detect a difference when there is a difference to detect. If a comparison study does not include enough samples from both groups it will risk concluding that there is no difference between the groups even when there is a difference in reality. If the difference between these groups is small, a study would need more samples to detect the difference than if the difference is large.

Since the *Building Performance Database* has more than 220,000 possible samples for our comparison, and data set from Higgins et al has only 23, our test used a different number of samples from each dataset. Of course, we wanted to use as many samples as possible, but had to limit the analysis to a subset from each dataset in an attempt to control for potentially confounding factors.

If we are comfortable with an 80% likelihood that a statistical test with our samples will be able detect a large difference between the groups (Cohens-$\delta = 0.8$) with 95% confidence, we would need a sample of at least 12 buildings from the Higgins' dataset, and at least 1,200 samples from the *Building Performance Database*. Explained another way, with this sample size we could expect that if there is a large difference between the groups, our study design will leave a 20% chance of failing to detect the difference. If the difference between the groups is actually small (Cohen's-$\delta = 0.2$) the same study design would have an 86% chance of failing to detect the difference. In such a case, to ensure a study design with Power = 0.8 we would need 198 samples of buildings with radiant cooling, and almost 20,000 samples from the *Building Performance Database* .

The Higgins dataset includes information from 23 commercial buildings with radiant cooling. These buildings include offices, offices with labs, education buildings, libraries, a visitors center, and a community center. The data set includes information about building location, floor area, and year of construction (and renovation if applicable). The 23 buildings included were from 19 different cities in North America, in 7 different ASHRAE climate zones.

In light of the need to control for potentially confounding variables, we subset both datasets so that they only include data from comparable populations. As discussed previously, to be comparable the two populations must have similar distributions of any factor that influences energy use intensity. Figure 1.3 demonstrates the need for this control – the first plot presents overlapping histograms of site-EUI for office buildings in Hawaii (orange) and office buildings in Michigan (purple). The second plot presents a histogram of the relative differences between the two datasets. Office buildings in Michigan have higher site-EUI – the median relative difference is 82% higher than in Hawaii and there is only a 19% chance that an office building in Hawaii has a lower site-EUI than an office building in Michigan. Clearly, it is important to control for climate (or possibly other locational factors).

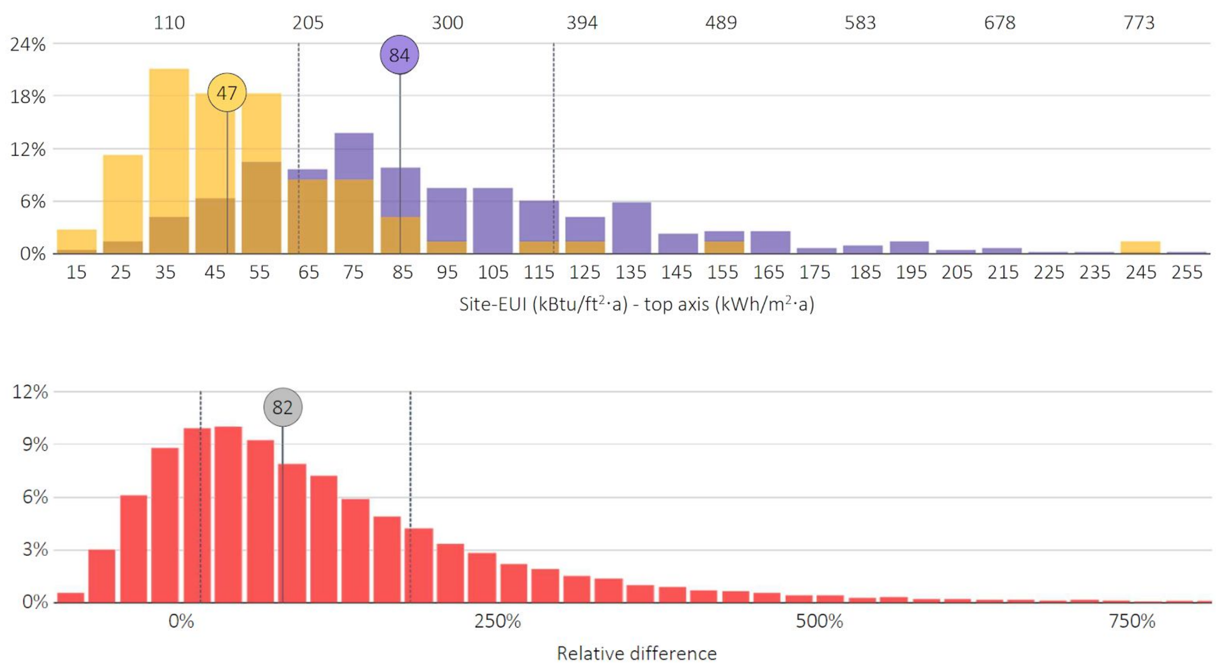


Figure 1.3: (top) Overlapping histograms of site-EUI for office buildings in Hawaii (orange) and Michigan (purple), and (bottom) histogram of the relative differences between site-EUI in each data subset. The medians of each distribution are indicated (circled number) as are the first and third quartiles (dash lines).

Although we could not control for all of the potentially confounding factors hypothesized in the previous Section 1.3.2, we subset the two datasets so that they were similar in several ways. From the Higgins dataset we subset for office buildings because it was the largest potential sample group from a single commercial building type category – with 12 buildings – and because multiple buildings within this category are likely to have similar exogenous characteristics. On the contrary, although the dataset also includes public assembly buildings:

- a subset for this building type would only include only five samples which is much less likely to detect a difference with statistical significance, and
- As a group, public assembly buildings are less likely to have similar exogenous characteristics. For example public assembly buildings may include libraries, community centers, arenas, churches, etc – each with unique activities, schedules, and end uses.

Further, we subset for offices in ASHRAE Climate Zones 3C and 4C (Warm Marine and Mixed Marine) because it was the largest potential sample group from similar climates. According to data from the *Building Performance Database*, the median site-EUI for all office buildings in climate zone 3C is exactly the same as in climate zone 4C. This resulted in 8 samples to represent commercial buildings with radiant cooling – smaller than preferred, with less than a 7% chance of detecting a small difference (Cohen's-δ = 0.2) between the groups, and only a 60% chance of detecting a large difference (Cohen's-δ = 0.8).

To create a symmetric sample group of baseline buildings, we subset the complete *Building Performance Database* for offices in climate zones 3C and 4C, with mechanical cooling, and floor area of 929–40,877 m^2 (10,000–440,000 ft^2) to match the range in the subset from the Higgins database. The resulting subset included 2,690 sample buildings and resulted in the site-EUI distribution shown in Figure 1.4.

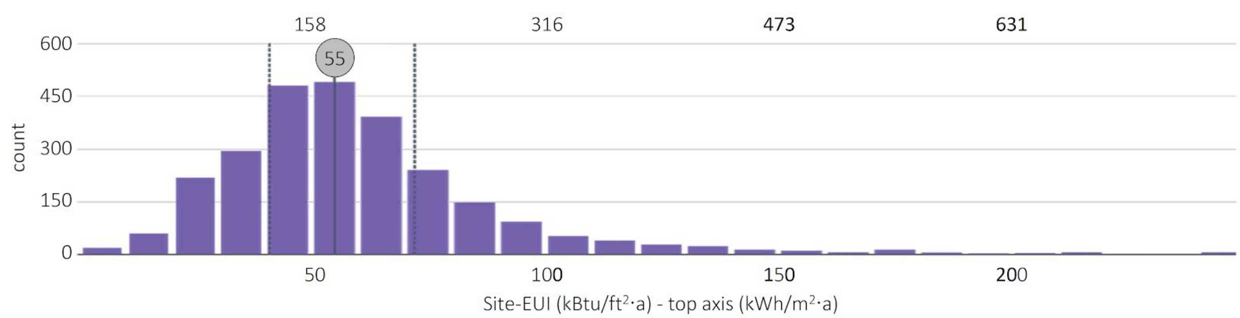


Figure 1.4: Histograms of site-EUI for office buildings from DOE Building Performance Database in climate zones 3C and 4C (Warm marine and Mixed Marine), and with floor area 929–40,877 m2 (10,000–440,000 ft2).

We observed that although a few buildings have site-EUI as large as 1,262 $kWh/m^2 \cdot a$ (400 $kBtu/ft^2 \cdot a$), the Shapiro-Wilk test confirmed that overall the distribution is normally distributed with median = 55 $kBtu/ft^2 \cdot a$ (173.5 $kWh/m^2 \cdot a$) and mean = 63 $kBtu/ft^2 \cdot a$ (198.7 $kWh/m^2 \cdot a$). This confirms that we can utilize conventional parametric statistical tests to compare the samples from the two groups. However, before making said comparison it is important to note that because our sample size for buildings with radiant cooling is very small, the result of a statistical test is highly sensitive to the presence of outliers in the small sample. Specifically, if any of our 8 samples were drawn by chance from the long tail of buildings with especially large site-EUI, their presence in the sample group would present a mean and standard deviation that does not realistically represent the population as a whole.

Figure 1.5 presents the distribution of site-EUI for the 8 sample subset. While 7 of the samples have site-EUI values in a typically expected range, one has a site-EUI = 454 kWh/m$^2 \cdot$a (144 kBtu/ft$^2 \cdot$a), which is larger than all but 2% of the samples in the baseline subset. The Shapiro-Wilk test confirms that the distribution is not normal when the sample is included in our subset, but that it is normal if the sample is excluded.

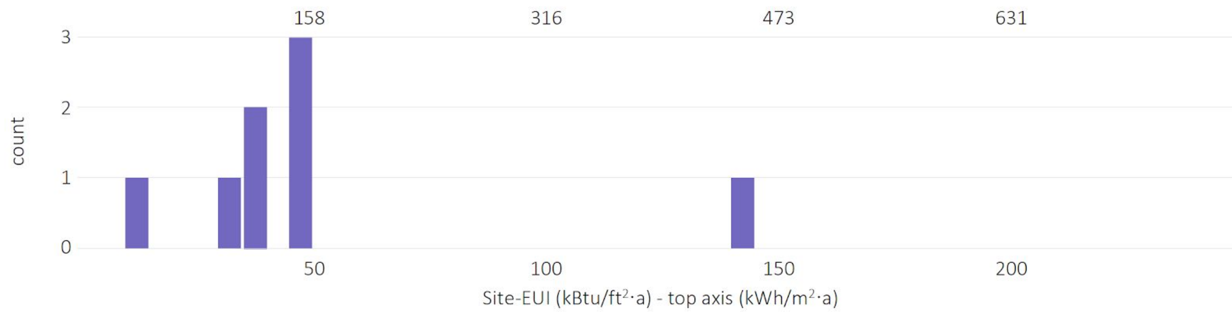


Figure 1.5: Distribution of site-EUI for an 8 sample subset of offices in climate zone 3C and 4C with radiant cooling (Warm marine and Mixed Marine), and with floor area 929–40,877 m^2 (10,000–440,000 ft^2)

Here, we require expert judgement and conceptual justification. If we proceed with the statistical test, such an outlier in a small sample group could result in a statistically insignificant result, and might even cause the site-EUI for radiant buildings to appear larger than that of baseline buildings. Although the specific reasons for very high site-EUI values may differ by building, we can safely claim that they are atypical cases. Some reasons for very high site-EUI include: poor maintenance or commissioning, special-case or out-of-category space-uses, and erroneous reporting. From a practical perspective, very high site-EUI buildings ought to be a type-category unto themselves as they defy comparison within the same sample as an otherwise regular population of buildings. We would not include data centers among a sample group to investigate the correlation between occupant density and site-EUI; similarly we should not include very high site-EUI buildings in samples to assess whether or not radiant cooled buildings have a lower site-EUI than the baseline.

Accordingly, we dropped the outlier from the sample group for radiant buildings, and restricted the range of site-EUI values included in the baseline sample group to a maximum of 442 kWh/m$^2 \cdot$a (140 kBtu/ft$^2 \cdot$a). The result was a sample group with seven radiant buildings and a sample group with 2,592 baseline buildings. To compare the two sample groups we used a two tailed t-test.

1.6.2 Results

The comparison of our samples reveals that office buildings with radiant cooling in western marine climate zones have a lower site-EUI than a comparable subset of the general building stock. The difference is large and statistically significant (p = 0.006, Cohen's-δ = 0.84). The difference in the medians is 53.6 kWh/m$^2 \cdot$a (17 kBtu/ft$^2 \cdot$a), which represents a relative difference of 45%. In these climate zones, offices with radiant cooling use 31% less site energy than comparable offices.

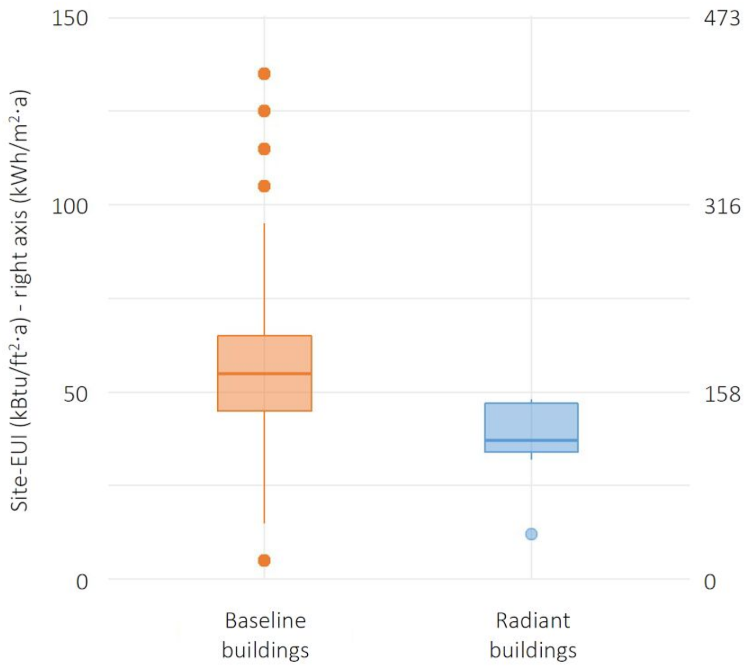


Figure 1.6: Distribution of site-EUI for baseline office buildings (orange), and office buildings with radiant cooling (orange). Data for baseline buildings is a subset from the US Department of Energy Building Performance Database (n=2,592). Data for radiant buildings is a subset from Higgins and Carbonnier (2017) (n=7).

1.7 CONCLUSIONS

Previous research has developed a strong case for the potential energy savings from radiant cooling. Many building energy simulations have compared radiant cooling and conventional all-air cooling; except for a few cases, researchers have predicted substantial advantages for radiant. However, most researchers have focused on the ideal cases, and very few have studied the energy performance achieved by radiant buildings in practice.

In this chapter we developed a consolidated inventory of the specific mechanisms by which radiant cooling can reduce energy use, and we considered counterposing factors that could erode the potential energy performance for buildings with radiant cooling if they are not proactively avoided.

Further, to assess whether or not buildings with radiant cooling use less energy in practice, we compared the measured site-EUI for a group of office buildings with radiant cooling to the site-EUI for a comparable subset of baseline buildings. We found that the buildings with radiant cooling use 31% less energy. Despite a relatively small sample group for radiant buildings, the analysis provides statistically significant evidence of a large difference between site-EUI for radiant buildings and site-EUI for comparable baseline building stock. However, we also explored many possible confounding factors that are likely to have played some role in the energy savings observed, or which make it difficult to claim – from a statistical basis – that radiant cooling reduces energy use any more than other advanced mechanical system strategies employed in high performance buildings. In conclusion, we find that buildings with radiant cooling have made relevant efficiency improvements over the general building stock, and that radiant cooling enables several outstanding efficiency opportunities on a path toward more responsible building energy systems.

CHAPTER 2

Structured interviews to catalog current practices for design and control of high thermal mass radiant cooling in North America

Reproduced in part from the following previously published co-authored works:

G. Paliaga, F. Farahmand, P. Raftery, J. Woolley, TABS radiant cooling design & control in North America: results from expert interviews, University of California, Berkeley; TRC Energy Services, 2017.

G. Paliaga, F. Farahmand, J. Woolley, Current Practice for Design and Control of High Thermal Mass Radiant Cooling Systems, and Opportunities for Future Improvements, in: ACEEE Summer Study on Energy Efficiency in Buildings, American Council for an Energy-Efficient Economy, 2018. https://escholarship.org/uc/item/4qg9276c.

Chapter Abstract

This chapter summarizes interviews with eleven professionals with substantial experience with design and operation of radiant buildings in North America. Interviews focused specifically on buildings with high thermal mass radiant cooling. Interviews revealed a diverse range of approaches for design and control of high thermal mass radiant systems. While interviewees expressed many similar approaches, they also have many unique preferences.

Examples of consistent themes include the use of dedicated outdoor air systems for ventilation and for supplemental cooling, and the use of relatively simple control schemes with constant setpoint for slab temperature (or indoor air temperature) for all times of the day and night. However, interviewees described unique preferences for space types that are appropriate for high thermal mass radiant, design and types of valves or pumps used for radiant zone control, the control of changeover between slab heating to slab cooling, and many other design considerations.

Preferences appear to be driven by project constraints and by personal experience. Interviewees report that their design preferences are effective, but there is no industry consensus about how alternatives compare for energy performance. This chapter outlines opportunities for further research, improvement radiant design and control, and the development of best practices.

2.1 INTRODUCTION

Radiant cooling and heating have the potential for improved energy efficiency, demand response, comfort, indoor environmental quality, and architectural design. Many radiant buildings have demonstrated outstanding performance in these regards, and the technology's application in commercial buildings appears to be expanding, especially among high performance buildings and zero net energy buildings (Higgins, 2016; Higgins & Carbonnier, 2017). However, most buildings industry professionals are unfamiliar with radiant systems.

The design and control of high thermal mass radiant cooling is especially unfamiliar to industry professionals. These systems cool building masses from the inside out, which uniquely decouples the time of cooling plant operation from the time that heat is extracted from the occupied space. Consequently, an air temperature thermostat cannot directly control the space cooling rate the way a typical forced air system operates. Control strategy choices for high thermal mass radiant cooling systems can be expected to impact comfort and energy use, yet there are not well-established guidelines for design of radiant buildings and their control systems.

Therefore, the objective of the research reported in this chapter was to document current design and control practices for high thermal mass radiant cooling systems by interviewing professionals with subject expertise. In this chapter, we review the project scope and methodology, and then present information about current practices related to: systems configuration design, sequences of operation (SOO), and system commissioning. In view of these findings, we identify ways that TABS radiant building design could be improved, and discuss needs for future research.

2.2 METHODOLOGY

As part of the California Energy Commission (CEC) Electric Program Investment Charge (EPIC) project Optimizing Radiant Systems for Energy Efficiency and Comfort, and in conjunction with the Center for the Built Environment (CBE) at University of California at Berkeley, TRC Energy Services conducted research to investigate best practices for design and control of TABS radiant cooling systems for commercial buildings. The goal of this portion of the EPIC project was to summarize current best practices as reported by experts. Research consisted primarily of interviews with radiant cooling design experts, with related exploration into previous literature and designer SOO. Radiant cooling was the focus of the research, but findings related to radiant heating are included when pertinent to radiant cooling design and control.

We conducted eleven interviews with radiant cooling experts in 2016. These interviewees were individuals within our industry networks, with practical experience in design, construction, and operation of TABS radiant buildings. We used a structured interview method to obtain responses to the same topic areas, and asked interviewees to share their typical and/or preferred design approaches, including motivations for each design approach, design tradeoffs, and challenges associated with implementation. We asked that responses focus on design and control of TABS (rather than radiant panels or embedded surface systems) for projects in North America. Interviews lasted one hour and included questions in each of the following topic areas:

- Interviewee background
- System configuration
- Controls and sequence of operation
- System commissioning

We analyzed and summarized interviewee responses for each question, paraphrased quotes that help capture key ideas, categorized the variety of responses to each question, and tabulated the count of responses in each category. The categorization of responses was developed after the interviews to group common themes – the categories were not a predetermined set of options. Response categorizations were emailed to the designers for review, and many provided both confirmation and corrections to our original categorizations.

2.3 INTERVIEW FINDINGS

We interviewed eleven prominent professionals who have designed a combined total of approximately 330 radiant cooling projects. Most often, interviewees were the lead design engineer, but occasionally served as the overseeing principal or consultant to the architect. We focused on experts working in North America, and therefore most of the radiant cooling projects discussed in these interviews were in the United States and Canada.

Interviewees consistently described the following key characteristics of radiant cooled buildings that underpin design approaches and how they control radiant systems:

- The upper limit of cooling capacity from TABS is lower than conventional air systems. It is important to reduce building envelope and internal loads, and supplemental cooling may be required too.
- TABS' high thermal inertia results in very slow changes in temperature at the indoor face of internally cooled surfaces. This is both an advantage and a challenge.
- The cooling capacity from TABS is somewhat self-regulating because heat transfer to the cooled slab surface naturally and instantaneously responds to changes in air and other surface temperatures.

In addition to these broad takeaways, the following sections summarize findings related to system configuration, control sequences, and commissioning. In each topic area, we highlight the common practices and then discuss the variety of practices.

2.3.1 System configuration

We asked interviewees about several different aspects of system configuration focused on:

- Slab configuration
- Supplemental cooling
- DOAS design
- Zoning

Common practices and variation in practices are discussed below.

Common practices

Most TABS designers prefer to embed radiant tubing within the structural slab. This approach is less costly than pouring a topping slab, and activates the entire thermal mass. However, sometimes tubing is located in a topping slab for various reasons; usually this is installed without insulation between the structural and topping slabs to maximize thermal mass.

Almost all radiant cooled buildings also use radiant systems for heating. Rarely, heating is provided with an alternate method, such as perimeter radiant panels. Not all radiant heated buildings use the radiant systems for cooling.

Interviewees consistently said that radiant systems have smaller cooling capacity than air systems and that indoor conditions respond slowly to changes in chilled water temperature or flow rate. Furthermore, many designers prefer large radiant zones – some even aim to control the entire floor plate as a single zone. For these reasons, interviewees emphasized a need for high-performance envelopes to minimize the magnitude and variation in heat gain, particularly to ensure that perimeter areas are not subjected to excessive variation in heat gain as compared to interior areas. At the same time, many interviewees noted that radiant cooling can remove direct solar radiation that strikes the indoor face of internally cooled surfaces much more rapidly than other types of heat gain; and for this reason, radiant floor cooling is sometimes specified in spaces with larger than normal solar gains.

Most radiant system designers include supplemental cooling – sometimes in select zones and sometimes in all zones. Supplemental cooling maintains comfort when gains exceed radiant system capacity, enables tighter temperature control in specific areas, and provides short term cooling capacity in spaces with highly variable gains (such as conference rooms). Designers use dedicated outside air systems (DOAS) ubiquitously to provide fresh air ventilation in radiant buildings and often also use the DOAS to provide supplemental cooling by adjusting volume flow, supply air temperature, or both together. Two general methods for supplemental cooling that most interviewees have used include:

1. In zones where radiant is expected to provide most of the needed cooling capacity, interviewees provided supplemental cooling with the DOAS ventilation system, either by increasing the delivered flow rate, or by decreasing the supply air temperature below space temperature. To enable supplemental cooling, the DOAS maximum airflow rate is typically sized above code minimum ventilation requirements by 20 to 30%, and cooling capacity is sized accordingly to achieve desired supply air conditions.
2. In zones where heat gains are expected to greatly exceed TABS cooling capacity, most interviewees include fan coils, radiant ceiling panels, or variable-air-volume air supply for supplemental cooling.

Variation in practices

Significant differences between interviewees related to system configuration design included:

Appropriate space types for radiant cooling. Interviewees were divided between those that have only included radiant cooling in specific space types (lobbies, atrium, open plan spaces, etc.) and those who have had success with radiant cooling in a wide variety of space types –

including private offices and high-density spaces with variable gains such as classrooms and art galleries.

- A few designers noted that offices pose difficulties for TABS, including acoustics, management of small individual thermal zones, and the need to accommodate flexibility for future tenant reconfiguration.
- Designers who feel that radiant systems are appropriate for most occupancy types, state that supplementary systems can be used to fine-tune conditions in individual spaces or when reconfigurations occur, even when radiant floor zoning has large zones by orientation and interior/perimeter.

Zone valves or pumps. Some designers prefer achieving zone control with valves, while others strongly prefer circulator pumps.

- Interviewees that preferred valves were divided in their preference for radiant floor control valve type. Most preferred 2-position on/off valves that effectively pulse water into the radiant zone with on/off control, while others prefer modulating valves that continually modulate flow.
- Interviewees that preferred pumps indicated that in buildings with a small number of zones that are fairly close to each other, valves can be used, but in instances where there are many zones (or large zones) circulator pumps are used.

Two-pipe versus four-pipe distribution systems. Approximately half of interviewees use two-pipe distribution for the entire building, meaning all radiant zones must be in the same mode, either heating or cooling. The other half of interviewees are evenly split between: (a) providing four-pipe distribution to the zone level, or (b) providing four-pipe distribution to sections of the building with two-pipe distribution continuing to groups of zones. Solution (b) is a way to balance first costs with level of control – by limiting four-pipe distribution to sections of the building that may need to be in different modes (heating or cooling) such as each floor, by orientation, or by floor and orientation. Interviewees who use two-pipe distribution explained that with a well-designed envelope, the need for heating and cooling should change slowly over the year, and that the slab setpoint will be essentially neutral during swing seasons.

Supplemental cooling design. Interviewees had a wide variety of approaches for zoning and controlling DOAS systems to provide supplemental cooling, and these were often designed in response to the unique needs of each building.

- On one extreme the DOAS system has VAV boxes at every zone, although most interviewees try to avoid this design because of the high initial cost.
- More commonly, the DOAS can vary flow or temperature at the air-handling unit (AHU) without any zone control, to provide supplemental cooling to all zones.
- The DOAS system may have a limited number of zone dampers that are either pressure independent (VAV boxes) or pressure dependent (simple zone dampers).

2.3.2 Controls and Sequences of Operation

We asked interviewees about several different aspects of design of controls and SOO:

- Slab temperature control
- Zone air temperature control
- Interaction with supplemental cooling and mode changeover
- Condensation control
- DOAS control

Common practices and variation in practices are discussed below.

Common Practices

Interviewees explained that the cooling capacity of TABS systems naturally adjusts to temporal and spatial variations in heat gains. This self-regulation occurs because heat transfer to the slab surface instantaneously responds to changes in the surrounding air temperature and changes in the temperature of other surfaces in the space. Interviewees noted that this characteristic is a critical design consideration. Self-regulation is the reason that radiant systems can maintain comfort throughout large zones even though slab surface temperatures respond slowly to changes in chilled water temperature or flow. The temporal and spatial granularity of zone control for radiant systems is typically much coarser than for typical variable-air-volume all-air systems.

Indoor conditions in TABS buildings do not respond quickly to changes in supply water temperature or flow rate; therefore, the type of reactive control strategies traditionally used for conventional variable-air-volume all-air systems are not especially useful for high mass radiant systems. Instead, controls for TABS radiant buildings are configured to target a slab temperature setpoint – measured with an embedded slab temperature sensor – by adjusting chilled water supply temperature or flow rate. Almost all interviewees shared that TABS buildings are controlled to maintain relatively constant slab temperature setpoint round-the-clock, instead of a zone air temperature set point. Each zone may have a unique slab temperature set point that is selected to result in comfort within the space throughout the day. These set points are usually programmed to change slowly over the course of the seasons, or as a function of recent outside temperatures. Only a few interviewees allow setbacks during vacant periods, and typically they are small setbacks. Choosing the appropriate relationship between slab temperature set point and outside air temperature typically requires tuning during the first few seasons of operation.

Several interviewees indicated that alternate plant designs could avoid the need to generate low temperature chilled water throughout the year, including use of night sky cooling, ground source or water source heat pumps, and water side economizing. While these were indicated as desirable strategies, most radiant buildings use conventional chillers that operate a typical low chilled water temperature. Further, most interviewees do not attempt active load shifting to reduce mechanical equipment operation during peak demand hours or take advantage of improved equipment efficiency overnight. One designer that had attempted load shifting explained that the strategy was ultimately abandoned to avoid the risk of morning discomfort, and instead adopted an approach that always maintains a constant slab temperature.

Usually, the control of radiant systems and supplemental cooling systems are not explicitly coordinated: radiant systems are controlled to maintain slab temperature and supplemental systems are controlled to maintain air temperature. None of our interviewees had controlled DOAS equipment to provide economizer cooling when outdoor conditions are appropriate. However, about half of our interviewees have used natural ventilation for thermal regulation in radiant buildings.

Avoiding condensation on internally cooled surfaces is important, but not difficult, and can be addressed through design and appropriate water temperature set points, surface temperature setpoints and DOAS systems dehumidification. Most avoid condensation by dehumidifying ventilation air to a dew point temperature or relative humidity target, then by maintaining chilled water temperature supplied to the slab above the dew point. Almost no interviewees had encountered condensation in practice. Those few that had encountered condensation problems attributed the issues to unusual situations (often during commissioning) or improper operation. The issues were resolved through operator training and control sequence revisions. In the collective experience of our interviewees, nobody had experienced ongoing issues with condensation.

Variation in Practices

Significant differences in design of controls and SOO between interviewees include

Space temperature set points. Some designers recommend space air temperature set points that are like those used in conventional HVAC systems, while others advocate for radiant systems to operate with a wider dead band between heating and cooling.

Condensation risk. Interviewees differed on whether or not to include active controls to reduce condensation risk. Condensation control is climate dependent, which may explain some of the variation in approaches, although some of these contrasting approaches were used in the same climate:

- Some interviewees emphasized that active control of slab supply water temperature and/or DOAS dew point are critical to prevent condensation. A few interviewees explained that as a fail-safe, they include simple moisture switches located on the CHW supply pipe near the zone manifold that disable CHW supply when condensation is detected.
- Other interviewees emphasized that they do not need active control when a system is engineered to never reach a condensation condition during normal operating conditions. Indoor humidity is used only for monitoring and alarming.
- Some designers ensure that chilled water temperature stays at least 2°F above the dew point, while others allow the chilled water temperature to drop below dew point as long as the slab surface temperature does not.

Slab temperature sensor location. All interviewees measure slab temperature, but were divided in their preference for slab temperature sensors located at the depth of radiant tubing versus near (or at) the surface of the slab. Most locate a sensor in-between the tubes, and at the same level as the tubes. Three interviewees locate the sensor near the slab surface, at a depth of 1-2 inches. For large zones some interviewees averaged multiple slab temperature sensors.

Chilled water plant size. About half of our interviewees shared that TABS buildings influence the sizing of a chilled water plant, while the other half specify a plant that is the same size as it would be for an equivalent building with conventional variable-air-volume all-air cooling. One interviewee explained that chiller equipment could be smaller if a TABS building were controlled to store thermal energy like a flywheel, but that it is difficult to control such a system without risking discomfort occasionally, and that customer and operator expectations do not usually allow for such a control strategy.

Chilled water plant supply temperature. Without exception, radiant cooling systems operate with higher chilled water temperature (at the zone) than typical air handling systems to help reduce the likelihood for condensation and discomfort. However, about half of interviewees design to generate chilled water at low temperatures typical of conventional buildings (mid-40°F), then blend with return water from radiant systems to achieve an appropriate radiant supply water temperature. This control decision is driven by a need for low temperature chilled water used for dehumidification, or for conventional systems (e.g., VAV, fan coils) in areas of the building where radiant cooling was not included. Interviewees that had designed central plants that supplied warmer chilled water for the radiant floor described the following strategies:

- Two chilled water plants that supply different temperatures,
- Chiller in series with the lead chiller generating warmer temperature water for radiant cooling, or
- Chilled water plant supplying warmer water to the radiant system and DX used for dehumidification in DOAS air handlers and/or conventional variable-air-volume all-air systems.

Heating and cooling mode changeover. Interviewees had a wide variety of approaches to control changeover between radiant slab cooling and heating modes, and emphasized the need for tuning changeover during the first year of occupancy. Interviewees stated several different ways to control changeover:

- Slab temperature and/or fluid temperature setpoints are reset based on season or trailing mean outside air temperature, resulting in slab temperatures often near space temperature when changeover occurs. This is usually combined with other strategies listed below.
- Delay changeover for multi-hour periods where the radiant slab is off. Interviewees and SOO we reviewed noted a slab lockout time in the range of 2 to 24 hours and that this parameter often needs to be adjusted in the field.
- Measure slab temperature to ensure it has reached space temperature (fully discharged) before changeover is allowed.
- Adhere to *ASHRAE Standard 90.1* requirements for two-pipe system changeover time delay requirements.
- Limit slab heating mode and cooling mode to operate within a certain range of outside air conditions (e.g., both modes may be disabled between 65°F and 75°F).
- Rely on natural ventilation to condition the space for the period in between active heating and cooling.

Although many interviewees mentioned that there are long periods (weeks or months) between the need for slab heating and the need for slab cooling, many designers allow the slab to changeover in a matter of days or hours depending on recent weather conditions and real-time demand. Interviewees were also divided in their concern that changes in mode can lead to energy waste when a slab changes mode too quickly, with some saying the situation should be avoided but is occasionally needed to maintain comfort. Other interviewees noted that a change in mode for the slab is only a difference of a few degrees, so that changeover in the same day is not of extreme consequence if there are appropriate delays that avoid shocking the central plant.

2.3.3 System commissioning

Interviewees told us that it is usually necessary to tune up radiant buildings during the first year of occupancy and to educate controls contractors and operations staff about proper system setup and management. Typically, buildings require unique settings based on building characteristics as well as climate zone that must be determined during occupancy and often require expert designer input to fine tune. Adjustments that typically occur include:

- Slab temperature set point for each zone
- Seasonal slab temperature set point reset
- Supplemental cooling quantity (usually determined by DOAS supply air temperature)
- Flow in individual radiant loops with manual balancing valves at the manifold.

All designers said that building operators need education to understand and operate a radiant cooled building properly, as radiant buildings are controlled very differently than buildings with other types of HVAC systems. Designers often stay engaged for the first year of occupancy to ensure project success even when they were not retained for ongoing commissioning services.

2.4 OPPORTUNITIES FOR FURTHER RESEARCH AND IMPROVEMENT IN COMMON PRACTICE

Interviewees often explained their different engineering solutions as being responsive to the varying needs of each application – including unique solutions for each building, owner, and climate. However, many interviewees seemed flexible in their approach, which yields opportunities for improving common practice. Where differences in design approach exist, there may be opportunities for refinement and improvement of design solutions. We provide commentary to identify opportunities to improve common practice and opportunities for further research. Note that this is a partial list suggested by the authors and not representative of the interviewees' opinions.

Self-regulation. The self-regulating nature of radiant TABS is a critical design consideration and key to the success of radiant buildings. Despite the importance of self-regulation, interviewees did not have quantitative information on the magnitude of the effect, response time, or specific approaches to design decision making (e.g., when self-regulation is acceptable versus when zonal supplemental cooling is required). Assumptions about self-regulation have a large impact on zoning and supplemental systems design, and it is likely that many radiant buildings rely too much on supplemental cooling when they could rely on self-regulation. Fully accounting for self-regulation could reduce system cost by avoiding unnecessary supplemental systems and control system complexity. Primary research and a literature review of published research on the self-regulation of radiant cooling should be used to develop quantitative design tools accessible to designers.

Heating and cooling mode changeover. Given the variety of methods used, a simple and standardized approach to heating/cooling changeover control would be helpful. Controls could address seasonal or weather based resets, lockouts between heating and cooling modes, and prevention of overshoot and energy waste when modes change too quickly.

Pre-cooling. The thermal mass and large response time for TABS can allow control sequences that strategically shift cooling plant operation to times when electricity is less expensive, or when outside temperature is better for cooling plant efficiency. However, we learned that very few TABS buildings actively employ these strategies. Many interviewees recognize this opportunity but have concerns such as risk of thermal discomfort, or limited energy savings potential because the slab temperature can only be reduced a small amount when considering the large thermal time lag of the mass. One of the challenges is a lack of algorithms to predict pre-cooling response. A few interviewees said that weather based predictive control would be useful for radiant cooling but also noted that there are no proven algorithms that they could rely on.

Chilled water plant supply temperature. Radiant cooling can reduce energy use by operating at a relatively warm chilled water temperature; this can improve chiller efficiency, and enable the use of chiller-less water-cooling strategies. Many TABS buildings do not take advantage of this opportunity. Our interviews revealed that chilled water is often generated at a low temperature to provide dehumidification, or to serve forced air cooling in portions of the building that do not include radiant, and then mixed with return water to supply higher temperature chilled water to the radiant systems. Some radiant designers use separate chillers for the separate purposes, but cost concerns often result in a single low temperature chiller plant for the whole building. Life-cycle cost analysis of various cooling plant solutions, including the variety of solutions used by interviewees, would reveal the most cost-effective solutions and help justify the investment in a more efficient cooling based on energy cost savings.

Chilled water temperature and condensation risk. Most designers are careful to keep chilled water temperature well above the dew point. However, since the slab surface temperature is always warmer than the chilled water supply temperature, at least one designer allows supply water temperature to drop below dew point, while remaining careful to keep the slab temperature above dew point. Operating at lower chilled water temperatures and associated lower slab surface temperature increases the cooling capacity of a radiant floor while increasing the risk of condensation. Condensation on chilled water piping is prevented by standard insulation and vapor barrier details, but it is unclear if there is risk of condensation on radiant manifolds and piping between the manifold and slab. Designer decisions for chilled water temperature have impacts on both the energy used to cool the slab and dehumidify ventilation air.

Two-pipe versus four-pipe distribution. Some interviewees used two-pipe hydronic distribution systems to reduce piping costs. They suggested that a high-performance envelope reduces the need for different zones to be in different modes, as well as rapid mode changeover in the same zone, thus eliminating the need for a four-pipe system. Interviewees that used four-pipe distribution systems either to the zone or groups of similar zones sought improved control and comfort at each zone level. Analysis of cost tradeoffs between four-pipe distribution and building heat gain management (improved envelope, reduced internal loads, etc.) may reveal the most cost-effective balance for specific types of buildings in specific climates. In addition, quantification of radiant self-regulation (discussed in the first bullet point) is required to determine when a two-pipe system is insufficient to maintain comfort.

Radiant and supplemental cooling interlocks. Regardless of how the radiant and supplemental control loops are interlocked, the control loop for supplemental cooling systems always responds more rapidly to changes in space temperature than does the control of massive radiant systems. In the end, it is not clear if the SOO typically used to interlock radiant and supplemental systems minimizes energy use for the two systems combined. Interviewees did not comment on the relative cooling energy cost between supplemental cooling versus radiant cooling and how to minimize it. In addition, many interviewees specify controls without any interaction or lockout between DOAS air systems controls and radiant slab controls – which we believe could lead to "fighting" between the DOAS and radiant floor. Interviewees were not concerned about potential fighting, often citing the very small temperature differences. None of the interviewees modulate DOAS supplemental cooling based on availability of free cooling (economizer operation) even though the energy cost of supplemental cooling in this situation may be less than radiant cooling energy cost. In all cases there appears to be an opportunity to reduce energy use with control sequences that consider the benefits and tradeoffs of both supplemental cooling and slab cooling.

Supplemental cooling control. Supplemental cooling design and control is usually a critical piece of the overall radiant system solution and interviewees had a wide variety of approaches. Many used novel solutions to reduce cost and avoid a fully variable-air-volume DOAS system. Interviews did not have sufficient time to delve into the nuance and variety of DOAS system design and how the DOAS is controlled to provide supplemental cooling (also see the previous item). Further investigation into this topic would be useful.

Supplemental heating. Only a few projects used supplemental heating in addition to radiant slab heating. We suspect that the need for supplemental heating only occurs in very cold climates, but we did not have time to determine if this design decision occurs only in cold climates, nor what the outdoor design condition threshold might be that triggers it.

Ceiling fans. Although many interviewees recognized that ceiling fans could extend the comfort envelope, reduce stratification, and increase the convective space cooling capacity for radiant systems, few had ever utilized the strategy. Several interviewees mentioned that many architects generally do not consider ceiling fans a viable design option, and others explained that better information is needed about the specific benefits to advance the design strategy. A couple interviewees mentioned that ceiling fans would disrupt stratification from radiant floor cooling and displacement ventilation. Those who have utilized ceiling fans said that they consider air movement as a factor that impacts thermal comfort, but that they have not specifically considered the increase in convection heat transfer when designing the radiant system. Others were hopeful about including the strategy in the appropriate circumstances. Further investigation into the energy impacts of ceiling fans in radiant applications would be useful.

System commissioning. Typically, radiant buildings require unique settings that need to be determined during occupancy and often require expert designer input to fine tune. Designers noted that these improved industry education, or development of self-tuning control sequences could help to address these challenges.

Terminology. There is a wide range of different terminology and understanding among experienced designers in the same field. For example, designers indicate that different terminology is used for design of TABS versus embedded surface systems (which have lower 'activated' mass because they are isolated from a structural slab by insulation). Variations in terminology and system design approach presented challenges during interviews and analysis of interview results. There is an opportunity to create a topology of radiant cooling that is more inclusive of the various aspects of radiant systems, and rigorously defined.

2.5 CONCLUSIONS

Radiant cooling can reduce energy use and electrical demand for heating, cooling and ventilation, but most design professionals are not familiar with the technology. We interviewed experts in the field to document the current practices for design and control of radiant cooling systems. This chapter documents the landscape of current practice for design and control of TABS buildings in North America.

We found that in some aspects there are common themes for how radiant cooling is normally employed, but in other aspects there are a wide variety of strategies and opinions among experts. There are significant differences between the various design and control strategies currently employed in radiant buildings which have clear implications for energy performance and comfort. Some of these differences are driven by project constraints, while others appear to be driven by designer preference, or by individual understanding about the behavior and capabilities of radiant systems. Radiant cooled buildings are often not designed or controlled to effectuate some of the well-recognized efficiency opportunities conferred by the technology. For example, more than half of radiant cooled buildings use conventional chillers and operate them at typical low chilled water temperatures. Furthermore, all of our interviewees designed radiant buildings that were controlled to maintain a slab temperature setpoint continuously, and none had proactively controlled radiant cooling systems for nighttime operation to avoid peak electrical demand periods, to improve cooling plant efficiency, or to reduce the required size for a cooling plant.

To support broader adoption of radiant cooling, we see a need for standardized sequences of operation and rigorous iterative improvement of these sequences based on feedback from completed projects. Our findings can inform the development of industry standards, design guidelines, resources for education and training, and best practices for design and control of radiant systems. Our interviews also revealed gaps in research where the most efficient strategies have not yet been evaluated, such as an assessment of the coordination between radiant systems and supplemental cooling systems.

CHAPTER 3

Side-by-side laboratory comparison of space heat extraction rates and thermal energy use for radiant and all-air systems

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Chapter Abstract

Radiant cooling systems extract heat from buildings differently than all-air cooling systems. These differences impact the time and rate at which heat is removed from a space, as well as the total amount of thermal energy that a mechanical system must process each day. In this chapter we present measurements from a series of multi-day side-by-side comparisons of radiant cooling and all-air cooling in a pair of experimental testbed buildings, with equal heat gains, and maintained at equivalent comfort conditions (operative temperature). The results show that radiant cooling must remove more heat than all-air cooling – 2% more in an experiment with constant internal heat gains, and 7% more with periodic scheduled internal heat gains. Moreover, the peak sensible space heat extraction rate for radiant cooling (heat transfer at the indoor face of the internally-cooled surface – not the cooling plant) must be larger than the peak sensible space heat extraction rate for all-air systems, and it must occur earlier. The daily peak sensible space heat extraction rate for the radiant system was 1–10% larger than for the all-air system, and it occurred 1–2 hours earlier. These findings have consequences for the design of radiant systems. In particular, this study confirms that space cooling load estimates for all-air systems will not represent the space heat extraction rates required for radiant systems.

3.1 INTRODUCTION

Radiant cooling and heating could be a pathway to reduce energy use and peak electrical demand in buildings compared to conventional all-air systems. A recent survey assessment of commercial building energy consumption in the United States indicated that the median energy use intensity for buildings with radiant cooling is 14–66% lower than standard buildings of comparable type and climate zone (Higgins & Carbonnier, 2017). Although radiant cooling is currently installed in a small portion of buildings overall, it is a common strategy among buildings with the lowest energy use intensity (Maor & Snyder, 2016; NBI, 2012; Paliaga et al., 2016). The number of high performance buildings with zero-net-energy aspirations has increased rapidly in recent years (Higgins, 2016), and consequently application of radiant cooling appears to be expanding.

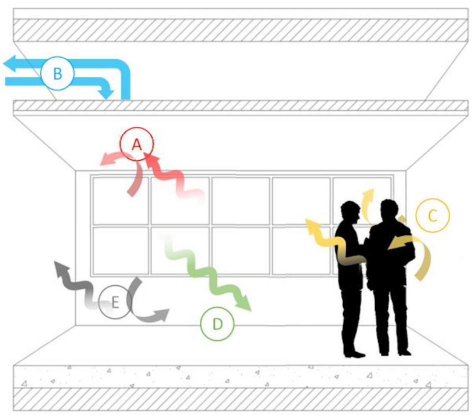
Several researchers have identified reasons that radiant cooling can reduce energy consumption and peak electrical demand compared to all-air systems. In Chapter 1 we consolidated the variety of explanations into a summary list of five fundamental energy advantages for radiant cooling, and discussed several ways that they avoid inefficiencies that are common for all-air systems in practice.When these advantages operate together, the potential savings for radiant cooling can be high. Numerous simulation studies and field evaluations have concluded that radiant cooling can consume much less energy than conventional all-air systems.

There has been substantial research to develop and validate building energy simulation tools that properly capture the fundamental heat transfer mechanisms involved with radiant cooling systems (Chantrasrisalai et al., 2003; Niu et al., 1995, 1997; Strand et al., 1999; Strand & Baumgartner, 2005; Strand & Pedersen, 2002; Yu et al., 2014). Yet despite the variety of simulation studies that have utilized these tools to compare the primary energy performance of radiant and all-air systems, only Feng et al. and Niu et al. have explicitly compared the dynamic space heat extraction requirements (space cooling load) for radiant and all-air cooling systems (Bauman et al., 2013; Feng, 2014; Feng et al., 2013, 2014b). Through simulation and laboratory experiments these researchers demonstrated that to maintain equal comfort:

1. Radiant cooling systems must extract heat from gains earlier than all-air cooling systems
2. Envelope heat transfer rates are different for radiant and all-air cooling systems
3. The daily maximum space heat extraction rate is larger for radiant cooling systems
4. The total amount of heat extracted each day is larger for radiant cooling systems

The dynamic space heat extraction requirement (space cooling load) is crucial for design, sizing, and control of any cooling system, yet as Feng et al. (Feng, 2014; Feng et al., 2014a) highlighted, industry common practice methods for design sizing of cooling systems do not properly capture the differences between radiant and all-air systems.

The space heat extraction rate is the rate at which heat is removed from a space by terminal heat transfer devices. The instantaneous space heat extraction rate required to maintain comfort is not equal to the instantaneous sum of heat gains in a space because a portion of the heat gains is absorbed by non-active masses and does not immediately result in a need for space cooling. For all-air systems the space heat extraction rate is the sensible enthalpy difference between supply and return (or room air outlet) air flows. For radiant systems the space heat extraction rate is the sum of convective and radiant (longwave and shortwave) heat transfer rates at the indoor face of the internally cooled surface. For high thermal mass radiant systems, the space heat extraction rate will be much different from the rate at which heat is transferred to the hydronic system. Generally, design of a cooling system should begin with an assessment of the space heat extraction rates that will be required to counterbalance the effect of expected heat gains in order to achieve desired comfort conditions. When this is known, mechanical systems and controls can be designed with the ability to provide the required space heat extraction rates. Each of these heat transfer rates are defined in Figure 3.1.



(A) **Space heat extraction rate:** the rate at which thermal energy is removed from a space by terminal heat transfer devices.

Radiant system: sum of the convective and radiant (longwave and shortwave) heat transfer rates at the indoor face of the internally cooled surface.

All-air system: sensible enthalpy difference between supply and return air flow (not shown in figure).

(B) **Hydronic heat extraction rate:** the rate at which thermal energy is removed by the hydronic system, calculated as the enthalpy difference between supply and return chilled water flow.

(C) **Internal heat gain rate:** the sum of convective and radiant (longwave and shortwave) heat gain from sources within a space (eg: people, lights, equipment).

(D) **Solar heat gain rate:** the rate at which electromagnetic energy from the sun is transmitted through glazing and absorbed by surfaces within or enclosing a space.

(E) **Envelope heat transfer rate:** the net sum of convective and radiant heat transfer rates at the indoor face of outdoor exposed surfaces.

Figure 3.1: Illustration and definition of terminology used throughout this chapter to describe various heat transfer rates. In this chapter we are principally concerned with the difference between the space heat extraction rate required by radiant and all-air systems to maintain equivalent comfort conditions (space cooling load). In all circumstances this chapter is limited to sensible heat transfer only.

In this chapter we expand on the current understanding of radiant cooling with observations from simultaneous tests of radiant cooling and all-air cooling in side-by-side experimental testbed buildings. The specific objectives of the comparison were to observe differences in:

1. The dynamic space heat extraction rates required to maintain equivalent comfort
2. The cumulative amount of thermal energy extracted by each system
3. The distribution of thermal energy in masses in each testbed

To be clear, this chapter is principally concerned with comparing the space heat extraction rates that are required by radiant cooling and all-air cooling systems to maintain a desired operative temperature (space cooling load). We do not address the multitude of considerations that must be made for design of an associated cooling plant, thermal distribution systems, and controls.

Prior to the work presented in this chapter, only one previous experiment study (Feng et al., 2014b) had compared the space heat extraction requirements (space cooling load) for radiant and all-air systems. That study provided foundational evidence about the differences between these systems, but it imposed atypical heat gains, used a relatively small adiabatic environmental chamber, and only observed differences over a single heat gain cycle. We build on the conclusions of Feng et al. (2014b) by comparing the two system types in more realistic circumstances, with various heat gain schedules, and over an extended period of time.

3.2 METHODOLOGY

We conducted a series of controlled experiments in a pair of equivalent testbed buildings – one with radiant cooling and one all-air cooling. The testbed buildings at Lawrence Berkeley National Laboratory (FLEXLAB) enable thorough assessment of building energy systems at a realistic physical scale, with naturally occurring solar gains, and natural interaction with the surrounding environment. For each experiment we operated the two testbeds simultaneously, imposed equivalent internal gains, and controlled each system to maintain equivalent operative temperatures.

In this chapter, we present results from two experiments, one with constant internal gains, and one with periodic internal gains. We operated each experiment for several days, during which we monitored thermodynamic states and heat transfer rates in both testbeds. It is important to compare these systems over the course of several days to ensure that the temperature of masses in each testbed reach steady-state oscillations that are no longer influenced by the initial states of each system.

For our comparison of radiant and all-air cooling we measured: air temperature distribution, operative temperature distribution, temperature of surfaces and masses, dynamic space heat extraction rates, and the cumulative amount of thermal energy extracted by each system. We did not assess the electrical performance for either system; our investigation focused on fundamental thermodynamic differences between radiant cooling and all-air cooling, regardless of the primary cooling sources and mechanical system elements that either may employ.

3.2.1 Experimental facility

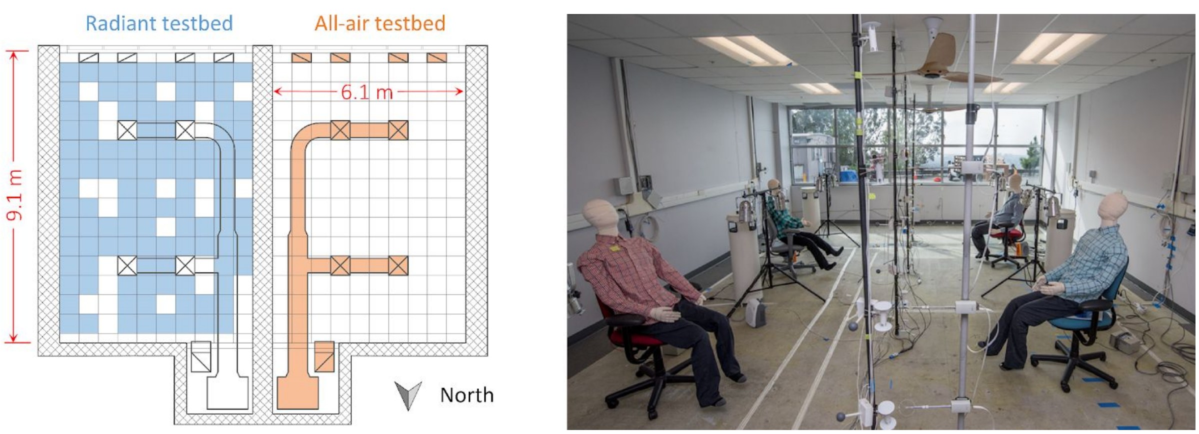


Figure 3.2: Plan view of testbed buildings (left), and photo of experimental setup (right). Air handler, overhead ductwork, supply diffusers, and return registers in the all-air testbed are highlighted in orange. Low thermal mass metal ceiling panels in the radiant testbed are highlighted in blue.

The experimental facility consisted of two side-by-side testbed buildings, illustrated in Figure 3.2. Each testbed had 57.6 m^2 (620 ft^2) floor area (6.1 m (20 ft) by 9.1 m (30 ft) interior dimensions, excluding the equipment room) and a 3.66 m (12 ft) high ceiling, with a drop ceiling at 2.74 m (9 ft). The floor was a 15.25 cm (0.5 ft) thick concrete slab with no additional floor covering. The southern wall conformed to *ASHRAE Standard 90.1* (2010), with 30% window-to-wall ratio and no exterior shading. All other walls, the ceiling, and the floor were very well insulated (U≤0.017 W/m^2-K); in this way each testbed approximated a single perimeter zone in a larger office building, where the majority of the zone boundary is adjacent to other similarly conditioned zones.

Both testbeds included an independent air handler with overhead supply air distribution and drop-ceiling return plenum. The air handlers were in equipment rooms within the thermal boundary of each testbed.

In the radiant cooling testbed the air handler circulated air at a constant 135 m^3/hr (80 cfm), a flow rate representative of typical ventilation rates in radiant buildings (ASHRAE, 2016b; CEC, 2016; Paliaga et al., 2017, 2018). The circulated air in the radiant testbed was not conditioned. We chose to include air circulation in the radiant testbed to mimic the air movement characteristics that could be expected in a real building with neutral temperature ventilation air flow. In the all-air testbed the air handler circulated air at a constant flow rate of 1000 m^3/hr (590 cfm) and a proportional integral control sequence adjusted supply air temperature to control the operative temperature. Neither testbed had ventilation air, and the infiltration rate in both testbeds was very small. Tracer gas decay tests indicated infiltration rates of 0.169 and 0.329 air changes per hour in the all-air and radiant testbeds respectively.

The radiant testbed was cooled by a low thermal mass metal ceiling panel system in the drop ceiling (Twa model MOD-RP1). The panels covered 73% of the floor area, as highlighted in blue in Figure 3.2. We covered as much of the ceiling area as possible to ensure even surface temperature distribution, and to reduce the surface temperature that would be required to extract heat from the testbed. We arranged the panels in six parallel loops with 19-20 panels in each. Water flowed through the ceiling constantly at 18.2 l/min (4.8 gal/min) and automated controls adjusted the supply water temperature to control the operative temperature in the space. For the experiment with constant internal gains, the median supply water temperature was 16.4 °C with an interquartile range of 3.2 °C (For the experiments presented in Chapter 4 the median supply water temperature was 14.5 °C with an interquartile range of 5.62 °C). We were careful to ensure that supply water temperature would not cause condensation. The median temperature rise across each loop was 2.2 °C with an interquartile range of 1 °C (For the experiments presented in Chapter 4 the median temperature rise across each loop was 2.6 °C with an interquartile range of 1.7 °C). Although a low mass radiant system has a distinctly different response time than a high thermal mass radiant system, the heat transfer rate for a surface is determined by the difference between the surface temperature and space air and surface temperatures. Consequently, the observations presented in this chapter should represent the surface temperatures and space heat extraction requirements for any type of radiant system – including high thermal mass radiant systems – to achieve the indoor conditions observed. Keep in mind that for a high thermal mass radiant system, the rate at which heat is transferred to the cooling plant will be considerably different from the space heat extraction rate – neither this chapter nor Chapter 4 address heat transfer rates at the cooling plant.

In the all-air testbed we used a constant volume variable temperature control scheme to provide cooling. We used this strategy instead of a variable-air-volume control scheme so that we could precisely balance heat gain from the fan in the all-air testbed with equivalent heat gain in the radiant testbed. Since we were focused on comparing the sensible space heat extraction rates by each system, we were careful to ensure that humidity in the all-air testbed remained low enough that supply water temperature would not cause condensation (latent space heat extraction).

We supplied equal internal heat gains to each testbed using a combination of different electric resistance heating apparatuses, selected to generate the radiant-to-total heat gain ratio desired for each experiment (see Section 3.2.4 and Section 4.2.2 for the experimental design associated with Chapter 3 and Chapter 4). We measured and balanced all internal heat gains located within the thermal boundary of each testbed, including electricity use for fans, pumps, controls, and data acquisition equipment.

We controlled both testbeds to maintain equal operative temperature setpoints. Although buildings are not regularly controlled to operative temperature, doing so for this comparison ensured equivalent comfort conditions in both testbeds – according to *ASHRAE Standard 55* (2017b). The controlled value in each system was the average of three operative temperature measurements, located along the centerline of each testbed, far enough from the south wall to avoid direct solar radiation (3.45 m, 5.3 m and 7.16 m from the south wall), and at 0.6 m height – according to *ASHRAE Standard 55* (2017b) for a seated occupant. We measured operative temperature with fast response thermistors placed at the center of 40 mm diameter grey plastic globes, in accordance with findings from various researchers (De Dear, 1987; Humphreys, 1977; Simone et al., 2007), and international standards for measurement of human thermal comfort (CEN, 2019; ISO, 2005).

3.2.2 Operative temperature and air velocity in each testbed

Human thermal comfort is influenced by several indoor environmental parameters including air temperature, mean radiant temperature, and air motion. Operative temperature is a concept that captures the combined effects of heat transfer by convection and radiation between a human occupant and a non-uniform thermal environment. The metric is used by all standards for thermal comfort in indoor environments as the best single predictor for comfort (ASHRAE, 2017b; CEN, 2007a, 2019; ISO, 2005).

Since radiant cooling exchanges heat with occupants differently than all-air cooling, it is imperative that performance of the two systems be compared at equivalent comfort conditions, not at equivalent air temperature. For this reason we averaged the measurement of operative temperature at three locations in each testbed and controlled both systems to an operative temperature set point of 26 °C.

Operative temperature is a calculated parameter that depends on measurement of globe temperature, air temperature, and air velocity:

$$T_O = \frac{(\bar{T}_R \cdot h_R + T_A \cdot h_C)}{(h_C + h_R)}$$

where T_O is the operative temperature, T_A is the air temperature, $\bar{T}_R$ is the mean radiant temperature, h_C is the convective heat exchange coefficient for a person, and h_R is the radiant heat exchange coefficient for a person.

For environments with low air velocity, the convective heat transfer coefficient and radiant heat transfer coefficient are similar, so the operative temperature can be approximated as the average of the mean radiant temperature and the air temperature (ASHRAE, 2017b; CEN, 2007a, 2019; ISO, 2005). In a study that compared various apparatus for measuring operative temperature at low air velocity Simone et al. (2007) showed that measuring temperature at the center of a small grey colored sphere approximates operative temperature to within 0.2 C, as long as the sensor is not located too close to a wall.

We used hot wire anemometers to measure air velocity in both testbeds to ensure that it was low enough to comply with the aforementioned assumptions about operative temperature. Figure 3.3 illustrates the results. The distributions represent velocity measurements at each position over a ten-minute period during steady operation. Air velocity was acceptably low throughout both testbeds.

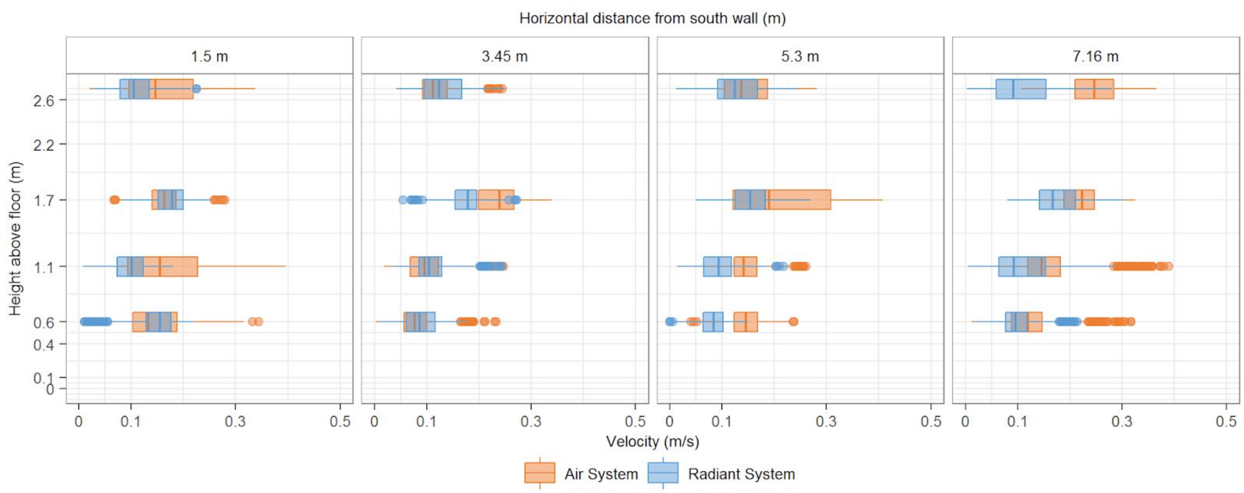


Figure 3.3: Distribution of air speed in each testbed. The comparisons presented in this chapter were conducted with both testbeds at equivalent comfort conditions. Air speed measurements indicate that air speed was similar in both testbeds, and low enough that it would not appreciably influence the measurement of operative temperature.

3.2.3 Measurements and uncertainty

We monitored more than 250 points in each testbed to assess thermodynamic states and heat transfer rates. We measured these points continuously throughout each experiment and recorded one-minute-average values on one-minute intervals. In summary, categories of measurements included

- Wall indoor surface temperatures
- Wall internal temperatures
- Slab indoor surface temperatures
- Slab internal temperatures
- Ceiling indoor surface temperatures
- Indoor air temperatures
- Indoor operative temperatures
- Hydronic system temperatures
- Hydronic system water flow rates
- Air system temperatures
- Air system airflow rates
- Internal heat gain rates
- Solar heat gain rates
- Surface heat flux rates
- System controlled variable

We calculated the sensible space heat extraction rates reported in this chapter, and in Chapter 4, from flow and temperature measurements in the chilled-water loops that served each testbed separately. Flow and temperature measurements were located at the point where chilled water circulating in the cooling plant loop was injected into the loop that serves terminal heat transfer devices. These measurements were located at the thermal boundary of each testbed, and therefore capture all of the thermal energy extracted from each testbed. Since the radiant system was a low thermal mass metal panel ceiling with a fast response time, the hydronic heat extraction rate was a close approximation of the instantaneous space heat extraction rate associated with convective and radiant heat transfer at the indoor face of the internally cooled ceiling surface. The mechanical ventilation system in buildings with radiant cooling often provides some amount of space heat extraction, but our assessment assumes that ventilation is provided at room-neutral conditions and that all space heat extraction is provided by the internally cooled surfaces.

Table 3.1 summarizes the uncertainty for key measurements and calculated metrics. We used propagation of error calculations to determine the uncertainty of the space heat extraction rate for each testbed and to determine the uncertainty of the difference between space heat extraction rates for the two testbeds.

Table 3.1: Calibrated uncertainty of measurements and calculated metrics

Measurement	Calibrated Uncertainty	Manufacturer and model
Water temperatures	± 0.02 °C	BAPI BA/10K
Air temperatures	± 0.02 °C	US Sensor Corp. PR103J2
Surface temperatures	± 0.02 °C	US Sensor Corp. PR103J2
Water flow rates	± 0.2% of measurement	Siemens MAG 6000 with MAG FM 1100
Internal heat gain rates (electric power)	± 1% of measurement	
Hydronic/space heat extraction rate	± <10 W	Calculated metric

The uncertainty values reported for temperature in Table 3.1 do not represent the absolute accuracy compared to a standard reference measurement; instead, they describe the calibrated repeatability among the group of measurements compared. Absolute uncertainty is important when values need to be compared to measurements from a separate study, in which case agreement with standard reference measurements is the only way to ensure accurate comparison. Since our experiments compared two cases side-by-side, we were able to calibrate all of our temperature sensors to one another in situ. Since our conclusions focus squarely on whether the space heat extraction rate for the radiant testbed was different from the space heat extraction rate for the all-air testbed, absolute uncertainty compared to a standard reference measurement is not especially relevant, while uncertainty of the difference is very important.

We conducted the in-situ calibration by placing all temperature sensors in a water bath to compare them against one another. The water bath used U.S. Sensor Corp USP 3021 (Littlefuse, Chicago, IL, USA) as reference (uncertainty ±0.01°C to standard reference measurement). We repeated the water bath comparison across a range of temperatures (18 steps between 0–70 °C). Then, we corrected the bias between sensors by adjusting the Steinhart-Hart coefficients for each sensor. This approach nearly eliminates bias between the sensors, consequently uncertainty of the difference between temperature measurements was reduced mainly to stochastic variation in repeated measurements.

Water flow rate measurements were factory calibrated to a standard reference measurement for a wide range of flow rates.

In parallel to propagation of error calculations, we also calibrated the testbeds to one another to improve our confidence in observing any difference between their space heat extraction rates. Prior to the experiments presented here, we conducted two baseline calibrations in which we operated both testbeds as identical all-air systems with constant internal gains for several days. Ultimately, these baseline calibrations yielded a difference in the daily average space heat extraction rates of 1 W – smaller than the magnitude of the uncertainty of the difference due to propagation of uncertainty from the associated measurements.

3.2.4 Design of experiments

We conducted two side-by-side experiments, one with constant internal gains, and one with periodic internal gains. We operated each experiment continuously for five days. In all cases the setpoint for operative temperature in both spaces was 26 °C.

The median value for internal heat gains in each testbed was 3,210 W (55.7 W/m^2 floor area), and ranged from 3,100–3,325 W as grid voltage varied. Heat gains in both chambers varied together, and the median difference between heat gain in each chamber was only 0.85 W. In the first experiment the internal heat gains were constant. In the second experiment we turned on the internal heat gains each day at 06:00 then off at 18:00. Internal gains during the off periods in the second experiment were approximately 400 W (6.9 W/m^2) due to controls and fan energy. Solar gains reached 1,000–1,500 W each day. Meteorological conditions during the experiments were mild, median outdoor temperature was 12.2 °C with interquartile range of 4.75 °C. Consequently, envelope heat transfer was relatively small, and flowed in both directions throughout each day; envelope heat transfer ranged from 15 W/m^2 gains–10 W/m^2 losses. Overall, envelope losses were more dominant than gains. Consequently, the cumulative thermal energy extracted by each system was much smaller than the cumulative thermal energy from internal and solar gains.

For reference, Figure 3.8 illustrates the solar and internal heat gain cycles for the two experiments.

3.3 RESULTS

3.3.1 Air, operative, surface and mass temperatures

In Figure 3.4, Figure 3.5, and Figure 3.6 we present a detailed comparison of temperature conditions in each testbed during the experiment with constant internal gains. Each figure presents violin plot distributions for temperature measurements at various locations throughout the entire five-day experiment. In addition to each distribution, the plots indicate median values and include whiskers to indicate the interquartile range.

Air temperature distribution

Figure 3.4 plots the air temperature distribution in each testbed. These measurements demonstrate that air temperature in the all-air testbed was cooler in almost every location. The median differences at each location were as much as 1.5 °C; the average of the median differences was 0.5 °C. Paired by observation time, the Wilcoxon signed rank test indicates a statistically significant difference at all but one location. Paired by location and by observation time, air temperature in the all-air testbed was cooler than the corresponding air temperature in the radiant testbed for 87% of instances ($p < 0.001$).

Among the four horizontal positions, air temperatures nearest the south wall were most different between the two testbeds, and cooler in the all-air testbed – the average of the median differences was 0.88 °C. In the all-air testbed the air temperatures at this horizontal location were also significantly lower than all other locations in the all-air testbed. We suspect that this was related to imperfect air distribution relative to the location of heat gains. All return registers were in the drop ceiling at the south wall (for reference see Figure 3.2), so the lower temperatures nearer the south wall suggest that some cooled air bypassed the heat gains as the bulk flow moved toward the return. This type of imperfect air distribution is common for all-air systems that use mixing air distribution. The specific pattern is sensitive to the relative locations of heat gains, control points, supply diffusers and return registers. One consequence is that comfort conditions and thermal energy use for the all-air system may be somewhat skewed compared to building energy simulations which typically assume perfect mixing within a zone.

Neither testbed developed strong temperature stratification, although there was some vertical temperature variation in each testbed. Generally, air temperature increased with height, but never by more than 1.5 °C over the 2.5 m range observed. There was a clear tendency for air temperature to decrease near the ceiling in the radiant testbed, while air temperature in the all-air testbed was usually warmest near the ceiling. This inversion is expected with internally cooled ceiling surfaces as natural convection draws cooler air downward from the ceiling surface and draws warmer air upward from heat gains. Other differences in the vertical temperature variation between the two testbeds might be related to the system type, but the differences are within a range that could be attributed to minor locational differences in heat gain, air distribution, or sensor position. For example, we suspect that the reversed inversion pattern at 3.5 m horizontal location in the all-air testbed may have occurred because supply air was not as well mixed at these sensor locations.

These results reinforce the general understanding that air temperature is warmer in radiant cooled buildings than in all-air cooled buildings at equivalent comfort conditions. Therefore, as others have asserted, radiant cooling can reduce heat gain from warm ventilation air compared to an all-air system (Kim & Olesen, 2015a, 2015b; Niu et al., 1995; Rhee et al., 2017; Rhee & Kim, 2015). For the same reason radiant cooling also increases the benefit of cooling from ventilation air when the outside temperature is cooler than indoors.

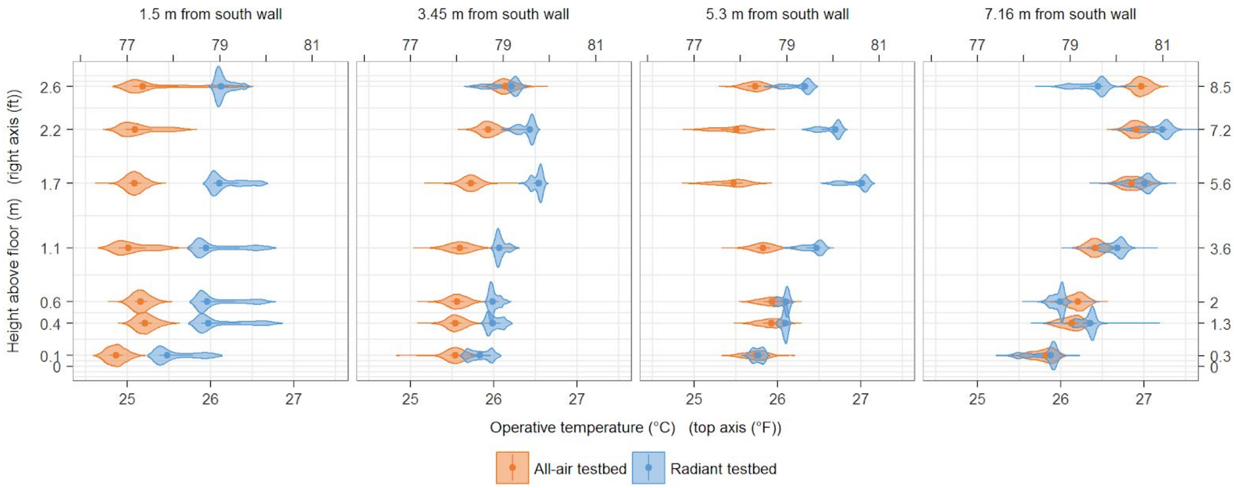


Figure 3.4: Distribution of air temperatures in each testbed. We measured air temperatures with radiation-shielded thermistors located on a 28-point grid that spanned the south-to-north centerline of each testbed with four horizontal positions and seven vertical positions. Air temperatures in the radiant testbed were warmer in almost every location. The median difference in air temperature was as much as 1.5 °C warmer in the radiant testbed – the average of the median differences was 0.5 °C. The figure plots the distribution of observations at each location, the point in each distribution indicates the median, and the whiskers extending from the median indicate the interquartile range.

Operative temperature distribution

We controlled both testbeds to maintain equal operative temperature measured as the average of three sensors at 0.6 m height at three horizontal positions (3.45 m, 5.3 m, and 7.16 m from the south wall). Figure 3.5 illustrates that operative temperature at the control points was essentially equal in both testbeds – the average of the three median deviations was less than 0.001 °C, which is one order of magnitude less than the sensor uncertainty (± 0.02 °C). However, operative temperatures at other locations were somewhat different from the control points, and there were differences between the two testbeds. In particular, above 1.1 m and nearer the south wall operative temperature was lower in the all-air testbed. The largest median difference between operative temperature at the control points and operative temperature elsewhere was 0.73 °C. The largest median difference between corresponding positions in the two testbeds was 1 °C. These differences were caused mainly by corresponding differences in air temperature. The variation in operative temperature throughout a room is a realistic consequence of nonuniform distribution of heat gains and uneven distribution of cooling. To some extent radiant cooling may be more resilient to non-uniform distribution of heat gains because heat exchange potential is spread out across large areas, and because the space heat extraction rate for an internally cooled surface is somewhat self-regulating – the local surface heat transfer rate naturally adjusts to local heat gains.

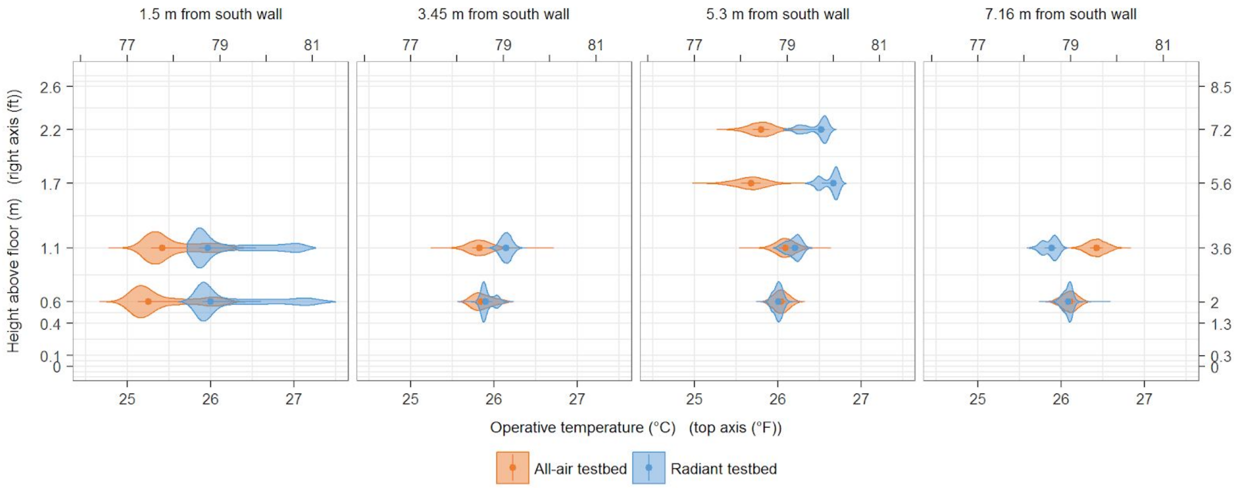


Figure 3.5: Distribution of operative temperature in each testbed. We measured operative temperature with thermistors placed at the center 40mm diameter plastic spheres, located at various heights and four horizontal positions along the south-to-north centerline of each testbed. The average of the median differences at each of the three control points (0.6 m height and 3.45, 5.3 and 7.16 m from the south wall) was less than 0.001 °C. The differences in operative temperature at other points is attributed mainly to corresponding differences in air temperature. The figure plots the distribution of observations at each location, the point in each distribution indicates the median, and the whiskers extending from the median indicate the interquartile range.

Surface and mass temperatures

Figure 3.6 and Figure 3.7 compare the temperature of surfaces and masses in each testbed. Figure 3.6 compares the distribution of temperature measurements from numerous locations, while Figure 3.7 compares infrared images of each testbed. Each of the distributions in Figure 3.6 aggregate measurements from between 3–15 thermistors distributed across the surfaces indicated. In every case temperatures of surfaces and masses in the radiant testbed were cooler than in the all-air testbed. Note that in the radiant testbed the indoor surface temperatures were typically cooler than the operative temperature setpoint, while in the all-air testbed the indoor surface temperatures were typically warmer than the setpoint. This occurred because in addition to removing heat directly from internal heat gains by radiant heat transfer, an internally cooled surface also removes heat from all indoor surfaces and masses to which it is exposed. The infrared images in Figure 3.7 clearly visualize this phenomenon.

Paired by observation time the Wilcoxon signed rank test indicates that each indoor surface in the radiant testbed was cooler than the corresponding surface in the all-air testbed for 100% of instances ($p<0.001$). The median temperature differences between corresponding indoor wall surfaces in each testbed were between 0.8–1.77 °C (minimum difference = 0.47 °C, maximum difference = 2.22 °C). The median temperature difference between the indoor floor surfaces was 1.85 °C (minimum difference = 1.34 °C, maximum difference = 2.38 °C).

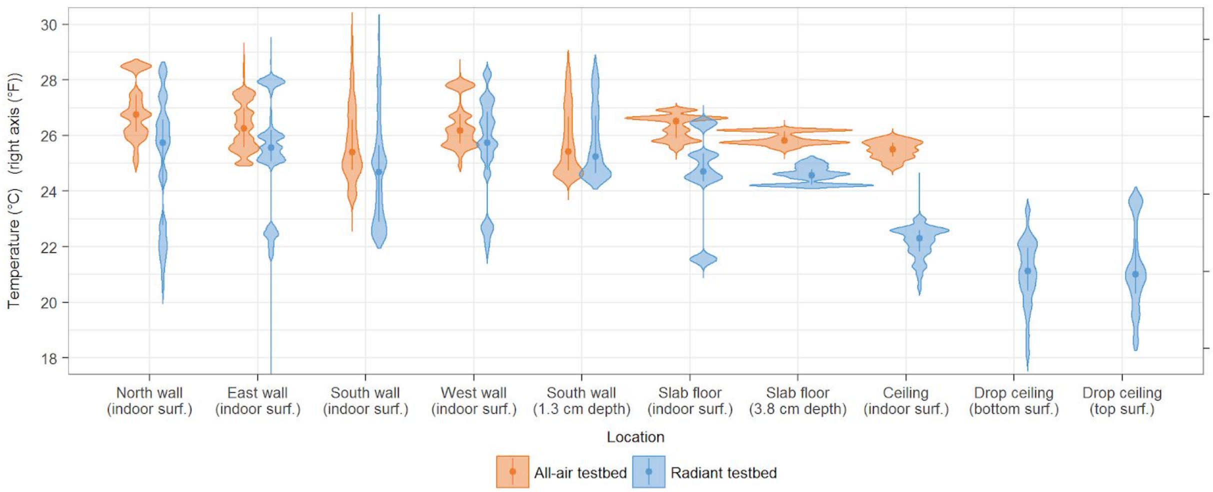


Figure 3.6:Distribution of surface and mass temperatures in each testbed. We measured surface temperatures with low mass thermistors taped to surfaces. Sensors for "South Wall (1.3 cm depth)" were located between the wall-baord and insulation. Sensors for "Slab Floor (3.8 cm depth)" were located in the middle of each concrete slab. Sensors for "Drop ceiling (bottom suf.)" were located on the bottom surface of the internally cooled low thermal mass metal panel ceiling, at the center of the first and last panels in each of 6 parallel loops. The internally cooled low thermal mass metal panel ceiling included 1" fiberglass insulation with radiation shielding; sensors for "Drop ceiling (top surf.)" were located on top of the insulation. Paired by observation time, indoor surfaces in the radiant testbed were cooler than corresponding surfaces in the all-air testbed for 100% of instances. The figure plots the distribution of observations at each location, the point in each distribution indicates the median, and the whiskers extending from the median indicate the interquartile range.

There are two consequences of these surface and mass temperature differences. First, radiant cooled buildings store less heat in non-active masses than all-air buildings. Second, although indoor air temperatures are warmer in radiant buildings, indoor surface temperatures are lower; which increases net gain by heat transfer through the envelope. The change in envelope heat transfer will be small for very well insulated, internal gain dominated buildings, but will be more substantial for buildings with large envelope to floor area ratios, large window to wall ratios, or poorly insulated walls and windows.

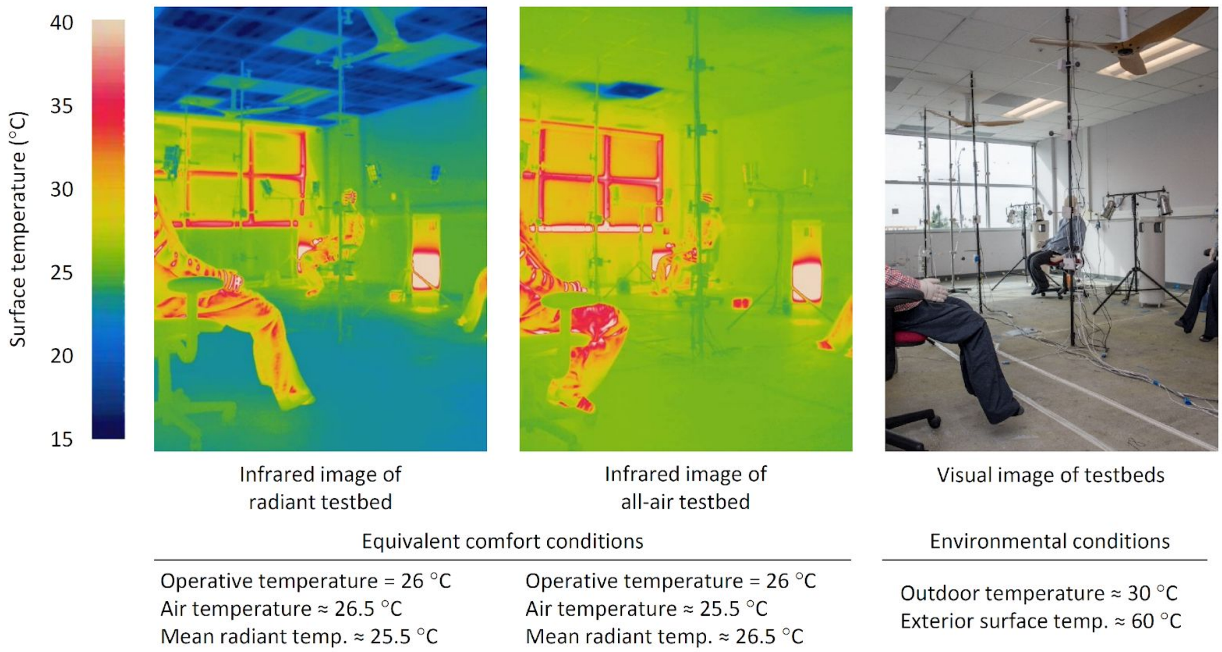


Figure 3.7: Infrared images of radiant testbed (left) and all-air testbeds (center) while cooling, with equal internal gains and equal solar gains, during experiment with constant internal gains. At the time of the infrared images (11:00–13:00 on 7 Sept. 2016), temperature of outdoor air was ~30°C, and temperature of the exterior cladding on the south wall was ~60°C. Although comfort (operative temperature) was equivalent in both testbeds, all surfaces and masses in the radiant testbed were cooler, which indicates that less heat was stored in the non-active masses of the radiant testbed.

3.3.2 Cooling rates and thermal energy

Figure 3.8, Figure 3.9 and Figure 3.10 compare the dynamic thermal response for each testbed in both experiments. Each figure shows results for the experiment with constant internal heat gains on the left, and results for the experiment with periodic internal heat gains on the right. The observations reveal fundamental differences between radiant and all-air systems.

Operative temperature response

Figure 3.8 compares the operative temperature response in each testbed for the two experiments. A more detailed view of two days for the experiment with periodic internal heat gains is included in Figure 3.10. For both experiments, each system type maintained operative temperature at 26 °C during all periods that required cooling. In the experiment with constant internal heat gains both cooling systems operated continuously. In the experiment with periodic internal heat gains both cooling systems turned off while the internal gains were at a reduced level because the operative temperature naturally remained lower than the setpoint.

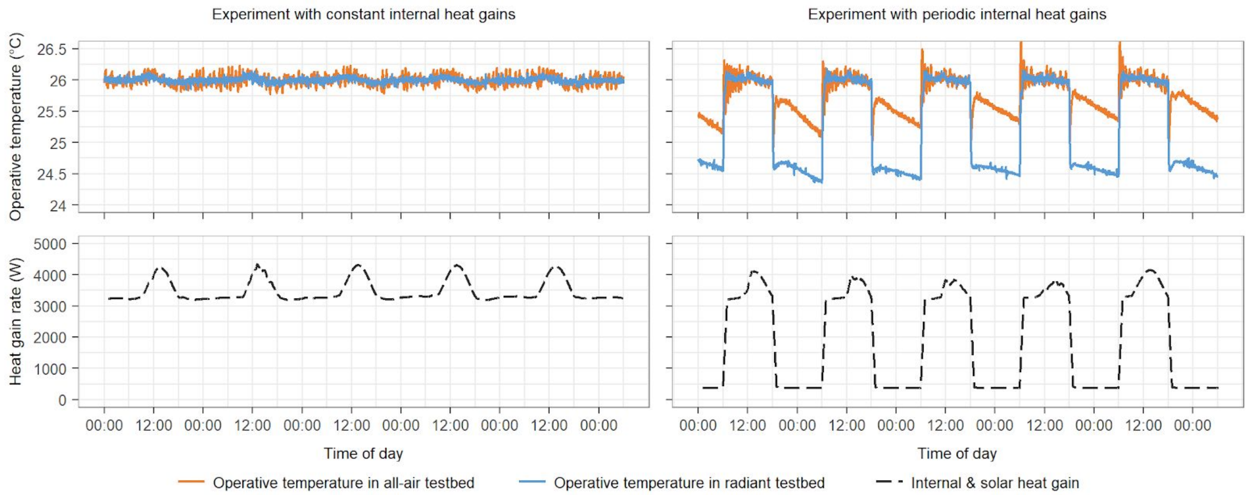


Figure 3.8: Operative temperature (top row) and combined internal and solar heat gain rate (bottom row) for experiment with constant internal gains (left) and experiment with periodic internal gains (right). The figure plots operative temperature at one-minute intervals, and heat gain rate as one-hour rolling averages at one-minute intervals.

In the experiment with periodic internal heat gains, operative temperatures in both testbeds decreased rapidly for 10–15 minutes after the internal gains were removed. Air temperatures – which are not shown in Figure 3.8 – also decreased at a similar rate. This temperature decrease was due to the thermal inertia of chilled-water and cooling equipment that remained after the internal gains were removed.

After the initial decline, operative temperatures and air temperatures in the all-air testbed increased for 1–2 hr because heat that had been absorbed in the building mass during the day was released into the space by convection to the air and radiation to the other surfaces. The operative temperatures and air temperatures returned to within 0.25 °C of the values from immediately before the heat gains were removed. Afterward, over the course of the night, operative temperature and air temperature declined steadily as heat stored in the building mass was rejected passively to the environment.

The corresponding operative temperature response for the radiant testbed was distinctly different. After the initial rapid decline, operative temperatures and air temperatures did not increase as they did in the all-air testbed. Since radiant buildings store less heat in masses, they release less heat into the room from masses after internal and solar gains diminish. Concomitantly, buildings with radiant cooling also reject less heat to the environment by passive means. A portion of the heat that would be absorbed by masses in an all-air building during the day then lost to the environment overnight is instead extracted by the radiant system earlier in the day. We expect that this pattern would be most pronounced in climates with large diurnal temperature variation, and especially in scenarios where natural ventilation is used overnight to pre-cool the building mass. It would be less pronounced in scenarios with fewer opportunities for passive heat rejection. Furthermore, we expect that the control strategy – such as whether or not pre-cooling or night setback are employed – will also influence the extent to which each system type enables passive heat rejection to the environment.

Dynamic space heat extraction rates

Figure 3.9 compares the dynamic space heat extraction rate generated by each system in the two experiments. A more detailed view of two days for the experiment with periodic internal gains is included in Figure 3.10. For both systems, we calculated the space heat extraction rate from flow and temperature measurements in the chilled-water loop serving each testbed. Since the radiant system was a low thermal mass metal panel ceiling with a fast response time, this measurement closely approximates the instantaneous space heat extraction rate associated with convective and radiant heat transfer at the indoor face of the internally cooled ceiling surfaces.

The results reveal that to maintain comfort conditions equivalent to an all-air system:

1. Radiant cooling must extract heat from gains earlier
2. The peak space heat extraction rate must be larger for radiant systems
3. The peak space heat extraction must occur earlier for radiant systems

These differences arise in response to dynamic heat gains, and would not occur at steady-state in an adiabatic system. To maintain consistent comfort conditions as heat gains increase, radiant cooling must extract heat earlier than an all-air system because a portion of the increasing heat gain that would be absorbed by masses in an all-air building is instead removed by radiant heat transfer to the internally cooled surfaces.

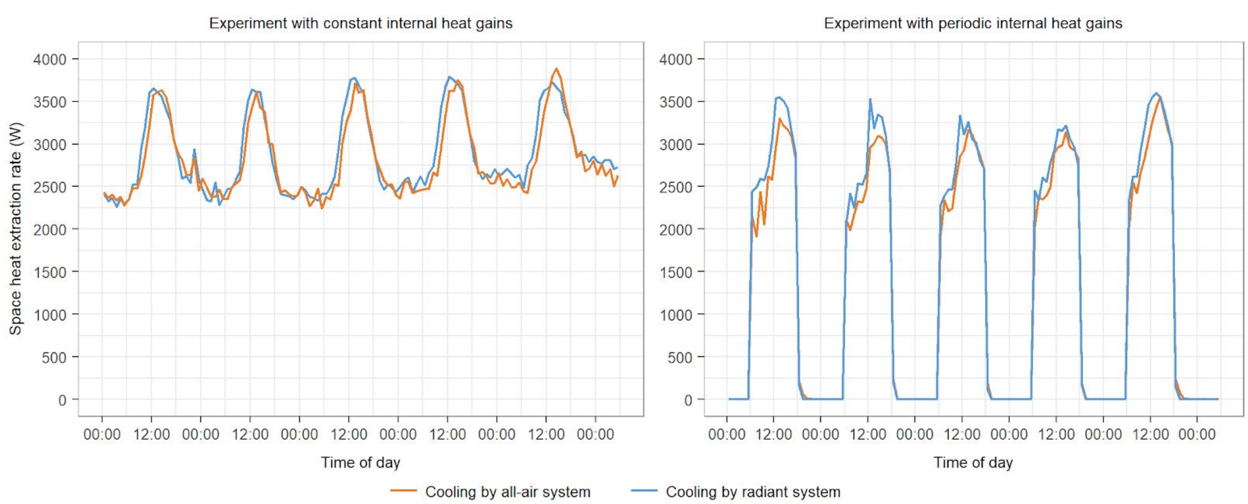


Figure 3.9: Space heat extraction rates for radiant (blue) and all-air systems (orange) in experiment with constant internal heat gains (left) and experiment with periodic internal heat gains (right). Data is plotted as one-hour rolling averages at one-hour intervals.

For clarity, Figure 3.10 repeats the data presented in Figure 3.8 and Figure 3.9 for the first two days from the experiment with periodic internal heat gains.

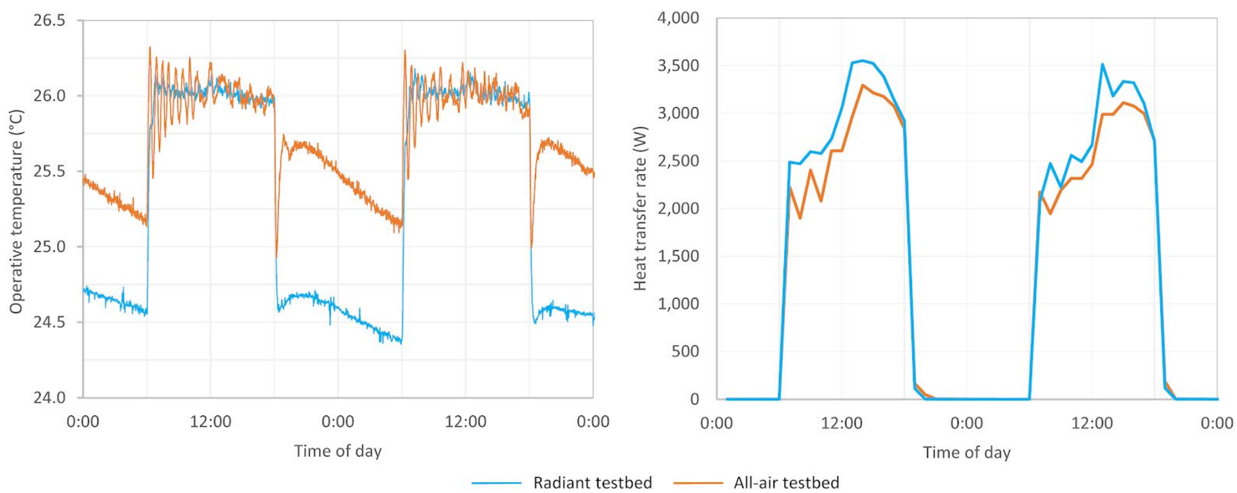


Figure 3.10: Operative temperature response (left) and space heat extraction rates (right) for the radiant testbed (blue) and the all-air testbed (orange) for the first two days from the experiment with periodic heat gains.). The figure plots operative temperature at one-minute intervals, and space cooling rate as one-hour rolling averages at one-hour intervals.

In the experiment with constant internal heat gains the differences in space heat extraction rate were driven by changes in solar gain each day. Except for the last day of the experiment, the peak space heat extraction rate in the radiant testbed was 1–3% larger (0.7–1.6 W/m^2) and occurred 1–2 hr earlier. In the experiment with periodic internal heat gains the peak space heat extraction rate in the radiant testbed was 2–10% larger (1.3–5.6 W/m^2) and occurred 1–2 hr earlier.

These results confirm that to maintain equivalent operative temperature the peak sensible space heat extraction rate for a radiant system must be larger than the peak sensible space heat extraction rate for an all-air system. This finding has consequences for the design of radiant systems; in particular, as Feng et al. (2014a) indicated, industry common practice methods for design sizing of the terminal cooling devices for all-air systems will underestimate the peak space heat extraction requirement (peak space cooling load) for radiant systems.

The scenarios presented here represent cases with realistic solar and internal gains, but they do not represent every reasonable possibility. The magnitude of the difference in space heat extraction requirements (space cooling load) will be smaller in some circumstances and larger in others. The magnitude of the difference will depend on:

1. The amplitude of the daily oscillation in heat gain
2. The ratio of radiant heat gains to convective heat gains
3. The thermal diffusivity and volume to surface area ratio of masses within and enclosing the space

We expect that the difference will be larger when the amplitude of the daily oscillation in heat gain is larger, and smaller when it is smaller. For example, in a hypothetical scenario with an adiabatic envelope and constant internal heat gains the steady-state space heat extraction rate would be equal for both system types.

We expect that the difference would be larger in scenarios with mainly radiant heat gains and smaller for scenarios with mainly convective heat gains. For example, in a hypothetical scenario where oscillating heat gains are composed entirely of shortwave radiation, the space heat extraction required for the all-air system would be limited to the heat that is shed by convection from non-active masses that have absorbed the heat gains. Whereas radiant cooling would extract a portion of the shortwave gains directly, and would rapidly extract heat absorbed by the non-active masses by way of longwave radiant heat transfer.

Lastly, we expect that the difference would be larger in scenarios where the temperature of surfaces within and enclosing a space are more resilient to heat gain – where exposed surfaces have high thermal diffusivity and high thermal capacity. These high thermal mass surfaces, such as exposed concrete construction, are readily available to absorb heat from the radiant component of heat gains without a substantial increase in surface temperature. Low thermal mass surfaces such as raised floors, drop ceilings, or furniture would tend to decrease the difference between the dynamic space heat extraction requirements (space cooling load) for radiant and all-air systems. These low thermal mass surfaces more rapidly convert radiant heat gains to convective heat gains.

Cumulative thermal energy use

Lastly, Figure 3.11 plots the cumulative thermal energy extracted by each system over the course of the two experiments. The results reveal that to maintain equivalent comfort radiant cooling must remove somewhat more heat from a building than all-air cooling. The difference is small but meaningful. In the experiment with constant internal heat gains radiant cooling extracted 2% more heat overall; and for the experiment with periodic internal heat gains radiant cooling extracted 7% more heat overall. These differences equate to an average difference in the space heat extraction rate of 60 W (1 W/m^2) and 183 W (3.2 W/m^2) respectively. For context, the uncertainty in the difference between space heat extraction rates for the two testbeds was approximately 15 W at the conditions observed – an order of magnitude lower than the observed difference.

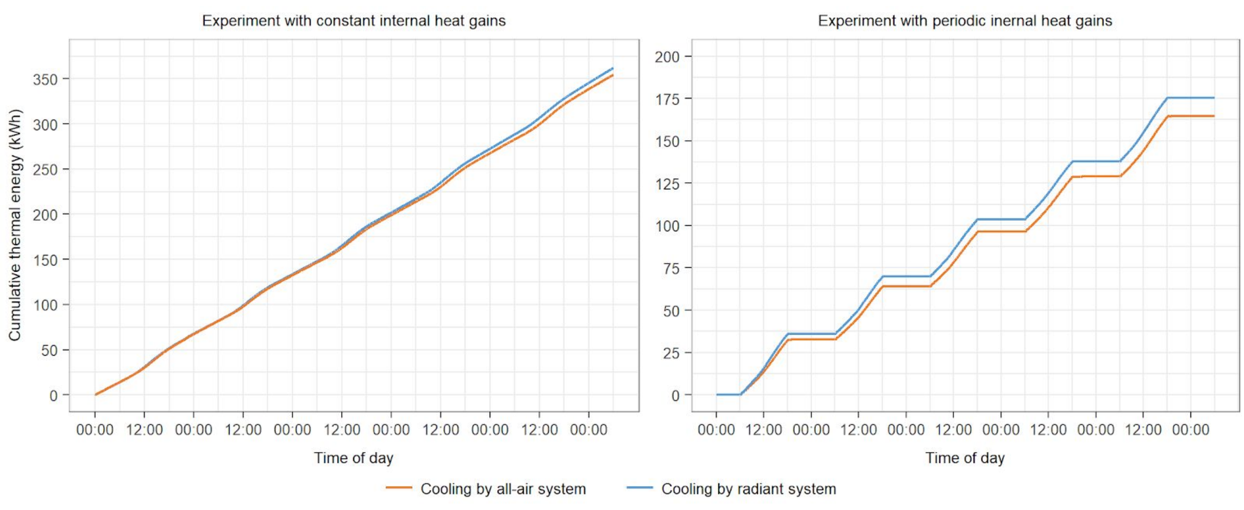


Figure 3.11: Cumulative space heat extraction energy for radiant (blue) and all-air systems (orange) in experiment with constant internal heat gains (left) and experiment with periodic internal heat gains (right). Data plotted on one-minute intervals.

These results reinforce the findings associated with Figure 3.9. Whereas the differences in the dynamic space heat extraction requirements (space cooling load) would exist even in an adiabatic system, the additional cumulative thermal burden for radiant cooling exists specifically because of interactions with the environment. The difference in the cumulative space heat extraction energy can be attributed to two factors:

1. Buildings with radiant cooling have lower interior surface temperatures which increases envelope heat gains.
2. Buildings with radiant cooling store less heat in non-active masses and subsequently reject less heat to the environment by passive means.

3.4 DISCUSSION

We have experimentally confirmed, far beyond the bounds of measurement uncertainty, that the dynamic space heat extraction requirement for radiant cooling is different from the dynamic space heat extraction requirement for all-air cooling when both systems maintain equal operative temperature. The differences observed in our experiments are generally similar to differences indicated by previous simulations and simplified laboratory experiments (Feng et al., 2013, 2014b) – and therefore provide conclusive evidence that these differences are present in realistic circumstances.

However, we expect that the magnitude and characteristics of the difference depend on many variables including: the type of heat gains, the magnitude of heat gains, the timing of heat gains, the location of heat gains, the physical and thermal characteristics of non-active masses, the sizing and layout of internally cooled surfaces, the outdoor conditions, and control of the cooling system.

Despite these many influences, the fundamental differences observed in our experiments should extend to other scenarios where radiant cooling and all-air cooling are compared at equivalent operative temperatures. For example, if operative temperature in both testbeds followed a dynamic profile throughout the day – as typically occurs in buildings with high thermal mass radiant systems – space heat extraction rates would be somewhat different from what we observed, but the peak space heat extraction rate would still be larger for the radiant system and it would occur earlier.

We assessed thermal energy extracted from the space, and we did not assess the way that high thermal mass radiant systems decouple the space heat extraction rate from operation of the cooling plant. For a high thermal mass radiant system, the space heat extraction rate is different from the hydronic heat extraction rate – see Figure 3.1 for further clarification. Although our results have confirmed that the peak space heat extraction rate for radiant systems must be larger than the peak space heat extraction rate for an all-air system, we also acknowledge that a strategically controlled high thermal mass radiant system could allow the cooling plant to be much smaller than what would be required for an all-air system in equivalent circumstances.

Although industry common practice methods for design sizing of terminal cooling devices do not properly capture the differences between radiant and all-air systems, researchers have developed building energy simulation tools that do account for the differences properly (Feng et al., 2013, 2014b). Many buildings with radiant cooling operate successfully every day despite the shortcoming of common design methods. This may be because designers for radiant buildings use building energy simulation tools that account for the differences, or it may be because designers employ a factor of safety in practice that is large enough to overcome the differences.

Finally, while our results have confirmed that radiant cooling must remove more heat than an all-air system in order to maintain equivalent comfort conditions, we also acknowledge that radiant cooling enables several efficiency opportunities that can result in overall primary energy savings compared to all-air cooling. Many building energy simulation studies have clearly established that properly designed radiant cooling systems can achieve substantial primary energy savings and peak electrical demand reduction compared to conventional all-air cooling systems.

Furthermore, although we have focused exclusively on cooling, similar differences should also exist between radiant and all-air systems in heating mode – the peak space heating rate would be larger for radiant heating systems, and more heat would be required each day.

3.5 CONCLUSIONS

Niu et al. (1995) and Feng et al. (2013, 2014b) previously indicated that the dynamic space heat extraction requirements (space cooling load) for radiant cooling and all-air cooling are different. However, industry common practices used for design sizing of terminal cooling devices do not recognize these differences. We conducted a series of laboratory tests to substantiate or refute the claimed differences. Whereas the previous studies were based on simulations and a simplified environmental chamber experiment, we conducted a series of experiments in much more realistic circumstances. We used a pair of equivalent side-by-side testbed buildings that enabled thorough assessment of building energy systems at a realistic physical scale, with naturally occurring solar gains, and natural interaction with the surrounding environment. For each experiment we operated the two testbeds simultaneously, imposed equivalent internal gains, and controlled each system to maintain equivalent operative temperatures.

We conclude that radiant cooling and all-air cooling remove heat from buildings in different ways. The experiments presented herein demonstrate that these differences influence the dynamic space heat extraction rates that are required to maintain equal comfort conditions in response to dynamic heat gains. We corroborate the previous claims by Niu et al. (1995) and Feng et al. (2013, 2014b): the time and rate at which heat must be extracted from a space depend on the type of terminal device used for cooling. To maintain comfort conditions equivalent to an all-air system: (1) radiant cooling must extract heat from gains earlier; (2) the peak space heat extraction rate must be larger for radiant systems; and (3) the peak space heat extraction must occur earlier for radiant systems.

The differences are mainly due to the way that heat gains are absorbed by, stored in, and released from non-active masses. Radiant cooling extracts heat from all surfaces in a building, so when internal or solar gains increase, a portion of the heat that would be absorbed by mass in an all-air building is extracted by the internally cooled instead. As a result, all interior surfaces in a radiant building are cooler than in an equivalent all-air building, and less heat is stored in mass. In our experiments non-active interior surfaces were as much as 2.38 °C cooler in the radiant testbed, and air temperatures were as much as 1.75 °C warmer.

These differences increase the cumulative amount of heat that must be extracted each day. Although the indoor air temperature is warmer in a radiant building, net gains by envelope heat transfer are larger than an all-air building because the interior surfaces of exterior walls are cooler. More importantly, since less heat is stored in non-active masses, radiant buildings reject less heat to the environment by passive means. As a result, in our experiments, despite having equal internal and solar gains, the cumulative amount of heat extracted from the radiant testbed was as much as 7% larger than the all-air testbed.

Finally, to maintain equivalent comfort conditions, the dynamic space heat extraction rate for radiant cooling must be different from that of all-air cooling. In our experiments, the peak space heat extraction rate was as much as 10% larger in the radiant testbed, and it occurred 1–2 hours earlier. These findings are of consequence for: the design and sizing of radiant systems, the choice and sizing of hydronic systems and cooling plants, the dynamic control of these systems, and the resulting potential for electricity savings and demand response. It is especially important to note that the peak sensible space heat extraction rate for radiant cooling – the heat transfer rate at the surface – must be larger than that of an all-air system in an equivalent building with equivalent operative temperature conditions. These findings should be true for all radiant system types, regardless of their thermal mass.

CHAPTER 4

Side-by-side laboratory comparison of radiant and all-air cooling: how natural ventilation cooling and heat gain characteristics impact space heat extraction rates and daily thermal energy use

Reproduced in part from the following previously published co-authored work:

J. Woolley, S. Schiavon, F. Bauman, P. Raftery, Side-by-side laboratory comparison of radiant and all-air cooling: How natural ventilation cooling and heat gain characteristics impact space heat extraction rates and daily thermal energy use, Energy and Buildings. 200 (2019) 68–85. https://doi.org/10.1016/j.enbuild.2019.07.020.

Chapter Abstract

For radiant cooling to maintain equivalent comfort conditions as all-air cooling it must remove more heat from a space, the peak space heat extraction rate must be larger, and the peak must occur earlier. In this chapter, we assess how the magnitudes of these differences are influenced by heat gain characteristics and by the availability of passive cooling. We present measurements from a series of multi-day side-by-side comparisons of radiant cooling and all-air cooling in a pair of experimental testbed buildings, with equal heat gains, and maintained at equivalent comfort conditions. In a five day experiment with mixed internal heat gains, solar gains, and natural ventilation night pre-cooling, radiant cooling had to remove 35% more heat than the all-air system in equivalent circumstances; and the peak heat extraction rate was 20% larger (median difference on multiple days). In a similar experiment with highly convective internal gains the differences were smaller (26% more thermal energy, 12% larger peak), while in an experiment with highly radiant gains the differences were larger (40% more thermal energy, and 21% larger peak). The differences were much smaller in an experiment without natural ventilation night pre-cooling (7% more thermal energy, 5% larger peak). These findings have consequences for the choice, design, and control of mechanical cooling systems, especially in buildings that also use passive cooling strategies such as natural ventilation night pre-cooling.

4.1 INTRODUCTION

Design, sizing, or simulation of any cooling system typically involves calculation of the dynamic space heat extraction rate that will be required to maintain desired comfort conditions over a particular range of time – this is commonly referred to as a "cooling load calculation". The space heat extraction rate is the rate at which heat is removed from a space by terminal heat transfer devices (ASHRAE, 2017a). For an all-air system, it is the enthalpy difference between airflow supplied to the space and air flow leaving the space. For a radiant system, it is the sum of convective and radiant (longwave and shortwave) heat transfer rates at the indoor face of the internally cooled surface.

Niu et al. (1995, 1997), Feng et al. (2013, 2014b) and Woolley et al. (2018a) (Chapter 3) have all shown that for radiant and all-air systems to maintain equal comfort conditions as one another: radiant cooling must remove more heat from a space than all-air cooling, and the peak space heat extraction rate for radiant must be larger than for all-air cooling.

The differences between required space heat extraction requirements (space cooling load) for radiant and all-air systems are mainly due to the ways that non-active surfaces and their thermal mass impact the dynamics of heat transfer and storage. In a space with all-air cooling, all radiant heat gains are absorbed by non-active surfaces. Whereas in a space with radiant cooling, a portion of the radiant heat gains is absorbed by non-active surfaces and a portion is absorbed by the internally cooled surfaces. Figure 4.1 compares the heat transfer pathways involved in radiant cooling and all-air cooling.

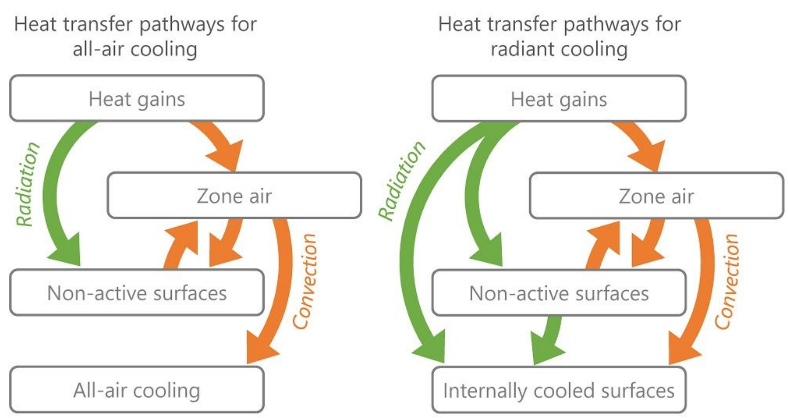


Figure 4.1: Simplified schematic comparison of the convective (orange) and radiant (green) heat transfer pathways involved in all-air cooling (left) and radiant cooling (right).

As a result of the differences illustrated in Figure 4.1, for the same operative temperature, all non-active surfaces in a radiant cooled space are cooler than surfaces in a space with all-air cooling and less heat is stored in non-active thermal masses (Woolley et al., 2018a). The consequences of this are twofold. First: conductive heat transfer through outdoor-exposed surfaces is larger for spaces with radiant cooling because the temperature difference across outdoor-exposed surfaces is larger. Second: since less heat is stored in non-active thermal masses, less heat can be released passively to the environment when there is an opportunity to do so – such as with natural ventilation night pre-cooling.

As these heat transfer mechanisms play out dynamically, they require that radiant cooling extract heat earlier, with a larger peak that occurs earlier. Figure 4.2 – adapted from the results in Chapter 3 (Woolley et al., 2018a) – illustrates the dynamic space heat extraction rates required (space cooling load) for an all-air system (left) and a radiant system (right) to maintain equal operative temperature in response to equal heat gains. Figure 4.2 shows the sum of internal-and-solar heat gains (grey dashed line), and the space heat extraction rate (orange line) for each system. The space heat extraction rate is divided into the amount of heat extracted by convection (orange hatched area), and the amount of heat extracted by radiation (green hatched area). The grey hatched area highlights the difference between the internal-and-solar heat gains (grey dashed line) and the space heat extraction rate (orange line); this area indicates heat that was absorbed by non-active surfaces, stored in thermal mass, then eventually released passively to the environment. Note that Figure 4.2 does not show the rate at which heat was transferred through outdoor-exposed surfaces. Since heat was lost to the environment through outdoor-exposed surfaces overnight, the cumulative thermal energy extracted from each space by the radiant and all-air systems was smaller than the cumulative internal-and-solar heat gains.

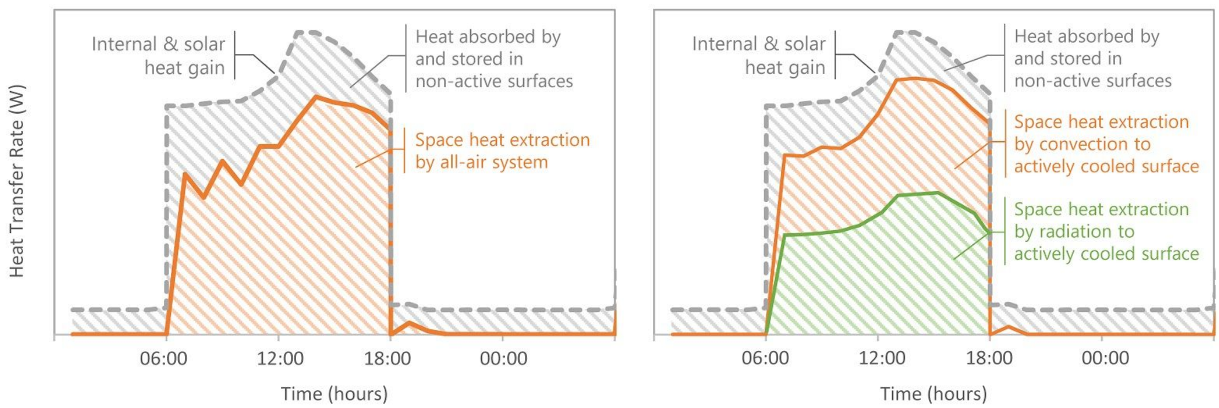


Figure 4.2: Conceptual example of the dynamic space heat extraction rates by convective heat transfer (orange) and radiant heat transfer (green) required by an all-air cooling system (left) and by a radiant cooling system (right) to maintain equal operative temperature in response to equal internal heat gains (grey). Adapted from the results in Chapter 3 (Woolley et al., 2018a)

Although researchers and practitioners generally understand that different types of terminal cooling devices extract heat from a space using different heat transfer mechanisms, they often do not recognize that these differences influence the rates at which heat must be extracted from a space to maintain desired comfort conditions (space cooling load). Consequently – as Feng et al. (2014a) showed – researchers and practitioners commonly size radiant cooling systems using cooling load calculations methods which assume that all space heat extraction occurs by convection with a well-mixed air volume.

This assumption is not accurate for radiant systems, yet is perpetuated by most industry standard cooling load calculation procedures (ASHRAE, 2017a), radiant system design procedures (ASHRAE, 2016a; Babiak et al., 2009), and by some building energy simulation tools. As discussed in the following paragraphs, each of these guiding resources: (1) fail to recognize that the required space heat extraction rate (space cooling load) depends on the type of terminal cooling device used, and (2) promote cooling load calculation methods that only account for space heat extraction by convection with a well-mixed air volume.

ASHRAE Fundamentals Chapter 18: Nonresidential Cooling and Heating Load Calculations (2017a) presents definitions and explanations that systemically fail to consider the implications of space heat extraction by any mechanisms other than convection with a well-mixed air volume. Further, the chapter presents two cooling load calculation methods that were developed for all-air systems and are mathematically limited to convection with a well-mixed air volume. Although *ASHRAE Systems and Equipment Chapter 6: Radiant Heating and Cooling* (2016a) clearly explains that radiant cooling transfers heat by convection and radiation, it does not recognize that the magnitude and timing of the required space heat extraction rate (space cooling load) is fundamentally different from that of all-air systems. Additionally, *ASHRAE Systems and Equipment Chapter 6* (2016a) specifically references the methods in *ASHRAE Fundamentals Chapter 18* (2017a), even though these methods do not account for the effects of space heat extraction by radiation with internally cooled surfaces.

In the widely referenced guidebook *Low Temperature Heating and High Temperature Cooling*, Babiak et al. (2009) thoroughly explain the combined radiant and convective heat transfer rates that a radiant system can be expected to produce for different steady-state conditions (space cooling capacity). However, the guidebook provides no explanation about how to determine the dynamic space heat extraction requirement (space cooling load) for a radiant system, and does not specifically recognize that it can differ substantially from that of all-air systems.

Among standards focused on the topic of space cooling loads, *ISO Standard 52016* (2017) – which superseded *CEN Standard prEn 15255* (2007b) – is the only resource we are aware of to explicitly state that the dynamic space heat extraction requirements (space cooling load) depends on the system type. In an equation for determining the space heat balance, the standard introduces a variable called the "convective fraction of the heating/cooling system". However, the standard currently provides no guidance on how to determine this fraction for different systems and circumstances.

Researchers have developed and validated numerical methods that properly estimate the fundamental heat transfer mechanisms involved with radiant cooling systems (Chantrasrisalai et al., 2003; Laouadi, 2004; Niu et al., 1995, 1997; Stetiu et al., 1995; Strand et al., 1999; Strand & Baumgartner, 2005; Strand & Pedersen, 2002; Yu et al., 2014). Although such methods have been incorporated into building energy simulation software, the problematic assumption – that all space heat extraction occurs by convection – still persists in some aspects. For example, for each simulation timestep EnergyPlus (*EnergyPlus*, 2020) uses the numerical methods developed by Strand et al. (1999; 2005; 2002) to calculate the rate at which internally cooled surfaces extract heat from a space, yet the cooling load calculations performed to autosize components of a radiant system and cooling plant only account for space heat extraction by convection with a well-mixed air volume. Additionally, several widely-used building energy simulation tools have not addressed the problematic assumption in any way, yet researchers and practitioners often use these tools for design and simulation of radiant cooling systems (J. Feng et al., 2014a).

Although previous laboratory and simulation work has carefully demonstrated the fundamental differences between required space heat extraction rates (space cooling load) for radiant and all-air systems, researchers have not thoroughly evaluated the factors that influence the magnitude of these differences. Woolley et al. (2018a) explained that the magnitude of the differences should be driven mainly by the extent to which heat is absorbed by, and stored in non-active masses, and the extent to which such heat can be released to the environment by passive means. Furthermore, in a simulation study, Feng et al. (2013) found that the difference between the required space heat extraction rates (space cooling load) is impacted by the presence of solar gains, by the characteristics of building construction, and by the radiant-to-total ratio for internal gains.

To build upon previous findings, we conducted a series of experiments designed to assess how the differences between required space heat extraction rates (space cooling load) for radiant and all-air systems are influenced by characteristics of heat gain and by the availability of passive cooling. In this chapter, we present measurements from a series of multi-day side-by-side comparisons of radiant cooling and all-air cooling in a pair of experimental testbed buildings, with equal heat gains, and maintained at equivalent comfort conditions (operative temperature). The experimental methods associated with this chapter overlap substantially with the methods associated with Chapter 3, and so are mainly documented in Section 3.2.1, Section 3.2.2, and Section 3.2.3. However, Section 4.2.1 provides an overview of the methods specific to experiments presented in this chapter, and Section 4.2.2 documents the experimental design..

Our first hypothesis was that natural ventilation cooling overnight would increase the differences between required space heat extraction rates (space cooling load) for the two systems. Consider a building in a climate with large diurnal temperature swings, that uses natural ventilation overnight to pre-cool thermal mass: in such a building, the non-active thermal masses absorb and store a portion of the heat from gains during the day, then these warm masses release heat passively overnight to the cool air from natural ventilation. Since radiant cooling preempts non-active masses from storing as much heat during the day, it also reduces the opportunity for passive heat rejection overnight. To assess this hypothesis, in Section 4.3.1 we present scenarios with and without natural ventilation cooling used to pre-cool building masses overnight.

Our second hypothesis was that the radiant-to-total heat gain ratio would impact the differences between required space heat extraction rates (space cooling load) for the two systems. Radiant heat gains include radiation from the sun, radiation from indoor lighting, and radiation emitted by objects within a space. As illustrated in Figure 4.1, radiant heat gains are absorbed directly by surfaces, and influence the heat transfer networks for each cooling system type differently than a similar magnitude of convective gains. We expect that more highly radiant heat gains would cause a larger difference between the required space heat extraction rates (space cooling load) for radiant and all-air systems. To assess this hypothesis, in Section 4.3.2, we present scenarios with highly convective internal heat gains, highly radiant internal heat gains, and mixed internal heat gains.

In Section 4.4 we discuss the practical implications of our findings and highlight needs for further research, then finally in Section 4.5 we conclude with a thorough summary of the key findings.

4.2 METHODOLOGY

4.2.1 Overview of methodology

We conducted six controlled experiments in a pair of equivalent testbed buildings – one with radiant cooling and one with all-air cooling – at FLEXLAB: the US DOE's building energy efficiency testbed at Lawrence Berkeley National Laboratory (FLEXLAB). This facility enables thorough assessment of building energy systems at a realistic physical scale, with naturally occurring solar gains, and natural interaction with the surrounding environment. For each experiment we operated the two testbeds simultaneously, imposed equivalent internal gains, and controlled each system to maintain equivalent operative temperatures during normally occupied hours. The facility is illustrated in Figure 3.2.

We operated each experiment for several days, during which we monitored thermodynamic states and heat transfer rates in both testbeds. This is essential when comparing these systems, as it ensures that the temperature of masses in each testbed reach steady-state oscillations and are no longer influenced by the initial states of each system.

In each testbed we measured: air temperature distribution, operative temperature distribution, temperature of surfaces and masses, dynamic space heat extraction rates, and the cumulative amount of thermal energy extracted by each system. We were focused on comparing the sensible space heat extraction rates by each system, and so we were careful to ensure that humidity in each testbed remained low enough that supply water temperature would not cause condensation (latent space heat extraction). We did not assess the electrical performance for either system; instead, our investigation focused on the fundamental thermodynamic differences between radiant cooling and all-air cooling, regardless of the primary cooling sources and mechanical system elements that either may employ.

The six experiments discussed in this chapter were part of a larger series of experiments which also included those presented in Chapter 3 (Woolley et al., 2018b). Chapter 3 included an extensive description of the experimental facility, details about measurements, uncertainty, and explanation of a baseline calibration of the two testbeds.

4.2.2 Design of experiments

In this chapter, we compare the dynamic space heat extraction rates required to maintain equivalent operative temperatures in each testbed (space cooling loads). We present results from six separate experiments with periodic heat gains. The first two experiments (exp. #1–2) assessed how natural ventilation night pre-cooling influenced the difference between dynamic space heat extraction requirements (space cooling load) for the two testbeds. The other four experiments (exp. #3–6) assessed how the radiant-to-total heat gain ratio for internal heat gains influenced the difference. Within the second set of experiments, we also investigated one way that the thermal properties of interior surfaces interact with heat gains so as to affect the space heat extraction rates required by each system type (exp. #5–6). For this later assessment, in one experiment (exp. #5) we oriented highly radiant internal heat gains downward toward the concrete slab floor, and in a similar experiment (exp. #6) we oriented the same heat gains upward toward the suspended ceiling. The following numbered list (used as reference throughout the chapter) describes the factors associated with each experiment:

1. Mixed internal heat gains + solar gains (no natural ventilation night pre-cooling)
2. Mixed internal heat gains + solar gains + natural ventilation night pre-cooling
3. Highly convective internal heat gains + solar gains + natural ventilation night pre-cooling
4. Mixed internal heat gains + natural ventilation night pre-cooling (no solar gains)
5. Highly radiant internal gains (oriented down) + solar gains + natural ventilation night pre-cooling
6. Highly radiant internal gains (oriented up) + solar gains + natural ventilation night pre-cooling

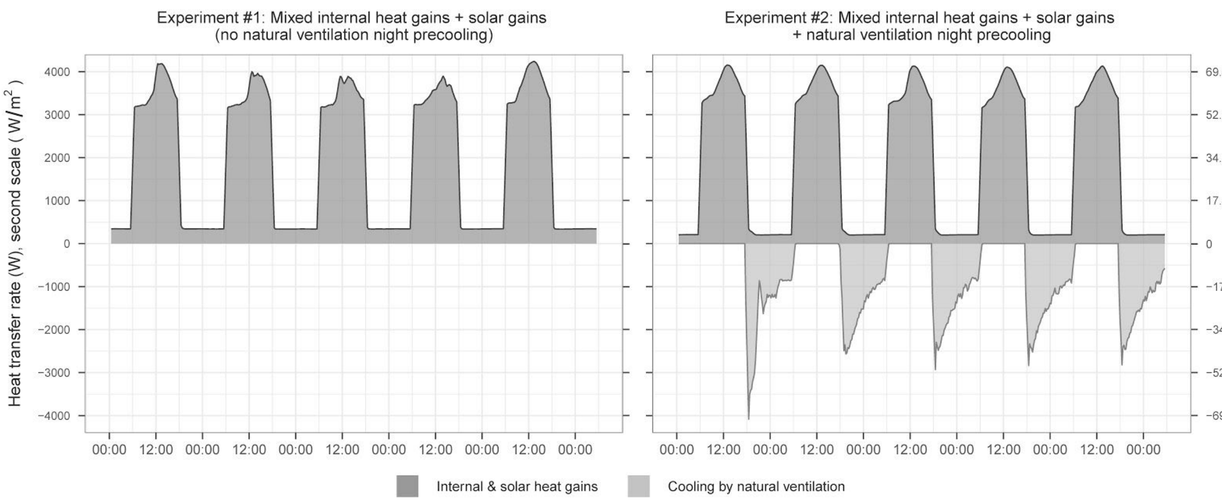


Figure 4.3: Internal and solar heat gain rates (positive axis) and natural ventilation cooling rates (negative axis) for an experiment without natural ventilation night pre-cooling (exp. #1, left) and an experiment with natural ventilation night pre-cooling (exp. #2, right). The figure plots heat transfer rates as one-hour rolling averages at fifteen-minute intervals. The patterns for the experiment with natural ventilation night pre-cooling (exp. #2, right) are typical of all five experiments that included natural ventilation night pre-cooling.

Figure 4.3 illustrates the patterns of periodic heat gains and night ventilation cooling for one experiment with natural ventilation night pre-cooling (exp. #2), and one experiment without natural ventilation night pre-cooling (exp. #1).

We supplied internal heat gains to each testbed equally. In every experiment, we turned on the internal heat gains from 06:00–18:00 each day. The median value for internal heat gains during that period in each experiment was between 3160–3760 W (55–65 W/m^2 floor area). During each experiment, internal heat gains varied from the median by as much as ± 150 W as grid voltage varied. Heat gains in both chambers varied together, so across all six experiments the median difference between internal heat gains in each chamber was only 2.5 W, and the percent difference in cumulative heat gain was only 0.05%. Heat gains during the off periods were approximately 200 W (3.5 W/m^2 floor area) due to controls and fan energy.

For the first experiment (exp. #1) the operative temperature setpoint in each testbed was 26 °C for all hours, and neither system required mechanical cooling from 18:00–06:00 because passive heat rejection to the environment exceeded the background internal heat gains. For the other five experiments, the operative temperature setpoint in each testbed was 26 °C from 06:00–18:00, then from 18:00–06:00 we cooled both testbeds to 20 °C operative temperature in mode designed to mimic natural ventilation night pre-cooling. We did not actually use natural ventilation.

Instead, we used the air handlers in each testbed to impose an idealized imitation of natural ventilation night pre-cooling that was more consistent, measureable, and repeatable than natural ventilation. We calculated the sensible space heat extraction rates in this mode ("cooling by natural ventilation" in Figure 4.3) exactly the same way that we calculated the sensible space heat extraction rates for each testbed during the 06:00–18:00 periods: by measuring the flow rate and temperature difference across the chilled-water loops that served each testbed separately. When comparing the dynamic space heat extraction rates and cumulative heat extraction (Figure 4.5, Figure 4.6, Figure 4.7, and Figure 4.8) we only counted the cooling in each testbed from 06:00–18:00. The heat extracted from each testbed between 18:00–06:00 was treated as if it were provided by natural ventilation, therefore it was not counted as a part of the mechanical heat extraction required in either testbed.

The experiments with mixed internal gains (exp. #1–2 & 4) used a combination of different electric resistance heating apparatuses – including thermal mannequins – to generate heat gains with radiant-to-total heat gain ratio ~0.5. The experiment with highly convective gains (exp. #3) used electric resistance fan heaters to generate heat gains with radiant-to-total heat gain ratio ~0, and the experiments with highly radiant gains (exp. #5–6) used an array of incandescent heat lamps to generate heat gains with radiant-to-total heat gain ratio ~0.8. We did not measure the radiant-to-total heat gain ratios for each heating apparatus; instead, we developed estimates based on other researchers' measurements of similar sources. We used *ASHRAE Fundamentals Chapter 18* (2017a) for representative rates of radiant, convective, and latent heat gain from human beings in different activities, and we referenced Jones et al. (1998) and Hosni et al. (1999) to guide estimates for other heating apparatuses used in the experiments with "mixed internal gains". We used results from Chantrasrisalai and Fisher (2007b, 2007a; 2006) to estimate the radiant-to-total heat gain ratio for infrared heat lamps in the experiments with highly radiant gains (exp. #5–6). Although Chantrasrisalai and Fisher did not measure heat lamps specifically, we based our estimate on the values measured for incandescent lamps because heat lamps are simply incandescent lamps with tungsten filaments tuned to operate at a lower temperature.

Solar gains reached 500–1,500 W each day, depending on the weather and on the sun angle. We conducted the series of experiments between August and October, when the solar altitude was changing most rapidly from day-to-day, so experiments at the end of the series received larger solar gains. Meteorological conditions during the experiments were mild, median outdoor temperature was 14.3 °C with interquartile range of 7.75 °C. Heat transfer through outdoor-exposed surfaces was relatively small; it ranged from 30 W/m^2 gains to 15 W/m^2 losses, and changed direction diurnally. Overall, losses through outdoor-exposed surfaces were more dominant than gains. Consequently, the cumulative thermal energy extracted from each testbed by mechanical cooling and natural ventilation cooling was 10-30% smaller than the cumulative thermal energy from internal and solar gains. For reference, Figure 4.4 presents a detailed disaggregated time series view of the different internal and solar heat gain components. Figure 4.9 and Figure 4.10 compare the cumulative thermal energy from internal and solar gains to the cumulative thermal energy extracted by each system type, and by natural ventilation night pre-cooling.

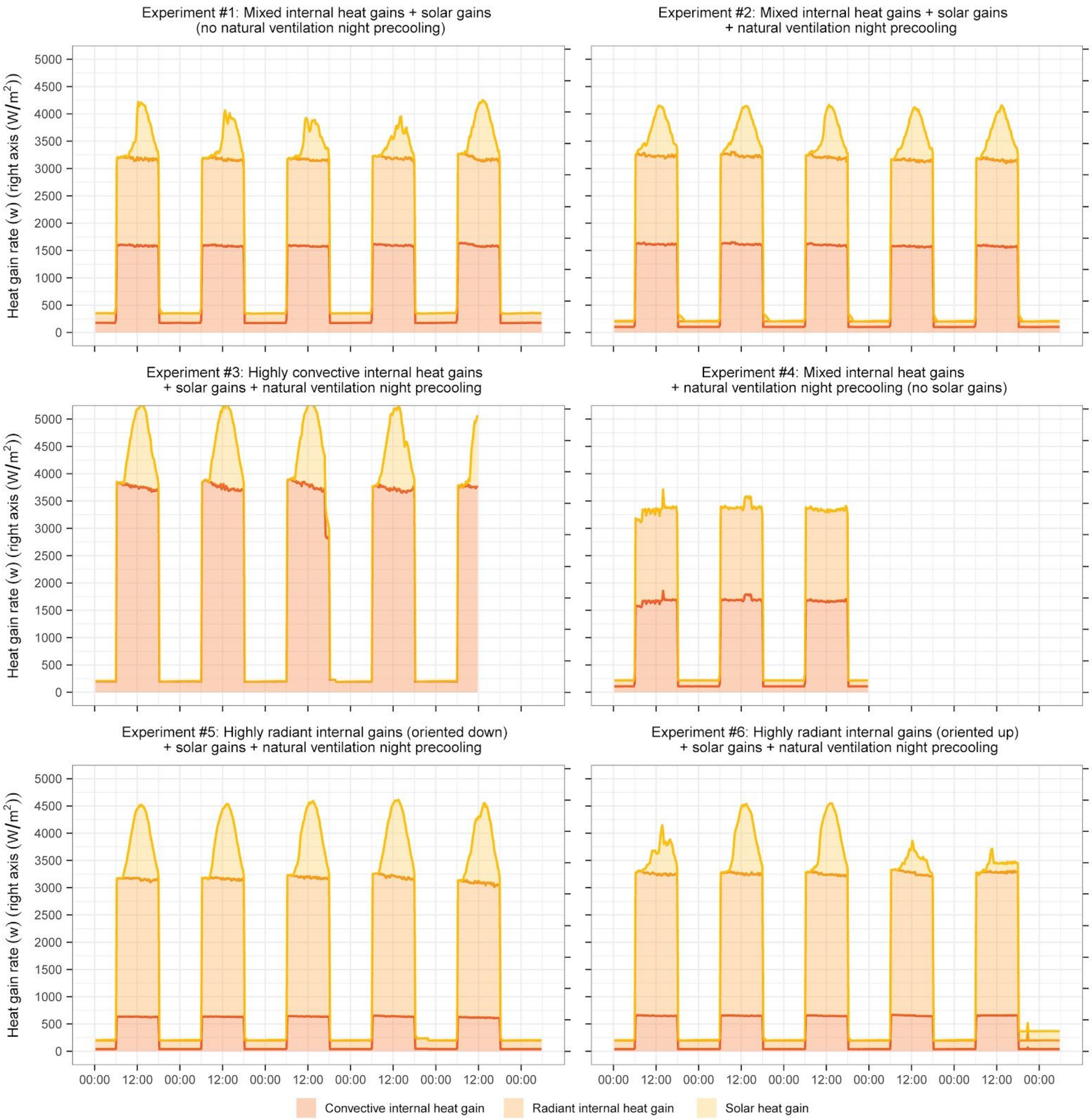


Figure 4.4: Internal and solar heat gain rates in each of the six experiments. The plots disaggregate the estimated radiant and convective components of internal heat gains, and present data across multiple days. Internal heat gain data are plotted at one-minute intervals, solar heat gain data are plotted as 30 minute rolling averages on 15 minute intervals.

4.3 RESULTS AND INTERPRETATIONS

Table 4.1: Summary metrics for each experiment

		Difference between peak space heat extraction rate				Difference between daily thermal energy use		
		min–max (median)				min–max (cumulative)		
	Experiment Description	%	W	W/m^2	minutes earlier	%	Wh	Wh/m^2
1	Mixed internal heat gains + solar gains (no n.v. night pre-cooling)	2–10% (5%)	72–331 (140)	1.25–5.7 (2.43)	30–51 (46)	5–11% (7%)	1,599–3,480 (10,992)	27.8–60.4 (192)
2	Mixed internal heat gains + solar gains + n.v. night pre-cooling	18–22% (20%)	376–472 (448)	6.5–8.2 (7.8)	45–100 (66)	31–37% (35%)	5,126–5,963 (27,482)	89–103 (477)
3	Highly convective internal heat gains + solar gains + n.v. night pre-cooling	10–17% (12%)	334–579 (393)	5.8–10.1 (6.8)	0–66 (61)	17–32% (26%)	4,555–8,399 (28,212)	79–145 (490)
4	Mixed internal heat gains + n.v. night pre-cooling (no solar gains)	19%	322	5.6	0	27–41% (33%)	3,901-4,878 (12,857)	68–85 (223)
5	Highly radiant internal gains (oriented down) + solar gains + n.v. night pre-cooling	18–29% (21%)	465–568 (501)	8.1–9.9 (8.7)	59–78 (67)	36–47% (40%)	5,638–6,633 (30,446)	98–115 (529)
6	Highly radiant internal gains (oriented up) + solar gains + n.v. night pre-cooling	15–23% (21%)	265–566 (517)	4.6–9.8 (9.0)	79–102 (90)	17–28% (21%)	1,971–4,926 (17,366)	34–85.5 (301)

4.3.1 Dynamic space heat extraction rates with and without natural ventilation night pre-cooling

Figure 4.5 compares the dynamic space heat extraction rates from each system in experiments with and without natural ventilation night pre-cooling. For an all-air system, the space heat extraction rate is the enthalpy difference between airflow supplied to the space and air flow leaving the space. For a radiant system, the space heat extraction rate is the sum of convective and radiant (longwave and shortwave) heat transfer rates at the indoor face of the internally cooled surface.

First, the results from both experiments in Figure 4.5 reveal that to maintain equal comfort conditions: (1) radiant cooling must extract heat from gains earlier than all-air cooling, (2) the peak space heat extraction rate for radiant must be larger than for all-air cooling, (3) the peak space heat extraction rate for radiant must occur earlier than for all-air cooling, and (4) radiant must remove more heat from a building than all-air cooling. This reinforces previous research findings (Feng et al., 2013, 2014b; Niu et al., 1995, 1997) including those presented in Chapter 3 (Woolley et al., 2018a).

Further, comparison of the two experiments in Figure 4.5 reveals that natural ventilation night pre-cooling strongly increases the magnitude of the difference between the dynamic space heat extraction requirements (space cooling load) for each system. We quantify the impact on each aspect of this difference in the following three paragraphs.

The comparison reveals that natural ventilation night pre-cooling increases the difference between the peak space heat extraction rates for each system type. In the experiment without natural ventilation night pre-cooling, the daily peak space heat extraction rate for the radiant testbed was 2–10% larger than for the all-air testbed; while in the experiment with natural ventilation night pre-cooling it was 16–22% larger. The median differences for each five day experiment were 5% and 20% respectively. In terms of heat transfer rates, these differences equate to 1.3–5.8 W/m^2 (median 2.4 W/m^2) and 6.5–8.2 W/m^2 (median 7.8 W/m^2) respectively.

Natural ventilation night pre-cooling also increased the difference between the times at which the peak space heat extraction rate occurred for each system type. In the experiment without natural ventilation night pre-cooling, the peak space heat extraction rate for the radiant testbed occurred 30–50 minutes earlier than for the all-air testbed; while in the experiment with natural ventilation night pre-cooling, the peak space heat extraction rate for the radiant testbed occurred 45–100 minutes earlier.

The comparison also reveals that natural ventilation night pre-cooling can have a very large impact on the difference between the total amount of thermal energy that each system must remove. In the experiment without natural ventilation night pre-cooling the radiant system extracted 7% more thermal energy over the course of five days, whereas in the experiment with natural ventilation night pre-cooling the radiant system extracted 35% more thermal energy. These differences equate to an average difference in the space heat extraction rate of 183 W (3.2 W/m^2) and 458 W (8.0 W/m^2) respectively.

Summary metrics for all six experiments are presented in Table 4.1. The multiple day time series results for all six experiments are presented in Figure 4.7, and multiple day time series for the cumulative space heat extraction energy for all six experiments are presented in Figure 4.8

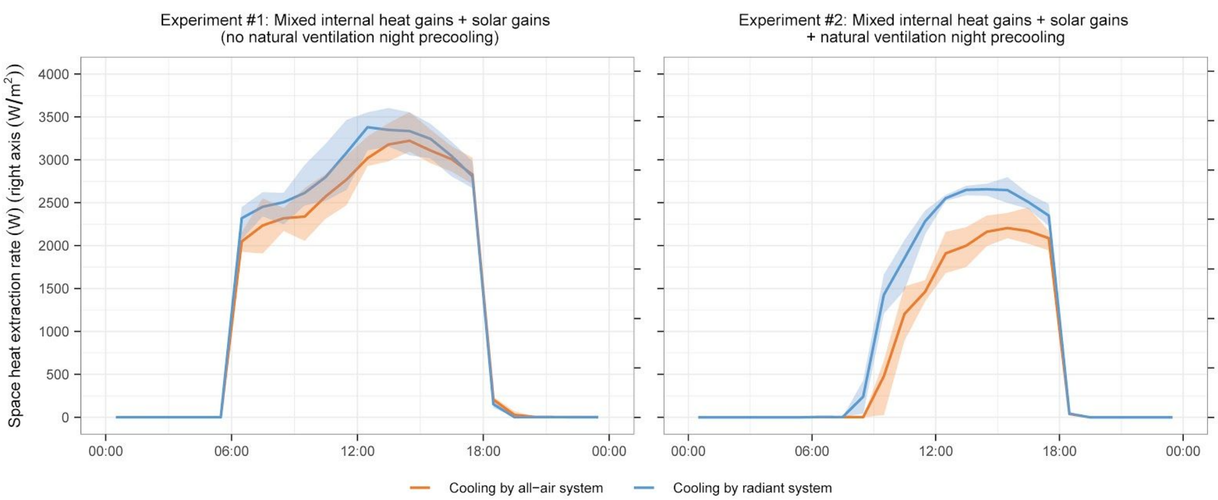


Figure 4.5: Space heat extraction rates for radiant (blue) and all-air systems (orange) in comparable experiments without natural ventilation night pre-cooling – exp. #1 (left) and with natural ventilation night pre-cooling – exp. #2 (right). Each 24 hour time series is a composite of data from five days; the lines indicate the mean of one-hour rolling means on one-hour intervals, while the ribbons indicate the minimum and maximum one-hour rolling mean values on one-hour intervals. The multiple day results for all six experiments are presented in Figure 4.7.

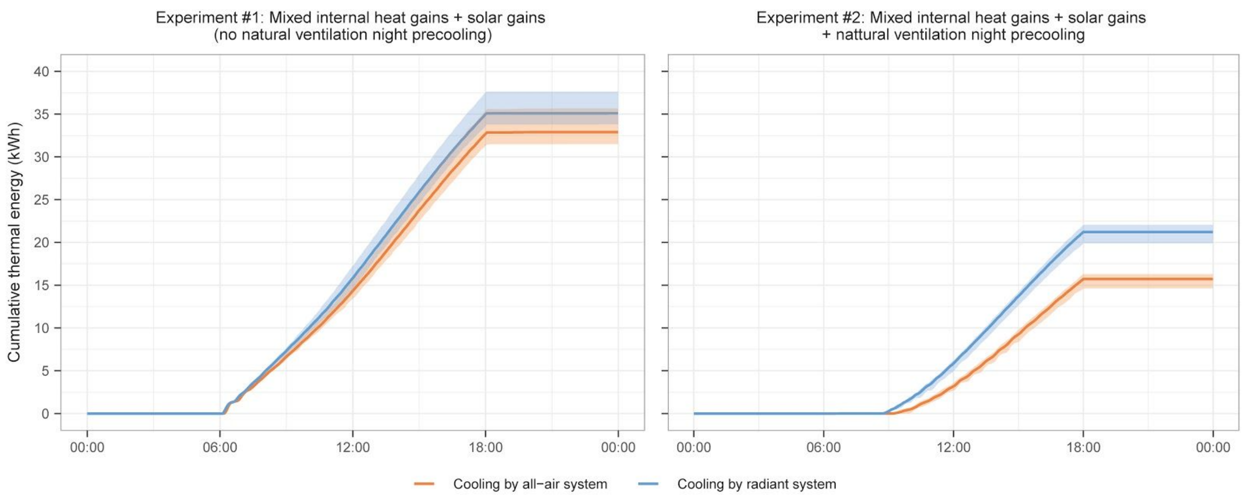


Figure 4.6: Cumulative space heat extraction energy for radiant (blue) and all-air systems (orange) in comparable experiments without natural ventilation night pre-cooling – exp. #1 (left), and with natural ventilation night pre-cooling – exp. #2 (right). Each 24 hour time series is a composite of data from five days; the lines indicate the mean of one-hour rolling means on one-hour intervals, while the ribbons indicate the minimum and maximum one-hour rolling mean values on one-hour intervals. The multiple day time series results for all six experiments are presented in Figure 4.8...

The fundamental difference between space heat extraction rates required by the two system types occurs because a portion of the heat gain that non-active masses would absorb (in a building with all-air cooling) is instead removed by radiant heat transfer to the internally cooled surfaces (in a building with radiant cooling). For the same reason, all interior surfaces in a space with radiant cooling are cooler than in a similar space with all-air cooling. As a result, spaces with radiant cooling experience somewhat larger heat gains due to conduction heat transfer through outdoor exposed surfaces, and since less heat is absorbed by and stored in non-active masses, less heat is rejected to the environment by passive means. These fundamental differences should exist in any scenario where radiant cooling and all-air cooling maintain equal operative temperatures, but the differences are greater when there is greater opportunity for non-active masses to reject heat passively – as demonstrated by the larger differences in the experiment with night ventilation pre-cooling.

The difference in the amount of heat absorbed by, stored in, and ultimately rejected passively from non-active masses each day is also demonstrated by comparison of the patterns of daily temperature change for non-active masses. For example: in the experiment with night ventilation pre-cooling, both testbeds started each day with practically equal slab core temperatures (median 21.5 °C with 0.0–0.2 °C difference); then over the course of each day, the slab core in the all-air testbed increased by 2.8 °C (5°F) – to 24.3 °C (75.7 °F) – whereas the slab core in the radiant testbed only increased by 1.8 °C (3.2 °F) – to 23.3 °C (73.9 °F). For comparison, in the experiment without night ventilation pre-cooling, the slab core temperature in the all-air testbed only increased by 0.5 °C – from 25.4 °C (77.7 °F) to 25.9 °C (78.6 °F) – and the slab core in the radiant testbed only increased 0.15 °C (0.27 °F) – from 24.5 °C (76.1 °F) to 24.65 °C (76.37 °F).

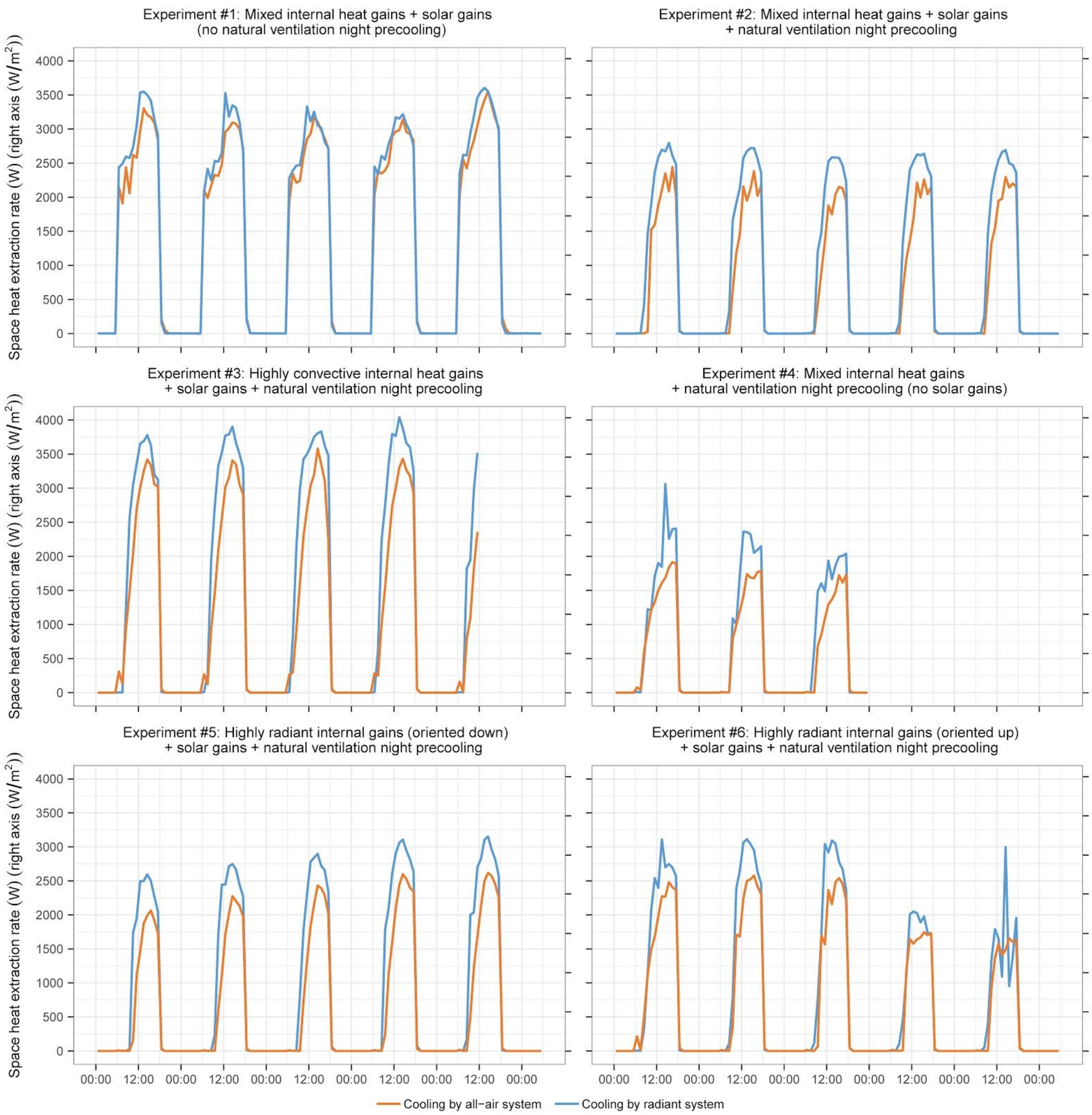


Figure 4.7: Space heat extraction rates for radiant (blue) and all-air systems (orange) in each of the six experiments. The plots present data across multiple days as one-hour rolling averages at one-hour intervals.

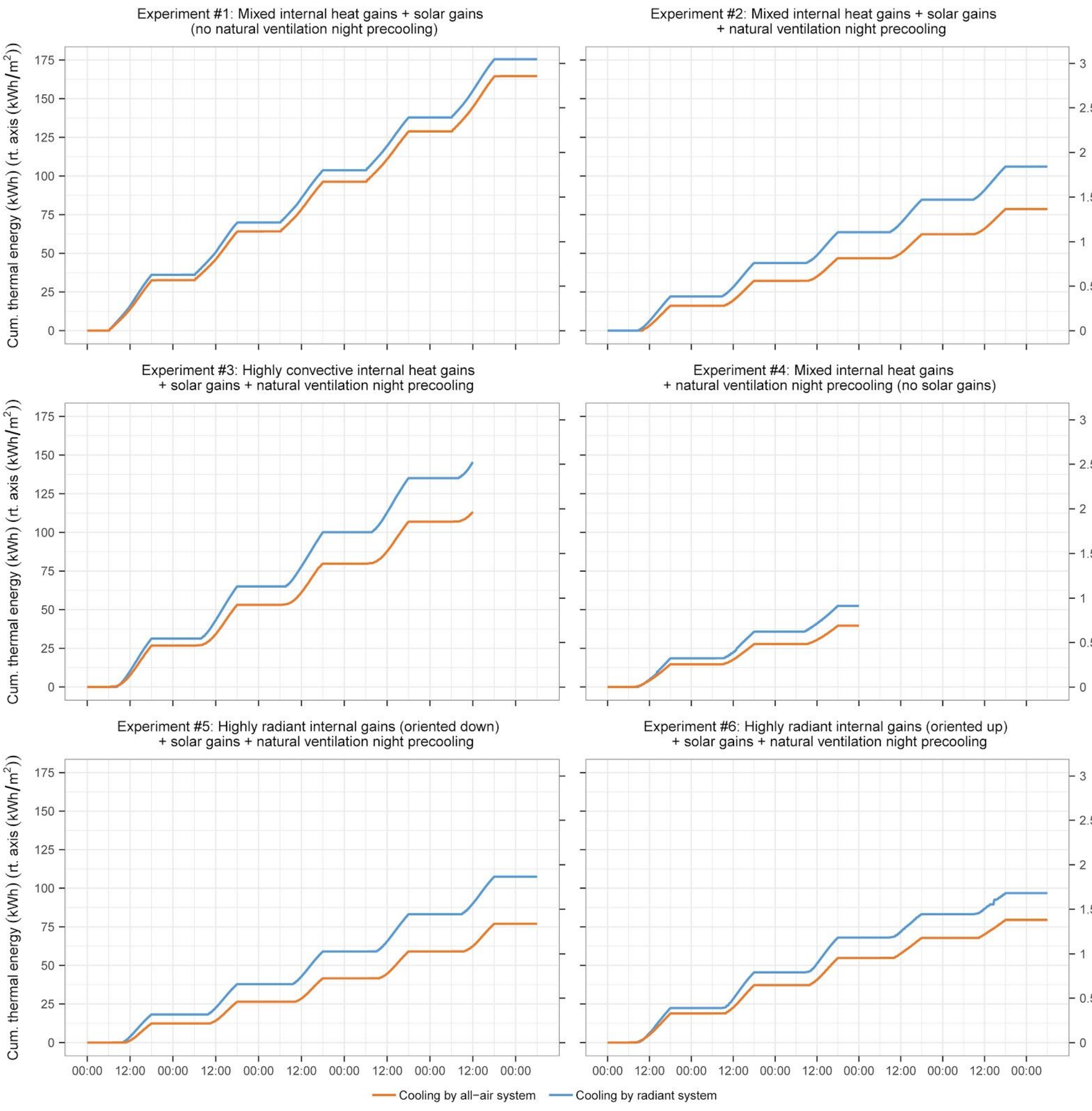


Figure 4.8: Cumulative space heat extraction energy for radiant (blue) and all-air systems (orange) in each of the six experiments. The plots present data across multiple days at one-minute intervals.

Figure 4.9 compares the disaggregated cumulative thermal energy flows in the all-air and radiant testbeds in an experiment with night ventilation cooling. This disaggregated view illustrates that the non-active masses in a space with all-air cooling absorb more heat during the day, and subsequently, can reject more heat to the environment by passive means. Therefore, in general, the percent difference in cumulative heat extraction for the two system types should increase as the opportunity for passive heat rejection increases, and as the proportion of heat gain due to heat transfer through outdoor-exposed surfaces increases.

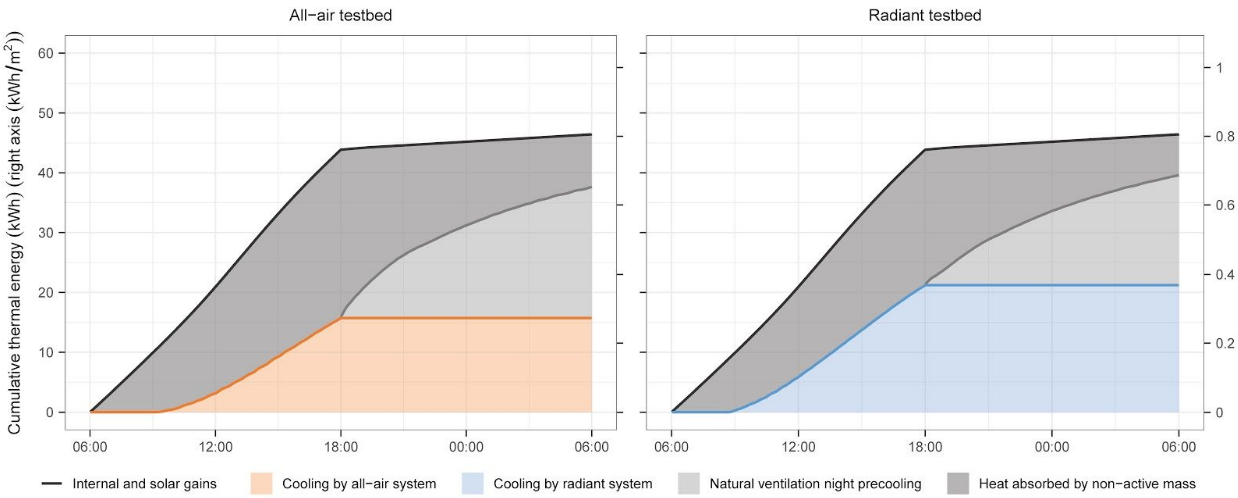


Figure 4.9: Cumulative thermal energy flows in the all-air (left) and radiant (right) testbeds in an experiment with natural ventilation night pre-cooling (exp. #2). Each plot indicates the cumulative thermal energy extracted from the space by mechanical systems (orange or blue), the cumulative thermal energy extracted from the space by natural ventilation night pre-cooling (light grey), and the cumulative thermal energy from internal or solar gains stored by non-active masses and/or released passively to the environment (dark grey). Data from multiple days in the experiment is plotted as composite 24-hour time series using median values on one-minute intervals. Figure 4.10 shows similar results across multiple days for all six experiments.

These findings have substantial consequences for the design and control of radiant systems, especially in coordination with natural ventilation or other passive cooling strategies. First, as Feng et al. (2014a) revealed, industry common practice “cooling load calculation” methods underestimate the peak space heat extraction rates required for radiant cooling systems. Second, radiant cooling must remove more thermal energy than all-air cooling; and in some cases the additional thermal burden can be very large. In the experiment with mixed internal gains, solar gains, and natural ventilation night pre-cooling, radiant cooling had to extract 35% more thermal energy than all-air cooling. To consume less primary energy than an all-air system, buildings with radiant cooling must be designed so that the advantages for cooling plant efficiency and thermal distribution efficiency overcome the additional thermal burden.

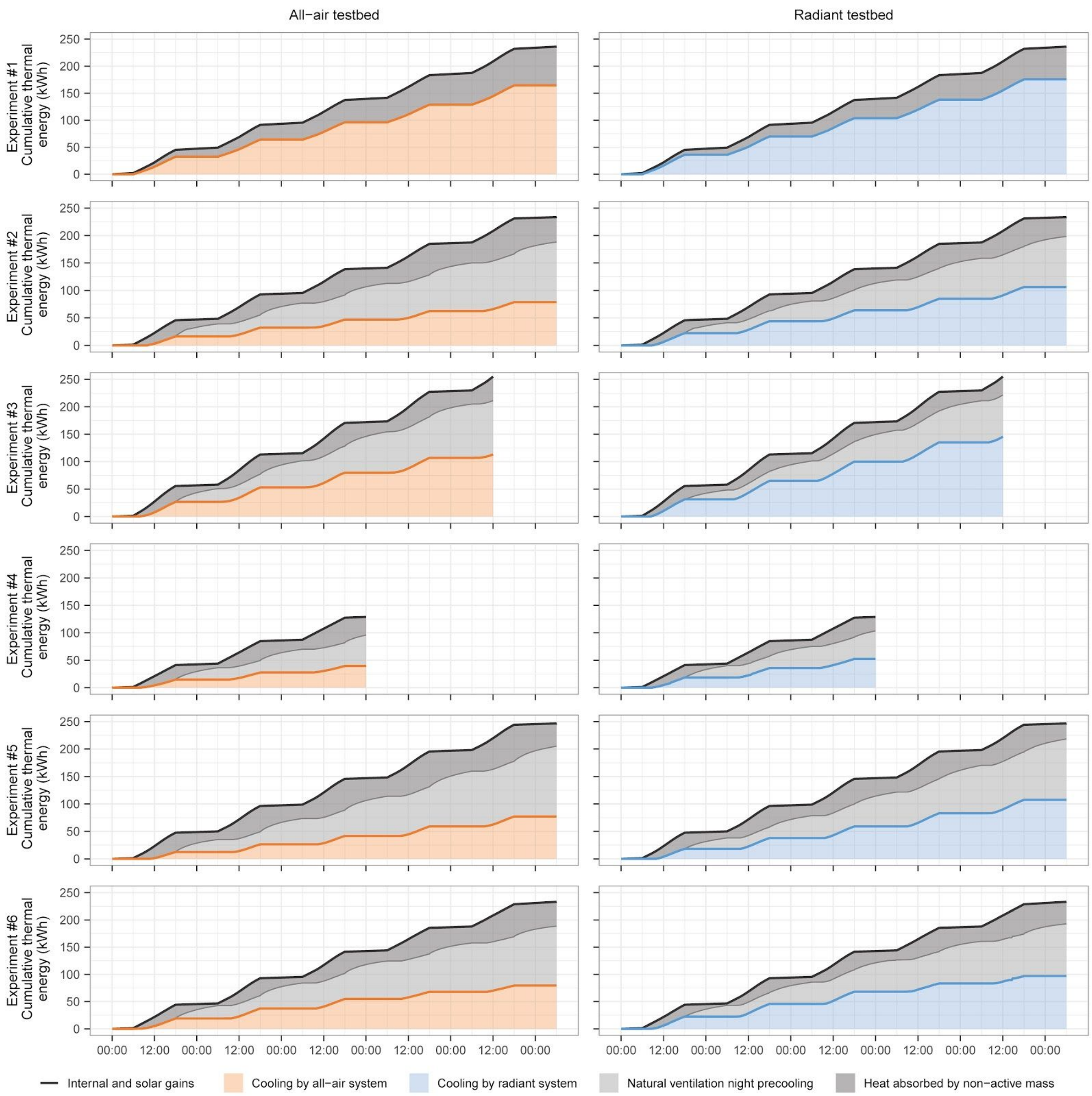


Figure 4.10: Cumulative thermal energy flows in the all-air (left) and radiant (right) testbeds in each of the six experiments. Each plot indicates the cumulative thermal energy extracted from the space by mechanical systems (orange or blue), the cumulative thermal energy extracted from the space by natural ventilation night pre-cooling (light grey), and the cumulative thermal energy from internal or solar gains stored by non-active masses and/or released passively to the environment (dark grey). Data is plotted across all days in each experiment on one-minute intervals.

4.3.2 The impact of heat gain characteristics

Comparison of the results from experiments with different radiant-to-total heat gain ratios reveals that heat gain characteristics can have a large impact on the difference between dynamic space heat extraction requirements (space cooling load) for the two system types.

Firstly, the difference between the peak space heat extraction rate for the two system types was larger when heat gains were more highly radiant. In the experiment with highly convective internal gains (exp. #3), the peak space heat extraction rate in the radiant testbed was 10–17% larger (median 12%), which equates to 5.8–10.1 W/m^2 (median 6.8 W/m^2) and occurred 0–60 minutes earlier. In an experiment with highly radiant internal gains (exp. #5), the peak space heat extraction rate was 18–29% larger (median 21%), which equates to 8.1–9.8 W/m^2 (median 8.7 W/m^2), and occured 60–80 minutes earlier.

Secondly, heat gain characteristics also have a large impact on the difference between the total amount of thermal energy that each system type must remove to maintain equal operative temperatures. In the experiment with highly convective internal gains (exp. #3) the radiant system extracted 29% more thermal energy over the course of the multi-day experiment, whereas in an experiment with highly radiant internal gains (exp. #5) the radiant system extracted 40% more thermal energy.

Heat gain characteristics impact the differences between the two systems because non-active masses absorb radiant gains more readily than convective gains, and because radiant cooling extracts heat from non-active masses more readily than all-air cooling. In a hypothetical scenario with only radiant gains, the space heat extraction requirement (space cooling load) for the all-air system would be limited to the heat that is shed by convection from non-active masses that have absorbed the heat gains. If the non-active masses have high thermal diffusivity, and a large thermal capacity, they will absorb and store a large amount of heat without large change in surface temperature, and therefore, will not shed much heat by convection. In the same scenario radiant cooling would extract heat through multiple mechanisms: by radiant transfer directly from gains, by radiant transfer with non-active surfaces, and by convective transfer with the room air.

Figure 4.11 illustrates a relationship between heat gain characteristics and the difference between space heat extraction requirements (space cooling loads) for radiant and all-air cooling systems. The figure plots the percent difference between cumulative space heat extraction energy (left) and peak space heat extraction rate (right) in the radiant and all-air testbeds during each 06:00–18:00 period for all five experiments with natural ventilation night pre-cooling. Each of these experiments imposed a different radiant-to-total ratio for internal heat gains, so we plotted the results as a function of the coincident cumulative radiant-to-total heat gain ratio over each day. Solar gains varied naturally from day-to-day, and in one experiment with mixed internal gains solar was omitted altogether by blocking windows entirely with rigid insulation. The values of the daily cumulative radiant-to-total heat gain ratio in Figure 4.11 include the contribution from solar gains, but do not include the contribution from gains by conduction through outdoor-exposed surfaces because we could not measure this heat gain component accurately.

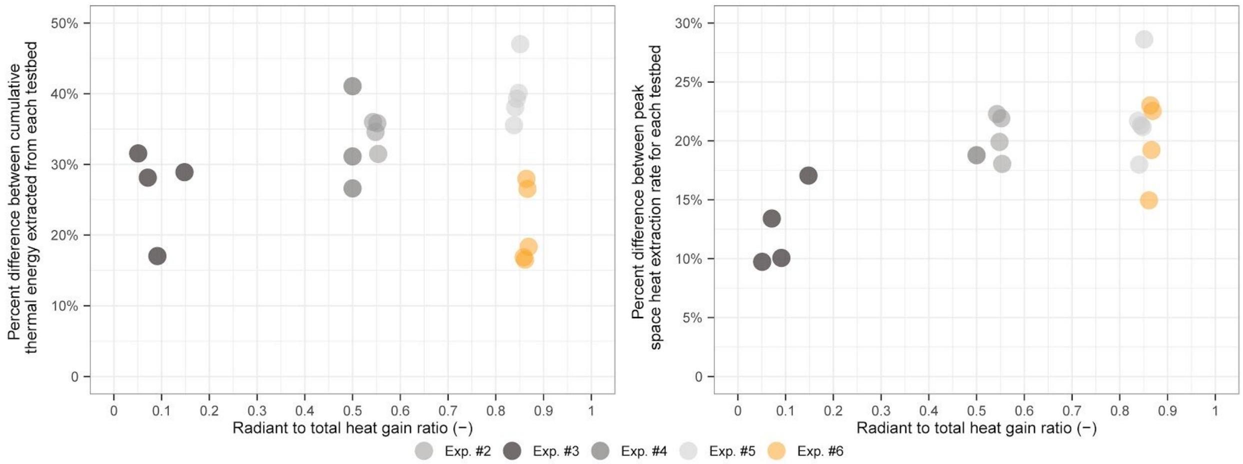


Figure 4.11: The percent difference between cumulative thermal energy extracted from each testbed (left) and the percent difference between the peak space heat extraction rate for each testbed (right) during each 06:00–18:00 period for all five experiments with natural ventilation night pre-cooling. Each point represents a single 06:00–18:00 period, and each is plotted as a function of the cumulative radiant-to-total heat gain ratio on the corresponding day. The significant difference between cumulative space heat extraction in the experiments with highly radiant gains oriented downward (exp. #5: grey) and upward (exp. #6: orange) illustrates that heat gains and surface thermal properties interact in a way that impacts the difference between space heat extraction requirements for radiant and all-air cooling systems. Excluding the results with highly radiant gains oriented upward (orange) there is a strong positive correlation (Pearson's r=0.75) between radiant-to-total heat gain ratio and the percent difference between cumulative space heat extraction energy. Figure 4.12, presents similar plots that also include results from the experiment without natural ventilation night pre-cooling (exp. #1).

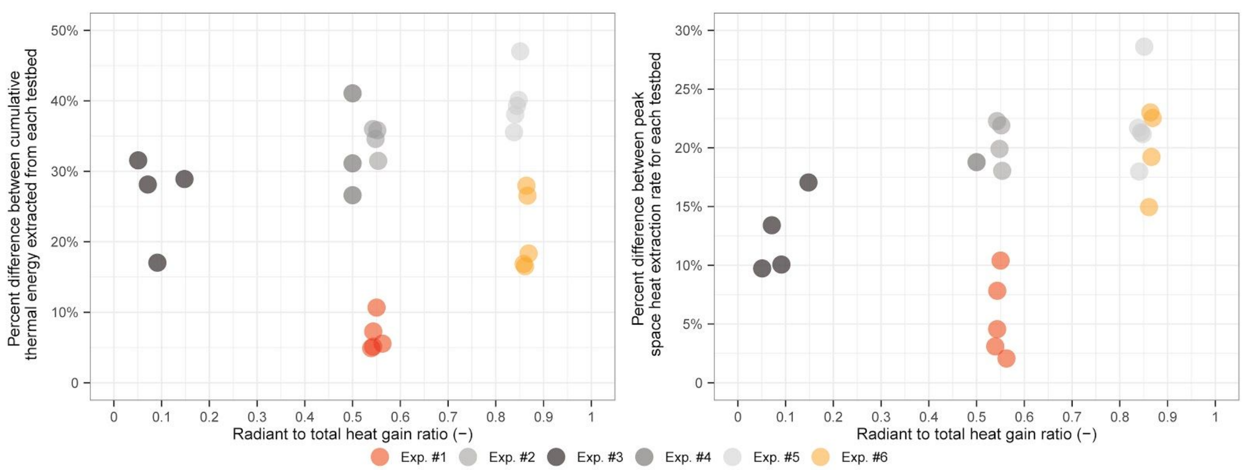


Figure 4.12: The percent difference between cumulative space heat extraction energy (left) and peak space heat extraction rate (right) in the radiant and all-air testbeds during each 06:00–18:00 period for all six experiments, presented as a function of the cumulative radiant-to-total heat gain ratio on the corresponding day.

The results confirm a strong positive correlation (Pearson's r=0.75) between radiant-to-total heat gain ratio and the differences between cumulative thermal energy extracted from each testbed. The differences tend to be larger for highly radiant heat gains, and smaller for highly convective heat gains. In light of the accuracy of measurements in this study, we can confidently conclude that the day-to-day differences within each multi-day experiment are mainly due to real variations in exogenous variables such: solar heat gains, heat transfer through outdoor-exposed surfaces, and infiltration.

However, the results also reveal that there are important interactions between heat gain type and the thermal properties of surfaces within and enclosing a space. We conducted two experiments with highly radiant heat gains. In one (exp. #5) we oriented infrared lamps downward – toward the non-active concrete slab floor – while in the other (exp. #6) we oriented infrared lamps upward – toward the suspended ceiling (an acoustic tile ceiling in the all-air testbed, and the internally cooled surface in the radiant testbed). When oriented downward, radiant gains in the all-air testbed were readily absorbed by and stored in the 15.25 cm (0.5 ft) thick concrete slab, but when oriented upward radiant gains absorbed by the suspended acoustic ceiling were rapidly converted to convective gains which were subsequently removed by the all-air cooling system. In the radiant testbed, radiant gains oriented upward were immediately removed by the internally cooled surface, and when oriented downward the direct exposure between floor and ceiling ensured that radiant gains were quickly removed by radiant heat transfer to the internally cooled surface. As a result, the percent difference between cumulative space heat extraction for radiant and all-air systems was larger when radiant heat gains were oriented downward. When radiant heat gains were oriented upward the percent difference in cumulative space heat extraction was more similar to the scenario with highly convective gains.

When the non-active surfaces within and enclosing a space have high thermal diffusivity and high thermal capacity, the temperature of those surfaces will be more resilient to heat gain and will shed less heat by convection. High thermal mass surfaces – such as exposed concrete construction – have high thermal diffusivity and high thermal capacity; they can readily absorb heat from the radiant component of heat gains without a substantial increase in surface temperature. Low thermal mass surfaces – such as raised floors, suspended ceilings, floor coverings, or furniture – have low thermal diffusivity and/or low thermal capacity; they tend to decrease the difference between the cumulative heat extraction requirements for radiant and all-air systems.

Although the orientation of radiant heat gains impacted the percent difference between the cumulative space heat extraction, it did not have a significant impact on the percent difference between the peak heat extraction rates. This may seem surprising; after all, a space with lots of non-active thermal mass would certainly be expected to reduce the peak cooling requirement for any system type, so it might seem natural to expect that orienting radiant heat gains toward the high thermal mass floor would decrease the peak space cooling requirement as well as the cumulative cooling requirement. However, we found that although the slab temperature increased by a greater magnitude each day in the case with radiant gains pointed toward the floor (it stored more heat overall), at peak the rate of temperature change for surfaces throughout the space was similar whether the radiant gains were oriented up or down. This indicates that most of the difference in thermal energy storage occurred early in the day. At peak, thermal conditions had equilibrated to a point that the division between gains stored and gains extracted by the mechanical system was similar whether the radiant gains were oriented up or down.

These findings have consequences for the practical design of radiant cooling systems. In particular, it is often noted that radiant cooling can have an especially large cooling capacity in spaces with solar gains, yet this characteristic can substantially increase the amount of thermal energy that a cooling plant must process each day when there is also an opportunity for passive heat rejection. Often, design of radiant cooling systems uses load calculation strategies intended for all-air systems, but in light of the results presented here, such practice could miss the actual cooling requirements by a substantial margin.

4.4 DISCUSSION

To maintain equal operative temperature in response to equal heat gains, the required space heat extraction rates (space cooling load) for radiant and all-air systems are different. Radiant cooling must extract more heat from a space than all-air cooling, and the peak space heat extraction rate (peak space cooling load) must be larger and occur earlier. Previous experiments proved this is a fundamental difference that must occur in any circumstance where radiant cooling and all-air cooling maintain equal operative temperature (Feng et al., 2014b; Woolley et al., 2018a). With the study presented in this chapter we have experimentally confirmed – far beyond the bounds of uncertainty – that heat gain characteristics, interior surface thermal properties, and the availability of passive cooling can have very large impacts on the relative difference between the dynamic space heat extraction requirements (space cooling load) for radiant and all-air systems.

The peak space heat extraction rate for radiant must be larger but the cooling plant can be smaller

In considering the results of this study, bear in mind that we compared the rates at which different cooling system types must extract thermal energy from a space; we did not assess cooling plant heat transfer rates. For a high thermal mass radiant system, the space heat extraction rate is very different from the heat extraction rate for the cooling plant, while for an all-air system, and for a low thermal mass radiant system, the two heat extraction rates may be nearly identical. So, while our results demonstrate that the peak space heat extraction rate for at the indoor face of internally cooled surfaces must be larger than the peak space heat extraction rate for an all-air system, we also acknowledge that a strategically controlled high thermal mass radiant system could allow the cooling plant to be much smaller than what would be required for an all-air system in equivalent circumstances. Bourdakis et al. (2015) assessed this issue directly with a simulation study. On the other hand, the cooling plant for low thermal mass radiant systems – like cooled metal ceiling panels – would need to have capacity to serve the larger peak space heat extraction requirements (peak space cooling load) revealed by our experiments. Importantly, in addition to space heat extraction requirements (space cooling load), the cooling plant must also have capacity to overcome system heat gains such as fan energy, pump energy, and duct heat transfer. In some cases, the difference between system heat gains for radiant and all-air systems may be even larger than the differences between required space heat extraction rates (space cooling load) revealed in this study.

Our experiments span a realistic range of radiant-to-total heat gain ratios

Our experiments included scenarios with internal heat gains across a wide range of radiant-to-total heat gain ratios. Although some computers, laboratory equipment, and cooking appliances may independently have radiant-to-total heat gain ratios of 15% or less, it is very unlikely that many spaces would have an aggregate radiant-to-total heat gain ratio as low as in our experiment with highly convective gains. Most buildings include a wide variety of different heat gains and subsequently have mid-range radiant-to-total heat gain ratios. However, spaces with extensive solar gains – such as lobbies or atria – could easily match the upper end of radiant-to-total heat gain ratios in our experiments. Conspicuously, radiant cooling and natural ventilation cooling are commonly used together in lobbies and atria (Paliaga et al., 2017, 2018) – often targeting reduced surface temperatures in spaces with large solar gains for thermal comfort reasons – yet our findings suggest that the cumulative mechanical cooling requirement for radiant systems can be 40% larger than for all-air systems in such a scenario.

The availability of passive cooling accentuates the impact of the radiant-to-total heat gain ratio

All of the experiments we used to test the impact of radiant-to-total heat gain ratio included natural ventilation night pre-cooling; this undeniably accentuated the magnitude of the difference between the dynamic space heat extraction requirements (space cooling load) for radiant and all-air cooling systems compared to what would occur in similar scenarios without night ventilation pre-cooling. Since the magnitude of the difference between the dynamic space heat extraction rates was smaller without night ventilation pre-cooling, we expect that the magnitude of change due to differences in the radiant-to-total heat gain ratio would also be smaller.

The thermal properties of interior surfaces are important for design of either cooling system type

In addition to demonstrating the impact of radiant-to-total heat gain ratio, the results presented here confirm that the thermal properties of surfaces within and enclosing a space impact the difference between dynamic space heat extraction requirements (space cooling load) for radiant and all-air systems. This impact is mainly due to the way that different surfaces absorb and store heat, or more specifically: how rapidly the temperature of surfaces rise in response to heat gains. Our results indicate that surfaces with low thermal diffusivity and/or low thermal capacity tend to diminish the influence that radiant-to-total heat gain ratio has on the difference between the dynamic space heat extraction requirements (space cooling load) for radiant and all-air systems. Notably, surfaces with low thermal diffusivity and low thermal capacity more rapidly convert radiant gains to convective gains. We showed that the interaction between surface properties and heat gains has a large impact, however we expect that the two scenarios we compared to make this assessment are among the most different that would reasonably be encountered: on the one hand radiant gains were incident on a concrete slab, and on the other hand they were incident on a low mass suspended ceiling. We recommend further simulation research to assess the extent to which typical construction practices and typical radiant-to-total heat gain ratios influence the difference.

The thermal properties of surfaces within and enclosing a space tend to be very important to the design of either radiant or all-air cooling systems; yet these properties are often not considered as a part of the design process, and generally are not fully accounted for by building energy simulation software. For example Raftery et al. (2014) showed that furnishings – which have low thermal diffusivity and low thermal capacity – tend to cover a substantial fraction of the floor area, block solar gains from being absorbed by the floor slab, rapidly convert solar gains to convective heat in the space, and consequently have substantial impact on the dynamic space heat extraction requirements (space cooling load). Furthermore, compared to buildings with all-air cooling, buildings with high thermal mass radiant cooling typically have fewer non-active surfaces with low thermal diffusivity and low thermal capacity – for example, they often do not have suspended acoustic ceilings, and often include exposed construction elements. Consequently we expect that the dynamic space heat extraction requirements (space cooling load) for 'typical' radiant buildings and 'typical' all-air buildings would be somewhat closer than what we have found through previous simulations and experiments, because low thermal diffusivity surfaces in buildings with all-air cooling would tend to decrease the amount of heat stored in masses, decrease the amount of thermal energy that can be rejected by passive means, and increase the peak space heat extraction requirement (peak space cooling load).

Supplemental cooling in radiant buildings would influence the dynamic space cooling requirements

Most buildings with high thermal mass radiant cooling also include supplemental cooling using some type of forced-air system. Very few researchers have considered the coordination between high thermal mass radiant cooling and supplemental cooling, and it is not known how dividing the space cooling between these two systems would influence the dynamic space heat extraction requirements (space cooling load) compared to an all-air system. We expect that the differences addressed in this chapter would be smaller for a radiant cooled space with supplemental cooling, because such a building is really a hybrid of radiant cooling and all-air cooling.

The magnitude of our findings will vary with climate characteristics

The magnitude of the differences highlighted in this chapter depend largely on the availability of passive cooling opportunities. Some climates have extensive passive cooling opportunities while others have limited opportunities. We have focused on the impact of natural ventilation night pre-cooling, but the idealized imitation of night ventilation cooling imposed for the laboratory experiments likely yielded the maximum impact that could be expected – we cooled the space toward the lower end of what is considered comfortable according to *ASHRAE Standard 55* (2017b). Further research could assess the way that night ventilation pre-cooling affects the difference between the annual cooling requirements for radiant and all-air systems, and to identify the geographic ranges where these differences are important.

Control for radiant cooling and natural ventilation pre-cooling should be coordinated strategically

In practice, there is rarely much strategy to the coordination of radiant cooling systems and natural ventilation cooling systems. Natural ventilation is common among buildings with high thermal mass radiant cooling, but these radiant systems are typically controlled to constant indoor air temperature or slab temperature setpoints for all hours and days of the week (Paliaga et al., 2017, 2018; Raftery, Duarte, & Dawe, 2019b, 2019a). Similar to the case observed in our experiments, this control strategy will cause the cooling plant to operate in response to heat gains

– rather than ahead of heat gains – which will preempt some of the benefits of natural ventilation night pre-cooling compared to an all-air system with natural ventilation night pre-cooling. It is likely that if a high thermal mass radiant system were controlled so that the cooling plant operated ahead of the typical heat gain period – as an "if needed" supplement to natural ventilation night pre-cooling – more heat could be absorbed by non-active masses during the day then rejected to natural ventilation night pre-cooling overnight. This issue deserves further research to develop, test, and demonstrate standard sequences of operation that optimize coordination between high thermal mass radiant and night ventilation cooling systems.

Evaporative water cooling could provide substantial efficiency benefits for radiant cooled buildings

Natural ventilation night pre-cooling is not the only passive or very-low-energy cooling strategy that could be used to reduce the need for vapor-compression equipment. In most climates, evaporative water cooling (waterside economizer cooling) can also be used to provide very-low-energy cooling for radiant systems. Similar to the control strategy recommended to coordinate radiant cooling with night ventilation pre-cooling, a high thermal mass radiant system could be controlled to allow the radiant slab to absorb and store heat during the day, then operate waterside economizer cooling to extract the heat overnight when wet bulb temperatures are lower. For some climates and scenarios, this strategy could completely eliminate the need for vapor-compression equipment (Duarte et al., 2018; Moore, 2008a, 2008b; Tian & Love, 2009b). Moreover, there are many climates where natural ventilation cooling at night is not quite adequate to provide substantial pre-cooling, but where evaporative water cooling could provide very low energy cooling. In such a scenario, radiant cooling could be very advantageous – even if it has additional thermal burden – because it can benefit from waterside economizer cooling.

Careful design and control of radiant cooling is essential to minimize electricity consumption

These findings have compelling implications for the design and ultimate energy use of buildings. Although radiant cooling enables several efficiency opportunities – especially improved cooling plant and distribution efficiency – the large additional thermal burden revealed by our experiments could definitely cause a radiant cooling system to consume more electricity than an all-air system in the same application. Therefore, to save electricity compared to all-air systems, radiant cooling must be carefully designed and controlled to leverage the advantages offered by the strategy. Most importantly, cooling plants for buildings with radiant should be operated at a warmer chilled water temperature, and cooling plant operation should be strategically coordinated with the availability of passive or very-low-energy cooling opportunities – such as evaporative water cooling– so as not to preempt the benefits offered by such strategies.

Finally, while our results have confirmed that radiant cooling must remove more heat than an all-air system in order to maintain equivalent comfort conditions, we expect that – when designed with climate-appropriate systems and control strategies – radiant cooling should be able to use substantially less electricity than all-air cooling. Many building energy simulation studies have concluded that in a wide variety of climates, radiant cooling can achieve substantial primary energy savings and peak electrical demand reduction compared to all-air cooling systems.

4.5 CONCLUSIONS

Previous research, including Chapter 3, proved that to maintain equal operative temperature as an all-air system: (1) radiant cooling must extract heat from gains earlier; (2) the peak space heat extraction rate must be larger for radiant systems; and (3) the peak space heat extraction must occur earlier for radiant systems (Feng et al., 2013, 2014b; Niu et al., 1995, 1997; Woolley et al., 2018a). We conducted a series of laboratory tests to investigate whether or not the radiant-to-total heat gain ratio, or the use of natural ventilation night pre-cooling, significantly affect the magnitude of these differences.

First of all, our results reaffirm previous findings, and underline the fact that the time and rate at which heat must be extracted from a space depend on the type of terminal heat transfer device used. Furthermore, we conclude that the difference between the dynamic space heat extraction requirements (space cooling load) for radiant and all-air systems depends on characteristics of the scenario. The use of natural ventilation night pre-cooling, and the radiant-to-total heat gain ratio both have large impacts on the difference between dynamic space heat extraction requirements (space cooling load) for the two systems. In an experiment with mixed internal heat gains and without natural ventilation night pre-cooling radiant cooling had to remove 7% more heat than the all-air system, and the peak space heat extraction rate was 2–10% larger (median 5%). Whereas in experiments with highly radiant gains and natural ventilation night pre-cooling, radiant cooling had to remove 40% more heat than the all-air system and the peak space heat extraction rate was 18–28% larger (median 21%). Summary metrics for all six experiments are presented in Table 4.1.

The differences found are mainly due to the way that each system type influences the time and rate at which heat is absorbed by, stored in, and released from non-active thermal masses within and enclosing a space. Radiant cooling extracts heat directly from all surfaces in a space; consequently, non-active masses absorb and store less heat than they would in a space with all-air cooling. This causes radiant cooling systems to extract heat from gains earlier, causes the peak space heat extraction rate to be larger, and causes non-active surfaces to be cooler. Moreover, since non-active masses in spaces with radiant cooling absorb and store less heat, there is less opportunity to reject heat from non-active masses by passive means. In climates with significant opportunity for natural ventilation night pre-cooling the difference between space heat extraction requirements (space cooling load) for radiant and all-air systems can be very large.

Finally, we also present results which reveal that the thermal properties of non-active surfaces within and enclosing a space interact with heat gains in ways that impact the differences between dynamic space heat extraction requirements (space cooling load) for radiant and all-air systems. In particular, surfaces with low thermal diffusivity and low thermal capacity will essentially convert radiant heat gains to convective heat gains, whereas surfaces with high thermal diffusivity and high thermal capacity will more readily absorb and store radiant heat gains.

Current practice for design and sizing of radiant cooling systems often utilizes cooling load calculation tools designed for all-air cooling systems, and thus fails to account for the differences we have observed. In some scenarios, the differences may be small, but – as we have demonstrated – in some scenarios the differences can be very large. Therefore, we encourage designers and researchers to use building energy simulation tools that properly represent the dynamic heat transfer characteristics of radiant cooling systems, so as to institute design strategies that can more fully capitalize on the potential energy benefits of radiant cooling. Further, we encourage standards organizations to develop more inclusive explanations and guidelines for cooling load calculations.

CHAPTER 5

A critical review – and proposed redefinition – of the industry standard definition of "cooling and heating loads" and the associated system design sizing procedure, with focus on high thermal mass radiant cooling systems

Reproduced in part from the following previously published co-authored work:

> *J. Woolley, F. Bauman, C. Duarte, P. Raftery, J. Pantelic, Optimizing radiant systems for energy efficiency and comfort – Cooling load and design sizing report, 2019.*

Chapter Abstract

The standard procedure for sizing cooling (and heating) systems was conceived for overhead-mixing all-air systems. It is not well suited for design of many other cooling (and heating) system types – it is especially problematic for design of radiant cooling (and heating) systems. There are several reasons that the standard cooling (heating) system design procedure is flawed, including that the standard definition of "space cooling (heating) load" is too narrowly constrained and omits fundamental principles that are essential to operation of various cooling (heating) system types. In this chapter, we address several shortcomings with the standard definition of "space cooling (heating) load" – as stipulated by *ASHRAE Fundamentals 2017 Chapter 18: Nonresidential Cooling and Heating Load Calculations* – and with the standard system design procedure. We focus especially on how the design procedure is applied to high thermal mass radiant systems – as stipulated by *ASHRAE Systems & Equipment 2016 Chapter 6: Radiant Heating and Cooling*. Although we focus on high thermal mass radiant cooling systems, we also consider how these issues relate to a range of different system types and control strategies. More specifically, in this chapter we: (1) identify several fundamental flaws with standard notion of "space cooling (heating) load", (2) explain how this standard definition has influenced building energy modeling, system sizing, and operation in practice, (3) discuss why addressing these shortcomings is especially important to the design, operation, and performance of high thermal mass radiant cooling systems, (4) propose a new definition for "space cooling (heating) load" and an associated cooling (heating) system design procedure that better suits a variety of systems and control strategies, and (5) present building performance simulation results to demonstrate the consequences of the standard procedure – compared to our recommended procedure – for the design of a high thermal mass radiant cooling systems. In summary, our assessment reveals that the fundamental flaws with the standard definition of "space cooling (heating) load" and the associated system design procedure are so significant that when used for design of high thermal mass radiant systems, they can lead designers to underestimate the peak space cooling load by 100%, yet to select cooling plant equipment that is 100% larger than necessary. Additionally, the standard approach can lead designers to control systems in a way that consumes more thermal energy during high tariff periods and causes more discomfort during occupied periods. We use our proposed design procedure to develop several example designs that reduce cooling plant equipment size as much as 50%, reduce annual thermal energy use during high tariff periods by as much as 100%, and reduce annual occupied discomfort hours by as much as 55%.

5.1 INTRODUCTION

To design a cooling (heating) system and properly size all of its subcomponents, a designer typically begins by calculating the dynamic "space heat extraction (input) rate"[3] that would be required to maintain intended indoor thermal conditions for each conditioned space in a building during a design period. This is generally referred to as a "cooling (heating) load calculation" – a process that is defined authoritatively by *ASHRAE Fundamentals Chapter 18: Nonresidential Cooling and Heating Load Calculations* (2017a). This standard cooling (heating) load calculation produces a singular time series of the – "ideal" – space heat extraction (input) rates required for each space during the design period, which is subsequently used as the basis for system design and sizing.

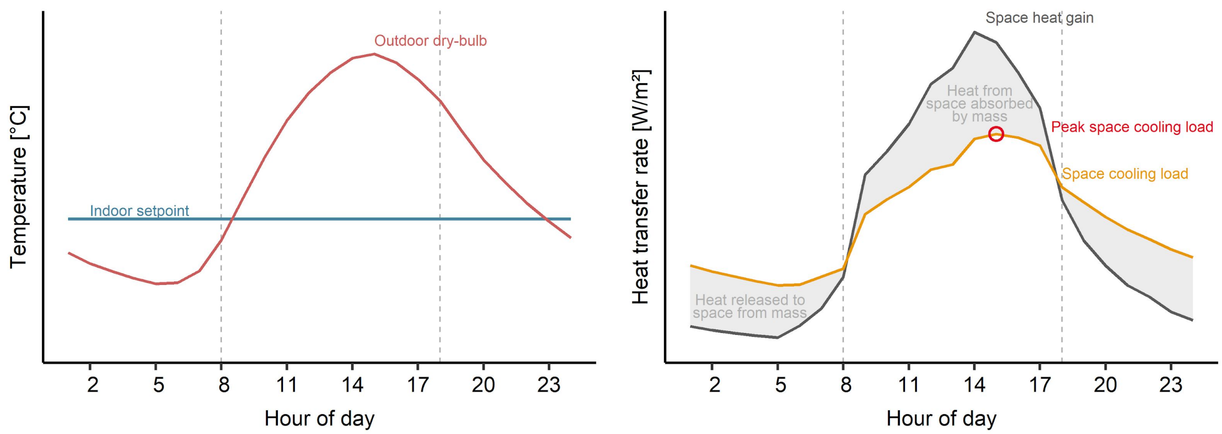


Figure 5.1: Conceptual illustration of the dynamic space cooling load for a cooling design day, and its relationship to outdoor temperature, indoor air temperature, and space heat gain rates. (Left): Outdoor temperature and indoor air temperature. (Right): Space heat extraction rate (space cooling load) and space heat gain rate (the sum of: internal heat gains, infiltration heat gain (loss), and solar heat gains). The red circle indicates the peak space cooling load used for sizing cooling systems. The gray hatched areas illustrate the rate at which space heat gains are absorbed by masses, and the rate at which heat stored in masses is released to the space. A portion of the space heat gains absorbed by masses may also be released to the environment (not indicated), and a portion of the heat released to the space from masses may have originated from the environment (not indicated)..

The rate at which heat must be extracted from (input to) a space is dynamic; it changes in time as the balance of heat gains to – and losses from – a space changes, and because surfaces within and enclosing a space absorb, store, and release heat dynamically. As a consequence of thermal storage, the space heat extraction rate required by a cooling system is generally smaller than the heat gain rate, yet it can persist after heat gains subside if heat stored in surfaces is released back to the space. Figure 5.1 illustrates the space heat extraction requirement (space cooling load) for a hypothetical design day, and its relationship to outdoor temperature, indoor air temperature, internal heat gain rates, solar heat gain rates, and net space heat gain (loss) rates.

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[3] The "space heat extraction rate" is the rate at which heat is removed from a space by terminal heat transfer devices. For all-air systems the "sensible space heat extraction rate" is the sensible enthalpy difference between supply and return (or room air outlet) air flows. For radiant systems the "sensible space heat extraction rate" is the sum of convective and radiant (longwave and shortwave) heat transfer rates at the indoor faces of the internally cooled surfaces.

Without bias toward any particular mathematical method, the standard cooling (heating) load calculation and associated system design procedure can be summarized as the following distinct steps:

1. **Space cooling (heating) load calculation**
    a. Define site location and meteorological information for a design period, including: (i) site latitude and longitude, (ii) outdoor air temperature (see Figure 5.1) and humidity, (iii) wind speed and direction, and (iv) direct and global horizontal solar irradiance.
    b. Define building characteristics, including: (i) building geometry, (ii) construction thermal characteristics, (iii) internal and external shading devices, and, (iv) envelope air tightness characteristics.
    c. Define internal heat gains for a design period of interest, including heat gains from (i) people, (ii) lighting, and (iii) equipment – this step should account for diversity in the timing and magnitude of internal heat gains in different spaces.
    d. Define other known heat gains (losses) to the space including: (i) infiltration, and (ii) direct to space ventilation.
    e. Perform calculations to determine: (i) solar heat gain rates and (ii) the distribution of solar heat gains.
    f. Define the constant indoor air temperature (see Figure 5.1) and humidity for the design period.
    g. Perform cooling (heating) load calculations to determine: (i) the rates of heat transfer and storage associated with boundary surfaces, (ii) the net heat gain to (loss from) the space, and (iii) the space heat extraction (input) rates required to maintain indoor environmental conditions during the design period – the space cooling (heating) load (see Figure 5.1).
2. **Design terminal heat transfer devices**
    a. Choose terminal heat transfer devices with steady-state cooling (heating) capacity to satisfy the peak space cooling (heating) load during the design period, at coincident operating conditions.
3. **Design cooling (heating) plant and distribution systems**
    a. Aggregate space cooling (heating) loads for all spaces to determine the rates at which heat must be transferred to (from) the distribution systems and cooling (heating) plants. This step must account for: (i) diversity in the timing and magnitude of cooling (heating) loads associated with different spaces, (ii) incidental dehumidification that occurs while generating required sensible heat transfer rates (or incidental sensible heat transfer that occurs while generated required dehumidification), and (iii) additional heat losses and gains to the system such as duct leakage, distribution losses, or ventilation that is cooled or heated by the system before it is supplied to the space.
    b. Select distribution systems and cooling (heating) plants with steady-state cooling (heating) capacity to match the peak aggregate cooling (heating) loads at coincident operating conditions.

Since the net heat gain to (loss from) a space changes from day-to-day, standards guide designers to size cooling (heating) systems based on the conditions in hypothetical "cooling (heating) design days". These design days are intended to represent the most extreme scenarios in which a cooling (heating) system should be expected to maintain the desired indoor conditions. *ASHRAE Fundamentals Chapter 14: Climatic Design Information* (2017a) is the standard reference for definition of climatic conditions on design days. As explained therein, it may be necessary to conduct load calculations for several different design days, to separately determine different operational extremes, such as: peak sensible cooling requirements and peak dehumidification requirements. Due mainly to seasonal variation in solar heat gains, different spaces within a building may require different design days, and the maximum aggregate load for the combined system may occur on yet a different design day. Following a cooling (heating) load calculation, standards guide designers to select and size terminal cooling devices that are capable of satisfying the maximum space cooling (heating) loads, then design and size distribution systems and cooling (heating) plant equipment that are capable of satisfying the maximum sum of coincident heat transfer rates from all associated terminal cooling (heating) devices. The "cooling (heating) capacity" for a system is usually not a constant value[4]; rather, it depends on coincident environmental conditions and system states. For example, the cooling capacity of an air-cooled chiller depends on the outdoor air temperature, the chiller entering water temperature, the refrigeration circuit part-capacity state – and to some extent – the water flow rate. Standards guide designers to account for the fact that the cooling (heating) capacity of a system changes with operating conditions. To do so, designers typically use models and system performance data that quantify cooling (heating) capacity as the steady-state heat extraction (input) rate that could be produced by continuous operation with steady conditions.

For most cooling (heating) system components, it is reasonable to assume that steady-state cooling (heating) capacity data is representative of the actual cooling (heating) rates that a system will produce in dynamic circumstances. The assumption is reasonable because the time required for most cooling (heating) system components to approach steady-state following a change to the inputs – the "response time" – is small relative to the time required for either the space cooling (heating) load or environmental conditions to change. Therefore, sizing a cooling (heating) system generally entails specifying a cooling (heating) plant, distribution system, and terminal heat transfer device(s) each with the steady-state cooling (heating) capacity to match the peak aggregate space cooling (heating) load at coincident conditions. This sizing process must account for: diversity in the timing and magnitude of cooling (heating) loads from different spaces, (ii) incidental dehumidification to generate sensible space cooling (or incidental sensible heat transfer to generate dehumidification), and (iii) additional heat losses and gains to the system such as duct leakage, distribution losses, or ventilation that is cooled or heated by the system before it is supplied to the space – so-called "system heating and cooling load effects" (ASHRAE, 2017a).

[4] Some systems, such as resistance heaters and gas furnaces may have a nearly constant heating capacity.

Although the standard approach for calculating space cooling (heating) load, and the associated system design procedure have been applied successfully for the design of many buildings, our research has revealed that because they were conceived for overhead-mixing all-air systems, they impose several assumptions and constraints that limit them from accurately representing all cooling (heating) systems types and applications. More specifically:

- The standard definition of "space cooling (heating) load":
  a. does not fully reflect all aspects of standards that address thermal comfort;
  b. only facilitates design of systems for basic applications;
  c. does not account for the way that cooling (heating) system type impacts the space cooling (heating) requirements;
  d. does not account for the way that system control strategies impact space cooling (heating) requirements, and;
  e. does not provide sufficient guidance on the selection of design periods
- The standard system design procedure:
  a. does not facilitate design for any performance metric other than indoor air temperature;
  b. is based solely on steady-state heat transfer and so is inaccurate for systems with long response time.

In this chapter, we explain why the standard definition of "space cooling (heating) load" is unsatisfactory, we propose a new definition for the concept, and we present simulation results to illustrate the practical impact of our proposed definition. In Section 5.2 we identify the specific shortcomings with the standard definition of "space cooling (heating) load", and explain how this notion has influenced building energy simulation tools and system sizing in practice. We consider how these issues relate to various system types, but we focus especially on why they are problematic for design and control of high thermal mass radiant cooling (heating) systems. In Section 5.3 we propose a new definition for "space cooling (heating) load", and a new system design procedure. In addition to the summary explanations in Section 5.3, the proposed redefinition is composed as a comprehensive revision to the explanatory sections of *ASHRAE Fundamentals Chapter 18* (2017a), which is included in Appendix A. Finally, in Section 5.4 we present simulation results to demonstrate the consequences of the standard procedure – compared to our recommended procedure – for the design of high thermal mass radiant cooling systems.

5.2 SHORTCOMINGS WITH THE STANDARD DEFINITION OF "COOLING (HEATING) LOAD" AND THE ASSOCIATED SYSTEM DESIGN PROCEDURE

5.2.1 The standard definition of "space cooling (heating) load" does not fully reflect all aspects of standards that address thermal comfort

ASHRAE Fundamentals Chapter 18 (2017a) defines "space cooling (heating) load" as the space heat extraction (input) rate that would be required "to maintain a constant space air temperature and humidity". There are two distinct problems with this definition as it relates to thermal comfort. First, it focuses *only* on indoor air temperature and humidity, which ignores the fact that many variables influence thermal comfort. Second, it imposes *constant* indoor air temperature and humidity, which is neither necessary nor realistic – especially for certain system types.

The standard definition of “space cooling (heating) load” implicitly discourages design based on prevailing standards that address acceptable thermal environmental conditions for human occupancy. *ASHRAE Standard 55* (2017b) and *EN 16798* (CEN, 2019) present comprehensive metrics to estimate thermal comfort based on many variables including: indoor air temperature, indoor mean radiant temperature, indoor humidity, indoor air speed, metabolic rate, clothing, and outdoor air temperature. These standards do not require constant indoor environmental conditions; rather, they describe a range of acceptable conditions during occupied periods and allow for substantial variation over time.

Moreover, the standard definition of “space cooling (heating) load” excludes some systems and control strategies that do not maintain a constant air temperature, yet still maintain a thermally comfortable indoor environment. For example, high thermal mass radiant cooling (heating) systems are really not capable of maintaining constant indoor air temperature. The response time (Ning et al., 2017) of these systems precludes them from using automated feedback control to instantaneously adjust the space heat extraction (input) rate in a way that is required to maintain constant indoor air temperature. Manual commissioning, or adaptive controls (Raftery et al., 2017) can select dynamic temperature setpoints and operating schedules for high thermal mass radiant systems that – together with their self-regulating characteristics – will consistently produce a comfortable indoor thermal environment; however indoor air temperature cannot be expected to remain constant.

Although designers familiar with radiant cooling (heating) understand that high thermal mass radiant systems behave differently than overhead-mixing all-air systems, these systems are often sized according to standard space cooling (heating) load calculation methods, and controlled with constant air temperature setpoints. This practice was revealed by Feng et al. (Feng et al., 2014a) who investigated what methods and tools designers use to size radiant cooling (heating) systems, then by Paliaga et al. (Paliaga et al., 2017, 2018), and Raftery et al. (Raftery, Duarte, & Dawe, 2019a, 2019b) who studied the design and control strategies commonly implemented in practice for high thermal mass radiant cooling (heating) systems. Unfortunately, there are various problems with sizing and operating high thermal mass radiant cooling (heating) systems in this way. First, the constant indoor air temperature constraint assumes an impossible behavior for high thermal mass radiant cooling (heating) systems, and produces an unrealistic estimate of the space heat extraction (input) rate (space cooling (heating) load) that the system will actually produce. Second – as discussed in Section 5.2.3 – standard space cooling (heating) load calculation methods do not properly account for the heat transfer pathways associated with space heat extraction (input) by radiant cooling (heating) systems. In Section 5.4, we present simulation results to demonstrate the practical impact of these combined problems.

Furthermore, the constant indoor air temperature constraint is especially peculiar because constant indoor air temperature does not necessarily indicate constant thermal comfort for any system type. For a typical overhead-mixing all-air system, the temperatures of surfaces within and enclosing a space change as they absorb and release heat, which causes operative temperature to change throughout the day. For example, in a space with substantial solar gains, constant air temperature may not adequately counteract the comfort impacts of insolation and increased surface temperatures (Arens et al., 2015).

The standard definition of "space cooling (heating) load" should not be constrained to a constant indoor air temperature. Instead, it ought to allow for dynamic thermal environments – within the limits established by consensus standards on the subject of thermal comfort. This would offer designers flexibility, and facilitate system design to reduce energy consumption, reduce equipment cost, and improve thermal comfort. For example, designers can substantially reduce energy use and equipment size if indoor temperature is allowed to drift somewhat over the course of the day (Hoyt et al., 2015; Schiavon, Melikov, et al., 2010; Schiavon & Melikov, 2008). The impact is more substantial if increased air motion, or personal comfort systems, are used to extend the range of acceptable indoor temperature (Hoyt et al., 2015; Huang et al., 2013; Lipczynska et al., 2018; Schiavon et al., 2017; Stefano Schiavon & Melikov, 2008; Sekhar, 1995).

5.2.2 The standard definition of "space cooling (heating) load" only facilitates design of systems for basic applications

ASHRAE Fundamentals Chapter 18 (2017a) presents a narrow conception of "cooling (heating) load" that is based solely on the design of overhead-mixing all-air systems for comfort conditioning. This standard definition overlooks how the notion of "cooling (heating) load" relates to design of heating, cooling, and ventilation systems for other objectives or applications. The following paragraphs give examples that demand a broader definition of the concept.

First, the design of cooling systems for data centers requires a different approach than what is represented in *ASHRAE Fundamentals Chapter 18* (2017a). For this application cooling systems must be sized to maintain an appropriate air temperature and flow rate at the inlet for computer equipment, and the relationship between air distribution and heat gains in such a space can have a dramatic impact on the amount of cooling that a mechanical system must generate. For example, if heat from computer equipment is mixed into the space – as described by *ASHRAE Fundamentals Chapter 18* (2017a) – system cooling requirements are much larger than if heat from computer equipment is captured and exhausted or recirculated to cooling equipment and handled directly.

Second, the design of heating, cooling, and ventilation systems may have multiple objectives that compete and interact in complex ways that are not captured by standard cooling (heating) load calculations. For example, a dehumidification system designed for a particular "latent space cooling load" can impact sensible space heat extraction requirements in a way that is not accounted for by a simple cooling load calculation presented in *ASHRAE Fundamentals Chapter 18* (2017a). The standard approach recommends that designers calculate sensible loads and latent loads separately, when in reality sensible and latent heat transfer rates are interrelated.

Additionally, the standard definition of "space cooling (heating) load" cannot accommodate the design of natural ventilation systems because it presumes a constant indoor air temperature, and a predetermined constant ventilation rate. In reality, natural ventilation systems must simultaneously satisfy several distinct objectives including: energy performance goals, and constraints on indoor temperature, indoor humidity, indoor air speed, and indoor air quality. These objectives interact and may compete with one another, making it impossible to maintain constant indoor temperature and humidity conditions, and impractical for a designer to specify the exact indoor thermal conditions that will occur.

Finally, the design of cooling (heating) systems for thermal comfort in outdoor spaces – such as stadiums or patio restaurants – completely eludes the standard definition of "cooling (heating) load". Yet, these systems merit a standard basis for system sizing with the same foundations that justify the design of any cooling (heating) system. These systems have different performance expectations, and operate with different heat transfer mechanisms than what is represented in *ASHRAE Fundamentals Chapter 18* (2017a).

5.2.3 The standard definition of "space cooling (heating) load" does not account for the way that cooling (heating) system type impacts space cooling (heating) requirements

ASHRAE Fundamentals Chapter 18 (2017a) recommends two different mathematical methods to calculate the space cooling (heating) load: a Heat Balance (HB) method, and the Radiant Time Series (RTS) method. However, neither of these methods – as currently implemented – fully account for the heat transfer pathways associated with space heat extraction (input) by various types of terminal heat transfer devices. Most importantly, both methods assume that convection with a well-mixed air volume is the only heat transfer mechanism by which heat can be removed from a space. Consequently, these methods do not properly estimate the space heat extraction rates by radiant cooling systems (Feng et al., 2013, 2014b; Niu et al., 1995, 1997; Novoselac et al., 2017; Woolley et al., 2018a, 2019), by underfloor air distribution systems (Lee et al., 2012; Schiavon et al., 2011; Schiavon, Lee, et al., 2010), or by displacement ventilation systems (Zhang, Cheng, et al., 2019; Zhang, Lin, et al., 2019). In addition to limitations with the mathematical methods, the definitions and explanations presented in *ASHRAE Fundamentals Chapter 18* (2017a) systemically fail to consider the implications of space heat extraction by any mechanisms other than convection with a well-mixed air volume. These issues are important because the presence of other heat transfer pathways – especially radiant exchange with the terminal heat transfer device – disrupts the network of heat transfer and storage, and impacts the time and rate at which heat must be extracted from a space.

Computational and experimental research (including Chapter 3 and Chapter 4) has proven that to maintain equal operative temperature as an overhead-mixing all-air system, a radiant cooling system must remove more heat overall, the peak space heat extraction rate must be larger, and it must occur earlier (Feng et al., 2013, 2014b; Niu et al., 1995, 1997; Novoselac et al., 2017; Woolley et al., 2018a, 2019). The differences are mainly due to the way that heat gains are absorbed by, stored in, and released from non-active masses – a process described thoroughly in Chapter 3 and Chapter 4 (Woolley et al., 2018a, 2019).

The magnitude of these differences depends on many factors. As documented by Woolley et al. (2019) (Chapter 4) and by Feng et al. (2013), the differences are larger for cases with highly radiant heat gains, and larger in scenarios that benefit from passive cooling of the thermal mass in a building. For some scenarios radiant cooling may have 25% larger peak space cooling load, and the cumulative daily space cooling load may be 40% larger. Since the ASHRAE conception of "space cooling (heating) load" does not account for heat extraction (input) by radiant heat transfer, it does not account for these differences in the heat gain (loss), and the space cooling (heat) load.

Similarly, Schiavon et al. (2011; 2010) and Lee et al. (2012) found that the space cooling load for underfloor air distribution systems was generally 19% higher than traditional overhead mixing systems. Researchers attributed these differences mainly to the fact that the raised floor in an underfloor air distribution system changes the interaction between heat gains and thermal storage from what normally happens in the absence of the raised floor.

The standard definition of "space cooling (heating) load" should account for all possible methods of heat transfer for space heat extraction (input) so that the calculations can properly assess the space heat extraction (input) requirements for different types of terminal heat transfer devices.

Researchers have developed and validated numerical methods that properly estimate the fundamental heat transfer mechanisms involved with high thermal mass radiant cooling (heating) systems (Chantrasrisalai et al., 2003; Fort, 1989, 2001; Laouadi, 2004; J. Niu et al., 1995, 1997; Stetiu et al., 1995; Strand et al., 1999; Strand & Baumgartner, 2005; Strand & Pedersen, 2002; Yu et al., 2014). Even though the most prominent numerical methods are direct descendents of the Heat Balance method presented in *ASHRAE Fundamentals Chapter 18* (2017a), the standard method has not been updated accordingly. The ASHRAE Heat Balance method was initially developed by Kusuda (1974) and implemented in the BLAST and TARP energy analysis programs (Walton, 1983). Later, the method was described in full by Liesen and Pedersen (1998), McClellan and Pedersen et al. (1997), and Pedersen et al. (Pedersen et al., 1997) as part of *ASHRAE RP-875*. Shortly afterward, Strand et al. (1999), Strand and Pedersen (2002), and Strand and Baumgartner (2005) extended the Heat Balance method to consider the heat transfer dynamics of high thermal mass radiant systems, and developed the requisite features to incorporate the methods into EnergyPlus (2020). Feng et al. (2014b) conducted laboratory experiments which validated the predictions from the more comprehensive Heat Balance method developed by Strand et al. (1999); and simultaneously, proved that the Radiant Time Series method and the ASHRAE Heat Balance method do not accurately predict the space heat extraction rates for radiant systems.

Although the methods developed by Strand et al. (1999; 2005; 2002) have been incorporated into some building energy simulation software, the problematic assumption perpetuated by *ASHRAE Fundamentals Chapter 18* (2017a) – that all space heat extraction occurs by convection – still persists in some aspects. For each simulation timestep EnergyPlus (2020) uses the numerical methods developed by Strand et al. (1999; 2005; 2002) to calculate the rate at which internally cooled (heated) surfaces extract (input) heat from (to) a space. However, the space cooling (heating) load calculations performed to estimate space heat extraction (input) requirements on a design day, and to autosize components of a radiant system and cooling plant still rely on the standard definition of "space cooling (heating) load". Specifically, the EnergyPlus (2020) radiant system autosizing subroutine sets the design water flow rate to achieve a hydronic heat extraction (input) rate that matches the space heat extraction (input) rate determined from a standard cooling (heating) load calculation (*EnergyPlus*, 2020). In addition to ignoring space heat extraction (input) by radiation in the initial load calculation, this approach fails to consider whether or not this hydronic heat extraction (input) rate will generate commensurate space heat extraction (input) rate. Moreover, several widely-used building energy simulation tools have not addressed the problematic assumption in any way, yet researchers and practitioners often use these tools for design and simulation of radiant cooling and heating systems (J. Feng et al., 2014a). In Section 5.4, we present simulation results to demonstrate the practical consequences of using the standard method for cooling load calculations to design high thermal mass radiant systems.

ASHRAE Fundamentals Chapter 18 (2017a) should be extensively revised to account for these issues, and the mathematical methods presented therein should be updated. However, the problem is more extensive than updates to this chapter. Although *ASHRAE Systems & Equipment Chapter 6: Radiant Heating and Cooling* (2016a) clearly explains that radiant cooling transfers heat by convection and radiation, it does not recognize that the magnitude and timing of the required space heat extraction rate (space cooling load) is fundamentally different from that of overhead-mixing all-air systems. Moreover, the system design procedure presented in *ASHRAE Systems & Equipment Chapter 6* (2016a) specifically references the methods in *ASHRAE Fundamentals Chapter 18* (2017a), even though these methods do not account for the effects of space heat extraction by radiation with internally cooled surfaces.

In the widely referenced guidebook *Low Temperature Heating and High Temperature Cooling*, Babiak et al. (2009) and ISO 11855-2 (2012) thoroughly explain the combined radiant and convective heat transfer rates that a radiant system can be expected to produce for different steady-state conditions (space cooling capacity). However, these references do not explain how to determine the dynamic space heat extraction (input) requirement (space cooling (heating) load) for a high thermal mass radiant system, do not specifically recognize that it can differ substantially from that of overhead-mixing all-air systems, do not explain how steady-state calculations ought to be used within a whole system design procedure, and do not offer any approach to adjust steady-state calculations for dynamic conditions. The guidebook does indicate that for TABS systems "dynamic simulations can be required to predict the thermal comfort in a conditioned zone", and ISO 11855-4 (2012) provides methods to perform such dynamic simulations, but neither reference frames the need for dynamic simulations in relation to the standard concept of "space cooling (heating) load".

Among standards focused on the topic of space cooling (heating) loads, *ISO 52016* (2017) – which supersedes *prEN 15255* (CEN, 2007b) – is the only resource we are aware of to explicitly state that the dynamic space heat extraction (input) requirements (space cooling (heating) load) depends on the system type. In an equation for determining the space heat balance, the standard introduces a variable called the "convective fraction of the cooling (heating) system". However, the standard currently provides no guidance on how to determine this fraction for different systems and circumstances.

5.2.4 The standard definition of "cooling (heating) load" does not account for the way that system control strategies impact space cooling (heating) requirements

ASHRAE Fundamentals Chapter 18 (2017a) defines "space cooling (heating) load" as the space heat extraction (input) rate that would be required "to maintain a constant space air temperature and humidity", and thus entirely overlooks the fact that the sequence of operations – including temperature setpoint schedules – will impact the space heat extraction (input) requirement. For example, scheduling a setback during unoccupied periods in the cooling season could cause a larger peak space cooling load than continuous setpoints, because surfaces will begin each day at a higher temperature, and so will store a smaller portion of the heat gains during the day. For the same reasons, a pre-cooling setpoint schedule would reduce the peak space cooling load (Braun, 2003; Henze et al., 2007; Keeney & Braun, 1997). Many designers account for the fact that the size for a heating system can be driven by morning warm up requirements following a setback period, yet this consideration is not accommodated by the standard definition.

This aspect of the standard "space cooling (heating) load" definition is especially problematic for high thermal mass radiant systems, for which the sequence of operations, choice of control feedback variable, temperature setpoint, the supply water temperature, and system availability schedule will substantially change the shape and magnitude of the space cooling (heating) load. It is also problematic for spaces that rely on multiple cooling (heating) systems. For example, the choice of control sequence to coordinate radiant cooling and supplemental air cooling will have a major impact on the space cooling load for each system. To this point, Chung et al. (2017)used building energy simulations to demonstrate that design and control of high thermal mass radiant systems, and supplemental air cooling systems substantially impacts the cooling load for each system. Similarly, the choice of control sequence in mixed-mode buildings that utilize natural ventilation for pre-cooling will have a major impact on the cooling load for mechanical systems. The research presented in Chapter 4 and Chapter 5 (Woolley et al., 2018a, 2019) clearly revealed that because radiant cooling extracts a significant amount of heat from non-active thermal masses, it can preempt some of the benefits of natural ventilation pre-cooling. For such a mixed-mode building, the sequence of operations would have a major impact on the space cooling load, as well as the cumulative mechanical cooling load.

The definitions, explanations, and mathematical methods in *ASHRAE Fundamentals Chapter 18* (2017a) effectively discourage design of systems that use strategic control strategies to shift electrical demand, reduce energy consumption, reduce equipment cost, and improve thermal comfort. As the expectations for dynamic control of electrical demand from buildings continue to accelerate, the dynamic control of cooling (heating) systems requires a more sophisticated approach for system design and sizing than what is promulgated by the standard definition of "space cooling (heating) load".

5.2.5 The standard definition of "space cooling (heating) load" does not provide sufficient guidance on the selection of design periods

ASHRAE Fundamentals Chapter 18 (2017a) defines the notion of cooling load and explains that cooling load calculations should be used as the basis for system design decisions. However it provides relatively little information about what constitutes an appropriate design period. *ASHRAE Fundamentals Chapter 14* (2017a) provides extensive climate summary statistics for locations around the world, and defines a standard method to use these summary statistics to generate a 24 hour times series for outdoor dry-bulb and wet-bulb temperatures for a design day. However although the chapter indicates that a designer should "use judgement" in choosing appropriate design conditions, it provides relatively little guidance about what constitutes an appropriate design period. In some cases, the largest heat gains to a space might occur during the spring or fall when solar gains are largest. A designer may choose to perform load calculations for various design days to find the constraining design condition, but this requires special effort, and is generally overlooked by the common practice use of a "cooling design day". For example, weather data for EnergyPlus and other modeling tools include a set of "cooling (heating) design days" developed from the methods in *ASHRAE Fundamentals Chapter 14* (2017a); but these are selected on the basis of outdoor dry-bulb and wet-bulb temperatures, so may not result in appropriate design conditions when maximum space cooling loads are actually driven by other factors. An iterative simulation on these design days is typically used to autosize equipment, even if there are ultimately other days in the year with large heat gains.

Additionally, as the results of this chapter indicate, the typical design day approach may not provide a sufficient basis for design of certain systems because a simulation on a single 24 hour period:

1. Requires convergence of iterative simulations, which can result in initial thermal conditions for masses that do not represent worst case scenarios.
2. Fail to capture the effect of multi-day transient oscillations caused by large thermal mass.
3. May not be adequate to assess the impact of system controls.

For example – in regard to system controls – Raftery et al. (2017) developed a control sequence for high thermal mass radiant systems that resets the slab temperature setpoint each day in response to indoor air temperature performance on the previous day. The thermal behavior that results from such a control sequence cannot be captured by a 24 hour design day simulation. An even simpler example of this issue is the impact that weekend setbacks have on temperature of thermal masses in a building. Design periods really ought to capture these effects, since they can significantly influence the heat extraction (input) requirements.

Additional research is necessary to quantify the impact of these issues and to recommend improvements to standards and current modeling practices.

5.2.6 The standard design procedure does not facilitate design for any performance metric other than indoor air temperature.

System design is inherently a type of multi-objective optimization. Designers are expected to make decisions based on numerous factors including: life cycle cost, greenhouse gas emissions, comfort, or indoor air quality. Unfortunately, since the standard design procedure expects that the standard space cooling (heating) load must be satisfied, it does not allow other objectives to enter into the system design process, except where design alternatives can still satisfy the standard space cooling load.

For example, if a designer wanted to develop a system design that simultaneously minimizes greenhouse gas emissions and capital costs by incorporating natural ventilation for pre-cooling, the standard design procedure would be incapable of determining the actual space cooling loads, and appropriate equipment sizing.

5.2.7 The standard design procedure is not satisfactory for systems with long response time

ASHRAE Systems & Equipment Chapter 6 (2016a) provides a step-by-step procedure to guide the design of radiant cooling (heating) systems. In general, this process directs engineers to calculate the "peak space cooling (heating) load" (determined according to *ASHRAE Fundamentals Chapter 18* (2017a)) then to design a radiant system with steady-state "space cooling (heating) capacity" to match the peak space cooling (heating) load. Apart from the limitations described previously, this design procedure is unsatisfactory because in practice high thermal mass radiant cooling (heating) systems do not operate at steady-state, so do not generate the space cooling (heating) capacity predicted by steady-state characterizations of performance.

High thermal mass radiant systems do not operate at steady-state because the thermal resistance and thermal capacitance of the internally cooled (heated) construction elements introduces a time delay between heat flux with the hydronic circuit, and heat flux with the space. As a result, the space heat extraction (input) rate differs considerably from the hydronic heat extraction (input) rate. Ning et al. (2017) evaluated the dynamic response for different types of radiant systems. The researchers conducted simulations which revealed that the current conceptual classification for radiant cooling (heating) system types (Babiak et al., 2009; ISO, 2012) does not provide adequate differentiation in regard to their dynamic behavior. Consequently, they developed a new classification scheme for radiant systems that quantified the delay for heat flux across an internally cooled (heated) surface as a "response time". This is the time it would take following a step change in the controlled inputs (supply water temperature or flow rate) for the temperature at the indoor face of an internally cooled (heated) surface in a space with constant heat gains to change by 95% of the difference between its initial and final values. Ning et al. (2017) showed that the response time is quick (<10 min) for radiant ceiling panels, medium (1–9 hours) for embedded surface radiant systems, and long (9–19 hours) for thermally active building systems For terminal cooling (heating) devices with a quick response time, it is reasonable to assume that the instantaneous space heat extraction (input) rate will be equal to the instantaneous hydronic heat extraction (input) rate and that the device will generate space heat extraction (input) rate that agrees with its steady-state cooling (heating) capacity at coincident conditions. However, this assumption is not reasonable for terminal cooling (heating) devices with a medium to long response time.

Although the space heat extraction rate is slow to change in response to a change in controlled inputs, it also changes rapidly in response to a change in heat gains without the need for an active change in controlled inputs. This phenomenon is often referred to as "self control" of high thermal mass radiant systems.

As a consequence of these two behaviors, in some conditions the instantaneous space heat extraction rate for a high thermal mass radiant cooling system may be smaller than predicted by steady-state assumptions, and in other conditions it may be larger. The difference depends on the initial thermal conditions of the internally cooled (heated) surface, slab, and the way conditions change at the boundaries of the surface. For example, when chilled water begins to flow through an internally cooled surface, it may take an hour or more before the space heat extraction rate begins to respond, yet such a surface can continue to extract heat from a space long after chilled water flow ends, and the space heat extraction rate will change naturally in response to changes in heat gains. We are not aware of any research that has quantified the range of response times for which steady-state capacity calculations may be acceptable, but it is clear that they are inaccurate for many radiant system configurations.

The standard design procedure for radiant cooling (heating) effectively directs engineers to assume that there is no delay or attenuation between hydronic heat extraction (input) rate and space heat extraction (input) rate. As a result, many designers often size hydronic systems, pumps, and cooling (heating) plants for high thermal mass radiant systems to handle the peak space heat extraction (input) rate predicted by standard cooling (heating) load calculation (Feng et al., 2014a; Paliaga et al., 2017, 2018), even though the space heat extraction (input) rate differs substantially from hydronic heat extraction rate. Research by Feng et al. (2014a) and Paliaga et al. (2017, 2018) has revealed that some designers use detailed numerical models to predict the dynamic performance of high thermal mass radiant systems and to size equipment, but that most do not. Researchers, manufacturers, and standards have advanced some simplified design methods that account for dynamic heat transfer behavior for high thermal mass radiant systems (Babiak et al., 2009; Koschenz & Lehmann, 2003; Olesen & Zöllner, 2007; Raftery, Duarte, Schiavon, et al., 2019; Uponor, 2013), but our research suggests that in practice, most designers do not explicitly account for these issues and instead size systems using steady state capacity estimates. In Section 5.4, we present simulation results to demonstrate the practical consequences of using the standard method for cooling load calculations and associated system design procedure to size and operate high thermal mass radiant systems.

5.3 PROPOSED REDEFINITION OF COOLING (HEATING) LOAD AND SYSTEM DESIGN PROCEDURE

To address the shortcomings described, we propose a comprehensive redefinition of “cooling (heating) load” and the associated system design procedure. Our approach expands the notion of “cooling (heating) load” so that it can accommodate the design of a variety of system types, control strategies, and performance objectives. Most significantly, our redefinition eliminates the idea of a singular – “ideal” – space cooling (heating) load as the objective for system design. Instead, we orient the system design procedure toward selecting and sizing components and their controls that best satisfy performance objectives such as thermal comfort, indoor air quality, resilience, grid-interactive responses, or energy cost minimization. Our approach requires that designers utilize modeling tools capable of accurately representing the systems and controls they design.

The problematic notion of “space cooling (heating) load” and the standard system design procedure is invoked by many standards and design guidelines, but it is defined authoritatively by *ASHRAE Fundamentals Chapter 18* (2017a). The shortcomings we’ve explained are pervasively entwined into almost every aspect of that chapter – from explanation about what input data is required for a load calculation, to definition about terms such as “heat gain”. Therefore, we have prepared an exhaustive revision to all definitions and explanatory sections within *ASHRAE Fundamentals Chapter 18* (2017a). The proposed revisions are included as Appendix A.

At the core of our amendments is a revision to the central problematic term:

Current definition:

> **Space cooling load** – the rate at which sensible and latent heat must be removed from the space to maintain a constant space air temperature and constant humidity in the space.

Revised definition:

> **Space cooling (heating) load** – the space cooling (heating) load at any point in time is the rate at which terminal heat transfer devices, with associated control sequences, must extract (input) sensible and/or latent heat such that associated thermal environmental conditions, and/or other performance metrics, comply with desired constraints during a design period (e.g.: limits on: operative temperature, peak electrical demand, etc).

Our redefinition does not require that a design be based on maintaining a constant indoor air temperature; in fact it doesn’t even require ex ante specification of the exact indoor thermal conditions that will occur. Instead, our redefinition allows designers to specify constraints on any ex post performance metric including: thermal comfort, indoor air quality, electrical demand, etc, . For example, a designer could specify an allowable range for operative temperature, or for the rate of change in operative temperature; then they would reject design alternatives for which simulation results do not comply with these constraints. In our view, standards should be impartial about what mathematical methods or tools are used for system models and simulations. Some designers may use numerical building energy simulation tools, while others may use simple diagrammatic design guidelines. Any type of "model and simulation" should be acceptable – as long as it represents the systems and controls it is used to design, with accuracy that is appropriate for the design phase in which it is utilized. For example, Koschenz and Lehmann (2003) and Raftery et al. (2019) have both developed simplified design tools that account for dynamic heat transfer behaviors for high thermal mass radiant systems..

Our proposed system design procedure includes the following steps. The first three steps align with standard system design procedure, but the rest differ:

1. **Describe all building and site characteristics and uncontrolled input values for a design period(s)**
    a. Define site location and meteorological information for a design period(s), including: (i) site latitude and longitude, (ii) outdoor air temperature (see Figure 5.1) and humidity, (iii) wind speed and direction, (iv) direct and global horizontal solar irradiance.
    b. Define building characteristics, including: (i) building geometry, (ii) construction thermal characteristics, (iii) internal and external shading devices, and (iv) envelope air tightness characteristics.
    c. Define site characteristics that impact building heat transfer, including: (i) shading by external objects (e.g.: trees and buildings), and (ii) reflection from external surfaces (e.g.: adjacent buildings, ground, water bodies)..
    d. Define internal heat gains for a design period(s) of interest (see Figure 5.1), including heat gains from: (i) people, (ii) lighting, and (iii) equipment – this step must account for diversity in the timing and magnitude of internal heat gains in different spaces.
    e. Define other known heat gains (losses) to the space including: (i) infiltration, (ii) direct to space ventilation.
2. **Describe performance objectives and constraints for the design period(s)**
    a. Define performance priorities for the design, which may include balancing multiple objectives such as achieving acceptable: (i) life cycle cost, (ii) le energy cost, and/or (iii) life cycle greenhouse gas emissions.
    b. Define constraints on performance metrics, which may include: (i) allowable range for air or operative temperature, (ii) allowable range for predicted mean vote (PMV), (iii) minimum required ventilation during occupied periods, (iv) maximum pollutant concentrations during occupied periods, (v) maximum peak electrical demand.
3. **Describe system design variables, controlled input variables, and control strategy**
    a. Define all terminal heat transfer devices, which may include: (i) sensible cooling (heating) devices, (ii) dehumidification devices, and (iii) sources of direct-to-space ventilation (including natural ventilation systems).
    b. Define all other factors and components associated with a space that may be controlled to influence performance (such as thermal comfort, or indoor air

quality) which may include: (i) ceiling fans, (ii) personal comfort systems, (iii) sources of ventilation, (iv) air cleaning devices, (v) occupant adaptive behaviors.

   c. Define all cooling (heating) systems within the scope of design, which may include: (i) distribution systems, (ii) air handlers, and (iii) cooling (heating) plants.
   d. Define all heat gains and losses from the cooling (heating) system, which may include: (i) duct leakage, (ii) fan heat, (iii) distribution losses.
   e. Define a sequence of operations for all controlled devices in a system (and occupant behaviors), which may include: (i) system operating schedules, (ii) feedback control loops and controlled variables, (iii) temperature setpoint schedules, and (iv) adaptive occupant responses.

4. **Simulation and design iteration**
   a. Perform simulation[5] of the building and systems model for the design period(s), and output values for any metrics needed to assess performance of the systems designed. This requires that designers utilize modeling tools capable of predicting these performance metrics, for the systems and controls to be designed, with accuracy that is appropriate for the scope and phase of design.
   b. Compare simulation results to the performance constraints (defined in step 2.b).
   c. Iterate on design definition and simulation (steps 3–4) so as to best achieve desired performance objectives (defined in step 2.a) subject to performance constraints (defined in step 2.b). Reject design alternatives that do not satisfy performance constraints, and choose among satisfactory design alternatives to best satisfy performance objectives.

Foremostly, it is important to recognize that our system design procedure does not result in a singular – "ideal" – space cooling (heating) load. Instead, it recognizes that there may be various system designs and control strategies that satisfy performance objectives and constraints – and each may have different cooling (heating) loads. Our procedure is generalized and intended to apply to any system type in a fundamental way. For design of many systems, it would likely be sufficient and expedient to abbreviate our procedure using common assumptions. However, such simplifying assumptions should not be expected to apply to design of all system types. With this in view, the standard definition of "space cooling (heating) load" is a special case that is permitted by our expanded definition. For example, if desired, a designer could use our procedure to design a cooling (heating) system with the simplifying assumptions that: (a) all space heat extraction (input) occurs by convection with the well-mixed air volume within a space, and (b) controls adjust space heat extraction (input) rates to maintain constant indoor air temperature. The resulting space heat extraction (input) rates would correspond exactly to the standard definition of "space cooling (heating) load".

[5] Note that this design procedure assumes that calculation of solar heat gains occurs as an integral part of the simulation step. In some cases, solar heat gains can be defined or calculated prior to simulation, but in other cases, solar heat gains can be impacted by controls and behavior – consider automated daylighting controls using blinds and dynamic electrochromic glazing, so can only be determined through simulation. If necessary, other parameters typically thought of as uncontrolled input variables (defined in step 1), could instead be determined as an integral part of the simulation step. For example, design of systems for demand response might require dynamic control of internal heat gains from lights and equipment.

In practice, our process may be repeated for different design periods, or different performance objectives to support final design decisions. Since the process does not result in a singular – "ideal" – space cooling (heating) load, it requires that designers develop and test various design alternatives, then choose between those that satisfy performance constraints.

Designers must also be careful to select design periods that are appropriate to assess whether or not a design will satisfy performance objectives and constraints in the course of operation. Simple system sizing could be based on a single "design day", and rule-of-thumb factors of safety, but design of systems that utilize more advanced controls might require a multi-day design period(s), and system sizing based on life cycle cost considerations generally requires an annual design period, or a multi-annual future forecast design period. Moreover, sizing of separate system components might require separate design days. Additionally, in many cases it can be difficult to make an ex ante determination what constitutes an appropriate design period. For example, because of solar gains, the maximum annual sensible space cooling loads in some spaces can occur in the autumn, not on the "cooling design day" typically specified by standard references such as *ASHRAE Fundamentals Chapter 14: Climatic Design Information* (2017a).

These may be challenging tasks because there are an immense number of possible designs that could be tested and because any project may have multiple competing performance objectives, such as: to minimize first cost, to minimize life cycle costs, to minimize greenhouse gas emissions, and to maximize thermal comfort. Challenging as it may be, this task is – and always has been – the charge and art of design.

Finally, it is also important to note that our proposed process focuses on the design of mechanical systems and controls. The process assumes that factors such as building physical characteristics and internal heat gains have been previously decided. However, where a building project embraces an integrated design approach, features such as façade elements, construction, and even internal heat gains may be treated as design variables, rather than as uncontrolled inputs. In this case, some variables described as parts of steps 1.b–1.d in our system design procedure would instead be defined in step 3, alongside other design variables.

5.4 PRACTICAL IMPACT OF OUR PROPOSED DESIGN PROCEDURE

In this section, we articulate the practical benefits of our proposed revisions for design of a high thermal mass radiant cooling system. We present a step-by-step example of each system design procedure, compare the resulting design decisions, then present results from cooling design day simulations and annual simulations to demonstrate the consequences for indoor thermal comfort and total thermal energy use. Although we discuss implications for cooling (heating) plant design and performance, we did not explicitly evaluate the performance of cooling (heating) plant equipment – we simply modeled plants with fixed maximum cooling (heating) capacity, controlled to target fixed cooling (heating) supply water temperature setpoints. We used EnergyPlus (2020) for all models and simulations.

Both examples design the internally cooled ceiling and floor surfaces for one southern exposed perimeter zone in a multi-zone multi-story office building, as illustrated in Figure 5.2. The model was based mainly on the the *US Department of Energy Commercial Reference Buildings* model for a large office (Deru et al., 2011). The zone had 175 m^2 (1884 ft^2) floor area (5 m (16 ft) by 35 m (115 ft) interior dimensions) and a 3 m (10 ft) high ceiling. The floor and ceiling were both

23.26 cm (0.76 ft) thick medium-weight concrete-slab with an additional covering on the floor surface with thermal resistance of 0.0206 K□m^2/W (0.117 °F□m^2□h/Btu). The outdoor exposed southern wall conformed to *California Title 24* (CEC, 2016), with 36% window-to-wall ratio and no exterior shading. The floor and ceiling were thermally interconnected to represent the heat transfer between multiple equivalent middle-story spaces. All other walls were represented with adiabatic boundary conditions.

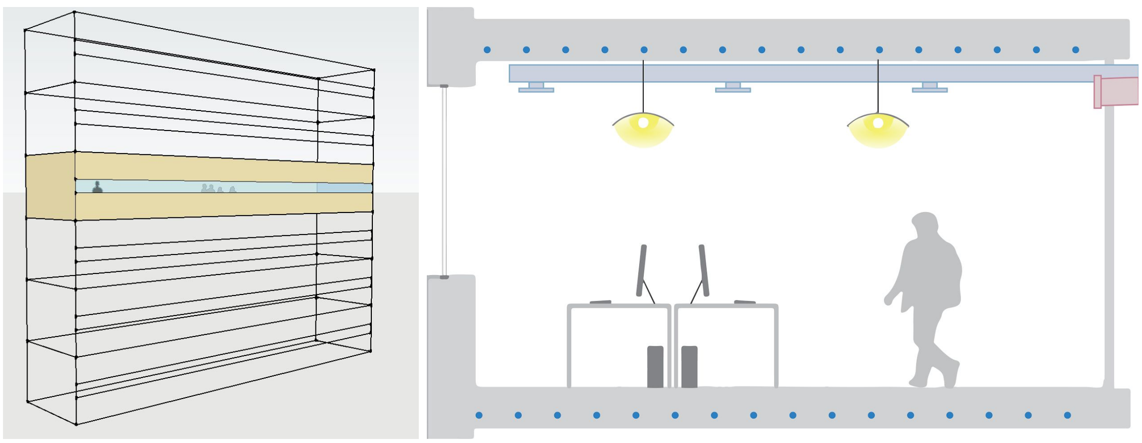

Figure 5.2: Isometric and cross-section illustrations of the zone used to demonstrate each system design procedure.

For both examples we imposed dynamic internal heat gains (composed of sensible heat from people, lights, and equipment) representing an office with weekday occupied hours 8:00–18:00. The internal heat gains were based on schedules from the *US Department of Energy Commercial Reference Buildings* model for a large office (Deru et al., 2011), with nominal internal heat gain rates somewhat smaller than those defined by *ASHRAE Standard 90.1* (2016c). The peak sensible internal heat gain rate was 16.9 W/m^2 (5.4 Btu/h□ft^2). We modeled infiltration to vary with wind speed, with a design infiltration flow rate of 0.56 l/s per m^2 exterior surface rate, and peak infiltration rate of 34.3 l/s (73 cfm) during the design day. The peak sum of sensible internal heat gains, infiltration, and solar heat gain on the cooling design day was 36.3 W/m^2 (11.5 Btu/h□ft^2).

For both examples, we imposed a continuous ventilation rate of 160 l/s (339 cfm) during occupied periods (08:00–18:00). This ventilation rate was approximately 20% larger than required by *California Title 24* (CEC, 2016), about 25% larger than required by *ASHRAE Standard 62.1* (2016b), and in agreement with the ventilation rates typically used for high thermal mass radiant buildings (Paliaga et al., 2017, 2018). For both examples, we used a dedicated outdoor air system to supply ventilation. The system heated ventilation air to 15 °C (59 °F), or cooled ventilation air to 25 °C (70 °F), or supplied unconditioned ventilation air between 15–25 °C (59–70 °F).

We performed system design calculations and annual simulations for the building located in Sacramento California – a climate with 0.4% cooling design condition outdoor dry-bulb temperature = 37.9 °C (100.2 °F) and mean coincident wet-bulb temperature = 21.3 °C (70.3 °F). For simplicity, both examples only design systems for sensible cooling requirements.

5.4.1 Example of the standard cooling load calculation and system design procedure:

Step #1 Space cooling load calculation

The first steps in the standard system design procedure are to define a site, and building characteristics, then to define climate conditions and internal heat gains for a design day, and to perform a standard space cooling load calculation. This process is described with greater detail in Section 5.1 – the introduction to this chapter. To be clear, although we are designing a radiant system, this standard space cooling load calculation only considers space heat extraction by convection. As discussed previously, Feng et al. (2014a) and Paliaga et al. (2017, 2018) showed that many designers commonly use this standard approach to size radiant cooling systems.

Figure 5.3 plots the results of the design day cooling load calculation for a constant indoor air temperature of 25 °C (77 °F). The calculation suggests that the peak sensible space cooling load should be 26.3 W/m² (8.3 Btu/h□ft²), and that it would occur at 15:00.

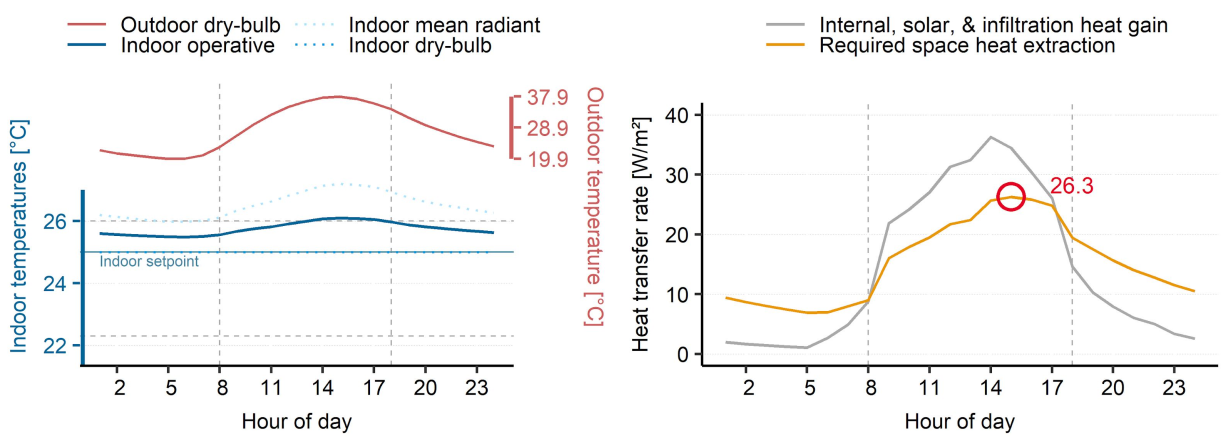

Figure 5.3: Standard cooling load calculation for the cooling design day. (Left): Outdoor dry-bulb air temperature and indoor temperatures. (Right): Sum of internal, solar, and infiltration heat gain (loss) rates, and the required space heat extraction rate (space cooling load).

1. *Heat transfer rates are normalized by the floor area for the zone analyzed.*
2. *The red circle indicates the peak sensible space cooling load of 26.3 W/m² (8.3 Btu/h□ft²).*
3. *The horizontal gray dashed lines indicate the minimum and maximum operative temperature that would achieve |PMV|<=0.5 during occupied hours for metabolic rate =1.15 met, and clothing = 0.67 clo, relative humidity = 55%, and indoor air speed <0.2 m/s (0.66 ft/s).*
4. *The vertical gray dashed lines indicate the start and end hours for occupancy in the example building.*

Step #2 Design internally cooled surfaces

The second step in the standard system design procedure is to design terminal heat transfer devices with steady-state cooling capacity to match the peak space cooling load at coincident conditions. In this case, our terminal heat transfer devices are the internally cooled 23.26 cm (0.76 ft) thick medium-weight concrete-slab ceiling and floor surfaces enclosing the space. As we've discussed, it is problematic to assume that this device operates with steady-state cooling capacity, but Feng et al. (2014a) and Paliaga et al. (2017, 2018) showed that many designers commonly use this standard approach to size radiant cooling systems.

This step requires that a designer select the configuration of internally cooled (heated) surfaces, including: the thickness and conductivity of these surfaces, the dimensions of tubes, the spacing between tubes, the depth of tubes, the number of parallel tubing loops, and the temperature and flow rate of chilled water supplied to the slab. Calculation of the steady-state cooling capacity for this terminal cooling device also requires information about the indoor operative temperature, which – for this step only – we assume is practically equal to the indoor air temperature for a space with radiant cooling (Dawe et al., 2020).

There are various combinations of design variable values for our internally cooled ceiling and floor surfaces that would generate commensurate steady-state space cooling capacity. For this example tube spacing = 22.86 cm (9 in), ASTM F876 ⅝" tubing, tube inside diameter = 17 mm (0.67 in), and tube depth = 57.15 mm (2.25 in) from the bottom face of each internally cooled surface. We selected these values because they are common design choices in practice, and because the typical range for these variables has a relatively small impact on steady-state space cooling capacity compared to water temperature and flow rate. These values result in 834 m (2736 ft) of tubing within each 175 m^2 (1884 ft^2) internally cooled surface, which we divided into 8 parallel, 104.2 m (342 ft) tubing loops.

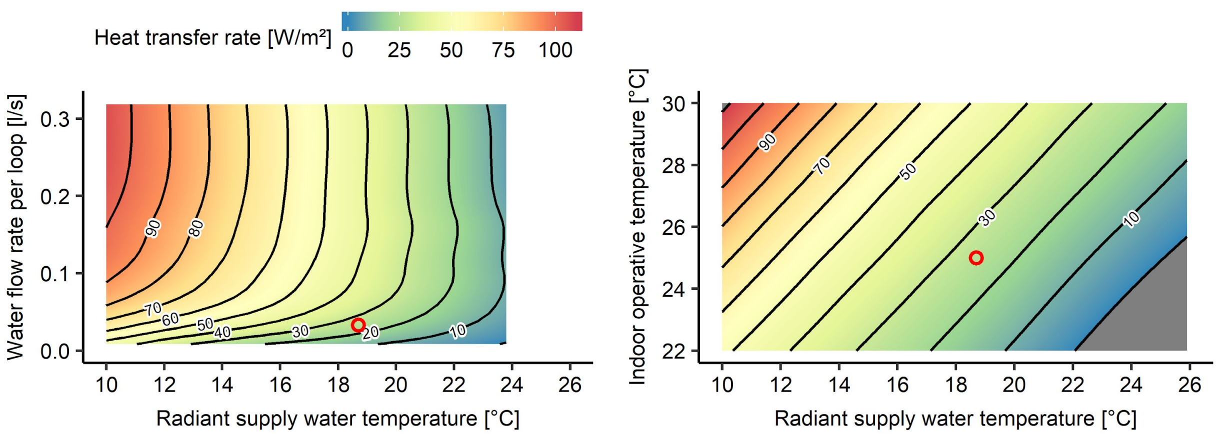

Figure 5.4: Steady-state space cooling capacity for an internally cooled 23.26 cm (0.76 ft) thick medium-weight concrete-slab floor and ceiling, with a thin covering on the floor surface, 8 parallel 104.2 m (342 ft) tubing loops, with tube spacing = 22.86 cm (9 in), tube inside diameter = 17 mm (0.67 in), tube depth = 57.15 mm (2.25 in). (Left): Steady state space cooling capacity as a function of supply water temperature and supply water flow rate. (Right): Steady state space cooling capacity as a function of supply water temperature and indoor operative temperature (ISO, 2012; Raftery, Duarte, Schiavon, et al., 2019).

1. *Heat transfer rates are normalized by the floor area for the zone analyzed, not by the total area at the indoor faces of the two internally cooled surfaces.*
2. *The red circles indicate design selected design conditions:*
    - *indoor operative temperature = 25 °C (77 °F)*
    - *supply water temperature = 18.7°C (65.7 °F)*
    - *supply water flow rate per loop = 0.0328 l/s (0.52 gpm)*
3. *The resulting steady-state sensible space cooling capacity = 26.5 W/m² (8.4 Btu/h□ft²), which matches the peak sensible space cooling load of 26.3 W/m² (8.3 Btu/h□ft²).*

Figure 5.4 plots the steady-state capacity values for these internally cooled ceiling and floor surfaces across a range of supply water temperatures, supply water flow rates, and indoor operative temperatures as calculated according to *ISO 11855* (2012) using the interactive web-based calculator developed by Raftery et al. (2019). We selected a point that would satisfy the peak sensible space cooling load of 26.3 W/m^2 (8.3 Btu/h□ft^2) with a manufacturer recommended supply-to-return water temperature difference between 2.77–4.44 °C (5–8°F) and water flow rate that would not exceed manufacturer recommended pressure drop of 30 kPa (10 ft of head) across each tubing loop (Uponor, 2013). As indicated in Figure 5.4, the resulting design uses 18.7 °C (65.7 °F) supply water temperature, and 0.0328 l/s (0.52 gpm) supply water flow rate per loop, and has 4.3 °C (7.74 °F) supply-to-return water temperature difference.

Step #3 Design cooling plant

The third and final step in the standard system design procedure is to design a cooling plant with steady-state cooling capacity to match the maximum simultaneous aggregate space cooling load from all associated zones at coincident outdoor conditions. Figure 5.5 plots the steady-state cooling capacity for a chiller, as a function of entering water temperature, and outdoor temperature.

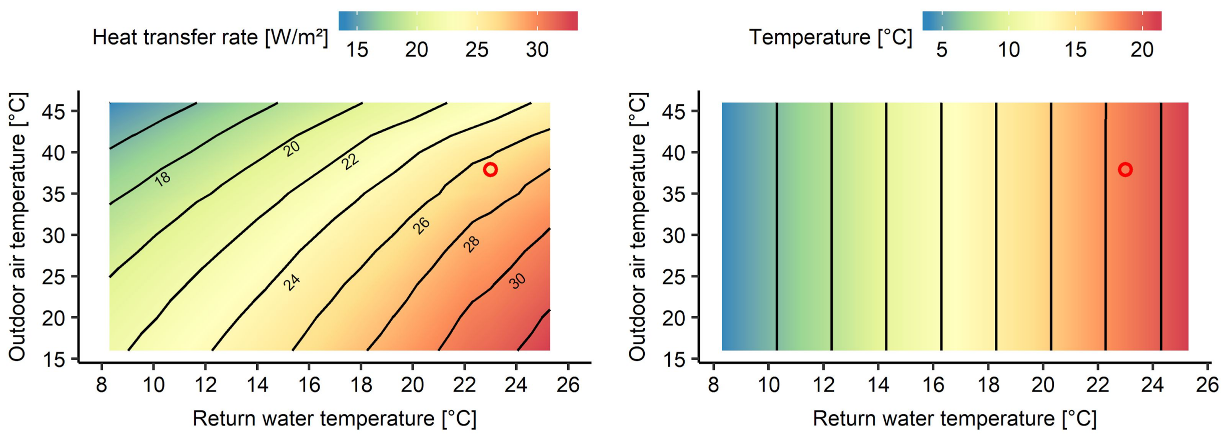

Figure 5.5: (Left): Steady-state cooling capacity for an air-cooled chiller as a function of return water temperature and outdoor air temperature. (Right): Steady-state supply water temperature for an air-cooled chiller as a function of return water temperature and outdoor air temperature.

1. *Water flow rate = 0.055 l/s□kW (3.1 gpm/ton).*
2. *Heat transfer rates are normalized by the floor area for the zone analyzed.*
3. *The red circles indicate the selected design conditions:*
   - *outdoor air temperature = 37.9 °C (100.2 °F)*
   - *return water temperature = 23 °C (73.4 °F)*
4. *The resulting steady-state cooling capacity = 26.5 W/m^2 (8.4 Btu/h□ft^2) which matches the peak space heat extraction rate (space cooling load) of 26.3 W/m^2 (8.3 Btu/h□ft^2)*
5. *The resulting supply water temperature = 18.7 °C (65.66 °F).*

5.4.2 Example of our proposed cooling load calculation and system design procedure:

Step #1 Define site, building, and design period conditions

The first step in our proposed system design procedure is to define a site, and building characteristics, then define climate conditions, and internal heat gains for a design period. This information corresponds exactly to the basic inputs for a standard cooling load calculation – corresponding to steps 1.a–1.c in our summary of the standard system design procedure presented in the Introduction. Accordingly, the values used for this example of our proposed system design procedure were described previously.

Step #2 Specify design objectives and performance constraints

Our proposed system design procedure does not restrict the cooling (heating) load calculation to a constant indoor air temperature, and does not require ex ante specification of the exact indoor thermal conditions that will occur. Instead, it allows designers to specify constraints on performance metrics. These constraints do not represent a system control strategy. Rather, the results of simulation for the design period will be compared to these constraints, and design variants that do not satisfy the constraints will be rejected. For this example design, we defined minimum and maximum constraints on the indoor operative temperature during occupied hours. We selected minimum indoor operative temperature = 22.3 °C (72 °F) and maximum indoor operative temperature = 26 °C (79 °F), which correspond to $|PMV|<=0.5$ for metabolic rate =1.15 met, clothing = 0.67 clo, relative humidity = 55%, and indoor air speed <0.2 m/s (0.66 ft/s) (ASHRAE, 2017b; CEN, 2019; ISO, 2005).

Step #3 Define terminal heat transfer devices, system details, and control sequence

The third step in our proposed system design procedure is to define the terminal heat transfer devices that will cool (heat) the space, the systems that will serve these devices, and a control sequence to manage these systems. This requires specification of all physical parameters and control sequences necessary to populate a mathematical model that adequately emulates the system thermodynamics. Since the proposed system design procedure requires iteration, and may allow for multiple solutions that satisfy performance constraints, we present four example design variants. In practice, a designer may test more design variants, as necessary to settle on a satisfactory design.

All four example design variants used tube spacing = 22.86 cm (9 in), tube inside diameter = 17 mm (0.67 in), and tube depth = 57.15 mm (2.25 in) from the bottom face of each internally cooled surface. These values are the same as what we selected for the example of the standard system design procedure. We selected these design variable values because they are common values in practice, and because the typical range for these variables has a relatively small impact on steady-state space cooling capacity compared to supply water temperature and flow rate. All four example design variants used 8 parallel 104 m (342 ft) tubing loops, with total water flow rate = 0.65 l/s (10.3 gpm) – the maximum flow rate that would not exceed manufacturer recommended pressure drop of 30 kPa (10 ft of head) across each tubing loop (Uponor, 2013).

Instead of using a feedback control loop to target an indoor air temperature setpoint – as is typical for most cooling and heating systems – these four example design variants use a feedback control loop that targets a floor surface temperature setpoint (measured at the top face of the floor). For calculations on the design day, we selected a floor surface temperature setpoint = 19 °C (66.2 °F) – the minimum floor surface temperature allowed by *ISO 7730* (2005) and *ASHRAE Standard 55* (2017b). Within a designer-specified water circulation availability period, the system controls two-position valves to allow water to circulate through parallel tubing loops at a designer-specified flow rate, and with chilled water from a plant with designer-specified capacity, and designer-specified supply water temperature setpoint. Outside of the availability period, the two-position valves remain closed, and the cooling plant is off.

Table 5.1: Design variable values for four example design variants tested in our recommended system design procedure, and for the preceding example design developed according to the standard system design procedure.

Design variable	System sized with standard design procedure	Systems sized with our recommended design procedure			
		Variant 1: plant=50%, avail.=0–24	Variant 2: plant=75%, avail.=0–24	Variant 3: plant=75%, avail.=18–12	Variant 4: plant=100%, avail.=18–12
Tube depth mm (in)	57.15 (2.25)				
Tube spacing mm (in)	22.86 (9)				
Tube inside diameter mm (in)	17 (0.68)				
Number of parallel tubing loops	8				
Length of ea. parallel loop m (ft)	104.2 (342)				
Total length of tubing m (ft)	834 (2736)				
DOAS supply air temp. °C (°F)	cooled to 25 (77), heated to 15 (59), floats from 15–25 (59–77)				
Supply water flow rate l/s (gpm)	0.262 (4.2)	0.65 (10.3)			
Supply water temp. stpnt. °C (°F)[A]	18.7(65.7)	20 (68)	20 (68)	18 (64.4)	19 (66.2)
Water circ. avail. period	00:00–24:00	00:00–24:00		18:00–12:00	
Indoor air temp. stpnt. °C (°F)[B]	25 (77)	–			
Floor srfc. temp. stpnt. °C (°F)[B,C]	–	19 (66.2)			
Plant capacity W/m^2 (Btu/hr ft^2)[D]	26.3 (8.3)	13.3 (4.2)	19.9 (6.3)	19.8 (6.3)	26.2 (8.3)
Plant capacity (as % of standard)	100%	50%	75%	75%	100%

A. *The supply water temperature setpoints recorded here are often not satisfied with the available plant capacity. In such a case, the supply water temperature values recorded in Table 5.2 will not match the supply water temperature setpoint values recorded here.*
B. *For the standard radiant system design, circulation through the internally cooled floor and ceiling is controlled by a constant indoor air temperature setpoint, whereas the design variants developed with our recommended procedure are controlled by a floor surface temperature setpoint.*
C. *The floor surface temperature setpoint recorded here is used for design day simulations, but annual simulations use an adaptive demand-shifting control sequence that adjusts the floor surface temperature setpoint each day based on feedback about the indoor air temperature on the previous day. In this case, the floor surface temperature setpoint recorded here is used as the minimum allowable floor surface temperature setpoint.*
D. *Value normalized by the floor area for the zone analyzed.*

With all of these preceding design variable values, we tested four example design variants, each with a different combination of cooling plant capacity, water circulation availability period, and supply water temperature setpoint. We tested cooling plant sizes that would be smaller than or equal to the cooling plant selected by the standard design procedure. We tested two different water circulation availability periods: one that allows operation during all hours, and one that reduces operation during a periods with high time-of-use electricity tariffs[6]. Then for each example combination, we adjusted the supply water temperature setpoint and floor surface temperature setpoint to find a setting that could satisfy the constraints on operative temperature during the cooling design day. We also ensured that supply water temperature would not cause condensation, but it was not necessary to invoke this constraint for any of the design variants. Table 5.1 summarizes all of the design variable values we selected for each design variant.

Step #4 Conduct design period simulation, compare results to constraints, and iterate

The last step in our proposed system design procedure is to simulate each design variant for the design period of interest, then compare the results to performance constraints, and iterate on design variants to select a system design and control sequence that best satisfies performance objectives. As results in Section 5.4.3 demonstrate, the traditional cooling design day may not be the most appropriate basis for system design selection, yet that is what we demonstrate here.

Figure 5.6 presents the results from simulation of the four example design variants on the cooling design day. These results show that on the cooling design day Variant 2–4 all satisfy the designer-specified constraints on operative temperature, but Variant 1 (plant=50%, avail.=0–24) does not. These results demonstrate that multiple design variants may satisfy the performance constraints. In practice, a designer may test more variants. For example, a designer may reassess Variant 1 (plant=50%, avail.=0–24) with the addition of ceiling fans, personal comfort systems, or different controls to find a design with 50% size cooling plant that will achieve acceptable PMV on the design day. Also, a designer must be careful to select a design period that is appropriate to test performance of each design variant. In Section 5.4.3, we present results from annual simulations which reveal that Variant 1 (plant=50%, avail.=0–24) actually performs reasonably well on an annual basis, demonstrating that a single day design period may not adequately represent differences between design variants.

Additionally, these dynamic simulations on the cooling design day reveal that the peak space heat extraction rate (peak space cooling load) can be much larger than what is predicted by a standard cooling load calculation, but that the cooling plant can be much smaller. In this case, the peak space heat extraction rates (peak cooling load) for the example design variants are 10–29% larger than the peak space heat extraction rate (peak cooling load) predicted by the standard cooling load calculations. At the same time, these design variants use cooling plants that are smaller than or equal to the peak space heat extraction rate (peak cooling load) predicted by the standard cooling load calculation. Of the four design variants tested here, the largest peak space cooling capacity is generated by the system with a 00:00–24:00 availability schedule and cooling plant that is 25% smaller than what is predicted by a standard space cooling load calculation.

[6] Time-of-use electricity tariff structures vary substantially between different utilities, and service types. We selected an availability schedule that would avoid operation from 12:00–18:00, which corresponds to the “summer peak pricing” period on PG&Es E-19 “General Demand” time of use tariff.

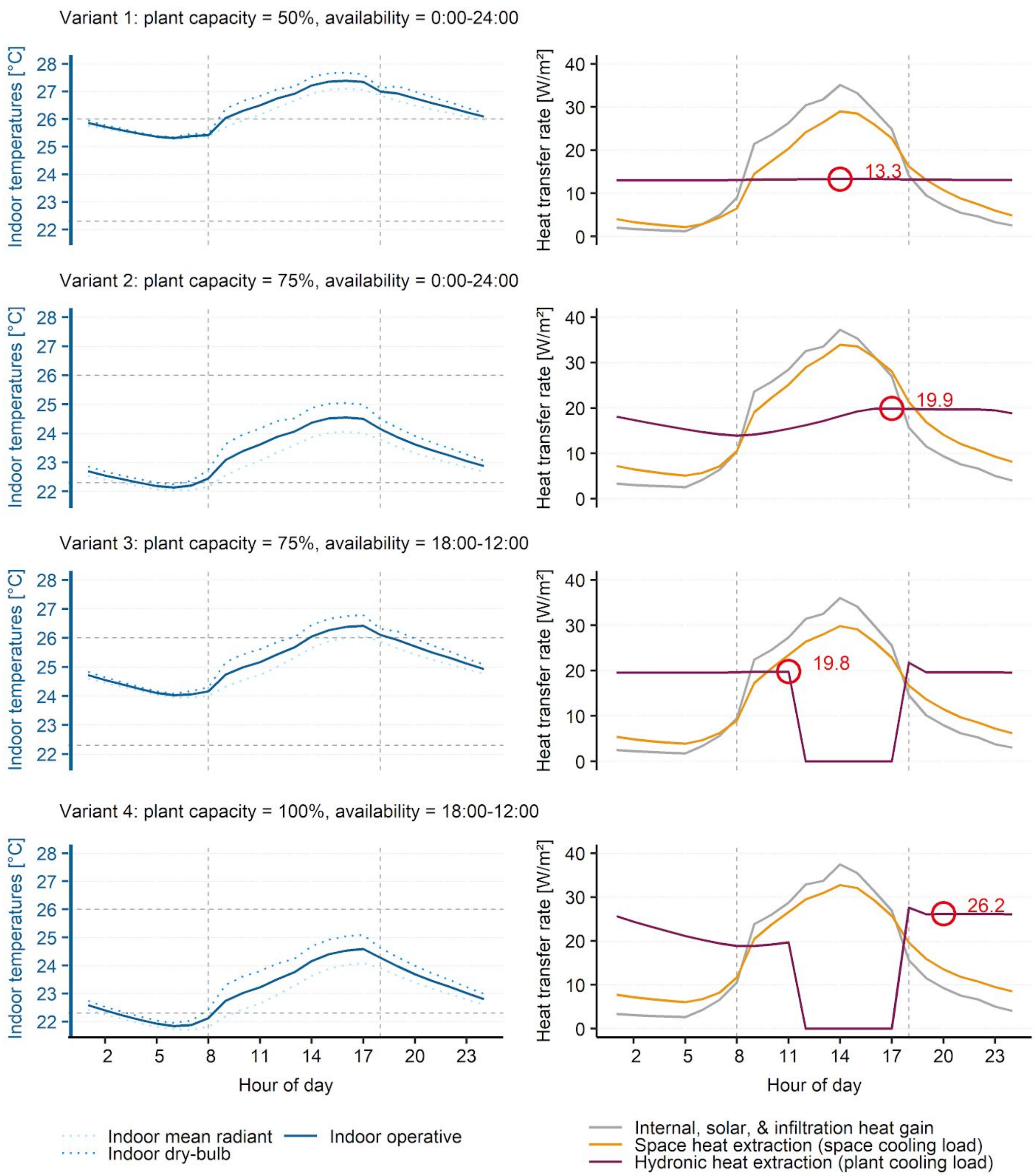

Figure 5.6: Inputs to and results from design day simulations for four example design variants. (Left): Outdoor dry-bulb air temperature and indoor temperatures. (Right): Sum of internal, solar, and infiltration heat gain (loss) rates, space heat extraction rate (space cooling load), and hydronic heat extraction rate (plant cooling load).

1. *The space heat extraction rate (space cooling load) is the sum of convective and radiant (longwave and shortwave) heat transfer rates at the indoor face of the internally cooled surfaces. Positive values indicate heat transfer from the space to the internally cooled surfaces.*

2. *The hydronic heat extraction rate (plant cooling load) is the heat transfer rate between water and the internally cooled surfaces. Due to thermal capacitance of water volume in the circuit, and transport time between the plant and internally cooled surfaces, this is very similar but not exactly the same as the heat transfer rate measured at the cooling plant.*
3. *Heat transfer rates are normalized by the floor area for the zone analyzed, not by the total area at the indoor faces of the two internally cooled surfaces*
4. *The red circles indicate the peak hydronic heat extraction rate (plant cooling load) for each design variant.*
5. *The horizontal gray dashed lines indicate designer-specified constraints on operative temperature. Minimum and maximum constraints on operative temperature were selected that would achieve |PMV|<=0.5 during occupied hours for metabolic rate =1.15 met, clothing = 0.67 clo, relative humidity = 55%, and indoor air speed <0.2 m/s (0.66 ft/s).*
6. *The vertical gray dashed lines indicate the start and end hours for occupancy in the example building*
7. *Outdoor air temperature for the cooling design day is shown in Figure 5.3*

5.4.3 Comparison of performance for systems designed according to each procedure

In this section, we assess and compare the systems designed in the preceding examples. First, we compare the space heat extraction rates predicted by a standard cooling load calculation to the space heat extraction rates that would actually be produced on the design day by the high thermal mass radiant system designed according to the standard procedure. Second, we compare the equipment size requirements, and design day system efficiency, for the systems designed according to each procedure. Third, we compare occupant thermal comfort, and thermal energy consumption for each system design predicted by annual simulation for the building in Sacramento, CA – using meteorological data representative of California Climate Zone 12 (CEC, 2019). All results are based on models developed in EnergyPlus (2020).

It is currently common to size high thermal mass radiant systems according to standard space cooling load calculations, then to control these systems with constant indoor air temperature setpoints on all days, regardless of occupancy (Feng et al., 2014a; Paliaga et al., 2017, 2018; Raftery, Duarte, & Dawe, 2019a, 2019b). Correspondingly, our models of the system sized according to the standard design procedure used two position valves controlled by constant indoor air temperature setpoints – heating setpoint = 22 °C (71.6 °F), and cooling setpoint = 25 °C (77 °F). Additionally, for annual simulations, we tested two common control variants:

1. The first approach allows changeover between heating and cooling whenever the indoor air temperature setpoints are not met
2. The second approach uses the same indoor air temperature setpoints, but imposes a 24 hour lockout between heating and cooling.

Our models of the four example design variants developed with our recommended procedure used two position valves controlled by a floor surface temperature setpoint (measured at the top face of the floor), and only allowed to open during a water circulation availability period – as described previously in Section 5.4.2 *Step #3*. Additionally, rather than operate with constant floor surface temperature setpoints for the entire year, we modeled an adaptive demand-shifting control sequence that adjusts the floor surface temperature setpoint each day based on feedback about the indoor air temperature on the previous day. The floor surface temperature setpoint is limited to 19 °C (66.2 °F) for cooling, and 29 °C (84.2 °F) for heating, in accordance with *ISO 7730* (2005) and *ASHRAE Standard 55* (2017b). This control sequence is modeled after one described by Raftery et al. (2017) and demonstrated in practice by Raftery et al. (2019a, 2019b).

Comparison of the standard cooling load calculation to simulated performance of the standard system design on the design day:

As discussed throughout this chapter, standard cooling load calculations do not properly estimate the space heat extraction rates that are generated by radiant systems. This inaccuracy is attributed in part to the fact that standard cooling load calculations do not consider space heat extraction by radiation, and in part to the fact that they do not account for dynamic variation in the indoor thermal conditions that typically occur with high thermal mass radiant systems. Figure 5.7 compares the space heat extraction rates (space cooling load) predicted by a standard cooling load calculation (also shown in Figure 5.3) to the space heat extraction rates that would actually be produced on the design day by the high thermal mass radiant system designed according to the standard procedure and controlled with constant indoor air temperature setpoints. The comparison reveals that on the cooling design day the actual peak space heat extraction rate is 22% larger, even though the cumulative space heat extraction is practically equal. This reconfirms findings from simulations by Niu et al. (1995, 1997) and Feng et al. (2013), and experiments by Feng et al. (2014b), Novoselac (2017), and Woolley et al. (2018a, 2019) (Chapter 3 and Chapter 4). Additionally, whereas previous research has made this comparison whilst radiant cooling and all-air cooling maintained equal operative temperatures or equal air temperatures, these results indicate that the peak space heat extraction rate for radiant cooling is larger than for all-air cooling even when the indoor air temperature for a high thermal mass radiant system drifts around the indoor air temperature setpoint, and the indoor air temperature for the all-air system remains constant.

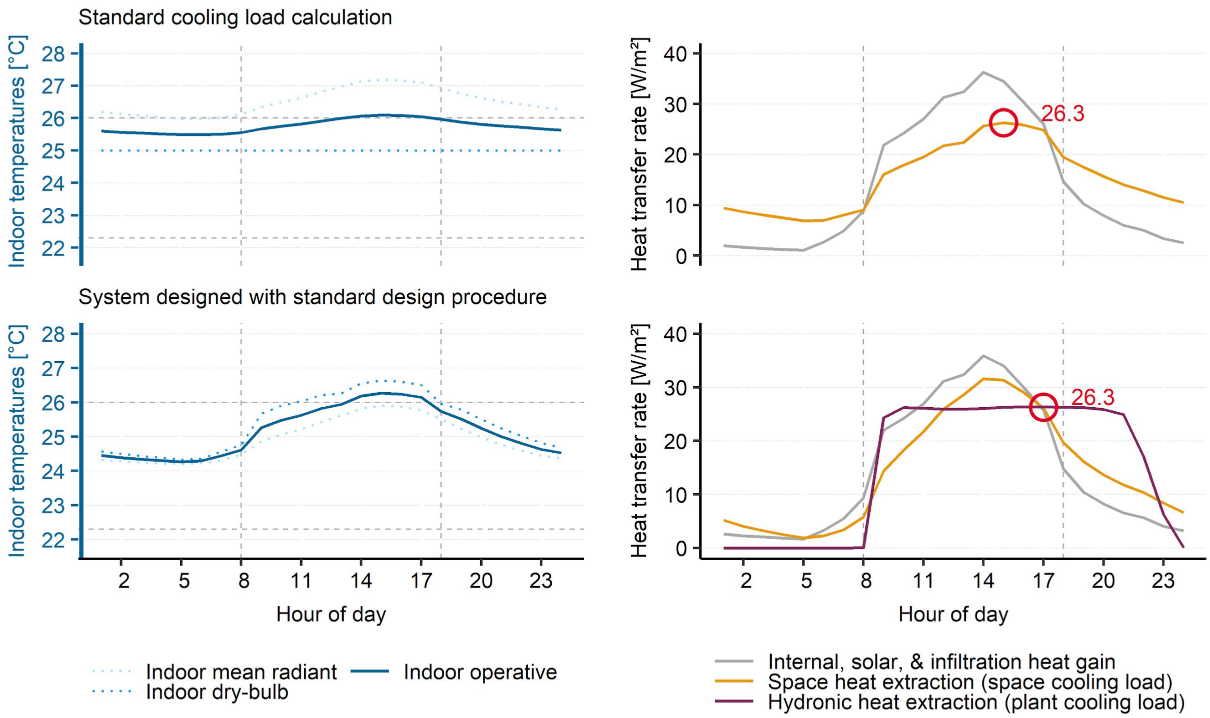

Figure 5.7: Comparison of the standard cooling load calculation to cooling design day simulation of the system designed according to the standard system design procedure.. (Left): Outdoor dry-bulb air temperature and indoor temperatures.(Right): Sum of internal, solar, and infiltration heat gain (loss) rates, space heat extraction rate (space cooling load), and hydronic heat extraction rate (plant cooling load).

1. *The space heat extraction rate (space cooling load) is the sum of convective and radiant (longwave and shortwave) heat transfer rates at the indoor face of the internally cooled (heated) surfaces. Positive values indicate heat transfer from the space to the internally cooled (heated) surfaces.*
2. *The hydronic heat extraction rate (plant cooling load) is the heat transfer rate between water and the internally cooled surfaces. Due to thermal capacitance of water volume in the circuit, and transport time between the plant and internally cooled surfaces, this is very similar but not exactly the same as the heat transfer rate measured at the cooling plant.*
3. *Heat transfer rates are normalized by the floor area for the zone analyzed, not by the total area at the indoor faces of the two internally cooled (heated) surfaces.*
4. *The standard design procedure does not account for hydronic heat extraction rate. Instead it assumes that the space heat extraction rate (space cooling load) = hydronic heat extraction rate (plant cooling load).*
5. *For the standard cooling load calculation (top), the red circle indicates the peak space heat extraction rate (space cooling load). This value was used to size the cooling plant. For the design day simulation of the resulting radiant system design (bottom), the red circle indicates the peak hydronic heat extraction rate (plant cooling load), which is constrained by the cooling plant capacity.*
6. *The horizontal gray dashed lines indicate minimum and maximum operative temperatures that would achieve |PMV|<=0.5 for metabolic rate =1.15 met, clothing = 0.67 clo, relative humidity = 55%, and indoor air speed <0.2 m/s (0.66 ft/s).*
7. *The vertical gray dashed lines indicate the start and end hours for occupancy in the example building*
8. *Outdoor air temperature for the cooling design day is shown in Figure 5.3*

Comparison of design variable values selected with each system design procedure, and simulation of of each system on the design day

In this subsection, we compare the equipment size requirements, and design day system efficiency, for the systems designed according to each procedure. Table 5.2 summarizes some consequential differences between the designs that result from each procedure. The results demonstrate that for high thermal mass radiant systems, the standard definition of "space cooling (heating) load" and the associated standard system design procedure can lead to: equipment that is much larger than necessary, supply water temperature that is colder than necessary, and operation mainly during periods with high electricity tariffs. In particular, these examples of our recommended design procedure result in as much as: 50% smaller cooling plant equipment, 5.2 °C (9.4 °F) warmer median supply water temperature on the cooling design day, and 100% reduction in chilled water consumption during periods with high electricity tariffs. Smaller cooling plant equipment translates directly to reduced capital expenses. Moreover, the warmer supply water temperatures that occur for designs with reduced sized cooling plants demonstrate that it would be possible to use very efficient cooling plant equipment, such as a cooling tower, instead of a conventional chiller. The standard design procedure indicated that the supply water temperature should be 18.7 °C (65.7 °F), whereas Variant 1 (plant=50%, avail.=0–24) has cooling supply water temperature 23.5–24.2 °C (74.3–75.7 °C) on the cooling design day. This comparison reveals that for design of high thermal mass radiant systems the standard design procedure can mislead designers by obscuring impactful design opportunities.

Table 5.2: Summary of the design variable values selected and consequential results from simulation on design day for: (A) high thermal mass radiant systems sized with the standard system design procedure, and controlled with constant indoor air temperature setpoints; and (B) high thermal mass radiant systems sized with our recommended system design procedure, and controlled with an adaptive demand-shifting control sequence.

	Design variable values			Results for simulation on design day		
	Cooling plant capacity (as % of standard)	Water circ. avail. period	Supply water flow rate l/s (gpm)	Supply water temperature range °C (°F)[A]	Chilled water use in high tariff hours[D] kWh/m^2 (kBtu/ft^2)[C]	Outdoor temp. range for chiller operation °C (°F)
(A) System sized with standard design procedure, and controlled with constant indoor air temperature setpoint						
System sized with standard design procedure	100%	00:00–24:00	0.262 (4.2)	18.7–18.9 (65.7–66.0)	0.157 (0.050)	23.4–37.9 (74.1–100)
(B) Systems sized with our recommended design procedure, and controlled with constant slab temperature setpoint						
Variant 1: plant=50%, avail.=0–24	50%	00:00–24:00	0.65 (10.3)	23.5–24.3 (74.3–75.7)	0.0799 (0.025)	19.9–37.9 (67.8–100)
Variant 2: plant=75%, avail=0–24	75%	00:00–24:00	0.65 (10.3)	20–20.2 (68–68.4)	0.110 (0.035)	19.9-37.9 (67.8-100)
Variant 3: plant=75%, avail=18–12	75%	18:00–12:00	0.65 (10.3)	21.2–22.5 (70.2–72.5)	0	19.9–34.2 (67.8–93.6)
Variant 4: plant=100%, avail=18–12	100%	18:00–12:00	0.65 (10.3)	19–19.7 (66.2–67.5)	0	19.9–34.2 (67.8–93.6)

A. *The supply water temperature setpoints recorderded in Table 5.1 are not always satisfied with the available plant capacity. This table records the actual supply water temperature range that occurs during the design day simulation due to limited cooling plant capacity.*
B. *Value is normalized by the floor area for the zone analyzed.*
C. *"High tariff hours" are assumed to be 12:00–18:00.*

Comparison of annual simulations of the system designs that result from the alternate system design procedures

In this subsection, we compare occupant thermal comfort, and thermal energy consumption predicted for each system design over the course of a year in Sacramento, CA – using meteorological data representative of California Climate Zone 12 (CEC, 2019). We performed annual simulations of each design variant in EnergyPlus (2020). The control sequences for these annual simulations are described in the introductory paragraphs for Section 5.4.3.

Table 5.3 summarizes the results from annual simulation of each system design.

Figure 5.8 presents the indoor operative temperature and the space heat extraction rates (space cooling load) for every work day of the year, as predicted by annual simulation of each system design. Each plot aggregates 260 daily time series traces into a composite 00:00–24:00 range. Each trace is colored to represent periods with chilled water circulation (blue), periods with heating water circulation (red), and periods with no water circulation (green). The comparison reveals that our proposed system design procedure and control strategy can enable substantial improvements in annual performance. Compared to the system designed according to the standard design procedure, the four example variants designed according to our proposed procedure reduced discomfort during occupied periods by as much as 55%, reduced cumulative thermal energy use for cooling by as much as 81% during periods with high electricity tariffs, increased median supply water temperature in cooling by as much as 3.3 °C (5.9 °F), and increased the minimum supply water temperature in cooling by as much as 1.6 °C (2.9 °F).

However, our four example design variants also increase annual thermal energy use for cooling by as much as 14%, and increase the relatively small amount of heating thermal energy use. We expect that these increases in thermal energy use occur because the systems controlled to a slab temperature setpoint with an adaptive demand shifting control sequence:

1. have lower operative temperatures which results in a larger cumulative indoor–outdoor temperature difference (i.e.: larger potential for envelope heat transfer);
2. derive less space cooling benefit from the DOAS system (a very small impact);
3. operate for longer hours with lower hydronic heat transfer rates, which reduces the amount of heat stored in surfaces and the amount of heat released to the environment.

Additionally, the median supply water temperature in cooling was lowest for Variant 4 (plant=100%, avail.=18:00–12:00) – despite the fact that the standard system designs had a lower supply water temperature setpoint. We expect that this occurred because the 18:00–12:00 availability schedule shifts operation to periods when building masses are naturally cooler.

Awareness of performance trade-offs such as these is essential to the design process, yet the standard design procedure does not examine trade-offs because it assumes a singular – “ideal” – space cooling (heating) load and simplified heat transfer.

Some designers impose a lockout period on changeover from heating to cooling to avoid energy use associated shifting the mass temperature, while other designers suggest that rapid changeover from heating to cooling is necessary to ensure comfort and that the impact on energy use is relatively small (Paliaga et al., 2017, 2018). For the single zone scenario we simulated, allowing rapid changeover from heating to cooling – based simply on a 3 °C (5.4 °F) deadband between indoor air temperature setpoints – reduced discomfort hours by 5%, while increasing annual thermal energy for cooling by 1% and heating by 0%.

Finally – and most pertinent to our critique of the standard system design procedure – these annual simulation results reveal that the cooling and heating design day simulations may not represent the most extreme behavior for a system. This result can be deduced from Figure 5.8.. Consequently, the comparison of design day simulation results for different design variants may not be indicative of differences in annual performance.

The annual peak space heat extraction rate (space cooling load) for each of the six example system designs was 60–76% larger than predicted by cooling design day simulations for each system. The discrepancy is even more significant when compared to standard cooling load calculations: the annual peak space heat extraction rate (space cooling load) for each of the system designs was 93–107% larger than what was predicted by standard cooling load calculations on the cooling design day.

Furthermore, the operative temperature response predicted by cooling design day simulations is not representative of what will occur annually. This issue is not simply that design day simulations fail to capture annual variation, but rather that the design day simulations of multiple design alternatives may not represent a consistent point within the respective annual variations. Specifically, although design day simulations indicate that Variant 1 (plant=50%, avail.=0–24) would not maintain indoor operative temperature within designer specified constraints, annual simulations reveal that this design would actually perform reasonably well, with fewer discomfort hours than the system designed according to the standard design procedure. On the other extreme, the cooling design day simulation for Variant 2 (plant=75%, avail.=0–24) and Variant 4 (plant=100%, avail.=18–12) largely underpredict the range of operative temperatures that would occur annually.

We see three factors underlying this problem with design day simulations.

1. The outdoor climate conditions, internal heat gains, and solar heat gains on the designated cooling design day may not represent the most extreme heat gain scenario that will occur annually.
2. Design day simulations may not capture real control variations that occur throughout the year.
3. Design day simulations typically repeat a 0:00–24:00 design period iteratively until dynamic heat transfer behavior converges to a stable daily profile (steady-state oscillation). This approach has a substantial and unrealistic influence on the initial thermal conditions predicted at the beginning of the design day. We expect this issue is especially pronounced for buildings with large thermal mass and cooling (heating) systems with long response time.

Consider, for example, that the design day simulations for Variant 1–Variant 4 included designer specified floor surface temperature setpoints, and supply water temperature setpoints; but our annual simulations used an adaptive control sequence which reset the floor surface temperature setpoint in response to performance the previous day. Consequently, the space heat extraction rates (space cooling load) and operative temperature response predicted on the design day within the annual simulation differ substantially from what is predicted by the iterative design day simulation with fixed setpoints.

These final observations highlight that in some circumstances, the conditions in the cooling design day prescribed by standards may not be the appropriate basis for system design decisions, and that the typical design day simulation procedure may not represent the realistic multi-day dynamics that a system and control sequence will encounter.

Table 5.3: Summary of results from annual simulations. (A) High thermal mass radiant systems sized with the standard system design procedure, and controlled with constant indoor air temperature setpoints. (B) High thermal mass radiant systems sized with our recommended system design procedure, and controlled with an adaptive demand-shifting control sequence developed by Raftery et al. (2017), and demonstrated by Raftery et al. (2019a, 2019b).

	24 hour changeover lockout[B]	Discomfort during occupied periods (hours)		Annual thermal energy use for cooling and heating plant[A]			
				Cooling kWh/m^2 (kBtu/ft^2)		Heating kWh/m^2 (kBtu/ft^2)	
		PMV>0.5 (too warm)	PMV<–0.5 (too cool)	Total	High tariff hours	Total	High tariff hours
(A) Systems sized with standard design procedure and controlled with constant indoor air temperature setpoint							
System sized with standard design procedure	No	494	13	59.9 (19.0)	27.7 (8.78)	2.8 (0.88)	0.01 (0.003)
	Yes	519	10	59.4 (18.8)	27.0 (8.57)	2.8 (0.88)	0.009 (0.003)
(B) Systems sized with our recommended design procedure and controlled with an adaptive floor surface temperature setpoint							
Variant 1: plant=50%, avail.=0–24	Yes	268	72	65.3 (20.7)	14.5 (4.6)	4.5 (1.4)	0.46 (0.146)
Variant 2: plant=75%, avail=0–24	Yes	157	81	67.9 (21.5)	20.9 (6.6)	5.1 (1.6)	0.47 (0.149)
Variant 3: plant=75%, avail=18–12	Yes	225	111	65.8 (20.9)	0	0	0.33 (0.11)
Variant 4: plant=100%, avail=18–12	Yes	176	144	66.9 (21.2)	0	0	0.30 (0.09)

A. *Annual thermal energy use for cooling and heating plants does not include the amount of thermal energy used by DOAS to heat ventilation air to 15 °C (59 °F) and cool ventilation air to 25 °C (70 °F). Annually, the DOAS system uses 4.7 kWh/m^2 (1.5 kBtu/ft^2) thermal energy for cooling and 4.8 kWh/m^2 (1.5 kBtu/ft^2) thermal energy for heating.*

B. *When the standard design does not include a 24 hour lockout on changeover between heating and cooling, changeover is only governed by a 3 °C (5.4 °F) deadband between heating and cooling setpoints which results in 46 days with heating and cooling on the same day. When the standard design includes a 24 hour lockout on changeover between heating and cooling, the two modes never operate with less than 24 hours separation.*

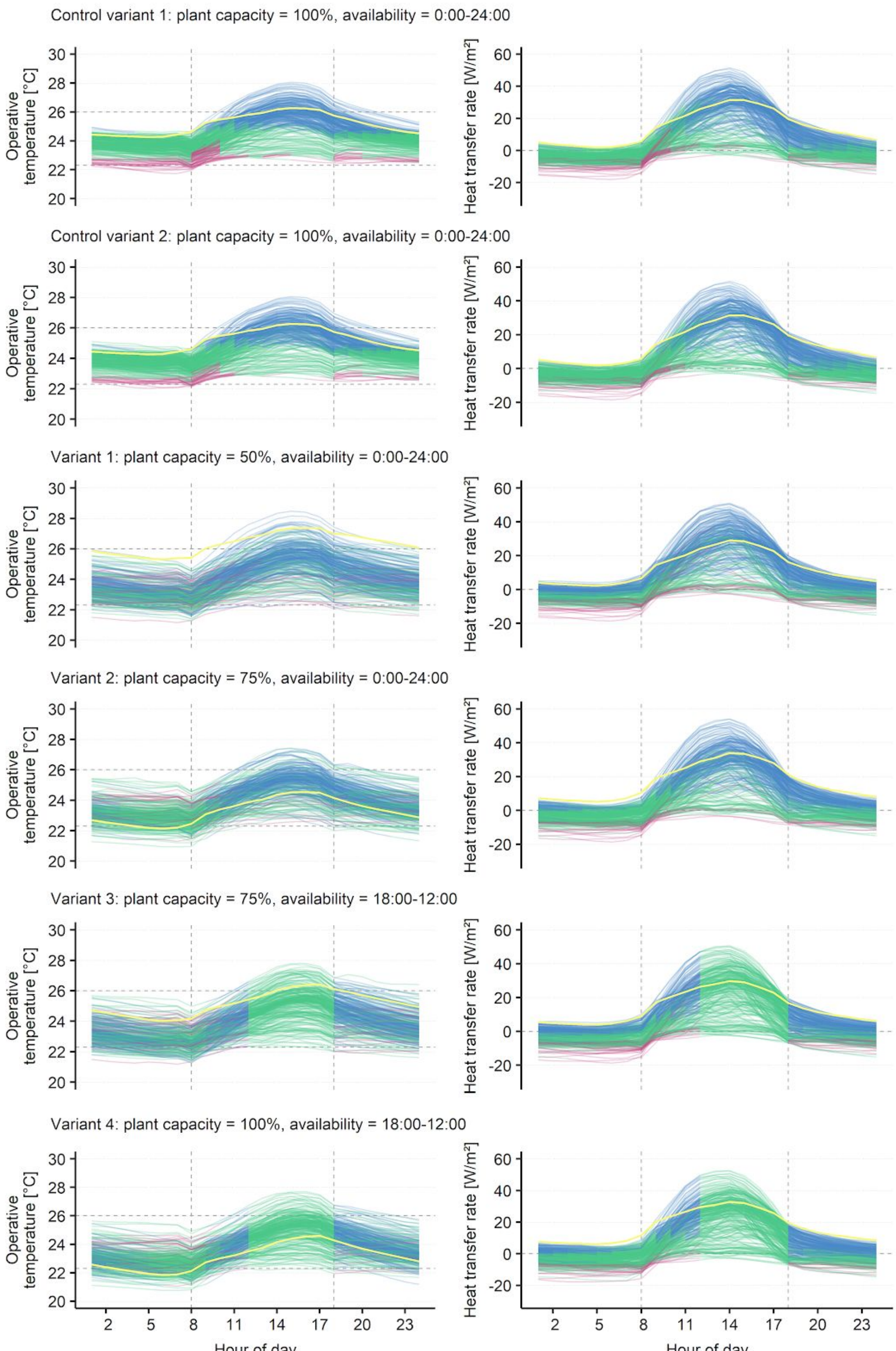
Control variant 1: plant capacity = 100%, availability = 0:00-24:00
Control variant 2: plant capacity = 100%, availability = 0:00-24:00
Variant 1: plant capacity = 50%, availability = 0:00-24:00
Variant 2: plant capacity = 75%, availability = 0:00-24:00
Variant 3: plant capacity = 75%, availability = 18:00-12:00
Variant 4: plant capacity = 100%, availability = 18:00-12:00
Operative temperature [°C]
Heat transfer rate [W/m²]
Hour of day
Hour of day

Figure 5.8: Annual simulation results for each example of system design and control. (Left): Indoor operative temperature.(Right): Space heat extraction rate (space cooling load).

1. *The space heat extraction rate (space cooling load) is the sum of convective and radiant (longwave and shortwave) heat transfer rates at the indoor face of the internally cooled (heated) surfaces. Positive values indicate heat transfer from the space to the internally cooled (heated) surfaces.*
2. *Heat transfer rates are normalized by the floor area for the zone analyzed, not by the total area at the indoor faces of the two internally cooled (heated) surfaces.*
3. *Each plot is a composite of 260 separate traces; the time series results from each work day in the annual simulation are overlaid onto a single 00:00–24:00 range.*
4. *Each trace is colored to represent periods with chilled water circulation (blue), periods with heating water circulation (red), and periods with no water circulation (green).*
5. *Each plot highlights the trace for the cooling design day.*
6. *The horizontal gray dashed lines indicate minimum and maximum operative temperatures that would achieve |PMV|<=0.5 for metabolic rate =1.15 met, clothing = 0.67clo, relative humidity = 55%, and indoor air speed <0.2 m/s (0.66 ft/s).*
7. *The vertical gray dashed lines indicate the start and end hours for occupancy in the example building.*

5.5 CONCLUSIONS

The standard definition of "cooling (heating) load" and the associated standard system design procedure is not appropriate for design of all cooling (heating) system types. The standard definition of "cooling (heating) load" omits important heat transfer fundamentals, fails to account for the impact of system controls, and imposes simple constraints that overlook fundamentals about thermal comfort. Consequently, use of the standard system design procedure obscures considerable opportunities to reduce costs, and improve energy efficiency and thermal comfort. In this chapter, we disentangled the many assumptions embedded within the standard definition of "cooling (heating) load" and examined the practical impacts these have on system design and performance in practice. We focused especially on the design of high thermal mass radiant cooling systems, but we also considered broader implications.

We recognize that there is a practical need for quick and simplified methods to estimate equipment sizing needs for cooling (heating) systems; however, the standard definition of "cooling (heating) load" ought to be a universal concept that facilitates design flexibility, and readily enables designers to consider strategies to improve building performance.

In this chapter, we proposed a broader and more flexible definition for "cooling (heating) load" and reinvisioned the standard system design procedure. Most significantly, our redefinition eliminates the idea of a singular – "ideal" – space cooling (heating) load as the objective for mechanical system design. Instead, our proposed approach orients the system design procedure toward selecting and sizing components and their controls that best satisfy designer-specified performance objectives such as: thermal comfort, indoor air quality, resilience, grid-interactive responses, greenhouse gas emissions, or life cycle energy cost minimization.

We used the standard definition of "cooling load" and the standard design procedure to design a high thermal mass radiant system. Then we performed design day simulations and annual simulations of the resulting system, and compared its performance to that of four different example systems designed with our recommended procedure. This comparison revealed large errors associated with the standard design procedure. In particular, in our examples, standard cooling load calculations underestimated peak space cooling loads by more than 100%, yet overestimated the required cooling plant capacity by as much as 100%. The comparison also demonstrated how the standard design procedure can lead designers to overlook considerable opportunities to improve performance. For example, the standard design procedure indicated that the cooling supply water temperature should be 18.7°C (65.7 °F), but using our proposed procedure we developed an example system design and control strategy that would operate with cooling supply water temperature 20.3–25.1 °C (68.5–77.2 °F) – median 22.2 °C (72 °F) – while also reducing discomfort during occupied periods. Additionally, the four examples developed with our design procedure reduced annual thermal energy consumption for cooling by as much as 81% during periods with high electricity tariffs, and reduced discomfort during occupied periods by as much as 55%.

Furthermore, our examples demonstrate that even when a designer employs accurate models of systems and controls, using a typical "cooling design day" simulation as the basis for system design may result in suboptimal equipment sizing, and may lead designers to reject design variants that would actually perform well on an annual basis.

Many critical global challenges hinge on improving performance of heating, cooling, and ventilation systems in buildings; and achieving such improvements demands system design that is more sophisticated than what is currently designated by the the standard definition of "space cooling (heating) load". Yet, the standard definition of "cooling (heating) load" is commonly used as the basis to design a wide range of cooling (heating) systems – including high thermal mass radiant systems. Therefore, we recommend that industry stakeholders update standards to address the shortcomings we have explained in this chapter. In Appendix A we include a comprehensive revision to the definitions and explanatory sections in *ASHRAE Fundamentals Chapter 18* (2017a). Finally, we recognize that custom models for buildings and systems and annual simulations are currently beyond the reach of many designers. We therefore think that there is an urgent need to develop design guidelines and user-friendly design tools that facilitate accurate comparison and optimization of system design and control alternatives.

CONCLUSIONS

The efficiency and demand responsiveness of heating, cooling, and ventilation systems is a critical factor in transition to more sustainable global energy systems. Radiant cooling and heating could play a substantial role in enhancing the energy performance of new and existing buildings. The technology is already common among zero-net-energy buildings, but it is not very common in general. There are not well established guidelines to support design of radiant systems and their controls, and most professionals in the buildings industry are unfamiliar with radiant systems. Some researchers and professionals have strong expertise with radiant cooling and heating systems, but even among this cohort we have shown that there is some disagreement about fundamentals and best practices – the technology is evolving and advancing. This dissertation builds upon the existing foundation of knowledge, and addresses several issues that are critical to design and control of radiant cooling and heating systems.

To begin with, we reviewed literature on the energy performance of radiant cooling systems and developed a statistical assessment of the energy use intensity for buildings with radiant cooling compared to that of comparable standard building stock. We showed that buildings with radiant cooling do use considerably less energy than standard buildings. However, we also explained why the observed savings – statistically significant with a large effect size – may not be attributed only to radiant cooling. As a part of this initial investigation, we also summarized the fundamental energy efficiency opportunities enabled by radiant cooling, and outlined numerous system design and control decisions that would be critical to actually achieve these potential benefits in practice.

Then, we collected qualitative information about how high thermal mass radiant cooling is most commonly understood and implemented by design professionals. We cataloged and described the design and control strategies regularly used in practice, and discovered that there are significant opportunities to improve on common approaches. Most importantly, we found that these systems (1) are typically designed using fundamentally flawed methods, (2) operate chiller plants at conventional low water temperature, (3) do not use water-side economizer (evaporative fluid cooler), and (4) do not use pre-cooling controls. The issue is not that designers are completely unaware of these performance gaps. In practice, many design decisions are made as compromises, and are motivated by an intent to avoid risk where there are not yet reliable standard practices, to satisfy cost constraints, and to alleviate other practical challenges. Although some designers are aware of the missed opportunities that we highlight, it also appears that their practice affords insufficient opportunity to quantify the energy consequences of these common decisions. These observations provided a tangible basis for the motivation behind other aspects of our research.

One significant knowledge gap in both theory and practice is associated with the methods, procedures, and tools used to design and size radiant cooling and heating systems. The epistemological basis for our standard approach to design of cooling and heating systems is based on immense experience with all-air systems – more than a century of collective industry focus on all-air systems – and there are several problems when the same methods are used for design or radiant systems. The work described in this dissertation helps to overcome the problems, particularly by resolving how the space heat extraction (input) requirements (space cooling (heating) load) for radiant systems should be dealt differently in system sizing and building energy modeling.

In Chapter 3 and Chapter 4 we presented results from an extensive series of high quality laboratory experiments in realistic conditions that prove that space heat extraction requirements (space cooling load) for radiant systems are larger than that of conventional all-air systems. These results confirm the indications from previous research, demonstrate the phenomenon in realistic circumstances, and reveal how the timing and magnitude of the differences between the space heat extraction requirements (space cooling load) for these two system types is impacted by factors such as heat gain characteristics and diurnal patterns of heat gain and passive heat loss to the environment. In a five day experiment with mixed internal heat gains, solar gains, and natural ventilation night pre-cooling, radiant cooling had to remove 35% more heat than the all-air system in equivalent circumstances; and the peak heat extraction rate was 20% larger (median difference on multiple days). In a similar experiment with highly convective internal gains the differences were smaller (26% more thermal energy, 12% larger peak), while in an experiment with highly radiant gains the differences were larger (40% more thermal energy, and 21% larger peak). The differences were much smaller in an experiment without natural ventilation night pre-cooling (7% more thermal energy, 5% larger peak).

Finally, in Chapter 5 we directly addressed the fact that the current standard procedure for sizing cooling (and heating) systems is not appropriate for design of radiant cooling and heating systems. In particular, we challenged the standard notion of "space cooling (heating) load" – as canonized by *ASHRAE Fundamentals 2017 Chapter 18: Nonresidential Cooling and Heating Load Calculations*. We critiqued several specific flaws with the current definition, and presented building energy simulation results to demonstrate the practical consequences of sizing radiant systems with the standard methods. Our assessment revealed that when used to design high thermal mass radiant systems, the standard design procedure can lead designers to underestimate peak the space cooling load by as much as 100%, yet oversize cooling plant equipment by as much as 100%, operate with a cooler water temperature than necessary, and control systems in a way that results in undesirable variations in indoor operative temperature. To resolve this, we also proposed a new definition for "space cooling (heating) load", associated concepts, and the standard cooling system design sizing procedure. Our redefinition is presented as a comprehensive revision to the explanatory sections of *ASHRAE Fundamentals Chapter 18: Nonresidential Cooling and Heating Load Calculations,* which is presented in Appendix A. Most importantly, our proposed definition shifts the focused objective of the system design procedure. Traditionally, designers select equipment and controls that satisfy an idealized space cooling load; our definition guides designers to select equipment and controls that satisfy indoor environmental objectives – such as thermal comfort, and indoor air quality.

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APPENDIX A

This appendix is reproduced in part and from an unpublished document presented to:

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as a proposed revision to:

ASHRAE Fundamentals 2017 Chapter 18:
Nonresidential Cooling and Heating Load Calculations.

CHAPTER 18

NONRESIDENTIAL COOLING AND HEATING LOAD CALCULATIONS

Cooling and heating loads are the rates of thermal energy transfer through a heating or cooling system that would be required to achieve desired indoor thermal environmental conditions, and/or other performance metrics. Heating and cooling systems are designed, sized, and controlled to produce commensurate thermal energy transfer rates. Heating and cooling loads can be determined for any point in time, but loads are inherently dynamic so can only be calculated across some period of time. Cooling and heating loads calculated across heating and cooling design periods serve as the primary basis for selection and design of most heating and cooling systems. These design load calculations affect the size of piping, ductwork, diffusers, air handlers, boilers, chillers, coils, compressors, fans, pumps, and every other component of systems designed to condition indoor environments. Therefore, cooling and heating design load calculations can have large impacts on first cost of building construction, comfort and productivity of occupants, and operating cost and energy consumption.

The amount of heating or cooling required for a particular space is dynamic and depends on many factors including: the type of system used to provide heating and cooling, diurnal patterns of outdoor temperature and humidity, patterns and distribution of internal sensible and latent heat gain, building construction, and system controls. To produce an appropriate rate of heat transfer to or from a space, heating and cooling systems must transfer thermal energy through a series of steps. Heat is transferred from to or from a space by terminal heat transfer devices, which transfer heat to or from a distribution system, which transfers heat to or from a cooling or heating plant. The heat transfer rates required at various points within the system can vary in time and magnitude. For example, thermal energy storage attenuates heat transfer through a system, and losses from ductwork and the need to heat or cool ventilation air require the plant heating or cooling load to be larger than the space heating or cooling load.

This chapter describes the principles underpinning heating and cooling loads generally, but focuses mainly on design load calculations for space heating and space cooling, which are typically intended to estimate the maximum rates at which thermal energy would ever need to be transferred to and from a space to achieve desired indoor thermal environmental conditions, and/or other performance metrics. The chapter also discusses factors that affect plant heating and cooling loads and system sizing, but whole system design is addressed with greater detail in other chapters. Similar principles can be used to estimate building energy consumption – the subject of Chapter 19. The main difference between design load calculations and building energy simulations is that the former facilitate the selection of design details (such as flow rates or supply temperature) based on conditions in a design period, whereas the latter estimate the heat transfer rates, indoor thermal conditions, energy use, and other performance metrics that a particular system would produce during a simulation period..

This chapter discusses the typical elements of heating and cooling load calculations. Section 1 provides an overview of the principles that govern the dynamics of heating and cooling loads, and provides definitions for terminology.. Section 2 provides practical guidance for how to prepare cooling load calculations, and explains how the concept of heating and cooling loads fits within a recommended conceptual system design procedure. Section 3 provides reference documentation for estimating heat gains and losses (e.g., internal heat gain, ventilation and infiltration, moisture migration, and fenestration heat gain). Then, the chapter describes two different methods for estimating cooling loads, and one method for estimating heating loads. Section 4 describes the heat balance (HB) method for cooling load calculations, Section 5 describes the radiant time series (RTS) method for cooling load calculations, and Section 6 describes a simplified method for heating load calculations.

1. COOLING AND HEATING LOAD PRINCIPLES

Cooling and heating loads depend on many complex dynamic heat transfer processes involving the environment, the building construction, its internal contents, internal heat sources and sinks, the heating cooling and ventilation systems, and their controls. These factors impact the timing and magnitude of space heating and cooling loads in the following ways:

Environment: Outdoor environmental variables including temperature, humidity, solar irradiance, wind speed and wind direction are significant periodic inputs to the dynamic heat balance for a space. Their diurnal patterns can have large impacts on the magnitude and timing of heating or cooling loads. For example, buildings often release heat to the environment passively overnight, thereby reducing space cooling loads for the following day.

Building construction: Building geometry and thermal properties of construction impact the magnitude and timing of heat gains and losses through outdoor exposed surfaces. These characteristics influence space heat gains from solar radiation, gains and losses by conduction to outdoors, and gains and losses by infiltration. The thermal properties of surfaces within and enclosing a space (whether or not they are exposed to outdoors) also impact the extent to which heat gains are absorbed, stored, and later released to the space (or to the environment), instead of immediately impacting the space cooling or heating load.

Internal factors: The magnitude and timing of heat gains from lights, people, appliances, and equipment have distinct impacts on space heating and space cooling loads. As discussed in Section 3.2 it is important to distinguish between convective heat gains, radiative heat gains and latent heat gains, because they each impact the timing and magnitude of space heating and cooling loads differently. The physical contents of a building can also have substantial impact on heating and cooling loads, especially because of the way they absorb, store, and release heat from gains.

Systems: The type of terminal heating and cooling devices used in a space impacts the magnitude and timing of space heating and cooling loads. Different terminal heating and cooling devices interact with the complex heat transfer network for a space in different ways, changing the extent to which heat is absorbed and stored in masses, as well the rates of conductive heat transfer through outdoor exposed surfaces. As a result, to achieve equivalent operative temperature and humidity conditions different systems require different space heat transfer rates.

System controls: The timing and magnitude of space heating and cooling loads are impacted by: the times at which heating or cooling systems are controlled to operate (i.e.: setpoint schedules), the way that parallel strategies are coordinated (i.e.: mechanical cooling and natural ventilation cooling), and the comfort range within which indoor operative temperature and humidity are allowed to drift over the course of a day (i.e.: *ASHRAE Standard 55* specifies that systems may allow indoor thermal environmental conditions to drift and change over the course of a day, as long as they are maintained within an acceptable range and with an acceptable rate of change.).

In light of these factors, it should be noted that accurate determination of space heating and cooling loads requires definition of the system and control strategy – just as it requires definition of the environmental conditions, building construction, and heat gain characteristics. Often, design space load calculations are made without a final definition of the heating and cooling system that will be used, so it is therefore important that inputs and assumptions for design load calculations reasonably reflect the system type and control strategy that is ultimately used..

1.1 Relationship Between Space Loads and Plant Loads

The design and control of systems also impacts the relationship between space loads and plant loads. For some systems the rate of heat transfer to or from a space translates almost immediately to equal loads for the heating or cooling plant. While for other systems, the thermal capacity of system components, transit time for thermal distribution, or active thermal storage delays the transfer of heat from one end of the system to the other, and spreads out heat transfer rates required by the heating or cooling plant. For example, when designed and controlled with these factors in mind the peak capacity of a cooling plant for a building with high thermal mass radiant cooling can be much smaller than the peak space cooling load – the plant can operate overnight, or extract heat from the slab slowly over a long period of time, while the actively cooled surfaces extract heat from the space more rapidly. Furthermore, leaks, thermal losses, and other factors cause plant heating and cooling loads to be larger than space heating cooling loads, and must be accounted for as part of design – these issues are discussed further in Section 3.7.

1.2 Relationship Between Heat Gains and Space Cooling Loads

Surfaces within and enclosing a space (walls, floor, furniture, etc.) absorb and store a portion of the thermal energy from heat gains to a space. Consequently, peak space cooling load on a particular day is generally smaller than the corresponding peak space heat gain rate, yet space cooling loads can be larger than space heat gains at other times, when heat gains subside and heat stored in surfaces is released to the space. As surfaces absorb heat their temperature increases, which impacts operative temperature in a space and shifts the balance of convective and radiant heat transfer between surfaces and the indoor air. As a result, as space heat gains increase, the space cooling load increases more slowly. In an adiabatic system the space cooling load would eventually increase to match the space heat gain rate, but the time scale for this thermal response in a real building is generally so long that it does not reach steady-state.

Figure A.1 illustrates the space cooling load for an air system to maintain constant indoor air temperature in an example south-facing space with typical internal gains, solar gains, and envelope heat transfer.

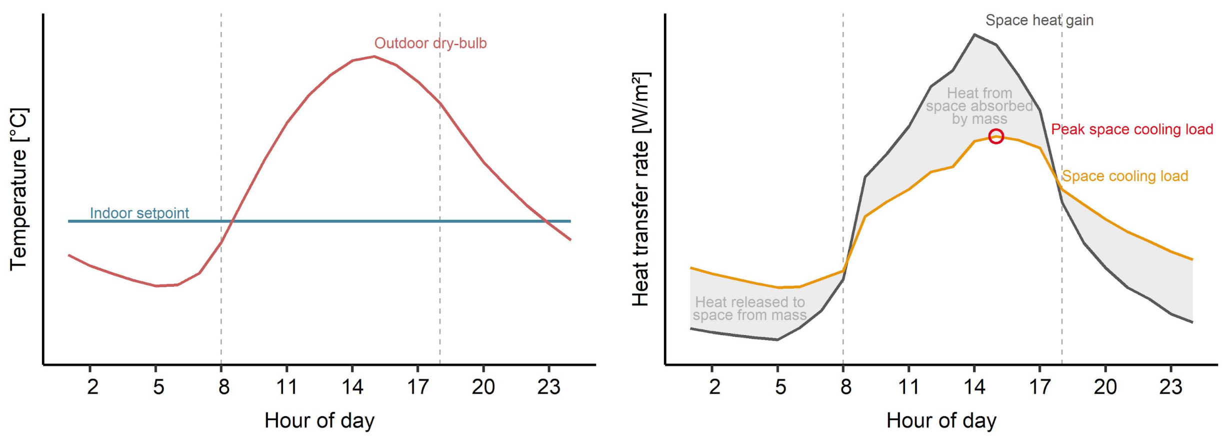

Figure A.1:The dynamic response for space cooling load in an adiabatic space..

This dynamic relationship between space heat gains and space cooling or heating loads must be considered when designing a cooling or heating system, and requires accounting for the complex heat transfer networks within the space.. Several mathematical methods may be used to model these effects; however, some methods are only accurate for particular scenarios. Cooling and heating load calculations can be performed with computer software; since each software may implement different methods, practitioners should carefully consider the assumptions and limitations associated with the software utilized. This chapter presents two mathematical methods: the Heat Balance method (Section 4), and the Radiant Time Series method (Section 5).

1.3 Sensible Heat Transfer Network

All of the factors affecting heating and cooling loads are dynamic, but their relationship to one another can be illustrated by the heat transfer network in Figure A.2. Each node in Figure A.2 represents a source of sensible heat entering a space, a physical element of the indoor thermal environment, or a route by which heat leaves a space. Each link in Figure A.2 represents a heat transfer pathway between two nodes, and can be characterized by a particular heat transfer mechanism (conduction, convection, or radiation), and a heat transfer rate. The network is arranged with all heat sources on the left and all heat sinks on the right (heat flows from left to right). Simply put, if the rate of heat entering a space outweighs the rate at which heat leaves a space, the temperatures of the indoor thermal environment will increase. The rate of heat entering and leaving a space need not be balanced at all times, as discussed previously, the space cooling load is generally smaller than the heat gain rate because surfaces can absorb and store a considerable amount of heat without exceeding the constraints of an acceptable indoor thermal environment. The space heating or cooling load at any moment is the rate at which terminal heat transfer devices must input or extract heat so that indoor thermal environmental conditions follow an acceptable trajectory and ultimately remain within desired constraints. If the actual space heat input or extraction rate does not match the heating or cooling load, the indoor thermal environment will change at an undesirable rate and may ultimately exceed the constraints of an acceptable indoor thermal environment.

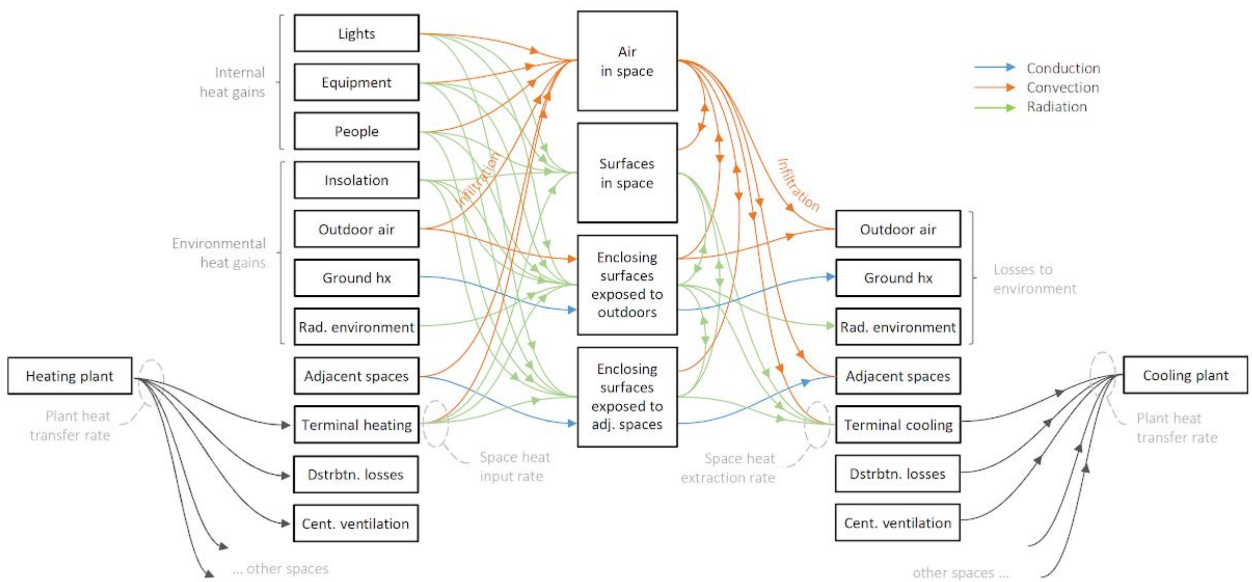

Figure A.2: Generalized sensible heat transfer network for a space, and the associated heating and cooling systems.

1.4 Terminology

Each of the heat transfer pathways indicated in Figure A.2 are intricately interrelated and their rates change with time. Since there are important differences between the timing and magnitude of each, practitioners should clearly differentiate between them using appropriate terminology.

Space Heat Gain Rate. The space heat gain rate is the rate at which heat is generated within and/or enters a space – except that which is intentionally added by heating systems. Heat gains are classified (1) as either sensible or latent; (2) by their source; and (3) by the heat transfer mechanism by which they enter a space (conduction, convection, or radiation). The boundary of a space includes the infinitesimally thin indoor face of surfaces that enclose a space. The space heat gain rate is not a net value – it does not account for the amount of heat lost from a space or stored in masses – it is simply a sum of the instantaneous heat addition to a space. As discussed further in Section 3, sources for space heat gain include (1) solar radiation through transparent surfaces; (2) conduction across outdoor exposed surfaces (floors, walls, and roofs); (3) conduction across surfaces that separate adjacent spaces; (4) convection and radiation across the boundaries that separate adjacent spaces; (5) heat generated within a space by occupants, lights, and equipment; (6) positive net heat transfer associated with direct-with-space ventilation and infiltration of outdoor air; and (6) miscellaneous heat gains.

Space Heating or Cooling Load:

The space cooling (heating) load at any point in time is the rate at which terminal heat transfer devices, with associated control sequences, must extract (input) sensible and/or latent heat such that associated thermal environmental conditions, and/or other performance metrics, comply with desired constraints during a design period (e.g.: limits on: operative temperature, ventilation rates, peak electrical demand, etc). *ASHRAE Standard 55* defines acceptable indoor thermal environmental conditions for human occupancy, and *ASHRAE Standard 62.1* defines ventilation for acceptable indoor air quality. The space heating or cooling load cannot be calculated for any point in time without context to thermal conditions at preceding times. As discussed previously, space heating or cooling loads are dynamic and depend on many factors including the system type and control strategy. Therefore, there may be more than one space heating or cooling load profile that satisfies desired constraints during a design period. Sensible and latent space heating and cooling loads must be accounted separately, yet with consideration for how they interact. For example, to generate sensible space heat extraction rates commensurate with sensible space cooling loads, a system may cause incidental latent cooling (dehumidification) that exceeds the amount of dehumidification that would otherwise be required. Consequently, the resulting total space cooling load is larger than the sum of the sensible and latent cooling requirements. This is one of many examples for how the system and control strategy influences space cooling heating loads. Importantly, the instantaneous space cooling load is not equivalent to the sum of all heat gains at the same time because surfaces within and enclosing a space absorb and store a portion of the heat gains (this is illustrated in Figure A.1).

Space Heat Input or Extraction Rate. The space heat input or extraction rate is the rate at which terminal heat transfer devices actually input or extract heat from a space. If the space heat input or extraction rate does not match the space heating or cooling load the indoor thermal environment will change at a rate different from the rate expected.

Plant (System) Heating or Cooling Load: The instantaneous plant (system) heating or cooling load is the rate at which a heating or cooling plant (or other point in the system) would need to transfer heat to or from the rest of a heating or cooling system in order to generate space heat input or extraction rates commensurate with the space heating or cooling loads. The plant heating or cooling load is not simply the sum of space heating or cooling loads. Practitioners must account for: heat gains and losses that occur outside of the space (e.g.: ventilation air, or duct leakage), diversity in timing and magnitude of aggregate space heating and cooling loads, the transit time for thermal distribution, system control sequences, and the dynamic thermal response of system components (e.g.: thermal energy storage, high thermal mass radiant slabs). Each of these factors decouple the timing and magnitude of space heating or cooling loads from that of the plant heating or cooling load.

Plant (System) Heat Input or Extraction Rate: The plant heat input or extraction rate is the rate at which a heating or cooling plant actually inputs or removes heat from a heating or cooling system. If the plant (system) heat input or extraction rate does not match the plant (system) heating or cooling load the space heat input or extraction rate will not match the space heating or cooling load.

Design Period (Space/System/Plant) Heating or Cooling Loads: Heating and cooling loads can be estimated for any point in time, but load calculations across heating and cooling design periods serve as the primary basis for selection and design of most heating and cooling systems. These design period heating or cooling loads usually represent the maximum rates at which thermal energy would ever need to be transferred to and from a space to achieve desired indoor thermal environmental conditions. However various design periods might be used to guide the design of systems that perform well across a variety of scenarios. The inputs used to define a design scenario – discussed further in Section 2.1 – may not represent regular operation; rather design period space heating and cooling loads are intended to bound the system design process. At the same time, a design scenario should represent realistic expectations, as overly conservative assumptions may lead to design and sizing of systems that are more costly than necessary or do not perform well under regular operating conditions.

Peak Design Heating or Cooling Load: While heating and cooling loads can be calculated for any point in time, and design period heating and cooling loads represent the dynamic loads during a design period, the peak design heating or cooling load is the single maximum heat input or extraction rate required during a design period. Commonly, the peak load is used as the basis for sizing the capacity of a heating or cooling system.

In summary, the terminology described here helps to differentiate between various heat flow rates that differ in time and magnitude. These terms can be combined logically to describe more detailed concepts associated with heating and cooling loads. For example, "cooling plant design period loads" are the heat transfer rates that would be required by a cooling plant during a design period to generate the space heat transfer rates commensurate with design period space heating or cooling loads. The same concepts may also be applied to intermediate points in a system. For example, the heat extraction rate for one hydronic zone in a high thermal mass radiant cooling system could be described as the "zone hydronic heat extraction rate", and this could be different from the associated "space heat extraction rate" because thermal capacity of the slab imposes considerable delay for heat transfer through the slab, and because a controlled hydronic zone may be associated with multiple spaces.

2. COOLING AND HEATING DESIGN LOAD CALCULATIONS IN PRACTICE

The affecting cooling load calculations are numerous, often difficult to define precisely, and always intricately interrelated. Many cooling load components vary widely in magnitude, and possibly direction, during a 24 h period. Because these cyclic changes in load components often are not in phase with each other, each component must be analyzed to establish the maximum cooling load for a building or zone. A **zoned system** (i.e., one serving several independent areas, each with its own temperature control) needs to provide no greater total cooling load capacity than the largest hourly sum of simultaneous zone loads throughout a design day; however, it must handle the peak cooling load for each zone at its individual peak hour. At some times of day during heating or intermediate seasons, some zones may require heating while others require cooling. The zones' ventilation, humidification, or dehumidification needs must also be considered.

Load calculations should accurately describe the building. All load calculation inputs should be as accurate as reasonable, without using safety factors. Introducing compounding safety factors at multiple levels in the load calculation results in an unrealistic and oversized load.

Variation in heat transmission coefficients of typical building materials and composite assemblies, differing motivations and skills of those who construct the building, unknown infiltration rates, and the manner in which the building is actually operated are some of the variables that make precise calculation impossible. Even if the designer uses reasonable procedures to account for these factors, the calculation can never be more than a good estimate of the actual load. Frequently, a cooling load must be calculated before every parameter in the conditioned space can be properly or completely defined. An example is a cooling load estimate for a new building with many floors of unleased spaces for which detailed partition requirements, furnishings, lighting, and layout cannot be predefined. Potential tenant modifications once the building is occupied also must be considered. Load estimating requires proper engineering judgment that includes a thorough understanding of heat balance fundamentals.

Perimeter spaces exposed to high solar heat gain often need cooling during sunlit portions of traditional heating months, as do completely interior spaces with significant internal heat gain. These spaces can also have significant heating loads during non sunlit hours or after periods of non occupancy, when adjacent spaces have cooled below interior design temperatures. The heating loads involved can be estimated conventionally to offset or to compensate for them and prevent overheating, but they have no direct relationship to the spaces' design heating loads.

Correct design and sizing of air-conditioning systems require more than calculation of the cooling load in the space to be conditioned. The type of air-conditioning system, ventilation rate, reheat, fan energy, fan location, duct heat loss and gain, duct leakage, heat extraction lighting systems, type of return air system, and any sensible or latent heat recovery all affect system load and component sizing. Adequate system design and component sizing require that system performance be analyzed as a series of psychrometric processes.

System design could be driven by either sensible or latent load, and both need to be checked. In a sensible-load-driven space (the most common case), the cooling supply air has surplus capacity to dehumidify, but this is usually permissible. For a space driven by latent load (e.g., an auditorium), supply airflow based on sensible load is likely not to have enough dehumidifying capability, so subcooling and reheating or some other dehumidification process is needed.

This chapter is primarily concerned with a given space or zone in a building. When estimating loads for a group of spaces (e.g., for an air-handling system that serves multiple zones), the assembled zones must be analyzed to consider (1) the simultaneous effects taking place; (2) any diversification of heat gains for occupants, lighting, or other internal load sources; (3) ventilation; and/or (4) any other unique circumstances. With large buildings that involve more than a single HVAC system, simultaneous loads and any additional diversity also must be considered when designing the central equipment that serves the systems. Methods presented in this chapter are expressed as hourly load summaries, reflecting 24 h input schedules and profiles of the individual load variables. Specific systems and applications may require different profiles.

This chapter presents two load calculation methods that vary significantly from previous methods. The technology involved, however (the principle of calculating a heat balance for a given space) is not new. The first of the two methods is the **heat balance (HB) method**; the second is **radiant time series (RTS)**, which is a simplification of the HB procedure. Both methods are explained in their respective sections.

Cooling load calculation of an actual, multiple-room building requires a complex computer program implementing the principles of either method.

2.1 Definition of Design Scenarios

Regardless of the mathematical method used to calculate design heating or cooling loads, designers must specify the inputs that define a design scenario. These inputs and assumptions have substantial influence on the result of design heating and cooling load calculations; they should be defined realistically, yet may also incorporate assumptions that intentionally introduce a margin of safety to ensure that the resulting heating or cooling loads represent the maximum rates at which thermal energy would ever conceivably need to be transferred to and from a space. It is often useful to conduct load calculations for various scenarios, to guide the design of systems that perform well for maximum conceivable loads, while also performing well on typical days and at low load conditions. In addition to load calculations during design periods, annual building system simulations can also be exceedingly useful for system design, whether they are performed to estimate energy use – the topic of Chapter 19 – or to predict and evaluate other aspects of system performance.

Generally, the following information must be specified to define a design scenario.

Site Location Information. Most load calculation methods model solar heat gains as a dynamic interaction between climatological data, a solar position model, site characteristics, the building geometry, and building construction thermal properties. Therefore, the design scenario must include information about site latitude, longitude, altitude, and orientation, as well as the specific range of time for which load calculations will be performed. Additionally, information such as local terrain roughness, external shading, or external reflectance help to estimate local outdoor conditions based on climatological data from a meteorological station. For example, external objects such as adjacent buildings, water, or parking lots may reflect solar radiation and increase direct solar gains to a space.

Building Characteristics. Not surprisingly, design load calculations must be based on information about the building geometry and thermal properties of building construction. Each load calculation software tool may require different level of detail, but generally information is required to describe: material thermal properties, the layered configuration of materials to form construction surfaces (opaque and transparent), the geometry and arrangement of construction surfaces to form spaces, the relationship to adjacent spaces, outdoor surfaces, and the outdoors environment, as well as the thermal properties and geometry of surfaces within these spaces. Important thermal properties of materials include: spectral absorptivity, reflectivity, transmissivity and emissivity, surface roughness, thermal conductivity, specific heat capacity, and density. Additional input requirements may include information to estimate the effects of thermal bridging, or corner heat transfer effects. Some software tools are setup to allow users to input common construction ratings for particular construction surfaces, such as SHGC and U-factor; in which case it is important to ensure that surface area definitions comply with standard rating procedures (as discussed in *ASHRAE Fundamentals Chapter 15: Fenestration*).

System Characteristics. Calculation of space heating or cooling loads requires definition of the terminal heating or cooling devices that will provide heat transfer with the space. Different terminal heating and cooling devices interact with the complex heat transfer network for a space in different ways, and therefore impact the timing and magnitude of space heat transfer rates required to achieve desired operative temperature and humidity conditions. Many design load calculation procedures presuppose that terminal cooling devices are idealized air systems that transfer heat by convection and mix air perfectly throughout a space. This assumption is appropriate in many cases, but will not accurately represent space heating and cooling loads for other systems including: displacement systems, underfloor air distribution systems, and radiant systems. Furthermore, calculation of plant heating or cooling loads requires definition of all sources of heat transfer throughout a system, including the terminal heat transfer devices in multiple spaces, losses in distribution, and ventilation heating and cooling requirements that occur outside of the space.

Outdoor Conditions. Time series climatological data is required for the site location during the design period, and typically must include: outdoor dry-bulb temperature, outdoor humidity ratio, direct solar irradiance, diffuse solar irradiance, wind speed, wind direction, and opaque sky cover percentage or horizontal infrared radiation intensity. There are many resources for historical, typical, and projected future climatological data. *ASHRAE Fundamentals Chapter 14: Climatic Design Information* provides climatological information for many locations, and discusses other data resources. Regardless of the source, designers should conscientiously appraise the appropriateness of available data, including consideration of potential differences between conditions at the source meteorological station and conditions likely at the project site. For example, dry-bulb temperatures or wind speeds in an urban area may vary considerably from measurements by a nearby rural meteorological station.

It is essential that designers choose an appropriate period to guide design load calculations. The peak sensible space cooling load often occurs during periods of peak outdoor dry-bulb temperature, or periods of peak solar gains through fenestration – which often occur in cool months with low solar altitude. However, because of combined sensible and latent loads, the peak plant cooling load can occur during periods of peak wet-bulb temperature.

Indoor Thermal Environment and System Controls. The instantaneous space heating or cooling load is the rate at which terminal heat transfer devices would need to input or extract heat so that indoor thermal environmental conditions comply with desired constraints (*ASHRAE Standard 55* defines acceptable indoor thermal environmental conditions for human occupancy). Therefore, definition of a design scenario requires specification of these desired constraints for the thermal environment. These may include minimum and maximum limits for all aspects of the thermal environment (air temperature, operative temperature, humidity ratio, etc), as well as limits on the rate of change for each, all of which might change in time as a function of other dynamic variables. Traditionally, design load calculations have specified constant indoor air temperature as the only constraint – an assumption that is absolutely integral to some load calculation procedures. However, modern objectives for design of high-performance building systems – such as the ability to actively shift electric demand – necessitate that design load calculations be guided by more liberal constraints.

Moreover, it is not sufficient to schedule an allowable envelope for indoor thermal conditions, a design scenario must also specify the control strategy used by a system. This is important because strategies like setback during vacant periods, or pre-cooling to avoid operation during peak electric demand periods, have substantial impact on the magnitude and timing of space cooling loads. Notice that definition of these constraints and controls are not the same as specifying the exact indoor thermal conditions that shall occur; rather they serve as inputs for a model to predict what indoor thermal conditions would occur and the corresponding space heat transfer rates that would be required by systems. For example, the design scenario for a system that employs pre-cooling would not specify the exact indoor thermal conditions that would occur; rather, it would specify a control strategy that precools to a particular setpoint, then load calculations would estimate how the indoor thermal conditions evolve in response to heat gains, control behaviors, and system characteristics. Often, load calculations specify very simple control strategies – such as a constant indoor air temperature setpoint; although this assumption is acceptable for design of many systems, it is not sufficient for some systems and controls. For example, high thermal mass radiant systems cannot be controlled to maintain constant indoor air temperature, and load calculations that impose such a simplifying assumption would overestimate the space heat extraction rates required by a radiant system to maintain acceptable indoor thermal comfort.

Internal Heat Gains. A design scenario must specify the magnitude, schedule, and characteristics of internal heat gains. Internal heat gains include heat from people, lights, and equipment (appliances, processes, etc), located within a space. As discussed in Section 7, these heat gains may have sensible and latent components, and the sensible part may enter the space as conduction, convection, or radiation. Some models differentiate between long-wave and short-wave radiant gains. All models must include some method to estimate the distribution of radiation between surfaces in a space – some methods simply assume uniform distribution across all surfaces. Direct-to-space ventilation should be scheduled as an internal heat gain, but heat gains from ventilation handled by central air handler should be accounted for in air handler system load calculations instead of space load calculations (as discussed in Section 7). Some load calculation methods calculate solar gains and infiltration rates as dynamic interactions with the environment, while others require these heat gains be scheduled as part of the design scenario.

2.2 System Design Procedure

1. **Describe all building and site characteristics and uncontrolled input values for a design period(s)**
    a. Define site location and meteorological information for a design period(s), including: (i) site latitude and longitude, (ii) outdoor air temperature and humidity, (iii) wind speed and direction, (iv) direct and global horizontal solar irradiance.
    b. Define building characteristics, including: (i) building geometry, (ii) construction thermal characteristics, (iii) internal and external shading devices, and (iv) envelope air tightness characteristics.
    c. Define site characteristics that impact building heat transfer, including: (i) shading by external objects (e.g.: trees and buildings), and (ii) reflection from external surfaces (e.g.: adjacent buildings, ground, water bodies)..
    d. Define internal heat gains for a design period(s) of interest, including heat gains from: (i) people, (ii) lighting, and (iii) equipment – this step must account for diversity in the timing and magnitude of internal heat gains in different spaces.
    e. Define other known heat gains (losses) to the space including: (i) infiltration, (ii) direct to space ventilation.
2. **Describe performance objectives and constraints for the design period(s)**
    a. Define performance priorities for the design, which may include balancing multiple objectives such as achieving acceptable: (i) life cycle cost, (ii) energy cost, and/or (iii) life cycle greenhouse gas emissions.
    b. Define constraints on performance metrics, which may include: (i) allowable range for air or operative temperature, (ii) allowable range for predicted mean vote (PMV), (iii) minimum required ventilation during occupied periods, (iv) maximum pollutant concentrations during occupied periods, (v) maximum peak electrical demand.
3. **Describe system design variables, controlled input variables, and control strategy**
    a. Define all terminal heat transfer devices, which may include: (i) sensible cooling (heating) devices, (ii) dehumidification devices, and (iii) sources of direct-to-space ventilation (including natural ventilation systems).
    b. Define all other factors and components associated with a space that may be controlled to influence performance (such as thermal comfort, or indoor air quality) which may include: (i) ceiling fans, (ii) personal comfort systems, (iii) sources of ventilation, (iv) air cleaning devices, (v) occupant adaptive behaviors.
    c. Define all cooling (heating) systems within the scope of design, which may include: (i) distribution systems, (ii) air handlers, and (iii) cooling (heating) plants.
    d. Define all heat gains and losses from the cooling (heating) system, which may include: (i) duct leakage, (ii) fan heat, (iii) distribution losses.
    e. Define a sequence of operations for all controlled devices in a system (and occupant behaviors), which may include: (i) system operating schedules, (ii) feedback control loops and controlled variables, (iii) temperature setpoint schedules, and (iv) adaptive occupant responses.

4. **Simulation and design iteration**
    a. Perform simulation of the building and systems model for the design period(s), and output values for any metrics needed to assess performance of the systems designed. This requires that designers utilize modeling tools capable of predicting these performance metrics, for the systems and controls to be designed, with accuracy that is appropriate for the scope and phase of design.
    b. Compare simulation results to the performance constraints (defined in step 2.b).
    c. Iterate on design definition and simulation (steps 3–4) so as to best achieve desired performance objectives (defined in step 2.a) subject to performance constraints (defined in step 2.b). Reject design alternatives that do not satisfy performance constraints, and choose among satisfactory design alternatives to best satisfy performance objectives.

3. HEAT GAINS AND LOSSES

As illustrated by Figure A.2, sources for sensible (and latent) space heat gain include: (1) solar radiation through transparent surfaces; (2) conduction (and moisture diffusion) from outdoor exposed surfaces; (3) conduction (and moisture diffusion) from interior partitions; (4) conduction (and moisture diffusion) from other surfaces within a space; (5) sensible heat (and moisture) generated in the space by occupants, lights, and appliances; (6) sensible heat (and moisture) associated with direct-to-space ventilation, infiltration, or transfer air from adjacent spaces; and (7) miscellaneous heat gains.

Sensible (and latent) heat losses from a space include: (1) conduction (and moisture diffusion) to outdoor exposed surfaces (floors, walls, and roofs); (2) conduction (and moisture diffusion) to interior partitions; (3) conduction (and moisture diffusion) to other surfaces within a space; (4) sensible heat (and moisture) associated with direct-to-space ventilation, infiltration, or transfer air from adjacent spaces, (5) miscellaneous losses, such as short-wave and long-wave radiative losses through transparent surfaces.

Section 3 provides standard reference information for estimating these different components of heat gain and loss – except for conduction into and out of surfaces within and enclosing a space, which is calculated differently by each of the cooling and heating load calculation methods described in this chapter. Section 3.1 provides information about the calculation of solar heat gains (a subject covered in greater detail by Chapter 15), Section 3.2 provides information about different sources of internal heat gains, Section 3.3 provides information to estimate gains and losses by infiltration (a subject covered in greater details by Chapter 16), Section 3.4 addresses latent gain from moisture diffusion through surfaces, and Section 3.5 addresses latent heat gains from other sources.

3.1 Solar Heat Gains

See existing *ASHRAE 2017 Fundamentals Chapter 18* section "Fenestration Heat Gain"

3.2 Internal Heat Gains

Internal heat gains from people, lights, motors, appliances, and equipment can comprise the majority of the space heat gains in a modern building. While building envelopes have improved in response to more restrictive energy codes, internal heat gains have increased because of factors such as increased use of computers and the advent of dense-occupancy spaces (e.g., call centers).

People

Table A.1 gives representative rates at which sensible heat and moisture are emitted by humans in different states of activity. A portion of the sensible heat emitted by people is transferred to air in the space by convection, and a portion is transferred by radiation to surfaces within and enclosing a space – both components are counted as space heat gains. In high-density spaces, such as auditoriums, these sensible and latent heat gains comprise a large fraction of the total heat gain to a space. Even for short-term occupancy, the extra sensible heat and moisture introduced by people may be significant. See ASHRAE Fundamentals Chapter 9 for detailed information; however, Table A.1 summarizes design data for common conditions.

Table A.1: Representative Rates at Which Heat and Moisture Are Given Off by Human Beings in Different Activities and Locations

Table 1 from *ASHRAE 2017 Fundamentals Chapter 18*

Electric Lighting

Electric lighting often comprises a major fraction of the total heat gain to a space. Most of the heat gain associated with lighting is produced by the lamps (the light-emitting elements), but a portion may be produced by ballasts and other appurtenances. A portion of the heat from each of these sources is transferred to air in the space by convection, and a portion is transferred by radiation to surfaces within and enclosing a space. The convective and radiant components from all parts of an electric lighting system are counted as space heat gains; however since electric lighting is often integrated into ceiling systems, a portion of the heat is transferred to the space, and a portion is transferred to the ceiling plenum. Therefore, the heat gain to a space from electric lighting can be calculated as:

$$Q_{L\text{-space}} = 3.41 \cdot W_{nom} \cdot F_{use} \cdot F_{space} \quad \text{(A.1)}$$

where

$Q_{L\text{-space}}$ = heat gain to space from lighting, Btu/hr {W}
W_{nom} = nominal lighting power, W
F_{use} = lighting use factor
F_{space} = space fraction
3.41 = conversion factor

The lighting use factor is the ratio of instantaneous lighting power consumption to the installed nominal lighting power. The factor may change in time to represent the pattern of use due to building operation, occupancy, and electric lighting needs. Design cooling load scenarios often set the lighting use factor to 1.0 to ensure cooling systems will be sized for a worst-case scenario.

The nominal lighting power is the rated electric power consumption of all lighting systems associated with a space, including lamps, ballasts, and controls. This can be estimated from values in Table A.2 – the maximum allowable lighting power densities (electric lighting power per square foot {meter}) specified by ASHRAE Standard 90.1 for different space types. Alternatively, when lighting plans and manufacturers technical information are available, the nominal lighting power can be calculated by:

$$W_{nom} = W_{lamp} \cdot F_{sa} \qquad \text{(A.2)}$$

where

W_{nom} = nominal lighting power, W
W_{lamp} = rated power of lamps, W
F_{sa} = special allowance factor

The rated power of lamps is obtained from manufacturer technical information for lamps, separate from their ballasts and controls. In some applications, the actual power consumed by lamps in operation may be smaller than their rated power..

The special allowance factor is the ratio of the lighting system's total power consumption, including lamps and ballast, to the rated power of the lamps. For incandescent lights, this factor is 1. For other types of lights, the factor can be greater than 1 to account for power consumed by the ballast, power supply, or controls. For example, metal halide and high-pressure sodium vapor lighting systems may have special allowance factors from about 1.3 (for low-wattage lamps) down to 1.1 (for high-wattage lamps)... The special allowance factor can be less than 1 when the ballasts used limit the power input to the lamps relative to their rated power. When available, use manufacturers' values for the nominal power of lighting systems, rather than estimating lamps and ballasts independently.

Table A.2

Table 2 from *ASHRAE 2017 Fundamentals Chapter 18*

The space fraction is the portion of heat emitted by a luminaire that is transferred to the space as either convection or radiation. For luminaires that are integrated into the ceiling, this faction is less than 1, and the remaining portion of the heat is transferred to the plenum above as convection. Table A.3 – composed of data from experimental research by Fisher and Chantrasrisalai (2006) and Zhou et al. (2016) – presents the space fraction for several types of luminaires installed in drop ceilings.

In addition to determining the space heat gain from lighting it is necessary to distinguish between the portion that is emitted to the space as convection, and the portion that is emitted as radiation. Accordingly, Table A.3 presents the radiant fraction for each type of luminaire studied. The radiant fraction is the portion of the heat transferred to the space that is emitted as radiation; the remaining portion is emitted as convection.

Table A.3

Table 3 from *ASHRAE 2017 Fundamentals Chapter 18*

The data in Table A.3 represents space fraction and radiative fraction for typical operating conditions: supply-to-return airflow rate of 1 cfm/ft² {5 L/(s·m2)}, supply air temperature between 59 and 62 °F {15 and 16.7 °C}, and room air temperature between 72 and 75 °F {22 and

24 °C}. and lighting power density of 0.9 to 2.6 W/ft2 {9.7 to 28 W/m2}. For design power input above this range, the lower bounds of the space fraction and radiant fractions should be used; for design power input below this range, the upper bounds should be used. Using values in the middle of the range yields sufficiently accurate results. However, values that better suit a specific situation may be determined according to the notes for Table A.3.

For a room with a non-ducted ceiling plenum return, the heat transferred from lighting systems to the plenum above the ceiling is basically transferred into the return air stream and therefore impacts heating and cooling loads for the air handler, but not space heating and cooling loads. Conversely, for a room with a ducted return, a large portion of the heat transferred from lighting systems to the ceiling plenum would eventually be transferred to the space by conduction through the suspended ceiling. Despite the difference in how heat transferred to the plenum ultimately impacts loads, the values in Table A.3 apply to luminaires in suspended ceilings whether the room uses ducted return, or non-ducted ceiling plenum return

If the space airflow rate is different from the typical condition (i.e., about 1 cfm/ft2) {[i.e., about 5 L/(s·m2)]}, Figure A.3 can be used to estimate the lighting heat gain parameters. Data shown in Figure A.3 are only applicable for the recessed fluorescent luminaire without lens.

Figure 3 from *ASHRAE 2017 Fundamentals Chapter 18*

Figure A.3

Although Table A.3 and Figure A.3 would accurately represent a vented luminaire with side-slot returns, they are likely not applicable for a vented luminaire with lamp compartment returns, because in the latter case, all heat emitted by convection is likely to go directly to the ceiling plenum. This would result in a much lower space fraction, and a radiative fraction of 1.

For luminaire types not listed in Table A.3, it may be necessary to use judgment to estimate each component of the heat emitted by lighting.

When using the radiant time series (RTS) method for cooling load calculations, note that because a major portion of the radiation emitted by downlight luminaires may be absorbed by the floor, it could be more appropriate to use the solar radiant time factors (RTFs) instead of the non solar RTFs. Solar RTFs are calculated assuming most solar radiation is absorbed by the floor, whilenonsolar RTFs assume uniform distribution by area over all interior surfaces. This effect may be significant for rooms where lighting heat gain is high and for which solar RTFs are significantly different from non solar RTFs.

This is the end of revisions drafted for *ASHRAE 2017 Fundamentals Chapter 18* as of May 29, 2020. The other sections in the existing chapter also deserve revisions. In particular:

1. Although Strand et al. (1999), Strand and Pedersen (2002), and Strand and Baumgartner (2005) have extended the Heat Balance Method to consider radiation heat transfer as a pathway for heat input to and extraction from a space, the explanations and mathematical representation of the method in *ASHRAE 2017 Fundamentals Chapter 18* only apply to all-air systems..
2. The existing section *Heating Load Calculations* outlines a simple method for sizing heating systems. It is an acceptable method, and it is used in practice, but it is only one approach that makes a lot of major assumptions. A fundamental representation of load calculations should allow the opportunity for more advanced methods, especially to facilitate design of high performance buildings. This section should remain in the chapter, but should be renamed "Simple Heating Load Calculation Method"
3. The existing section *Previous Load Calculation Methods* should be updated to identify the changes made in this revision. In particular, this revision introduces a flexible and forward-facing approach for load calculations that allows for a variety of solutions, whereas the existing approach is backward-facing, highly constrained, and produces a singular – "ideal" – estimate of space cooling load.
4. The existing section *Example Cooling and Heating Load Calculations* should be updated to reflect the flexible forward-facing approach.

Otherwise, the major content of the *ASHRAE 2017 Fundamentals Chapter 18* could remain as is, with the following outline: