Thermal Comfort and Acoustic Quality in Buildings Using Radiant Systems

by

Caroline Karmann

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Committee in charge:

Professor Stefano Schiavon, Chair

Professor Gail Brager

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Abstract

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In the US, people spend about 90% of their time indoors. This long exposure to indoor conditions affects our well-being, performance and health. Design and operation of these spaces also impacts energy use in building which, in the US, accounts for 40% of primary energy use. With these dual challenges, researchers and building professionals seek design strategies to simultaneously address the challenge of indoor environmental quality (IEQ) and energy use. Radiant heating and cooling systems have the potential to achieve significant energy savings primarily due to the use of lower temperature differences between the space and the heating or cooling source. Compared to buildings with all-air systems, buildings with radiant systems have been commonly associated with increased thermal comfort but decreased acoustic quality. The concern of reduced acoustics is particularly the case with regard to massive radiant systems and the need to preserve heat transfer, thereby keeping radiant surfaces uncovered. Achieving improved IEQ is fundamental for the successful adoption of radiant technologies in buildings. This dissertation proposed to address thermal comfort and acoustic quality in spaces using radiant systems through two major questions:

- How do spaces with radiant systems compare to spaces with all-air systems in terms of thermal comfort and acoustic quality?
- Can the combination of free-hanging acoustical clouds and fans below a massive radiant ceiling address simultaneously thermal comfort, cooling capacity and acoustical performance issues?

As part of a larger research team I utilized literature review, occupant surveys, statistical analysis, and laboratory experiments of a market-ready solution to address these questions.

We performed a literature review to assess if there was existing evidence that radiant systems provide better, equal or lower thermal comfort than all-air systems. This review identified five studies that could not establish a preference between the two systems and three studies showing a preference for radiant systems. These studies used multiple methods to demonstrate their findings and, in addition, several types of all-air and radiant systems were tested. This limited number of available studies did not allow us to draw a conclusion about the effectiveness of radiant systems for thermal comfort.

Following this review, we conducted occupant surveys in buildings using radiant systems. We gathered responses from 1284 occupants (20 buildings) that we complemented with responses form 361 occupants (6 buildings) previously surveyed. We used an existing database to extract a subset of occupant responses from all-air buildings whose key characteristics match those radiant buildings. This comparison involved 3892 responses from 60 buildings total. All-air and radiant buildings have equal indoor environmental quality, including acoustical satisfaction (assessed for noise and sound privacy), with a tendency towards improved thermal comfort in radiant buildings. There is a 16% probability of temperature satisfaction superiority for occupants exposed to radiant systems.

In a third phase, we experimentally assessed the combined effect of free-hanging acoustic clouds and fans for an office room. Free-hanging acoustic clouds are intended to reduce acoustical issues of a radiant massive ceiling, yet, they will also reduce its cooling capacity. Fan-induced air movement can be used to compensate for the cooling capacity reduction and enhance thermal comfort. In the test configuration, we installed a ceiling fan between the clouds (blowing in the upward or downward direction) and small fans above the clouds (blowing horizontally) both at the ceiling level. The two types of fans were tested independently. The acoustical results showed that if the clouds covered 40-50% of the ceiling area, acceptable reverberation time (one of the metrics for acoustical quality) was achieved. The cooling capacity experiments conducted without fans showed that for 47% cloud coverage, the cooling capacity only decreased by 11%. The ceiling fan increased cooling capacity by up to 22% when blowing upward and up to 12% when blowing downward, compared to the reference case and over the different cloud coverage ratios. For variants with small fans, the cooling increases with coverage proving that the combination of a cloud and a small fan has a positive effect on cooling capacity. Elevated air motion in the occupied space can provide further advantages from a thermal comfort perspective for the ceiling fan variants. Combining fans with acoustical absorbents close to the radiant surface has the potential to increase cooling capacity while simultaneously providing improved acoustic quality.

In summary, the dissertation work has: (1) summarized the state of current research on thermal comfort for radiant systems compared to all-air systems; (2) developed and analyzed the largest database of occupant responses in buildings using radiant systems; (3) used occupant feedback to show that radiant and all-air buildings have overall equal indoor environmental quality, including acoustical satisfaction, but with a tendency towards improved thermal comfort in radiant spaces; and (4) experimentally tested two practical solutions of combined acoustic clouds and fans below a radiant chilled ceiling, herein showing that it is possible to overcome the limitation of ceiling sound absorption often associated with radiant slab systems.

Table of Contents

List of Figures

List of Tables

List of Symbols

Energy symbols

Symbol	Quantity	Unit
A	Area	m^2
c_p	Water specific heat capacity	$KJ \cdot kg^{-1} \cdot K^{-1}$
h_c	Convective heat transfer coefficient	$W \cdot m^{-2} \cdot K^{-1}$
h_r	Radiative heat transfer coefficient	$W \cdot m^{-2} \cdot K^{-1}$
q	*Heat transfer rate*	*W*
q''	Heat flux ($= q/A_{rad}$)	$W \cdot m^{-2}$
P	Power	W
$\dot{m}$	Water mass flow rate	$Kg \cdot s^{-1}$
U_{cc}	Cooling capacity coefficient	$W \cdot m^{-2} \cdot K^{-1}$
t	Temperature in ºC	°C
T	Temperature in K	K
t_a	Air temperature	°C
t_{op}	Operative temperature	°C
t_{mr}	Mean radiant temperature	°C
t_g	Globe temperature	°C
$t_{w,s}$	Water supply temperature	°C
$t_{w,r}$	Water return temperature	°C
$t_{w,m}$	Mean water temperature	°C
ΔT_w	Water temperature difference ($\Delta T_W = t_{w,r} - t_{w,s}$)	K
v_a	Air speed	$m \cdot s^{-1}$

Acoustical symbols

Symbol	Quantity	Unit
$A_{absorption}$	Equivalent sound absorption area	Sabin, m^2
c	Speed of sound	$m \cdot s^{-1}$
d	Decay rate	$dB \cdot s^{-1}$
T_{60}	Reverberation time	s
V	Volume	m^3

Acknowledgements

I often came across the comparison of a PhD with a journey that can often seem long and of which one may often wonder where it leads. This impression is felt all the more deeply as it coincides with a more personal journey, moving away from Europe to Berkeley, CA with much changes, losses but also wonderful surprises. This period has been marked by so much changes and so much uncertainties that there is no way I would have been able to make it without the permanent and extraordinaire support of many people.

I would like to thank Stefano Schiavon for his never-ending and generous support, and passion for our field. I am not sure what my experience at UC Berkeley would have been without Stefano's friendly supervision, patience, insightful feedback, and contagious optimism. Certainly this would have been a different story and I feel very fortunate for having been there at that point of time.

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I would like to thank Fred Bauman, my direct supervisor at the Center for the Built Environment (CBE) for being such an exceptional mentor and person. I am grateful for his daily support and uncountable time helping me writing, conducting experiments, advising me with his unique wisdom and generosity. I would like to thank Paul Raftery for his smart and passionate advices, and for the many beers, coffees and bibimbaps that supported all sorts of enjoyable discussions. I would like to thank everyone else at the CBE for their kindness, colorfulness and dedication, for making this group be such a supportive community with beautiful celebrations.

I would like to thank everyone who was involved, initiated ideas, influenced or directly participated in my research. In particular: Cathy Higgins and Kevin Carbonnier from NBI, Gwelen Paliaga from TRC, and Lindsay Graham, Max Pittman, Sheila Shin, Gabriela Dutra De Vasconcellos, Veronika Földváry, and Prabhmeet Randhawa from UC Berkeley for their support with the survey study; Wilmer Pasut from Eurac Research, Yuanda Cheng from Taiyuan University of Technology, and Hui Zhang from CBE for conducting the preliminary study the radiant, acoustic clouds and fan project; and Mike Koupriyanov, Harmanpreet Virk and Jared Young from Price Industries, William H. Frantz and Kenneth P. Roy from Armstrong World Industries, and Thomas Lesser from Big Ass Fans for their in-kind and material support, and their tremendous help with the laboratory studies.

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I would like to thank everyone who helped me make this journey possible, from the very beginning, my friends from Europe who helped me embrace this incredible trip and never doubted about it. A particular thanks goes to Nadir, KC, Anna, Josh, Pierre-Yves, Thomas, Charlotte, and so many many others including my previous colleagues and friends at Transsolar, who pushed me to do my best.

I would like to thank my parents and my sister for their love and for all these hours spent online skyping rather than in real. I wished we would have been able to share meals rather than screens.

Beyond anyone, I would like to thank Burak, his love and his beautiful soul. I would not have made it without his infinite support. This was not my journey, but ours, and more broadly our life here in Berkeley, with all its ingredients, its colors, its dreams, its wonders and its struggles. Today, words are missing to express the extend of my gratitude and the depth of my feelings.
I will never forget. I will love you forever.

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1 Introduction

In the US, people spend nearly 90% of their time indoors (Klepeis et al. 2001). This long exposure to indoor conditions affects our well-being, performance and health. Design and operation of these spaces also impacts energy use in buildings which, in the US, accounts for 40% of primary energy use (US Department of Energy 2011). With these dual challenges, researchers and building professionals seek design strategies to simultaneously address the goal of achieving high levels of indoor environmental quality while minimizing energy use.

Climate change and predicted scarcity of energy resources have pushed governments to set aspirational goals towards Zero Net Energy (ZNE) buildings. In the U.S., the zero net energy target is set for 50% of commercial buildings by 2040 and for all commercial buildings by 2050 (US Congress 2007). More locally, the California Public Utilities Commission has an energy action plan to achieve zero net energy for all new commercial construction by 2030 (CPUC 2011). Heating, cooling and ventilation are responsible for 27% of overall building energy consumption (US Department of Energy 2011). Focusing on these systems offers a great potential for energy reduction.

Radiant heating and cooling systems are thermally controlled surfaces that exchange heat mainly through thermal radiation. Within the larger family of radiant systems, low-exergy hydronic systems operate at relatively low-temperature for heating and high-temperature for cooling. These systems have the potential to achieve significant energy savings primarily due to the use of lower temperature differences between the space and the heating or cooling source (Babiak, Olesen, and Petras 2009). Radiant systems are seen as a market ready alternative to conventional all-air systems to help achieve up to 50% reduction in primary energy use in buildings (Thornton et al. 2009, 2010; Leach et al. 2010)

Indoor environmental quality refers to the quality of a building's environment in relation to the health and wellbeing of those who occupy space within it (NIOSH CDC 2013). In practice, IEQ is commonly defined based on four tangible categories: thermal comfort, air quality, acoustic quality, and lighting quality (ANSI/ASHRAE/USGBC 2009). Salary costs in commercial buildings greatly exceed investment and operational expenses (Bendewald et al. 2014), leading to an economic incentive for improving indoor conditions in buildings.
A large occupant survey study conducted in 351 buildings and 52,980 observations has shown high levels of dissatisfaction with temperature, noise level and sound privacy (see Figure 1). This survey study was conducted mainly in the U.S., Australia and Canada, where all-air systems are largely dominating the market. Achieving improved indoor environmental quality is fundamental for successful adoption of any new technologies in buildings. New technologies should anticipate and address known issues in buildings.

Radiant systems have been commonly associated with increased thermal comfort but decreased acoustic quality. Researchers and practitioners have brought multiple arguments to support these claims. For thermal comfort consideration: reduced air movement and draft problems, active control of mean radiant temperature (MRT), more homogeneous conditioning provided to the space, positive influence on the human 'body-exergy' balance, and enhanced comfort for floor systems due to highest view factor to the occupants. Acoustical considerations include: exposed hard surfaces (acoustically reflective) that come in direct conflict with acoustic absorption area, and reduced ventilation background noise (sound masking) causing issues of sound privacy. While these arguments are reasonable, they often lack clear evidence on how thermal and acoustic conditions are actually experienced by building occupants exposed to radiant systems.

This dissertation focuses on questions of thermal comfort and acoustic quality in spaces using radiant systems. With the growing interest for radiant systems, we want to learn how they impact occupant satisfaction in buildings and investigate solutions for enhanced indoor quality in spaces using these systems.

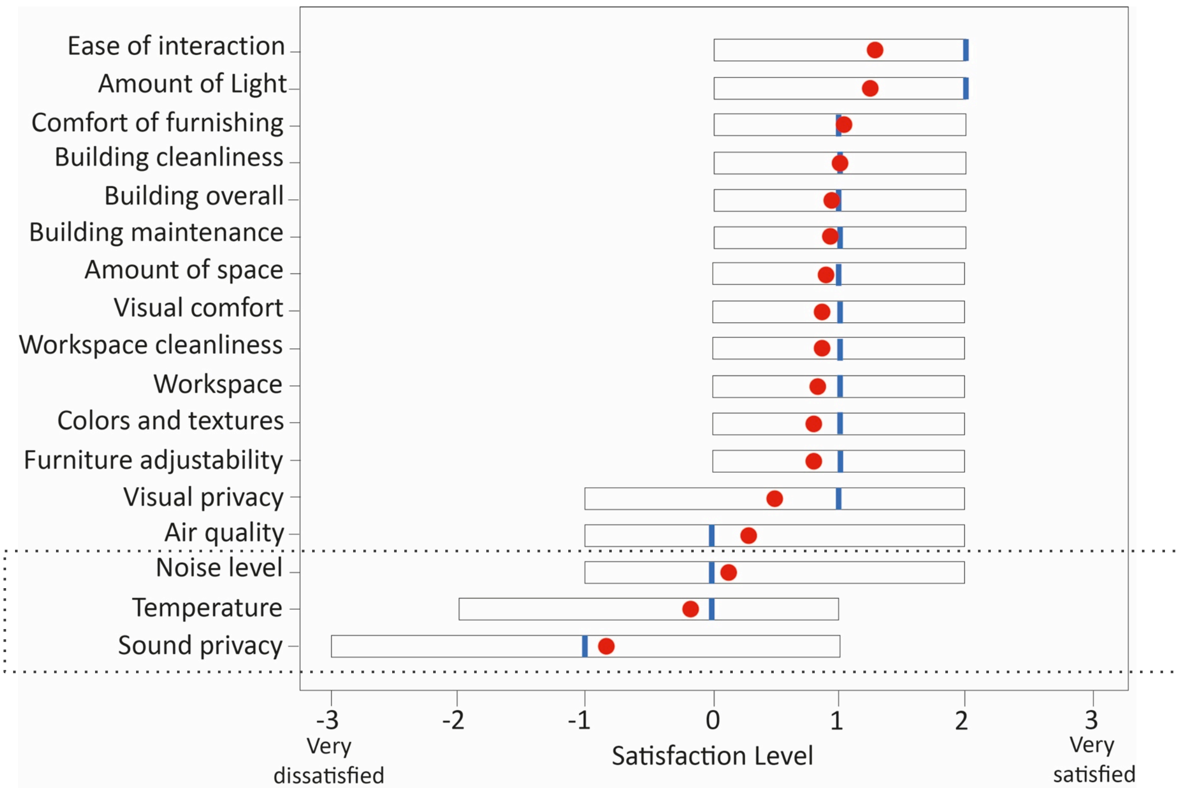


Figure 1: Results of the CBE IEQ Occupant Satisfaction Survey based on 351 buildings and 52,980 occupants. Source: (Frontczak et al. 2012)

1.1 Radiant systems

Historical background

The first forms of radiant systems emerged in Northern China and Korea for heating purposes. They are represented by *kang* (larger bed-stove), *dikang* (variant of kang that would cover the entire floor -‘di’ means floor) and korean *ondol*, depending on the region and time period. Archaeological excavation conducted at a Neolithic building remains of Xinle in Shenyang (c.5300-4800 BC) revealed that some floors dried by repeated heating, turned into baked clay, suggesting the possible use of heated floor very early on (Guo 2005). More affirming archeological work documented Korean *ondol* from the 11th century B.C. The same principle was independently used by the Romans since the 3rd century B.C. for baths. These systems were all based on the same principle of using combustion gases as the heated medium. Their architecture consists of three parts: a fireplace outside of or below the room, a double floor that allows the flow of the hot combustion gases, and a chimney on the opposite side of the room (see Figure 2).

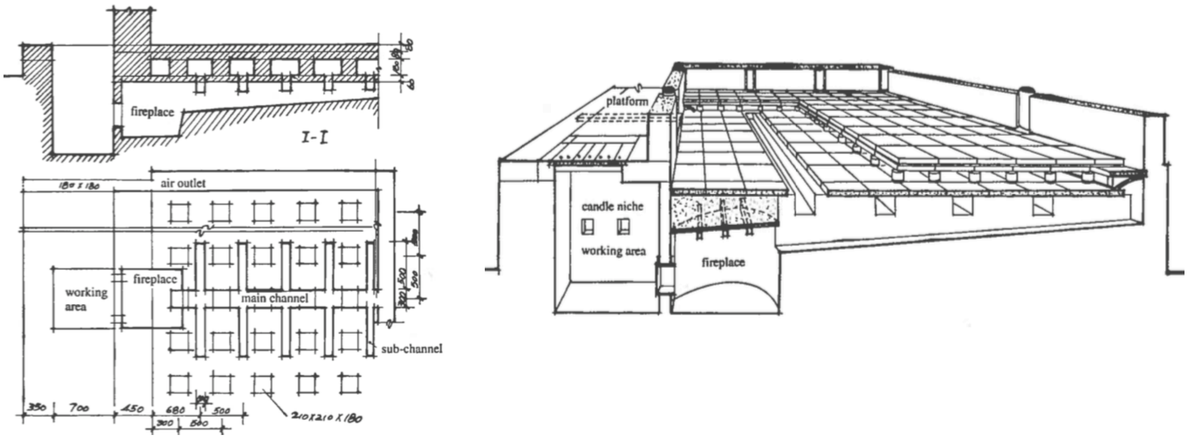


Figure 2: The structure of dikang: plan and section of the Jingyi Xuan Building (left) and axonometric drawing after Zhang Yuhuan *Zhongghuo Gidai Jianzshu Kishu* (right) (Guo 2005)

Radiant systems have since largely evolved. Most changes occurred during the 20th century. The earliest type of radiant system in the U.S. was a poured-in-place concrete floor with embedded copper piping that circulated warm water to heat the space above. This system was installed in residential constructions, and was first introduced to the U.S. market by Frank Lloyd Wright after he experienced it in Japan. Yet, the growth of this technology remained limited, primarily due to fears of pipe failures. Radiant floor heating became more popular in the late 1980s when a new type of plastic tubing (PEX) became widely available.

Hydronic radiant systems use water as the circulating heat transfer fluid. Other types include air-based and electrical systems (which use electrical resistance for heating purpose only). Large, exposed areas of building surfaces are usually required for the radiant exchange. Hydronic radiant heating and cooling systems, specifically in commercial buildings, are the focus of this thesis. For simplicity, I will often refer to those as ‘radiant systems’. While the early examples of radiant systems were focused on heating applications, radiant systems can be used as well for cooling applications (they would require a more careful design and operation due to humidity concerns and risk of condensation on cold surfaces). Radiant systems are not evenly distributed around the world. They are more commonly adopted in Europe but their design and application are still considered in development in the U.S. (Thornton et al. 2009). Besides this geographical disparity, there are variations in terms of type used and conditioning mode; Olesen (2012) observed that thermally activated building systems are not yet widely distributed in the U.S. and Asia, and according to Tian and Love (2009), radiant cooling applications are still rare in North America.

Radiant systems definition and classification

Radiant heating and cooling systems are defined as thermally controlled surfaces that exchange at least 50% of their heat through thermal radiation (ASHRAE 2012a). Unlike all-air systems, radiant systems utilize all three types of heat transfer mechanisms:

- **thermal radiation**: transfer of heat through electromagnetic waves emitted from one object with greater energy intensity to an absorbing object with less energy intensity (by definition over 50% of the heat exchange)
- **thermal convection**: transfer of heat through the movement of molecules within a liquid or gas, resulting in a current or flow of energy (most of the remaining heat exchange)
- **thermal conduction**: transfer of heat through the contact of molecules in directed connected objects (in case of contact between the heated/cooled surface and the human body)

There are multiple types of hydronic radiant systems. Based on standards and guidelines, we can identify four main families, and multiple sub-types, of radiant systems, illustrated in Figure 3.

(1) **Radiant panels**, where the pipes are attached to metal panels which are fixed to the construction by means of hangers (Babiak, Olesen, and Petras 2009; ASHRAE 2012a). The heat carrier is close to the surface. This type of system generally covers ceiling applications.

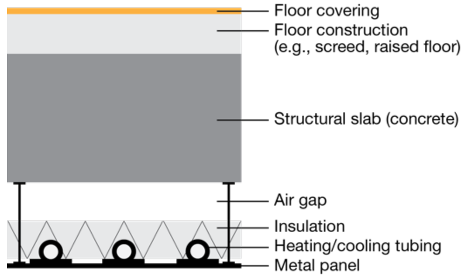
(2) **Embedded surface systems (ESS)**, where the pipes are embedded in the outer layer of the slab/wall, but are insulated from the structure. This family of systems generally covers floor applications, for example in a topping slab. Depending on their construction details, the EN 15377 (CEN 2008) / ISO 11855 (ISO 2012) standards distinguished 5 types of systems that fall within this family of radiant systems:

- *Type A* with pipes embedded in the screed or concrete ("wet" system)
- *Type B* with pipes embedded outside the screed (in the thermal insulation layer, "dry" system)
- *Type C* with pipes embedded in the leveling layer, above which the second screed layer is placed
- *Type D* include plane section systems (extruded plastic / group of capillary grids)
- *Type G* with pipes embedded in a wooden floor construction

(3) **Thermally activated building systems (TABS)**, where the pipes are embedded in a massive concrete slab/mass (the structure). This type is also referred to as *Type E* in EN 15377 (CEN 2008) / ISO 11855 (ISO 2012). As they are integrated within slabs, TABS are both associated with floor and ceiling applications. Nevertheless, pipes are often positioned toward the lower side of the slab while the upper side includes additional services (such as underfloor ventilation). In such cases, TABS would be dominantly associated with ceiling applications.

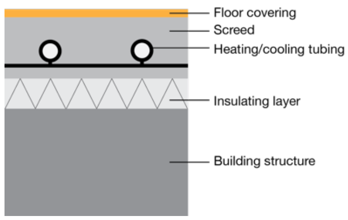
(4) **Capillary systems**, that are made out of a pre-fabricated mat made of thin plastic tubing embedded in a layer at the inner ceiling or as a separate layer in gypsum. This type is also referred as *Type F* in the EN 15377 (CEN 2008) / ISO 11855 (ISO 2012). Capillary system covers ceiling and wall applications.

Radiant panels

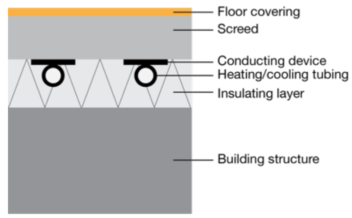


Section diagram of a radiant panel

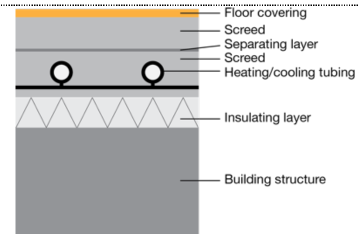
Embedded surface systems (ESS)



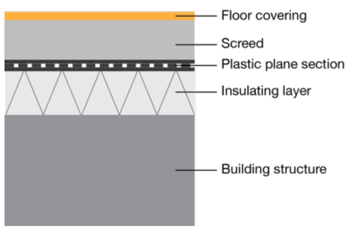
Section diagram of a radiant ESS (ISO 11855, type A)



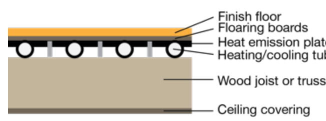
Section diagram of a radiant ESS (ISO 11855, type B)



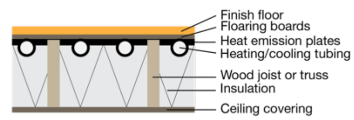
Section diagram of a radiant ESS (ISO 11855, type C)



Section diagram of a radiant ESS (ISO 11855, type D)

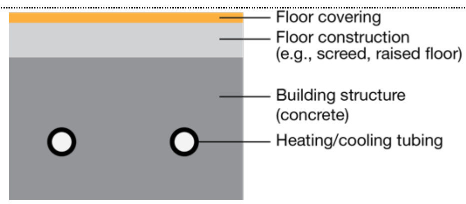


Section diagram of a radiant ESS (ISO 11855, type G)



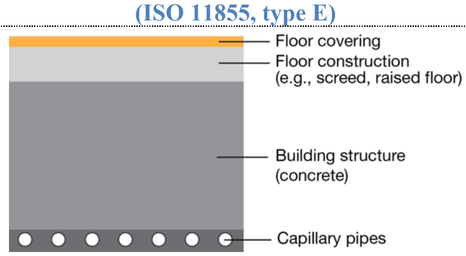
Section diagram of a radiant ESS (ISO 11855, type G variant)

Thermally Active Building Systems (TABS)



Section diagram of thermally activated building system (ISO 11855, type E)

Capillary systems



Section diagram of radiant capillary system (ISO 11855, type F)

Figure 3: Classification type of radiant systems

Applications

Because they are detached from structural slabs of buildings, radiant panels are probably the most common type of radiant systems. They are usually made out of metal tubing and, depending on their configuration, they can integrate acoustical absorbents, and in some cases other building services such as fire protection, return air ducts, and lighting. Radiant panels are decoupled from the thermal mass and therefore they tend to respond faster to a change in settings on the supply side. Their application covers a large range of buildings (e.g., offices, schools, laboratories and other commercial buildings), but they tend to be rare for residential applications. ESS were the first type of hydronic radiant systems. They were (and are still) most popular for residential applications. Meanwhile, we also see these systems for both heating and cooling applications in office environments (e.g., IDeAs Z2 design facility in San Jose, see Figure 11) or within large spaces such as transportation terminals, hotel lobbies or malls (e.g. Suvarnabhumi Airport in Bangkok, ARTIC intermodal transit center in Anaheim, Hearst Tower in New York, Walmart malls). TABS is a more recent type of radiant system. They were implemented for the first time in 1990 in Horgen, Switzerland for the Dow Headquarters whose mechanical systems was designed by Meierhans and Partners (Meierhans 1993). TABS have been slowly penetrating the European market but they remain the least common type of radiant system in North America. A recent seminar conducted with practitioners revealed that they represented about 2% of radiant projects in U.S. (TC 6.5 participants 2016). The U.S. market remains dominated by dropped ceilings, that allows for more flexibility and opportunity for reconfiguration. TABS requires the pipes to be built within the structural slabs. Therefore, these systems are not a common solution for retrofits.

Low-energy systems

Hydronic radiant systems operate at relatively low-temperature for heating and high-temperature for cooling. These systems are also referred to as 'low-exergy' systems having the potential to achieve significant energy savings primarily due to the use of lower temperature differences between the space and the heating or cooling source (Babiak, Olesen, and Petras 2009). This allows for higher efficiency at the device that generates hot or cold water, such as a boiler or chiller. In some cases, the relatively lower chilled water temperatures may eliminate the need for a chiller altogether, thereby enabling the use of non-refrigerant cooling sources, such as ground water, cooling towers, or indirect evaporative coolers. The use of water as the circulating fluid instead of air also reduces the energy costs of transferring heat to or from the space. Lastly, in the case of massive systems, the thermal inertia inherent in a radiant slab system also provides opportunities for peak demand reduction and load shifting (Meierhans 1993; Feustel and Stetiu 1995; B. W. Olesen et al. 2006; Lehmann, Dorer, and Koschenz 2007; Gwerder et al. 2008; Babiak, Olesen, and Petras 2009)

Architectural relevance

A key asset of hydronic radiant heating and cooling systems lies in its integration within building surfaces. Water pipes are hidden within the ceiling, floor or walls or within built-in panels. When implemented as ESS or TABS, they become invisible. Entire surfaces are being activated without noticeable change in their appearance. They provide comfort to spaces, thereby covering a primary function of architecture. This feature has attracted renown architects in the early stages of these system's development. Frank Lloyd Wright used radiant floors (ESS) for many of his houses in the first part of the 20th century and Peter Zumthor used TABS for the Kunsthaus in Bregenz (see Figure 4 and Figure 5) as its second application worldwide.

Radiant systems offer opportunities to re-think energy use in buildings in a synergistic way. In that sense, these systems allow more holistic narrative on sustainability. Primarily, radiant surfaces are heat exchangers. The heat contained in its circulating fluid moves from warmer to colder sinks with the goal to condition living spaces. Comfort can only be guaranteed at a relatively low temperature difference between the space and the circulating fluid. Therefore, the circulating fluid never gets too warm or too cold, offering greater opportunities for the re-use of waste energy or use of local renewable resources. Sanaa's Zollverein communicates this concept well. The building is located above a closed mining site of the Ruhr region of Germany. To prevent mining galleries from getting filled with ground water, a series of pumps are permanently taking out the water from the mining shafts to the Emscher river. This waste water is at a temperature of approximately 35ºC all year-round, which is close to an ideal temperature to operate radiant systems. Considering this resource, the building incorporated radiant TABS system in its slabs but also in its exterior walls (see Figure 6 and Figure 7). The whole envelope became active and, considering this free resource, designers decided not to add insulation and to keep the walls very thin considering the scale of the building. This climate and energy concept is unique and could only be considered in combination with its local and free energy resource.

Although Sanaa's Zollverein is a rather extreme example of synergies, radiant conditioning remains an appropriate system for enhanced design integration. This quality is in part related to the constraints radiant systems bring. Compared to all-air systems, radiant systems are limited in terms of capacity (the fluid temperature is bounded for comfort and health reasons and the surface area is limited to what's available). This limitation requires designers to pay greater attention to building orientation, envelope and internal loads (e.g., lighting). Further, radiant systems do not need high-grade heating and cooling sources and therefore they give designers an opportunity to find more innovative energy sources, such as geothermal or lake/sea water (e.g. Exploratorium in San Francisco). While such energy concepts obviously require a stronger engineering knowledge, the narrative is also appealing from a designer's perspective: it is a holistic story of design integration and energy use that may root the building in its context.

Radiant systems also possess a fascinating character that can drive many creative re-appropriations. The analogy between radiant systems in buildings and blood vessels within the human body is a common theme that confers radiant systems with an intrinsic organic quality (see Figure 8). In either case, a fluid is being pumped throughout a system to exchange energy. Without this movement, comfort may be lost and the building loses one of its primary functions. The fact that radiant systems are invisible is all the more intriguing. There is a certain abstraction to a simple building surface that can be activated through the effect of heat. Further, radiant systems cover the three types of heat transfer (radiation, convection and conduction) which triggers the potential for thermal experiences and added narratives. This theme is particularly present in the work of Philip Rahm (see Figure 9 where the polarization in the space caused by cold and warm thermal surfaces triggers experiences, migrations and seasonal movements within the house). These narratives are sometimes more impregnated by the poetic power than they are an accurate description of physical phenomenon. And yet, they generate fascination and genuine interest from designers on radiant technologies.

For all these reasons, radiant systems are generally appreciated by architects. They open doors to new opportunities for bridging between fields, with enhanced integration of energy efficient systems and emerging creative narratives.

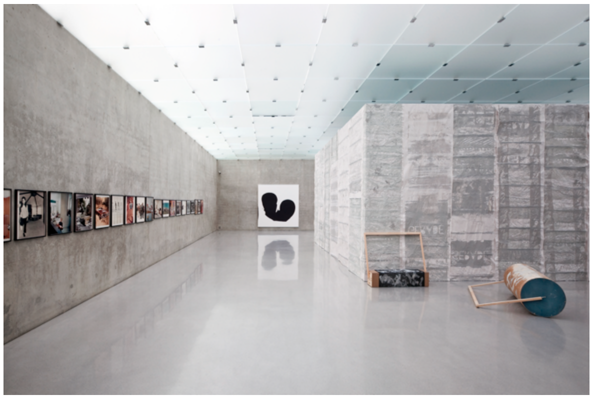


Figure 4: Inside view of the Kunsthaus Bregenz in Austria by Peter Zumthor (architect) and Meierhans+Partners (MEP). Image source: www.contemporaryartdaily.com

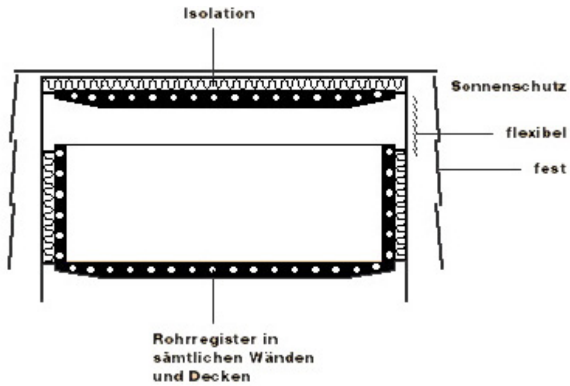


Figure 5: Section diagram of the Kunsthaus Bregenz with the location of the radiant embedded TABS, inside and outside view. Image source: Helene Binet

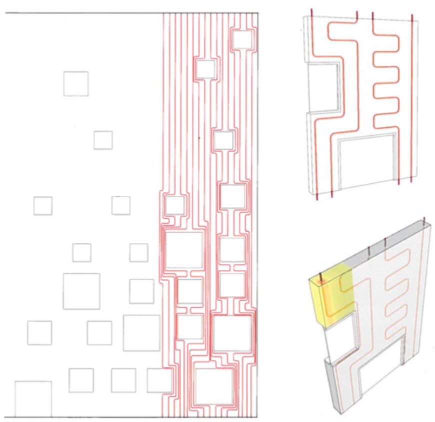


Figure 6: Sanaa Zollverein – Radiant system. Image source: Kiel Moe

Figure 7: Sanaa Zollverein – Wall section – Image: http://www.arcspace.com

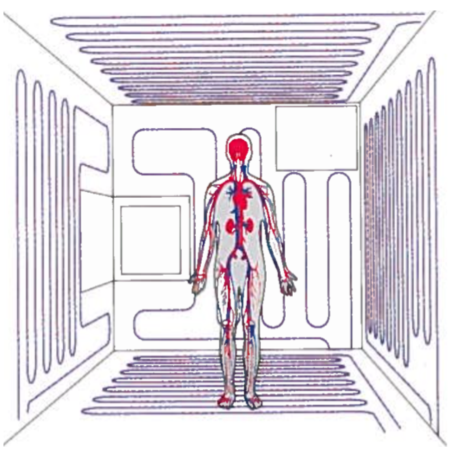


Figure 8: Kiel Moe's Human body analogy

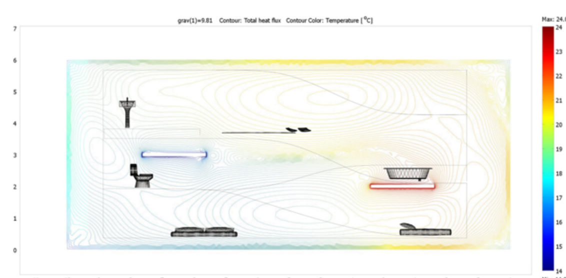


Figure 9: Philippe Rahm's 'Interior Gulf Stream' that uses two radiant surfaces to rethink functions- Housing and studio for Dominique Gonzalez-Foerster

Radiant systems of this study

Aside from the architectural icons mentioned above, we can find radiant systems in more common commercial buildings. This thesis focus on thermal comfort and acoustic quality in building using radiant heating and cooling systems in commercial buildings. The pictures below illustrate some of the spaces we studied.

Figure 10: Space using TABS: office in David Brower Center, Berkeley, CA - WRT-Solomon E.T.C. - Image: Tom Griffith

Figure 11: Space using ESS: office in IDeAs Z2 design facility, San Jose, CA - EHDD Architecture - Image source: Wakely David

Figure 12: Space using radiant panels: office in Port of Portland, Portland, OR - ZGF Architects - Image: Nick Merrick, Hedrich Blessing

Figure 13: Space using radiant panels: classroom in Lane Community College, Eugene, OR - SRG Partnership Inc. - Image: Lara Swimmer, Steve Wanke

1.2 Thermal comfort

Thermal comfort is defined as the “condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation” (ANSI/ASHRAE, 2013). While this definition may appear at first intangible, building scientists have found ways to translate it towards quantitative metrics. The most common ones are the Predicted Mean Vote (PMV) / Predicted Percent Dissatisfied (PPD), which offer a high commodity by integrating a full set of physical measures: air temperature, mean radiant temperature, relative humidity and air speed (for a given activity and clothing level). This allows designers to predict comfort and building operators to rely on zone measurements.

Radiant systems have been largely studied in laboratory chambers. Most of these studies were oriented towards guidelines to avoid potential sources of discomfort: temperature asymmetry, vertical temperature gradient and floor surface temperature. The practical implications of these laboratory tests are included in thermal comfort standards (ANSI/ASHRAE 2013; ISO, EN 2005). Meanwhile, radiant systems have been implemented in a great number of buildings. These systems are generally believed to offer improved thermal comfort conditions compared to other air-based conditioning systems. The theoretical arguments put forward to justify the claims include: reduced air movement and draft problems (Stetiu 1999; Zhang and Niu 2003), active control of mean radiant temperature (MRT) (Simmonds 1996; Simmonds et al. 2000; Watson and Chapman 2002), more homogeneous conditioning provided to the space (Zhang and Niu 2003; Bozkır and Canbazoğlu 2004), positive influence on the human ‘body-exergy’ balance for both radiant heating and cooling cases (Shukuya 2009), and comfort for floor systems due to highest view factor to the occupants (Causone et al. 2010). Sometimes, researchers have referred to many of these arguments (Baker 1960; Boerstra, Opt Veld, and Eijdems 2000; Schmidt and Ala-Juusela 2004; Sattari and Farhanieh 2006; Moe 2010). While these justifications appear reasonable, they fail to provide clear evidence on improved thermal comfort for radiant systems based on field studies.

Per ASHRAE 55, the comfort zone is intended to achieve 80% satisfaction with thermal comfort. Yet, despite code compliance mechanisms and the tremendous energy consumption of HVAC systems, buildings fail at providing comfortable environment to most occupants. Surveys assessing long-term satisfaction (215 building, 34000 responses) has shown that only 11% of buildings provided satisfactory thermal comfort level to at least 80% of their occupants (Huizenga et al. 2006). This long-term satisfaction survey study was conducted with a poll representative of all-air buildings. To date we don’t know if radiant systems would be able to provide higher satisfaction.

While there are many reasons why an individual building might fail in providing thermal comfort, building scientists commonly refer to two main arguments (ASHRAE/CIBSE/USGBC 2010). First, building design and operation follows recommendations in standards that were developed in controlled laboratory experiments, and then applied universally across the globe, but these often do not adequately represent real building environments or their occupancies, and differences that might occur across different climates and cultures. Secondly, buildings are operated with minimal feedback about the comfort conditions they are providing (which can take many forms), and therefore, we don’t know how systems perform in buildings and facility managers have limited information in knowing how to respond. An additional reason for the high dissatisfaction with thermal comfort is the difference in perception between occupants within a same building or a same building zone. Personal comfort systems (e.g. foot warmer, desk fan, heated/cooled chair) and personalized control offer new ways to address individual differences.

The study of thermal comfort for radiant systems has followed a similar path as the one of all-air systems. With this dissertation, I would like to address the following questions:

- Do spaces using radiant system provide a better, lower or equal thermal comfort than spaces using an all-air systems?
- How does this compare to ASHRAE objectives?

1.3 Acoustic quality

The issue of space acoustical quality is commonly divided into two types of concerns: noise and sound privacy. Noise is defined as an unwanted, unpleasant and in some other ways distracting sound. Noise does not depend only upon the intensity or sound level. In that sense, it is not an objective fact. Sound privacy refers to both the freedom from intrusive noise (such as telephone ringing, footsteps) and the freedom from being overheard or overhearing others (speech privacy).

(Frontczak et al. 2012) identified acoustic categories (noise and sound privacy) as the greatest source of occupant dissatisfaction in offices (see Figure 1). This confirms the outcome of previous occupant-based studies (Abbaszadeh et al. 2006; Jensen and Arens 2005; Navai and Veitch 2003; Charles Salter et al. 2003; C. Salter and Waldeck 2006). Sound privacy is a particular concern in open space offices (Kim and de Dear 2013; Bradley and Gover 2004), which became widely adopted due to the higher flexibility and cost savings they allow. (Leather, Zarola, and Santos 2010) referred to many researchers in the fields of health and occupational psychology to show that exposure to noise is associated with a range of negative outcomes including: impaired physical health; poorer psychological health; impaired quality of life (e.g., disturbed daily activities and sleep disruption); and impaired language development, cognition, and learning in children. As a result, noise is associated with increased stress and a source of fatigue for occupants (Sundstrom et al. 1994; Evans and Johnson 2000; Witterseh, Wyon, and Clausen 2004).

Radiant systems are associated with heated/cooled surfaces that need to be exposed to allow the heat exchange with the space. To avoid compromising view factors (inherent to radiant systems), it is desirable to have the direct view between the occupants and the active surface unobstructed, which often comes in direct conflict with having acoustic absorption areas in the ceiling plane. Radiant systems are also associated with quieter air systems (reduced to only cover ventilation requirements). While this can at first be seen as positive for radiant systems, it appears that background noise from ventilation equipment may help towards sound privacy by masking conversation or other undesired sounds. Therefore, spaces with radiant systems are often pointed out for the lack of sound privacy they provide. In some cases, these spaces resort to sound masking strategies (Veitch et al. 2002) to cope with this issue (e.g. white noise generators, see Figure 14). Although these critiques on acoustic quality in spaces using radiant systems sound reasonable, they again fail to provide clear evidence of the actual effect radiant systems have on acoustic satisfaction.

Figure 14: Interior view of Manitoba Hydro Place (Image: (iiSBE Canada, 2014)) and white noise generator in the raised floor (Image: Leif Norman)

The design of buildings is mainly based on programmatic requirements, architectural qualities, construction constraints (structural), and now, energy compliance and sustainability. Yet, none of these concerns fully embrace the problem of architectural acoustics that are often relegated to secondary interests (Charles Salter et al. 2003; ASHRAE/CIBSE/USGBC 2010; GSA Public Buildings Service 2011). Considering acoustic dissatisfaction in offices (in general) as well as the added critique against radiant systems, part of my objective is to verify how buildings using radiant systems actually perform in terms of acoustic quality, and to investigate an acoustic solution compatible with a radiant slab system.

1.4 Free-hanging acoustical panels and ceiling fans

In 2013, the Center for the Built Environment (CBE) conducted a preliminary simulation study of free-hanging acoustical panels (see example Figure 15 left) combined with a ceiling fan. This solution was envisioned to address sound reflection and noise in spaces with exposed massive radiant slabs. The ceiling fan would cover two functions in the space: enhance convective heat exchange of the radiant slab and provide elevated air movement in the occupied zone for thermal comfort purpose. The simulation was based on computational fluids dynamics (CFD). The fan was modeled for both upward and downward directions (see Figure 15 right) and multiple variants of ceiling coverage were tested (see Figure 16). The results showed up to 54% increase of the cooling capacity compared to a baseline case with no panels and no fan.

The results appear extremely promising. Yet, they are based on a non-validated CFD simulation study and there is no published peer-reviewed article that fully describes the model, methods and results obtained. Further, such acoustic and thermal combinations would benefit from additional analysis and full-scale measurements to be validated. As part of my dissertation, I built on this research idea and conducted a laboratory testing to investigate the effect of free-hanging acoustic clouds and fans on cooling capacity, thermal comfort and acoustic performance. Such testing also offered me the opportunity to re-think ceiling coverage (toward more simple coverage layout) and fan integration.

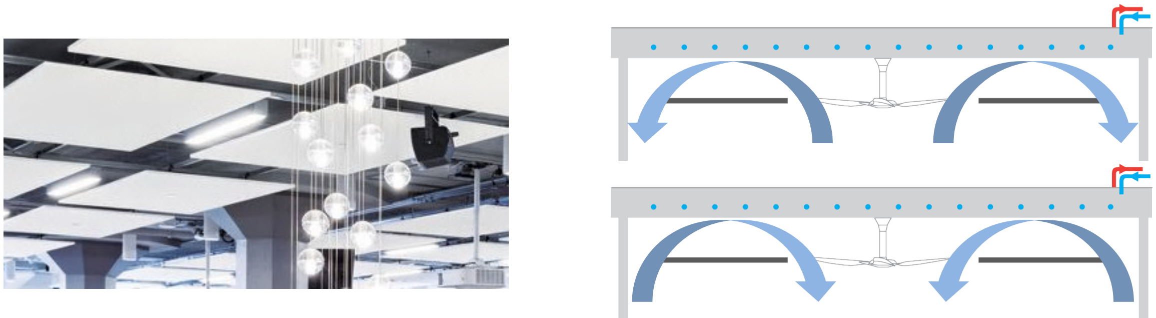


Figure 15: (Left) Example of free-hanging acoustic clouds (image: Armstrong World Industries); (Right) Section schematic of a ceiling with acoustic clouds and ceiling fan in upward and downward blowing directions

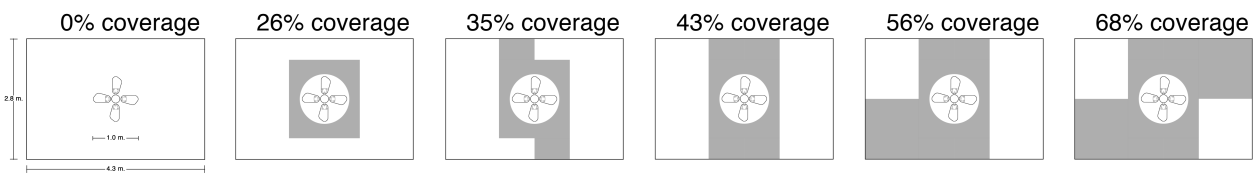


Figure 16: Plans of the ceiling with ceiling fan and different acoustic coverage going from 0% to 68% (in gray)

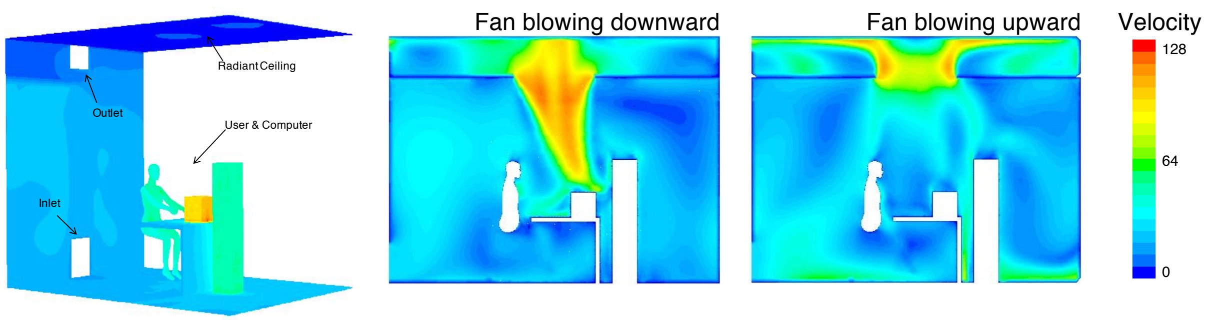


Figure 17: CFD visualization: rendering of the room with location of ventilation inlet / outlet (left) and air speed patterns on a section view for 68% coverage and ceiling fan in downward and upward and directions (center & right)

2 Objectives

Statement of purpose

In the US, people spend about 90% of their time indoors. This long exposure to indoor conditions has the potential to affect the well-being, performance and health of the occupants residing within a space. Design and operation of buildings also impact energy use in buildings which (in the US) accounts for 40% of a building's primary energy use. With these dual challenges, researchers and building professionals seek design strategies that simultaneously address the challenge of improving IEQ and reducing energy use. Radiant conditioning systems offer opportunities to achieve higher energy efficiency. These systems are different than the more conventional all-air systems in terms of their architectural integration, heat transfer mechanisms, control, and operation.

Radiant systems have been commonly associated with increased thermal comfort and decreased acoustical quality. However, there has been minimal scholarly evidence to support these assertions. Little is known on how radiant heating and cooling systems affect IEQ in buildings, especially from the perspective of perceived comfort from building occupants. Further, there are multiple types of radiant systems; among them, thermally activated building systems (TABS) are made out of exposed concrete, which creates acoustical challenges due to the high reflectivity of concrete. Free-hanging acoustical clouds and fans appear as a relevant combination to be added below TABS. The space above the acoustical cloud allows for increased air speed (higher convection) below the ceiling. Adding fans offers new opportunities to offset and increase the cooling capacity. Balancing building IEQ factors is important in the design of an effective workspace for the occupants, and so we need to consider the interactions between thermal and acoustical comfort and investigate new solutions.

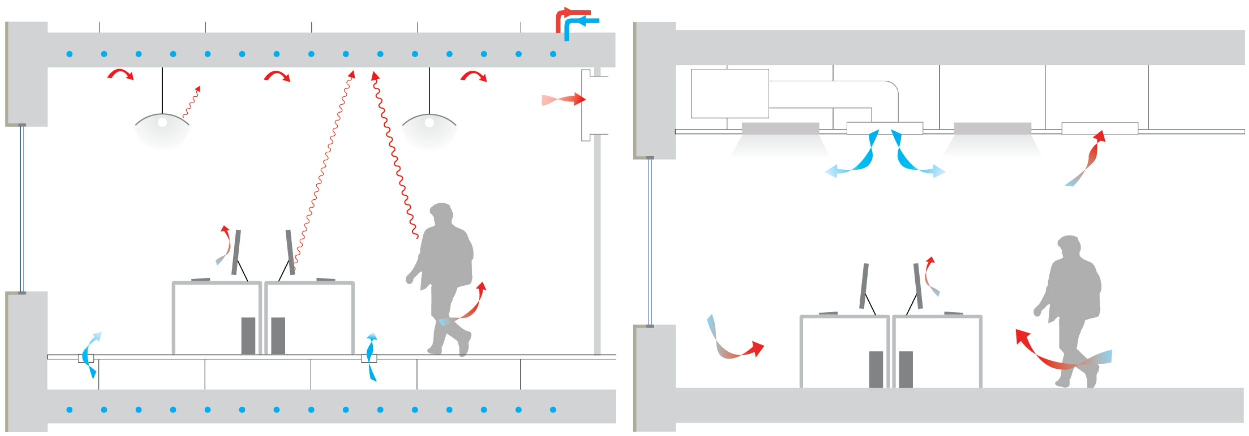


Figure 18: Sections of a room with radiant and all-air systems. The radiant system described (left) is a TABS supplemented by underfloor air distribution, and the all-air system (right) is an overhead/mixing ventilation system. The arrows represent the mode of heat transfer (radiation and convection)

Objectives

This dissertation addresses thermal comfort and acoustic quality in buildings using radiant systems. It is structured around two primary questions:

- How do spaces with radiant systems compare to spaces with all-air systems in terms of thermal comfort and acoustic quality?
- Can the combination of free-hanging acoustical clouds and fans below a massive radiant ceiling address simultaneously thermal comfort, cooling capacity and acoustical performance issues?

3 Overview

Methods

To answer my first question, I administered the CBE Occupant IEQ Survey in buildings conditioned by radiant systems. I then used the existing CBE occupant survey database to create a subset of responses from all-air buildings, against which I could statistically compare occupant responses from radiant buildings, focusing on the perception of thermal comfort and acoustic quality. This study was preceded by a critical literature review on thermal comfort for radiant compared to all-air systems.

For my second question, I followed up on the preliminary simulation study of free-hanging acoustical clouds and fans below a radiant chilled ceiling. This combination is intended to provide sound absorption, increased convective heat transfer along the chilled ceiling and, depending on the variants, elevated air movement in the occupied zone. I led two laboratory experiments to test this solution. The experiments were conducted in the Hydronic Test Chamber at Price Industries in Winnipeg (Canada), and at the Armstrong World Industries Reverberant Chamber in Lancaster, PA (USA).

Organization of the dissertation

Chapter 4 presents a critical literature review of thermal comfort for radiant vs. all-air systems. After a brief description of the thermal comfort metrics commonly used for this comparison, I describe the relevance of the different research methods used in previous studies, and highlight the remaining research needs.

Chapter 5 focuses on occupant satisfaction with thermal comfort and acoustics in buildings using radiant compared to all-air systems. This study involved the collection of occupant IEQ survey responses from North American commercial and educational buildings using radiant systems. The analysis was conducted using multiple statistical methods including an assessment of effect size interpretation.

Chapters 6-7 describe the laboratory experiments on cooling capacity and acoustic performance of radiant slab systems with free-hanging acoustical clouds and clouds. Chapter 5 reports on the acoustic testing results and interpretation alongside the radiant cooling testing, while Chapter 6 more specifically reports on the tests that involved air movement.

Chapter 8 (conclusions) summarizes the key findings, limitations and practical relevance of this work, and suggests directions for future work.

This dissertation includes content from four publications. Most of the content of chapters 3-6 is extracted from articles that are published, under review or in preparation. These articles were written using the plural pronoun "we". I decided to keep the plural form within this dissertation as it better reflects the valuable collaborations undertaken during my PhD. The list of these publications is added in Chapter 10.

4 Literature review on thermal comfort for radiant vs. all-air systems

4.1 Background

While the first examples of radiant systems (using hot air as medium) go as far back as the 11th century B.C. in Korea (Guo 2005), studies of these radiant heating and cooling applications in terms of thermal comfort waited until the 20th century. This interest started in Europe and was focused on subjective impressions of warmth and freshness (Munro and Chrenko 1949), on theoretical heat perceived at head level (Kollmar and Liese 1957), and on thermal sensation and skin temperature of feet for radiant floors (Chrenko 1957). The data from these studies were further analysed to define comfort requirements for both radiant floors and ceilings (Missenard 1955). This work on thermal comfort for radiant systems caught the attention of researchers at Kansas State University who then started an extensive research program on radiant systems for heated and cooled floors (Ralph G. Nevins and Flinner 1958; R. G. Nevins, Michaels, and Feyerherm 1964; Michaels, Nevins, and Feyerherm 1964; Springer et al. 1966; R. G. Nevins and Feyerherm 1967), and on the effect of asymmetrical conditions (McNall and Schlegel 1968; Schlegel and McNall 1968; McNall and Biddison 1970; McIntyre and Griffiths 1972; Griffiths and McIntyre 1974). In Denmark, Fanger and Olesen contributed to this effort by investigating the limitation of radiant systems at providing homogenous (isothermal) thermal environments (B. W. Olesen et al. 1972; P. O. Fanger et al. 1980, 1985). These studies were based on human subject testing in laboratory chambers and were oriented towards guidelines to avoid potential sources of discomfort identified as temperature asymmetry (see Figure 19), vertical temperature gradient and floor surface temperature. The practical implications of these studies are included in thermal comfort standards (ANSI/ASHRAE 2013; ISO, EN 2005).

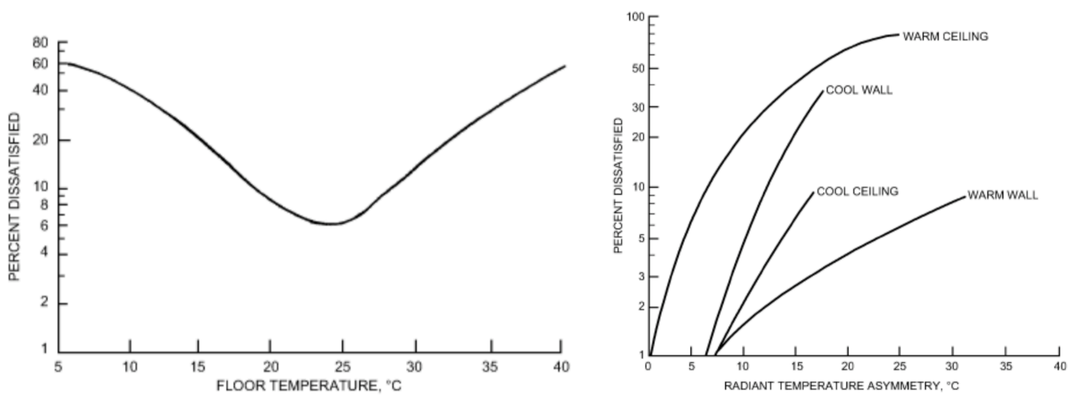


Figure 19: Percentage dissatisfied people defining floor temperature limits and radiant temperature asymmetry limits ((ANSI/ASHRAE 2013; ISO, EN 2005, 2001) same as (Fanger, 1985))

Researchers have used theoretical arguments to justify why radiant systems can provide better comfort than all-air systems. The main arguments are: reduced air movement and draft problems (Stetiu 1999; Zhang and Niu 2003), active control of mean radiant temperature (MRT) (Simmonds 1996; Simmonds et al. 2000; Watson and Chapman 2002), more homogeneous conditioning provided to the space (Zhang and Niu 2003; Bozkır and Canbazoğlu 2004), positive influence on the human 'body-exergy' balance for both radiant heating and cooling cases

(Shukuya 2009), and comfort for floors systems due to highest view factor to the occupants (Causone et al. 2010). Sometimes, researchers have referred to many of these arguments (Baker 1960; Boerstra, Opt Veld, and Eijdems 2000; Schmidt and Ala-Juusela 2004; Sattari and Farhanieh 2006; Moe 2010). While these arguments are reasonable, they fail to provide clear evidence on improved thermal comfort for radiant compared to all-air systems. Additionally, both radiant and all-air systems comprise a variety of types, strategies and design. Radiant systems need to be combined with a ventilation strategy to fulfil fresh-air requirements. All this adds complexity to the comfort assessment of the two systems.

This paper presents the results of a literature review on thermal comfort for radiant vs. all-air systems. After a brief description of the thermal comfort metrics commonly used for this comparison, we will detail the studies found based on the methods used. This review also questions the relevance of the methods used and highlights research needs. Beyond the comparison, we will address a few more concerns and findings regarding thermal comfort for radiant systems.

4.2 Methods

We performed a literature search using the key terms: "thermal comfort", "radiant systems", "hydronic systems", "thermo-active building systems", "thermally activated building components", "concrete core slab", "concrete core conditioning", "thermally activated building systems", "in-slab heating, floor surface radiant systems, radiant panel, low temperature heating and high temperature cooling systems", "chilled ceiling" and "water-based floor heating" in the following databases: Google Scholar and Web of Knowledge. We also used the reference sections of the papers we gathered to find additional publications. Selected proceedings, conference papers were also screened. We included only peer-reviewed articles and articles published in proceedings of scientific conferences. We excluded from our final selection all the publications that were based on grey literature or not comparing radiant to all-air systems.

We decided to classify the publications based on the research methods used: (1) building performance simulation (BPS), (2) physical measurements (in laboratory test chambers and in buildings) and (3) human subject testing / occupant based surveys. We use this classification scheme because it allowed us to distinguish simulated, measured and subjectively perceived comfort. When one article had more than one method, we decided to classify the publication based on the most robust method used (see discussion section for the comparison of the different methods).

4.3 Classification scheme

4.3.1 Review of the metrics used to assess thermal comfort

In ASHRAE Standard 55-2013 (ANSI/ASHRAE 2013), thermal comfort is defined as "that condition of mind which expresses satisfaction with the thermal environment". This definition brings forward the delicate question of which metrics can be used to assess thermal comfort. In this section we go over the comfort metrics that are relevant for our review. These include the metrics we came across during our literature review as well as the key comfort metrics used in radiant systems assessment. Metrics are classified into two categories: objective metrics (based on physical measurements) and subjective metrics (based on occupant feedback).

Objective metrics

One common way to quantify thermal comfort is through the measure of dry-bulb air temperature, globe temperature, mean radiant temperature (MRT) (derived from the globe temperature), and operative temperature (calculated using dry-bulb air temperature and MRT). The globe temperature also exists as 'half-globe' accounting for only half of the space.

The predicted mean vote (PMV) is a comfort model established to predict thermal sensation from "cold" to "hot" (Povl Ole Fanger 1970). This objective metric was developed using human subject testing in laboratory conditions and is based on a heat balance model applied to the human body. It uses six parameters: dry-bulb air temperature, MRT, air velocity, relative humidity, clothing level and metabolic rate and ranges from -3 (cold) to +3 (hot) with the value of 0 set as neutral. This metric has been translated into a predicted percentage of dissatisfied (PPD). Highest thermal comfort (i.e. lowest PPD) is associated with a neutral body sensation (PMV of 0).

The range of indoor conditions (e.g., temperature dead band or PMV/PPD values) can be used to characterize thermal comfort. EN ISO 7730 (ISO, EN 2005) and EN 15251 (CEN 2007) are using this method to define three categories of thermal requirements for mechanically cooled buildings: category I (or class A) (PPD < 6%, i.e. -0.2 < PMV < +0.2), category II (or class B) (PPD < 10%, i.e. -0.5 < PMV < +0.5) and category III (or class C) (PPD < 15%, i.e. -0.7 < PMV < +0.7). To account for time, EN ISO 7730 and EN 15251 propose frequency of exceeding a given category in percent time. There is some debate about the interpretation of these categories as aligned with levels of thermal comfort quality (Humphreys 2005; Nicol and Wilson 2011). Large field experiments have shown that the tightly air-temperature-controlled space (class A) did not provide higher acceptability for occupants than non-tightly air-temperature-controlled spaces (class B and C) (Arens et al. 2010). Based on these arguments, ASHRAE 55 (ANSI/ASHRAE 2013) did not include the classification of building categories. Yet, this metric was often used in the papers found for this review.

We found four local discomfort factors that are particularly relevant for radiant systems:

(1) Radiant asymmetry is defined as difference between the plane radiant temperature of the two opposite sides of a small plane element (P. O. Fanger et al. 1980). It is usually measured using half-globes to compare temperatures of two opposing surfaces of a room. Both EN ISO 7730 (ISO, EN 2005) and ASHRAE 55 (ANSI/ASHRAE 2013) define limits of radiant asymmetry when using radiant walls, floors and ceilings. These limits originate from (P. O. Fanger et al. 1985) and are based on a percent dissatisfied curve.

(2) Floor temperature that may be too low or too high can cause discomfort. Therefore, international standards have defined intervals of recommended temperatures based on a percent dissatisfied curve. EN ISO 7730 (ISO, EN 2005) and ASHRAE 55 (ANSI/ASHRAE 2013) specify limits for rooms occupied by sedentary or/and standing people wearing shoes. Both standards recommend floor surface temperatures within the occupied zone to be kept between 19°C to 29°C.

(3) A high vertical air temperature difference between head and ankles (stratification) can cause discomfort. (B. W. Olesen, Schøler, and Fanger 1978) have established a correlation between vertical air temperature difference between head and ankles (PDvertical) that has been further spread through the EN ISO 7730 (ISO, EN 2005) and ASHRAE 55 (ANSI/ASHRAE 2013). This metric only applies for head temperature being higher than feet temperatures (people are less sensitive under opposite conditions).
(4) Draft is defined as an unwanted local cooling of the body caused by air movement. (Povl Ole Fanger et al. 1988) developed a draft model using three variables (air temperature, mean air velocity, and turbulence intensity). Based on human subject testing this model was converted into percentage of dissatisfied for draft (PDdraft). This index is further defined within the EN ISO 7730 (ISO, EN 2005) but it has been removed from ASHRAE 55 because it was found to overestimate the draft risk (Toftum et al. 2003).

Another discomfort metric related to non-steady-state thermal environments is the 'temperature drift'. This metric is defined as a steady, non-cyclic change in operative temperature of an enclosed space. Temperature drift is associated with discomfort and is reported in [K/h]. Standard EN ISO 7730 (ISO, EN 2005) allows a maximum drift of 2 K/h. ASHRAE Standard 55 (ANSI/ASHRAE 2013) allows for 2.2 K/h for drift duration of 1 hour, but not more than 2.6 K/h during any 0.25 h period within that 1.0 h period. ASHRAE 55 also requires drift lasting 4 hours to be reduced to 0.8 K/h.

Human physiological measurements could also be taken. In our review, we found laboratory studies focusing on the different body part temperatures. These measurements are usually done according to standards (e.g., EN ISO 7726 (ISO, EN 2001)). Physiological measurements may also include core body measurement using an ingestible telemetry pill. These measurements can be used as input to detailed comfort models (such as the Advanced Thermal Comfort Model (Huizenga, Hui, and Arens 2001)), which can then be used to predict local and overall thermal comfort.

Subjective metrics

Thermal sensation vote (TSV) is a scale to rate thermal sensation from "cold" to "hot". Vote refers to human subjects filling out a thermal sensation scale during the exposure to certain thermal conditions at a given point in time. This metric was used to develop the PMV index and is sometimes referred to as 'actual mean vote'. Most researchers use a continuous 7-point ASHRAE interval scale going from -3 (cold) to +3 (hot) with the value of 0 set as neutral (ANSI/ASHRAE 2013). TSV can be conducted for whole body (global) sensation as well as for local sensation. The latter allows a comparison with the physiological measurements of local body parts.

Thermal comfort vote (TCV) is a scale to rate thermal comfort from "uncomfortable" to "comfortable". This vote requires human subjects or building occupants to fill out a thermal comfort scale. We commonly find this metric in right-now survey (at a given point in time) or background surveys (in general). This vote is commonly set on the ISO-defined 4-point scale ("uncomfortable", "slightly uncomfortable", "slightly comfortable", "comfortable"), where the value of 0 is unavailable (ISO 1995). We however found in our review a publication using a 5-

point scale that included a "neutral" comfort vote (Imanari, Omori, and Bogaki 1999). TCV can be conducted for whole body (global) as well as for local body parts.

Occupant satisfaction votes are often conducted in the framework of indoor environmental quality (IEQ) surveys in buildings. Typical questions on thermal comfort include satisfaction with temperature and self-reported performance/productivity in relationship to temperature. These surveys usually use 5- or 7-point scales ranging from "(very) dissatisfied" to "(very) satisfied" and with the value of 0 set as neutral (e.g., CBE Occupant IEQ Survey (Zagreus et al. 2004)). Additional subjective metrics on thermal comfort include thermal preference and thermal acceptability. Yet we did not come across these metrics in our review.

4.3.2 Conditioning systems classification

The terminology used for the various radiant and all-air systems is not always consistent across publications. In order to ease and better compare the systems, we decided to use a common classification scheme based on the current standards and construction of the systems.

Types of radiant systems

By definition, radiant systems provide at least 50% of the total sensible heat flux for space conditioning by thermal radiation. We looked at the international standard ISO 11855 (ISO 2012), the European standard EN 15377 (CEN 2008), the ASHRAE Handbook on HVAC Systems and Equipment (chapter 6) (ASHRAE 2012a) and the REHVA guidebook (Babiak, Olesen, and Petras 2009). Based on these standards and guidelines, we identified three main types of radiant systems: (1) radiant panels, where the pipes are attached to metal panels which are fixed to the construction by means of hangers (Babiak, Olesen, and Petras 2009; ASHRAE 2012a); (2) embedded surface systems (ESS) where the pipes are embedded in the surface of the slab/wall, but are insulated from the structure (EN 15377 / ISO 11855, type A, B, C, D, G), and (3) thermally activated building systems (TABS), where the pipes are embedded in a massive concrete slab/mass (within the structure) (EN 15377 / ISO 11855, type E). In our review, we classified the systems according to these three types (see Figure 20) and will report the surface activated (floor or ceiling).

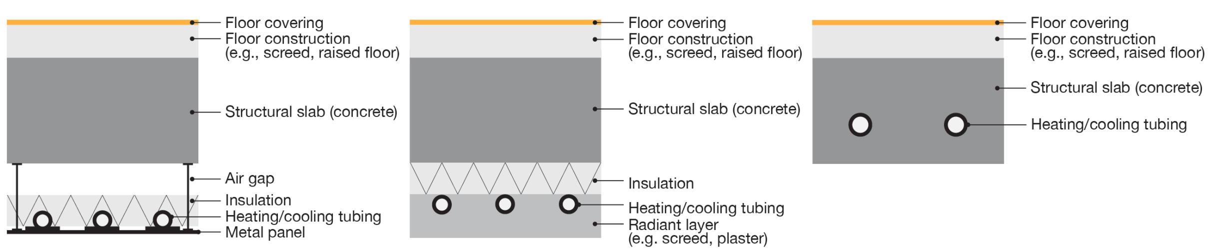


Figure 20: Illustration of radiant panels (left), embedded surface systems (ESS) (center) and thermally activated building systems (TABS) (right)

Types of all-air systems

In an 'all-air system', the extraction rate is mainly convective. While we found a few publications referring to natural ventilation (NV), most articles compared radiant to buildings that include a mechanical ventilation systems (MV). In some cases, both natural and mechanical ventilation could be activated (hybrid ventilation systems). Many studies are about cooling conditions in which case, the air system is more commonly referred to as 'air-conditioned' (AC).

MV can be further characterized by the design of air distribution strategies, that can have a large impact on the thermal comfort. We identified 3 common types of all-air distribution strategies (see Figure 21):

- Overhead (or mixing systems): supply air is delivered at a high velocity outside the occupied zone, usually at the ceiling level (overhead).
- Underfloor air distribution (UFAD): supply air is delivered from a raised access floor through floor diffusers that provide partial mixing of the room air, typically confined to the occupied zone.
- Displacement ventilation (DV): supply air is delivered within or close to the occupied zone (at or near the floor level). DV is sometimes classified as a subcategory of UFAD. What distinguishes the two systems is that DV does not necessarily require a raised access floor (air can be supplied through low side-wall diffusers) and the DV inlet velocity is very low to minimize mixing.

Chilled beams are a combined hydronic/air system that uses convection as the primary heat transfer mechanism. Thus and because this review focuses on thermal comfort, we decided to classify chilled beams as an 'all-air' system.

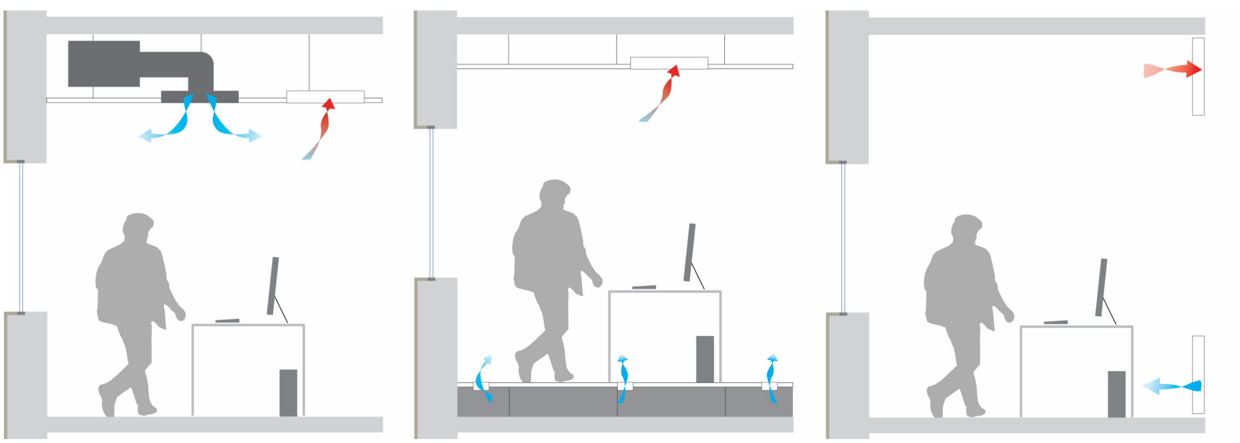


Figure 21: Illustration of mechanical ventilation types: Overhead/mixing systems (left), underfloor air distribution (center) and displacement ventilation (right)

4.4 Comparison of the systems

Of the 73 papers reviewed, 53 papers were excluded: 29 were not based on an actual comparison between the two systems; 16 were 'earlier studies' on thermal comfort for radiant systems (they were used to establish thermal comfort criteria and the testing conditions were beyond what is currently recommended); four were focused on exergy aspects without digging much into thermal comfort for the two systems; two were on transient conditions (rather than on radiant); one was not peer-reviewed
and one did not provide a proper description of the method and assumptions used. Of the remaining 20 papers, eight were judged conclusive (e.g., fair and realistic in the assumptions or laboratory set-up, comfort models used to assess the two systems).

4.4.1 Studies using building performance simulation

We found 9 papers comparing thermal comfort in radiant versus all-air systems that were based on computer simulation programs. Among the software used, we found: computational fluids dynamics (CFD) (e.g., Fluent) able to simulate the detailed airflow patterns and temperature distribution within the space, and whole building energy simulation (e.g., EnergyPlus (Crawley et al. 2001) or TRNSYS (University of Wisconsin, Madison, Solar Energy Laboratory and Klein 1979)) used to model zones and systems and predict indoor conditions and energy use for buildings. (Niu and Kooi 1994) used CFD to assess thermal comfort for radiant in comparison to all-air systems. They used vertical temperature distributions (stratification) and PD_{draft} as the main metrics. No clear preference for either system was found based on the two metrics. Building energy simulation offers researchers an effective method to simultaneously investigate thermal comfort and energy consumption in buildings. In many cases, the papers compared energy use under equivalent comfort conditions between the multiple radiant and all-air variants (Olsen and Chen 2003; Henze et al. 2008; Raftery et al. 2012; Fabrizio et al. 2012; Feng, Schiavon, and Bauman 2013). These studies were not retained as they focused on energy savings given thermal comfort constraints. Three studies using building energy simulation had a larger focus on thermal comfort and how this could satisfy our requirement for the selection of articles as described above. Chowdhury et al. (Chowdhury, Rasul, and Khan 2008) reported the study of the existing building located in Queensland Australia using a VAV system with air-conditioning (AC) and three low-energy upgrade variants: radiant ceiling panels (33% ceiling area), economizer and pre-cooling (cooling of the thermal mass through air-conditioning during off-peak hours). In all cases, the existing mechanical ventilation system is retained, but the strategies to cool the building are different. All simulations were conducted using DesignBuilder (with fine-tuning on EnergyPlus). The metric used was the PMV model. Although the radiant system appeared as the most comfortable of the three refurbishment options, the results showed that it did not bring thermal comfort improvement in comparison to the original AC system: the two systems would bring equivalent comfort. Overall, we found that this study did not bring conclusive evidence for improved thermal comfort for either system: the goal of the study was to find the best refurbishment variant and the PMV output was highly dependent on simulation input. Olesen and Mattarolo (B. W. Olesen and Mattarolo 2009) used EnergyPlus to compare ten different radiant system configurations (TABS, radiant panel, and ESS located on either floor or ceiling) to a reference (conventional) variable air volume (VAV) system with active heating and cooling. The simulation was done for a 4-story building located in Copenhagen, Denmark. The comfort metric used was the percentage of time during which indoor conditions (operative temperature) falls within categories I and II of EN15251 (CEN 2007). It was concluded that all radiant system variants enhanced the thermal comfort conditions. Yet, the input details of the simulations variants (including geometry of the building zones, controls, description of the radiant types, etc.) have not been provided in this conference paper. Also we could not track a more robust journal publication version of this paper. Therefore, we decided not to include this paper among the conclusive references. Salvalai et al. (Salvalai, Pfafferott, and Sesana 2013) used TRNSYS to compare five cooling strategies for a typical office for six different European climates. Radiant strategies included suspended ceiling panels and TABS (both combined with MV). All-air systems included a MV with fan coil. Additional passive based variants included NV and MV with night time ventilation cooling. The metric used to compare the variants is the percentage of time during which indoor conditions exceed the comfort limit of category II. For colder climates (represented by the cities of Stockholm, Hamburg and Stuttgart), both radiant and fan coils were within the standards requirements. For warmer climates (cities of Palermo, Rome, Milan) radiant systems variants could stay below 10% exceedence, while the fan coils variant reached approximately 35% in the worst case (Palermo). This study shows favourable thermal comfort for radiant systems compared to air systems. Yet we note that the scenario using radiant panels and TABS were assessed using the adaptive approach (based on EN 15251 (CEN 2007)) while the fan coil was assessed using the static approach (based on ISO 7730 (ISO, EN 2005)). Radiant panels and TABS were here combined with a MV system (and not a NV) and using the adaptive comfort model may not be totally correct for a fair comparison. Furthermore, it seems unlikely that a properly sized fan coil would bring this level of exceedence in the warmer climates. Based on these limitations, we found this study was unfair for our comparison purposes. From all studies based on BPS we thus retained one publication that did not show a clear preference for either systems (Niu and Kooi 1994).

4.4.2 Studies based on physical measurements

Studies involving physical measurement in laboratory conditions

We found five studies comparing radiant and all-air systems that were fully or partially based on physical measurement in laboratories. Olesen et al. (B. W. Olesen et al. 1980) conducted a full-scale experiment of a small office with one simulated outside wall. Nine heating systems were tested, including: radiant ceiling, radiant floor (electric system with an aluminium plate used for uniformity), air distribution system (different diffusor positions, air velocity in the room, air changes and air temperatures), convectors, radiators. The chamber included an adjacent controlled space that could simulate winter conditions through an outside wall (temperature down to -5º C and air infiltration rates up to 0.8 air-changes/h). The heat input was adjusted so that the room reference point nearby the frontage (assumed to be the most common place for an occupant to be seated) showed thermal neutrality. During steady-state conditions, air temperature, air velocities and surface temperatures were measured at several points. All nine heating systems proved in all tests to be capable of creating a remarkably uniform thermal environment (PPD ~ 5%) in the entire occupied zone. The vertical air temperature difference between 1.2 and 0.1 m level was less than 1.8K in the whole occupied zone in all tests. The floor temperature in the occupied zone with floor heating was always less than 27.5 ºC. It was concluded that all nine heating methods investigated are able to create an acceptable thermal environment. Kulpmann (Kulpmann 1993) performed thermal comfort and air quality experiments in a laboratory chamber equipped with a radiant ceiling and DV system. The internal loads were simulated through lighting and two workstations (thermal manikin and computer displays). The authors investigated the effect of varying the cooling capacity shares of the cooled ceiling and the ventilation system. The vertical profile of the room temperature was more pronounced when the load was covered with the ventilation system. An uncomfortable temperature difference of 5 ºC between ankle (0.1 m) and head (1.7 m) was measured when only DV was active. Little to no stratification was observed when the cooling load was handled (mainly/fully) by the radiant system. Air quality investigation showed that combination of DV with cooled ceiling induced a mixing of air within the space and could not ensure a safe displacement of air-transported pollution into the respiration area. Overall, the authors concluded that a cooled ceiling surface was 'best qualified' to maintain thermal comfort. Schiavon et al. (Schiavon et al. 2012, 2015) tested a similar combination a radiant ceiling and DV. In the first paper, they tested two different radiant coverage areas of the ceiling in addition to a baseline with only DV. The experiment was set up to keep the operative temperature fixed at 24º C for all configurations tested. For the pure DV test, the temperature profile suggests that the stratification height is between 1.1 m and 1.7 m. When the chilled ceiling was turned on, the stratification height appears to be reduced to a height close to 0.6 m (23 in.). The authors also observed that room air stratification in the occupied zone decreases when a larger portion of the cooling load is removed by the chilled ceiling. In all cases, we note that the temperature difference between head and ankle stayed below 3º C which satisfies standards requirements. It is thus delicate to conclude on one system achieving a better comfort. In the second paper, the same authors investigated the influence of very high cooling load (91 W/m^2) and two different heat source heights on thermal stratification (and air change effectiveness). The DV was tested for the higher heat source height only. Increased stratification was observed in the case of DV only compared to DV and radiant scenarios. The temperature difference between head and ankle exceeded standards requirements for the DV only case as well as for some of the DV and radiant combinations. While this experiment could bring some evidence for increased comfort in favour of the radiant system, we shall point out the very specific setting with high internal loads at a certain height. Such indoor layout seems quite specific and thus we will not include this result for our final assessment. Corgnati et al. (Corgnati et al. 2009) used a combination of experimental and numerical methods to assess an all-air mixing ventilation system alone or coupled with radiant ceiling panels in an office environment. The comfort metrics were related to the risk of draft (including PD_{draft}). The experiment was used to validate the CFD model. The radiant system was not part of the experimental set-up. The results showed that coupling air mixing and cold radiant ceiling panels with air jet supplied at low Archimedes numbers improves comfort in comparison to a system without radiant panels. The radiant cooling panels are increasing the jet longitudinal throw and reducing the vertical drop. This brings a significant decrease of the PD_{draft} due to the jet direct drop for the radiant configuration. Although this study shows an advantage for the radiant system variant, we decided not to include this study within our final count because it refers to a combination with a very specific air systems and because the metric used for the analysis (PD_{draft}) may overestimate discomfort. Mustakallio et al. (Mustakallio et al. 2016) studied thermal comfort conditions of a 17.3 m^2 room (modelled as a 2-person office and as a 6-person meeting room) including thermal manikins, two types of internal loads (medium and high) and a façade with a window. Four cooling variants were tested: (1) radiant panels with mixing ventilation using two linear diffusers located right below the radiant ceiling; (2) radiant panels with chilled beams using suspended radiant panels centred above the desk area, (3) chilled beam, and (4) mixing ventilation with desk-integrated cooling radiators. For consistency, we do not account the last variant as the system modelled is not a traditional radiant system (see section 0). Chilled beams (without radiant) use convective heat exchange and are here classified as an all-air system (see section 0). Results showed that the differences in thermal conditions achieved across the variants were not significant. The type and location of the diffusors in variant 1 bring questions regarding the primary heat exchange (convective or radiant) involved. Variant 2 remains radiant and therefore we decided to keep this study for our final assessment. From all studies based on physical measurements in laboratory settings we thus retain two lab testing of multiple systems in heating mode that showed comfortable conditions for both all-air and radiant systems (Mustakallio et al. 2016; B. W. Olesen et al. 1980) and one experiment of a DV system combined with a radiant chilled ceiling making a positive case in favour of radiant systems (Kulpmann 1993). We include further laboratory studies based on both physical and human subject testing in section 0.

Studies involving physical measurement in buildings

Field-studies based on both objective and occupant-based feedback are reported in section 0. Pfafferott et al. (Pfafferott et al. 2007) studied 12 low energy buildings located in Germany that included 4 buildings with thermally activated building systems (TABS). No buildings in this study had a compressor-based chiller. Cooling strategies include night ventilation for pre-cooling and earth-to-air heat exchangers. The indoor monitoring of thermal conditions was conducted over 2-3 years (between 2001 and 2005) in each building. This study was aimed at comparing the thermal comfort output of international and German standards and not comparing radiant to other conditioning systems. Therefore, the results may be considered carefully. The metric used was the frequency of exceeding standard requirements. The results showed the lowest frequency of exceeding for buildings using TABS in comparison to NV or hybrid systems (none of the buildings of this study were fully mechanically ventilated). This study was however not intended towards a comparison of conditioning systems and it would require specific analysis on that aspect to be able to draw conclusive answers on thermal comfort. Therefore, we will not include its result for our final assessment. Besides this study, we found multiple case-studies based on physical measurements in buildings and focusing on thermal comfort for radiant systems (e.g., (De Carli and Olesen 2002; Kalz, Pfafferott, and Herkel 2006; Rey Martínez et al. 2015; Kolarik, Toftum, and Olesen 2015)). These studies commonly show positive results in regard to thermal comfort in radiantly conditioned buildings. Yet none of these studies included a comparison of thermal comfort between radiant and all-air systems. To conclude, we won't retain any studies based on physical measurement in buildings for our final assessment.

4.4.3 Human subject testing / occupant based surveys

Studies involving human subject laboratory experiments

We found two laboratory studies involving human subjects directly comparing radiant and all-air systems (Schellen et al. 2013, 2012). In the first study, Schellen et al. (Schellen et al. 2012) wanted to focus on gender differences in thermophysiology, thermal comfort and productivity during convective and radiant cooling. Twenty college-age subjects (ten female and ten male) were exposed to the two cooling systems consecutively: convective and radiant. All tests were kept at neutral and comparable PMV levels. The results showed that under non-uniform conditions, the thermal sensation votes (TSV) significantly differ from the PMV: all tests showed a difference of 0.4 to 0.6 on a 7-point scale (p<0.001) for both conditioning systems and genders (this represents a change in PPD of about 10%). The experiment was conducted over a four-hour testing period and the authors found that for females the occupant responses changed over time for both radiant and convective conditioning. While the authors found different explanations for the time effect of the two conditioning systems, they concluded that radiant and air systems are equal in their ability to provide comfort. Overall, this study did not show preferences for either radiant cooling or all-air cooling systems. In the second study, Schellen et al. (Schellen et al. 2013) directly addressed the comparison of radiant and convective cooling systems. The authors explored three all-air scenarios: mixing ventilation with increased air velocities (no active cooling for this first configuration –all other configurations include active cooling), mixing ventilation, displacement ventilation; and three radiant scenario: radiant ceiling with mixing ventilation, radiant floor with mixing ventilation, radiant floor with displacement ventilation. Ten college-age male subjects were exposed to all six conditions during a two hours testing period.

All tests were kept at neutral and comparable PMV levels. The difference between PMV and TSV stayed within the accuracy of 0.5 (except for the passive cooling variant). Based on the physical skin temperature measurements, the authors noted that these differences between PMV and TSV were likely to be caused by local effects and local discomfort. The highest TSV was observed for the radiant floor cooling cases (with a floor temperature measured at 19.5-20 ºC against roughly 24 ºC for all-air cases). Subjects voted highest thermal comfort for active cooling by both displacement ventilation alone and chilled floor with displacement ventilation (75% of the votes for 'comfortable') and lowest thermal comfort for the passive cooling variant (increased air velocities) (55% of the votes differed from 'comfortable'). This study showed that vertical temperature gradients (up to 4 ºC/m) and lower temperatures near the floor even in combination with radiant floor cooling can result in acceptable thermal conditions. With respect to ventilation strategies, a clear preference was found for displacement ventilation. This study showed that non-uniform environments can achieve comparable or even more comfortable conditions compared to uniform environments. Yet, it did not prove preferences for any of the two (radiant or convective) systems. From all studies based on human subject laboratory experiments we retain two studies that could not show a preference for either system (Schellen et al. 2013, 2012).

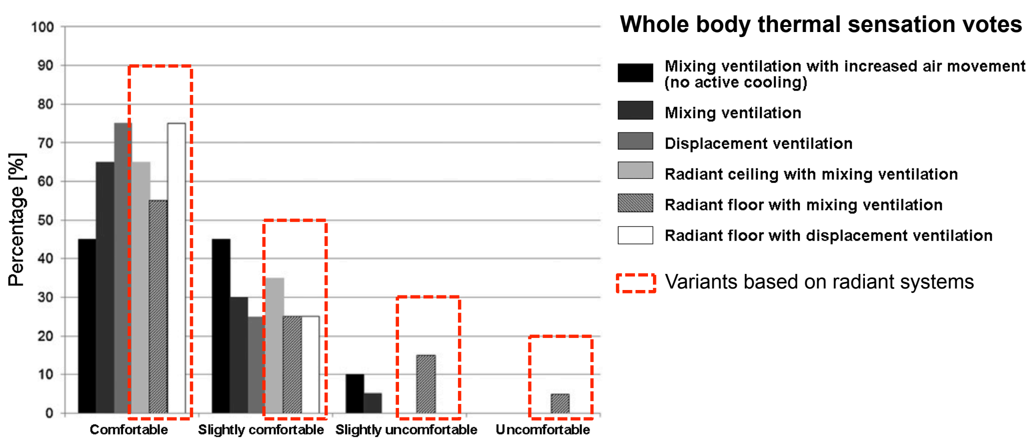


Figure 22: Frequency plot of whole body thermal comfort votes. Thermal stress assessment for the six cases tested. Affective evaluation: "Do you find it …?" ('it' refers here to subjects' personal thermal state). Answers based on the ISO-defined 4-points scale going from 'comfortable' to 'uncomfortable' (ISO 1995). Graphic adapted from (Schellen et al. 2013); the red dashed box indicate the tests conducted with radiant systems.

Studies involving occupant surveys

We found one study by (Imanari, Omori, and Bogaki 1999) comparing a radiant to an all-air system based on occupant feedback and simultaneous indoor condition monitoring. The comparison was performed in a meeting room of a building in Tokyo, Japan. This meeting room was built to include radiant ceiling panels and an overhead ventilation system (with and without reheat). The air change was double in the case of the all-air experiments (7.7 against 3.8 ACH for radiant panels for the same air supply and intake diffusers in the room), and therefore the air speed can be expected to be higher. Male and female experiments were conducted separately. For males, the room was used during a normal meeting and the subjects were asked to complete a thermal comfort survey at the end of their meeting (after a minimum stay in the room of 1 hour). For females, the room was used for the purpose of the experiment; the testing time was longer (2 hours) and the questionnaire included thermal comfort, thermal sensation (both at regular intervals) and work performance (measured through accuracy and achievement testing). Both males and females were tested under radiant and all-air conditions. Three series of experiments (for a total of seven cases) were tested. Males were tested for both cooling and heating cases; females were tested for cooling only. The number of subjects varied with each experiment. The PMV for all 3 series of tests was set at comparable levels and close to neutral. The results of this study showed a higher thermal comfort for radiant systems, more neutral thermal sensation votes for radiant and slightly improved work efficiency under the environment created by the radiant cooled ceiling. The draft risk measured in the room was also much smaller for the radiant cases (PD_{draft} estimated at 4.4-5.6% for radiant tests and 7.8-12.7% for all-air tests). This difference is likely to be associated with the sizing of the air system. As the draft risk metric has been rather criticized in our assessment so far, we decided to keep this study among the conclusive ones. We found the set-up involving an office with occupants relevant and decided to keep this study among the conclusive ones. Moving towards full building scale, the building "Software Development Block 1" (SDB-1), completed in 2011 located in Hyderabad, India offered a pretty unique setting as it is divided into two equivalent halves that comprise two optimized cooling systems: a mixing ventilation system (variable air volume (VAV) system) and a TABS with mixing ventilation (dedicated outdoor air system (DOAS)). The case study article from Sastry and Rumsey (Sastry and Rumsey 2014) included two thermal comfort aspects: objective measurements using a portable cart (dry-bulb air temperature, relative humidity, air velocity, MRT) and occupant feedback based on the Indoor Environmental Quality Occupant Survey developed at UC Berkeley (Zagreus et al. 2004). Objective comfort measurements showed that the radiant side of the building had a PPD rating of 7.9% as compared to 8.7% for the VAV portion of the building (based on the EN ISO 7730 (ISO, EN 2005)). Both sides stayed at a pretty high level of predicted thermal comfort with a slight advantage for the radiant side. Yet, the publication did not inform us on the measurement details and resolution. About 150 occupants answered the survey on each side of the building. The survey results showed that the group that fell in the "satisfied" or "very satisfied" categories grew from 45% on the VAV portion of the building to 63% on the radiant portion. As a side note, energy use was found to be lower on the radiant side (34% less energy as compared to the VAV system based on the first two years of operation). This study is the only one we found involving such a side-by-side comparison. From all studies based on occupant surveys in buildings, we retain one study showing increased thermal comfort in favour of radiant system (Sastry and Rumsey 2014).

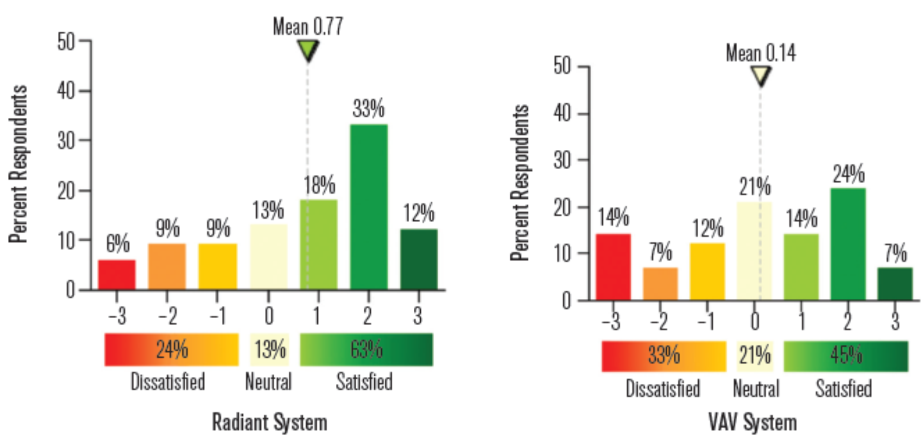


Figure 23: CBE occupant survey results for the Infosys building (Sastry and Rumsey 2014)

4.4.4 Summary of the comparison

Table 1 summarizes the conclusive studies found in this comparison. One study using BPS brought comparable thermal comfort results for thermal comfort in all-air vs. radiant systems (Niu and Kooi 1994). One study using physical measurements in laboratory conditions showed that a chilled ceiling with DV offers improved comfort compared to DV only (Kulpmann 1993). Using the same method, (B. W. Olesen et al. 1980; Mustakallio et al. 2016) did not find increased comfort for either radiant and convective systems. Rigorous laboratory studies from Schellen et al. (Schellen et al. 2013, 2012) did not prove preferences for either of the two systems. The study from Imanari et al. (Imanari, Omori, and Bogaki 1999) showed a thermal preference for radiant systems. A building case-study reported by Sastry and Rumsey (Sastry and Rumsey 2014) used both physical and subjective measures to assess thermal comfort in the building. The results showed that the radiant side of the building is able to provide improved comfort conditions in comparison to the all-air side. In summary, this literature review identified five studies that could not establish a thermal comfort preference between all-air and radiant systems and three studies showing a preference for radiant systems. The methods used to demonstrate this were multiple and so were the types of all-air and radiant systems tested.

Table 1: Summary of conclusive studies for our thermal comfort comparison

Publication	Method	Cond. mode	Radiant system	Convective (all-air) systems	Preferred system
Niu and Kooi (Niu and Kooi 1994)	BPS	Cooling	Radiant ceiling panels with DV	DV, ceiling “air panels”	No preference found
Olesen et al. (B. W. Olesen et al. 1980)	Lab testing (measurements)	Heating	Radiant ceiling, radiant floor	Convectors, Mixing ventilation (air supplied from the top and down nearby the façade)	No preference found
Kulpmann (Kulpmann 1993)	Lab testing (measurements)	Cooling	Radiant ceiling panels with DV	DV	Radiant
Mustakallio et al. (Mustakallio et al. 2016)	Lab testing (measurements)	Cooling	Radiant ceiling panels with chilled beams	Mixing ventilation, chilled beams	No preference found
Schellen et al. (Schellen et al. 2012)	Lab/Human subject testing	Cooling	Radiant ceiling panels with mixing	Mixing ventilation	No preference found
Schellen et al. (Schellen et al. 2013)	Lab/Human subject testing	Cooling	Radiant ceiling panels with DV and mixing ventilation	Mixing ventilation, DV (multiple conditioning strategies)	No preference found
Imanari et al. (Imanari, Omori, and Bogaki 1999)	Occupant surveys	Heating and cooling	Radiant ceiling panels combined with mixing ventilation	Mixing ventilation	Radiant
Sastry and Rumsey (Sastry and Rumsey 2014)	Physical measurements and occupant surveys	Cooling	TABS	Mixing ventilation	Radiant

4.5 Discussion

4.5.1 Relevance of the methods used and future research

This critical literature review covered a wide variety of metrics and methods used to assess thermal comfort. We noticed that metrics and models are often embedded and limited by the methods used. For instance, the PMV/PPD can be computed based on simulation output and can be physically measured. On the other side, subjective assessment requires human subjects or occupants. Thus, metrics, models and methods cannot be totally isolated from each other. We further found that the different methods were not offering us the same level of relevance in terms of thermal comfort assessment. We found that simulation-based methods are great to predict and compare variants and to verify standard compliance. Yet they are based upon the assumptions of idealized models, including the limitations of the comfort model used. Physical measurements are fundamental to verify indoor conditions and monitor buildings, but they often are unable to provide feedback on how a particular type of system impacts perceived comfort. Thus, we found that studies involving human subject testing and occupant based surveys were the most relevant for assessing thermal comfort.

Human subject testing in laboratories allows researchers to precisely control indoor conditions and assess perceived comfort at a great level of detail based on both objective and subjective metrics. On this aspect the laboratory studies from Schellen et al. (Schellen et al. 2013, 2012) were particularly relevant. They did not predict significant differences between the whole-body and body-part comfort levels. This outcome addresses the common perception of a non-uniform thermal environment being less comfortable than a uniform one. The fact that these studies did not conclude on a preference for either system is pretty informative.

Field studies do not allow the same level of control as laboratory studies but they provide evidence of how building systems perform under realistic and practical conditions. Results from occupant based surveys ultimately provide feedback on what buildings, systems, and other aspects of their indoor environment do occupants prefer compared to others, which is a key aspect for successful implementation of building technologies. Yet, occupant survey studies also incorporate one key limitation: they rely on subjective answers collected while many uncontrolled variable change at the same time and thus larger samples are required to overcome singularities and to capture responses of average occupants. The case of SDB-1 brought a strong case for the thesis that radiant systems provides better thermal comfort than air systems, but only one example of this comparison is available.

Comparing multiple cases of radiant buildings against multiple cases of non-radiant buildings may offer us a very relevant method to assess thermal comfort in real conditions. This method was used in Brager and Baker (Brager and Baker 2009) when comparing mechanically conditioned and mixed-mode buildings (i.e. buildings that use a hybrid approach to space conditioning that uses a combination of natural ventilation from operable windows, and air conditioning) or by Altomonte and Schiavon (Altomonte and Schiavon 2013; Schiavon and Altomonte 2014) when comparing LEED to non-LEED buildings. This method seems extremely appropriate for bringing new answers to our question and it does not require buildings to be built as a side-by-side comparison. In fact, there are plenty of radiant and all-air systems within the existing building stock from which we can learn and that potentially offer us many resources.

Because buildings tend to be different from each other (and not only based on the mechanical systems) we need to isolate confounding factors and therefore there is a need to conduct a large sample assessment. These studies will help us identify thermal comfort satisfaction patterns, and thus, provide answers to our questions while taking into account the practical constraints and robustness of HVAC system implementation and operation as well as the influence of further design aspects on thermal comfort.

A more general limitation of this review is the multiplicity of systems and conditioning strategies involved in both radiant and all-air cases. All-air systems comprised stratified and mixing systems; radiant systems included metal ceiling panels and TABS ranging from partial to full covering of surfaces, and located on both floors or ceilings. Additionally, radiant systems are required to be supplemented with a ventilation strategy, typically a dedicated outdoor air system. Both systems were tested under heating and/or cooling conditions at different supply temperatures. However, in our simplified classification scheme, the 19 studies we retained for our final analysis investigated over 20 different systems and associated controls. This variability adds noise within our assessment. It seems that multiple systems (including both radiant and all-air) are able to provide acceptable thermal comfort, depending on several factors, including operation and control.

4.5.2 Additional observations on thermal comfort for radiant systems

Temperature drifts for massive radiant types

Our final selection of publications did not include studies using temperature drifts because we could not find a paper comparing radiant to all-air systems that would be based on this metric. Temperature drift is still believed to be an issue for radiant systems that incorporate high thermal mass (e.g., TABS). During warm days these massive systems absorb heat until saturation. As a result, the rising temperature towards the end of the day may exceed the upper boundary of the comfort zone. Kolarik et al. (Kolarik, Toftum, and Olesen 2015) conducted a study of three European buildings using TABS located in Spain, Italy and Denmark. Physical measurements showed that the limit for 4-hour operative temperature drift (0.8 K/h) was exceeded in all buildings. While temperature satisfaction slightly decreased when the rate of temperature change increased, the median value of these votes stayed positive ("satisfied" and "just satisfied") even for the most extreme drifts. The authors reported that the data collected did not allow for robust statistical analysis. Thus, additional field studies would be needed to validate or challenge previous laboratory studies that did not include a radiant system (artificially controlled drifts) and where occupants were not allowed to modify their clothing during drifts (Kolarik et al. 2009).

Applicability of the human body exergy concept

The human body exergy concept is based on the assumption that one reaches thermal comfort when one's metabolic emission equals the energy outflow (due to radiation, convection, evaporation, conduction). Thus, having an influence on the mean radiant temperature increases the chances to target lowest human body exergy by design. Simone and Olesen (Simone and Olesen 2009) conducted a human subject laboratory experiment to address this question. 30 subjects were exposed to three different combinations of air and mean radiant temperature with an operative temperature around 23 ºC. Yet this study could not confirm any preference regarding air and mean radiant temperature ranges. Using multiple datasets from radiant system laboratory studies (McNall and Schlegel 1968; McNall and Biddison 1970; McIntyre and Griffiths 1972) and (Simone and Olesen 2009), Simone et al. (Simone et al. 2011) further investigated the relationship between thermal sensation (used as proxy of thermal comfort) and human body exergy consumption. Their statistical analysis showed that the lowest human body exergy was correlated with neutral and slightly cool thermal sensations, yet with a moderate correlation coefficient (R^2=0.68), regardless the fact that TSV was averaged for each indoor condition. We also noted a very high difference between air and operative temperatures within the data (up to 7ºC difference and 2.6ºC on average). Overall, the assumptions and outcomes of these studies lead us to question the applicability of the human body exergy concept for increasing comfort in spaces that use radiant systems.

4.6 Conclusions

We performed a literature review to assess if radiant systems provide better, equal or lower thermal comfort than all-air systems. Studies focusing only on radiant systems or only on all-air systems were not included as they did not inform our comparison. This literature review brought five studies that could not establish a thermal comfort preference between all-air and radiant systems and three studies showing a preference for radiant systems. These studies used multiple methods to demonstrate their findings and, in addition, several types of all-air and radiant systems were tested. The two systems performed similarly when compared based on building energy simulation (Niu and Kooi 1994), laboratory studies (B. W. Olesen et al. 1980; Mustakallio et al. 2016) and human subject testing in laboratory conditions (Schellen et al. 2012, 2013). Radiant cooling ceiling panels showed better results than all-air systems based on laboratory studies (Kulpmann 1993) and occupant responses in a building (Imanari, Omori, and Bogaki 1999). A side-by-side field comparison between an all-air system and TABS with DOAS that was based on occupant survey responses showed increased satisfaction with thermal comfort for the radiant system (Sastry and Rumsey 2014). All these studies were fully (or mainly) about cooling applications (only one heating variant in (Imanari, Omori, and Bogaki 1999)). Overall, we found that a limited number of studies are available and therefore a solid answer cannot be given. Nevertheless, there is suggestive evidence that radiant systems may provide equal or better comfort than all-air systems. Further studies are needed to confirm this statement. Both systems are able to provide acceptable thermal comfort, depending on several factors, including operation and control.

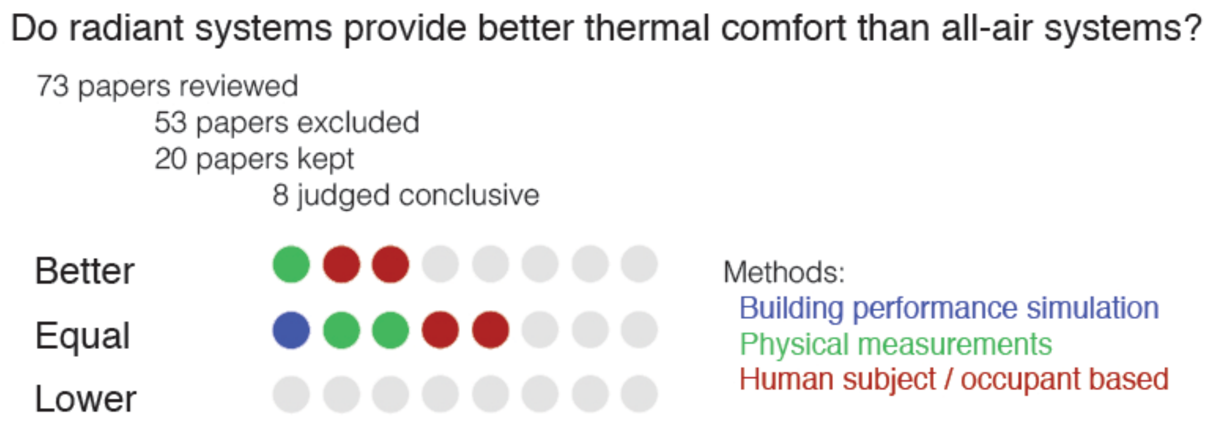


Figure 24: Graphical illustration of the findings from the literature review

5 Occupant satisfaction with thermal comfort and acoustic quality in buildings using radiant and all-air systems

5.1 Introduction

In the U.S., people spend almost 90% of their time indoors (Klepeis et al. 2001). This long exposure to indoor conditions has the potential to affect the well-being, performance and health of the occupants residing within a space. To date, little is known about how radiant systems affect the IEQ in buildings. The previous chapter has shown suggestive evidence that radiant systems may provide equal or better comfort than all-air systems. Aside from thermal comfort, acoustic quality can be critical in buildings. Radiant systems are implemented on large surfaces, generally ceilings or floors. Keeping these surfaces exposed is a challenge when integrating acoustical absorbents; this can be especially challenging for thermally activated building systems (TABS). Results from additional survey studies on 180 occupants (seven radiant cooled buildings including six using TABS) confirmed low acoustic satisfaction (Bauman et al. 2012). The use of radiant systems may also indirectly affect other aspects of the building design and its indoor environment. For instance, using a radiant system may affect the design of the envelope, lighting equipment, and the integration of air systems for ventilation.

The goal of this study is to compare IEQ - in particular, thermal comfort and acoustic quality – as reported by the occupants within a large set of buildings using radiant and all-air systems.

5.2 Method

5.2.1 Occupant survey method

We used the online Occupant Indoor Environmental Quality Survey administered by the Center for the Built Environment (CBE), University of California, Berkeley, to perform our data collection (Zagreus et al. 2004). The survey asks a set of basic questions about occupant demographics followed by nine core categories of indoor environmental quality, including thermal comfort, air quality, acoustics, lighting, cleanliness/maintenance, spatial layout, office furnishing, and general building and workspace satisfaction. We invited occupants in each building to take the survey through an invitation e-mail that included a URL to the survey for their building. The survey measures occupant satisfaction in each of the above categories using a 7-point Likert scale with answers ranging from 'very satisfied' to 'very dissatisfied' with 'neither satisfied nor dissatisfied' as the middle option (see Figure 25). This survey is a web-based survey and, depending on the question, dissatisfied responses trigger branching questions targeting the source of that dissatisfaction. The survey takes participants approximately 10-15 minutes to complete, depending on the number of branching questions received, and open-ended comments provided.

For each building where we administered an occupant survey, we also conducted a survey of the building's characteristics, specifically: location, size, year of construction or renovation, type of HVAC system(s), green building certification, energy use, etc.). Either the building manager, the facilities manager, or a member of the design team provided this information in each case.

CBE has conducted surveys in over 1000 buildings with about 100,000 individual occupant responses since 2001, primarily in the U.S.. Building types cover commercial offices, healthcare, laboratories, educational buildings, libraries, residential, etc. This database represents a unique research resource that has been used, for instance, to assess which parameters have the largest effect on occupant satisfaction (Frontczak et al. 2012), to evaluate the effectiveness of office layout (Kim and de Dear 2013), HVAC strategies (Huizenga et al. 2006; Brager and Baker 2009; Kim and de Dear 2012), or building rating systems (Altomonte and Schiavon 2013; Schiavon and Altomonte 2014).

Table 2: Core modules (categories) included in the CBE occupant IEQ survey

Category	Description
Spatial layout	Amount of space available for individual work and storage Level of visual privacy Ease of interaction with co-workers
Office furnishings	Comfort of office furnishings (chair, desk, computer, equipment, etc.) Ability to adjust furniture to meet your needs Colors and textures of flooring, furniture and surface finishes
Thermal comfort	Temperature in your workspace
Air quality	Air quality in your workspace (i.e. stuffy/stale air, air cleanliness, odors)
Lighting	Amount of light in your workspace Visual comfort of the lighting (e.g., glare, reflections, contrast)
Acoustic quality	Noise level in your workspace
	Sound privacy in your workspace (ability to have conversations without neighbors overhearing and vice versa)
Cleanliness and maintenance	General cleanliness of the overall building General maintenance of the building
General comments	Your personal workspace Building overall

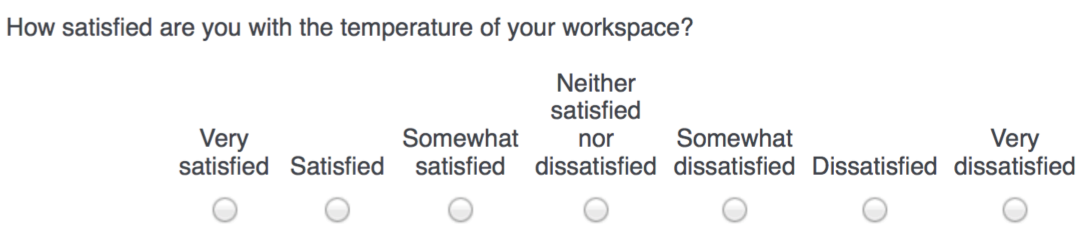


Figure 25: Sample of survey question (satisfaction with temperature) using a 7-point Likert scale

5.2.2 Selection of buildings and collection of data

While the existing CBE survey database represents quite a large sample of buildings, only a few of those buildings used radiant systems as the primary conditioning system. This disparity reaffirms all-air as being the most common conditioning system in areas where we have conducted surveys in the past. To fill this gap, the current study involved a large effort to collect new survey data in radiant buildings. This section details the building selection criteria for both radiant and all-air groups.

Collection of data from radiant buildings

The current study focused on radiant cooling (either as a single conditioning mode or in combination with heating) for primarily office and educational buildings in the U.S. and Canada (regardless of whether they had been renovated or were new construction). Specifically, this included: regular offices, higher education and learning centers, libraries, and government buildings. There was no restriction on the ownership type; we include both private and publicly owned buildings. We included the three major types of radiant systems: embedded surface systems (ESS); thermally activated building systems (TABS); and radiant ceiling panels (RCP). We did not include other hydronic systems that are not focused on radiant heat exchange. For buildings using mixed conditioning strategies, we made sure that the workstations for the surveyed occupants were in a radiantly conditioned area. We only included buildings that had a minimum of 15 occupants and we did not survey transient occupants (such as students or library visitors) due to their lack of continuous experience with the spaces in question.

We looked at the existing CBE survey database and found 8 buildings (568 occupants) that met the criteria above, and then sought out additional candidate radiant buildings to increase the sample size. We began with a target building set for the study based on CBE's online database of radiant buildings (Center for the Built Environment 2016) to which we added other potential building candidates from NBI's Getting to Zero (GtZ) database (New Building Institute 2015). We identified 146 potential building candidate and tried to reach all of them. For the buildings that met our selection criteria and agreed to participate, we administered both the CBE Occupant IEQ Survey and the building characteristics survey. As an incentive, upon completion of the occupant survey, participants were entered into a raffle to win a gift card. Building managers, facilities managers, and design team members who completed the building characteristics form also received gift cards as compensation for their support of the study. We collected this additional data between November 2015 and March 2017, and discarded the oldest survey from two buildings that we surveyed twice (i.e., we discarded two surveys from the existing CBE survey database).

Table 3: Number of buildings and occupants surveyed available for this analysis

	Existing CBE IEQ survey data		Data collected	Data used for this study	
	Building from (Altomonte and Schiavon 2013)	Radiant buildings	Radiant buildings	All-air buildings	Radiant buildings
Buildings surveyed	144	6	20	34	26
Occupants surveyed	21,477	361	1,284	2,247	1,645

Data for all-air buildings

The data from the all-air buildings came from a subset of the CBE survey database, as described above (Altomonte and Schiavon 2013). This subset consists of commercial buildings surveyed up until 2010 and whose building characteristics our team verified. We wanted our all-air subset to conform with the characteristics of the radiant buildings collected. As a first step, we established a list of key criteria for selecting the all-air buildings that matched those of the radiant dataset. This included buildings that: (1) are located in the U.S. or Canada; (2) are offices, educational or government buildings (only office spaces surveyed); (3) use active all-air mechanical cooling systems, (4) are no older than the oldest radiant building of the subset, and (5) are of comparable size (building area) to the radiant subset (range of minimum and maximum area based on the radiant building subset). We then created a subset of all-air buildings that met these criteria.

5.2.3 Statistical analysis methods

The survey records satisfaction votes on an ordinal scale. Depending on the statistical methods, we sometimes treated our data as an interval scale, which makes the implicit assumption that the intervals between votes are treated equally. This may not be the case with satisfaction data; for example, people's perception of the difference between 'neutral' and 'slightly dissatisfied' may be larger or smaller than the difference between, say, 'moderately satisfied' to 'very satisfied'. We formatted the building's characteristics using either factors or a binary structure.

We used an occupant's individual responses as the main unit of our analysis, as it has the advantage of correctly accounting for the number of people that have answered the survey. The use of individual responses also prevents one from artificially reducing the variance and, consequentially, increases accuracy. An alternative is to use one average value for each building. In a few cases, we refer to buildings, which is a more common scale in the field (e.g., building design, operation, energy use, etc.). Using responses aggregated by building does not reduce the building design process to a collection of individual characteristics and independent decisions. Therefore, the building scale can sometimes be more representative of design intent and reveal the differences between building performance. It also prevents buildings with higher numbers of occupants from bringing bias to the overall result. We report the number of occupants per building in the results throughout this paper. We used the R statistical software v. 3.3.2 (R Development Core Team 2017) for all statistical analysis.

We used multiple statistical methods in our analysis. We compared both mean and median values. Median values are more relevant in the case of interval data. Yet, in some cases, median values do not provide sufficient information about the granularity of the differences in the data, and, in some cases, they tend to describe a 'jump' in the data. Therefore, we commonly reported and discussed both mean and median.

We tested the statistical significance of the difference between independent groups using the Wilcox rank test, where *p-value* < 0.05 is considered statistically significant. *P-values* are sensitive to sample size and larger samples can lead to possible over-interpretation of the results. Therefore, we complement our results with effect sizes. We used boxplots as one form of graphical representation of the results. Boxplots indicate the mean, the median, and the 1st and 3rd quartiles of satisfaction votes for each group considered.

Effect size is a quantitative measure of the strength of a phenomenon, and reflects practical significance (Kraemer et al. 2003). We used Spearman rank correlations (ρ) because of our data structure (ordinal scale) which is a rank-based measure of association that evaluates the monotonic relationship between two continuous or ordinal variables. This type of effect size describes the magnitude of shared variance between two or more variables. Spearman's ρ is kept within the interval [-1, +1] with 0 indicating no association. While the calculation of ρ is straightforward, the thresholds for interpretation of effect sizes (i.e., what is meaningful or not) vary by author. Cohen (Cohen 1988) was the first to propose thresholds. He used 0.1, 0.3 and 0.5

to define 'small', 'medium, and 'large' effects, respectively. Cohen's values have been later increased by Ferguson (Ferguson 2009) to the more conservative thresholds of 0.2, 0.5 and 0.8, respectively, to prevent over-interpretation of effects. Both authors commonly warn about the challenge of interpretation of effect sizes, which vary from one field to another. We could not find interpretation schemes of effect sizes commonly used in our field, and thus, we present more detail on how we address effect size thresholds in this paper. We used two separate approaches for this analysis:

1. We used the dataset of this study (60 buildings) and generated groups of best/worst buildings based on the upper and lower 3-5 buildings of the dataset (enough to include about 120 occupants on either side). We conducted this analysis for the IEQ category showing the highest effect size (which, as shown later, was temperature satisfaction). As median values tend to overlap for a wide range of buildings, we chose the best/worst buildings based on their mean values. These extreme scenarios provided a set of values against which we compared what is defined as a 'large' effect.

2. We used the dataset of this study and compared the effect size obtained for conditioning type (radiant/all-air) to the effect size obtained for other binary variables of our survey such as type of ventilation (mixed-mode/mechanical), type of office (enclosed/open), gender (male/female), etc. This analysis provided us with a different perspective to compare the outcome of conditioning type and it allowed us to put the discussion of effect size into context. We used several IEQ categories in this assessment.

We also used Cliff's delta (∂) to report effect sizes. Cliff's ∂ explains the probability of superiority of one variable against the other: probability that a randomly selected observation from one group is larger than a randomly selected observation from another group, minus the reverse probability (i.e., for this study, ∂ = P(radiant > all-air) – P(all-air < radiant)) (Cliff 1996). Cliff's ∂ is a non-parametric test; it is not affected by the distribution of the dependent variable. Cliff's ∂ ranges between -1 and +1, where 0 indicates overlapping distributions. We could not find references for interpreting Cliff's ∂ values, but will discuss the values obtained.

Our analysis also includes linear models with mixed effects. This type of model recognizes the relationship between serial observations scaled on the same unit. We used building ID and type of office as the random effect and report between-group variability.

5.3 Results and discussion

5.3.1 Description of the final dataset

We contacted a total of 141 buildings for the radiant dataset, and we obtained new data from 20 buildings (1284 occupants), which we combined with the radiant buildings from the existing CBE dataset. Our study involved 26 radiant surveys and 34 all-air buildings, with 1,645 and 2,247 occupants, respectively. Table 4 summarizes the final dataset.

Table 4: Description of the dataset used for the analysis of this paper

Criteria		Radiant subset	All-air subset	Total
General	Occupant responses (% of total)	1,645 (42%)	2,247 (58%)	3,892 (100%)
	Building surveys count (% of total)	26 (43%)	34 (57%)	58 (100%)
	Occupant responses added	1,284 (35%)	-	1284 (35%)
	Building surveys added	20 (34%)	-	20 (34%)
Type of radiant system	Radiant panels	478 (12%)	-	478 (12%)
	In-slab (TABS & ESS)	1,167 (30%)	-	1,167 (30%)
	Non-radiant	-	2247 (58%)	1,978 (58%)
Ventilation systems	Mechanical ventilation (MV)	1,038 (27%)	1,185 (30%)	2,036 (57%)
	Mixed-mode ventilation (MM)	607 (16%)	969 (25%)	1,487 (40%)
	NA	-	93 (2%)	234 (2%)
Climates	Cold (ASHRAE zone 6A, 7)	55 (1%)	395 (11%)	450 (12%)
	Cool (ASHRAE zone 5, 5A, 5B)	384 (10%)	477 (12%)	861 (22%)
	Mixed (ASHRAE zone 3C, 4A, 4C)	813 (21%)	803 (21%)	1,616 (42%)
	Warm (ASHRAE zone 3A, 3B)	393 (10%)	572 (16%)	965 (25%)
	NA	-	-	-
Type of offices	Cubicles with high partitions	157 (4%)	336 (9%)	493 (13%)
	Cubicles with low partitions	665 (18%)	974 (25%)	1639 (42%)
	Enclosed private office	256 (7%)	547 (14%)	803 (21%)
	Enclosed shared office	80 (2%)	173 (4%)	253 (7%)
	Open office with no partitions	295 (8%)	35 (1%)	330 (8%)
	NA	192 (5%)	182 (5%)	374 (10%)
Year of occupancy (construction/renovation)	1st Quartile	2010	2005	2006
	2nd Quartile (median)	2012	2006	2008
	3rd Quartile	2013	2008	2012
	Max	2015	2009	2015
Building size m^2	1st Quartile (m^2)	5,574	2,764	4,095
	2nd Quartile (median)	16,020	6,132	6,763
	3rd Quartile (m^2)	18,860	7,990	16,350
	Max	20,440	17,190	20,440

5.3.2 Occupant satisfaction in buildings using radiant systems

Figure 26 shows the distribution of the responses for each survey category. We observe that the ranking of the survey categories for occupants exposed to radiant systems follows the patterns of larger survey studies conducted (e.g. Frontzcak et al. (Frontczak et al. 2012)), where ease of interaction, maintenance/cleanliness, furnishing, visual comfort categories show higher results compared to temperature and acoustic questions. For the first 13 categories of this radiant systems survey (including: building cleanliness, ease of interaction, building maintenance, colors and textures, overall building satisfaction, comfort of furnishing, workspace cleanliness, workspace satisfaction, amount of light, air quality, amount of space, adjustment of furniture, visual comfort), a minimum of 79% of occupants answered neutral or satisfied with their environment on the 7-point satisfaction scale (votes ranging from "neither satisfied nor dissatisfied" up to "very satisfied" are represented in white and green tones). The focus of this paper, thermal comfort and acoustic quality, were among the four questions that received the lowest satisfaction votes: temperature, visual privacy, noise, and sound privacy. The distributions of interest are:

- temperature: 31.5% dissatisfied, 10.5% neutral and 58% satisfied
- noise: 40% dissatisfied, 14% neutral and 46% satisfied
- sound privacy: 59% dissatisfied, 14% neutral and 27% satisfied.

Aside from sound privacy, the "satisfied" votes were larger than the "neutral" or "dissatisfied" votes for each of the survey categories. The distribution of votes is not normal and is skewed by the disproportionate "satisfied" votes.

We looked at the source of temperature dissatisfaction (see Figure 27). The respondents rated control and access to control among the highest four sources of dissatisfaction. Occupants also referred to the air movement being too low (20% of dissatisfied answers), which is consistent with previous studies. Interestingly, the sources of dissatisfaction that many may expect in spaces using radiant systems, "hot/cold floor surfaces" and "hot/cold ceiling surfaces" were not highly reported: just 8% and 2% of dissatisfied answers, respectively.

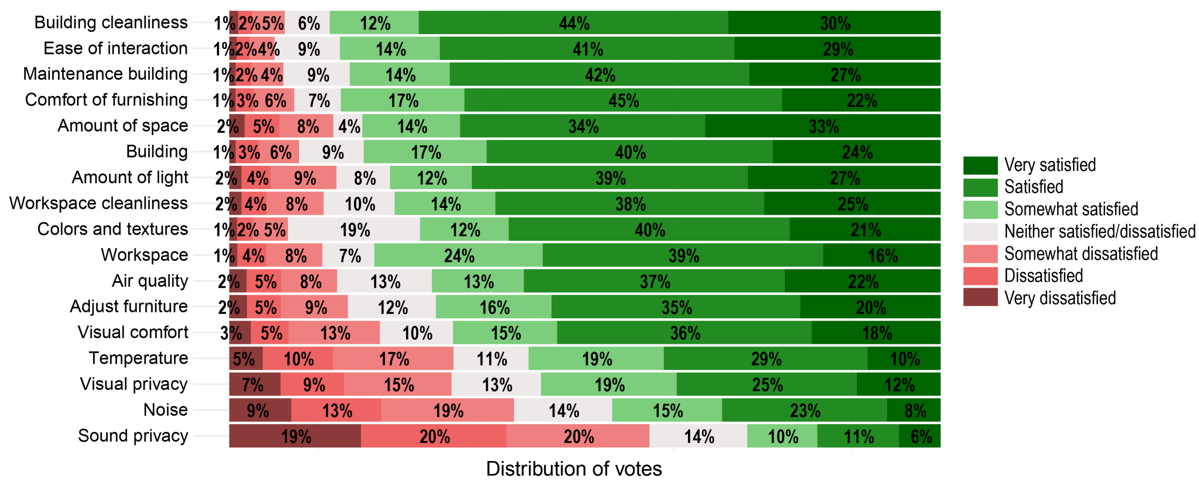


Figure 26: Distribution of occupant satisfaction votes in buildings using radiant systems for all survey categories ordered by mean satisfaction score

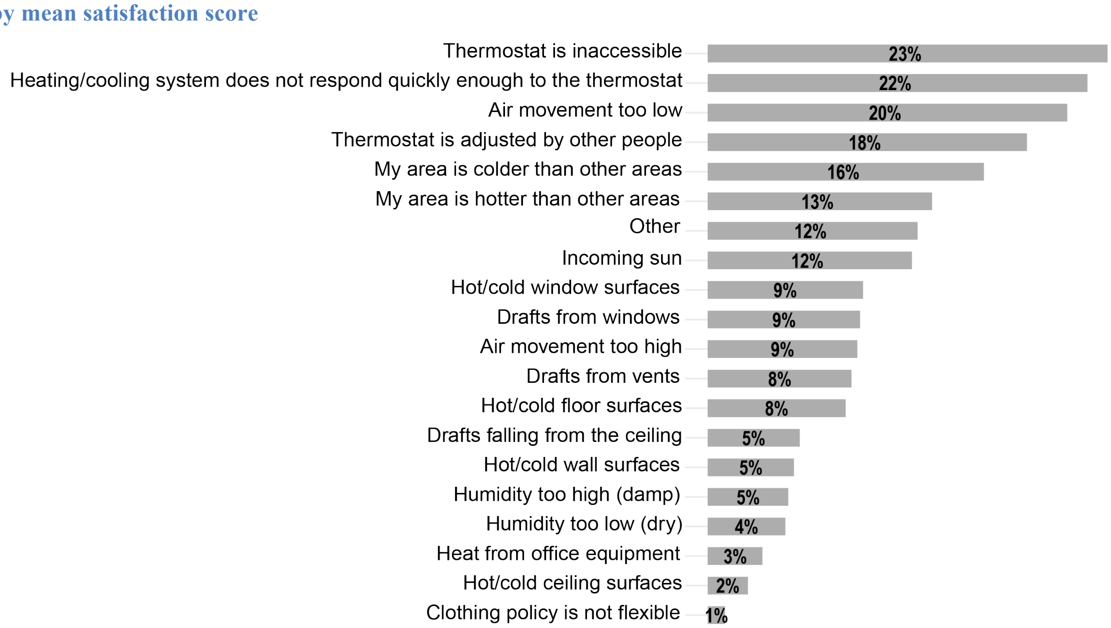


Figure 27: Source of temperature satisfaction of 611 negative votes with temperature

5.3.3 Occupant satisfaction in radiant vs. all-air buildings

Comparison of occupant satisfaction with IEQ in radiant and all-air buildings

Table 5 summarizes the key results of the comparison between radiant and all-air buildings. We provide the following metrics for each surveyed satisfaction question: mean, median (Mdn) and standard deviations (SD) of scores for occupants of radiant and all-air buildings; the difference in mean (ΔM) and median (ΔMdn) between the two groups; the statistical significance of the difference (*p-value*), and the effect size (Spearman's rho (ρ) and Cliff's delta (∂)).

Table 5: Selected results of statistical analysis between the radiant and all-air groups

	Radiant group			All-air group			Comparison			Effect size	
Satisfaction with: [a]	**Mean**	**Mdn**	**SD**	**Mean**	**Mdn**	**SD**	**ΔM**	**ΔMdn**	**p-value**	**Spearman's ρ**	**Cliff's ∂**
building cleanliness	1.77	2	1.29	1.57	2	1.43	0.20	0	<0.001 ***	0.06	0.07
ease of interaction	1.74	2	1.26	1.46	2	1.46	0.28	0	<0.001 ***	0.09	0.1
building maintenance	1.67	2	1.29	1.38	2	1.5	0.29	0	<0.001 ***	0.09	0.1
amount of light	1.48	2	1.53	1.42	2	1.6	0.06	0	0.552	0.01	0.01
workspace cleanliness	1.44	2	1.48	1.41	2	1.54	0.03	0	0.977	0	0
comfort of furnishing	1.6	2	1.28	1.31	2	1.53	0.29	0	<0.001 ***	0.08	0.09
building	1.54	2	1.35	1.28	2	1.5	0.26	0	<0.001 ***	0.08	0.09
amount of space	1.58	2	1.57	1.23	2	1.72	0.35	0	<0.001 ***	0.10	0.12
colors and textures	1.42	2	1.35	1.27	2	1.59	0.15	0	0.146	0.02	0.03
workspace	1.33	2	1.37	1.15	2	1.47	0.18	0	0.001 **	0.06	0.06
air quality	1.27	2	1.56	1.13	2	1.59	0.14	0	0.002 **	0.05	0.06
adjustment of furniture	1.19	2	1.56	1.08	2	1.65	0.11	0	0.095	0.03	0.03
visual comfort	1.08	2	1.63	1.04	2	1.69	0.04	0	0.732	0.01	0.01
visual privacy	0.5	1	1.78	0.38	1	1.96	0.12	0	0.19	0.02	0.03
temperature	**0.56**	**1**	**1.71**	**0.05**	**0**	**1.82**	**0.51**	**1**	**<0.001 *****	**0.14**	**0.16**
noise	0.14	0	1.79	0.22	0	1.82	-0.08	0	0.223	-0.02	-0.02
sound privacy	-0.66	-1	1.83	-0.64	-1	1.94	-0.02	0	0.876	0	0

[a] We ordered the results by mean satisfaction score for each category based on the full database. We indicate in bold the variable for which there is the largest difference between the two groups.

When comparing the two types of building systems, temperature satisfaction shows the largest difference in all these measures (ΔM=0.51, p<0.001, ΔMdn=1, ρ=0.14, ∂=0.16) in favor of the radiant subset. In Figure 28 (left), we show boxplots of temperature satisfaction for radiant and all-air systems. Although the effect size based on Spearman's ρ was larger for temperature satisfaction than the other survey categories, it could be considered as either negligible ($\rho < 0.2$) or small ($0.1 \le \rho < 0.3$) depending on the reference used for effect size thresholds (Ferguson 2009; Cohen 1988), or otherwise given the lack of established effect size thresholds for our field. In section 0, we conduct more analysis on the interpretation schemes of this index.

After temperature satisfaction, the second biggest difference in means is for satisfaction with perceived amount of space, but with no difference in median values (ΔM=0.35, p<0.001, ΔMdn=0. ρ=0.1, ∂=0.12). Aside from temperature satisfaction and perceived amount of space, the overall differences observed between the radiant and all-air groups are very small, with no difference in median, and negligible effect size. Overall building satisfaction shows a difference in means of 0.26 (ΔM=0.26, p<0.001, ΔMdn=0, ρ=0.08, ∂=0.09) in favor of the radiant subset. Acoustic satisfaction (noise and sound privacy) did not show statistically significant differences between the radiant and all-air groups. This is noteworthy because previous survey results had indicated lower acoustic satisfaction with radiant buildings due to large areas of exposed concrete surfaces [19]. Additional analysis by type of office is reported in section 0 to provide further insights into the acoustic satisfaction.

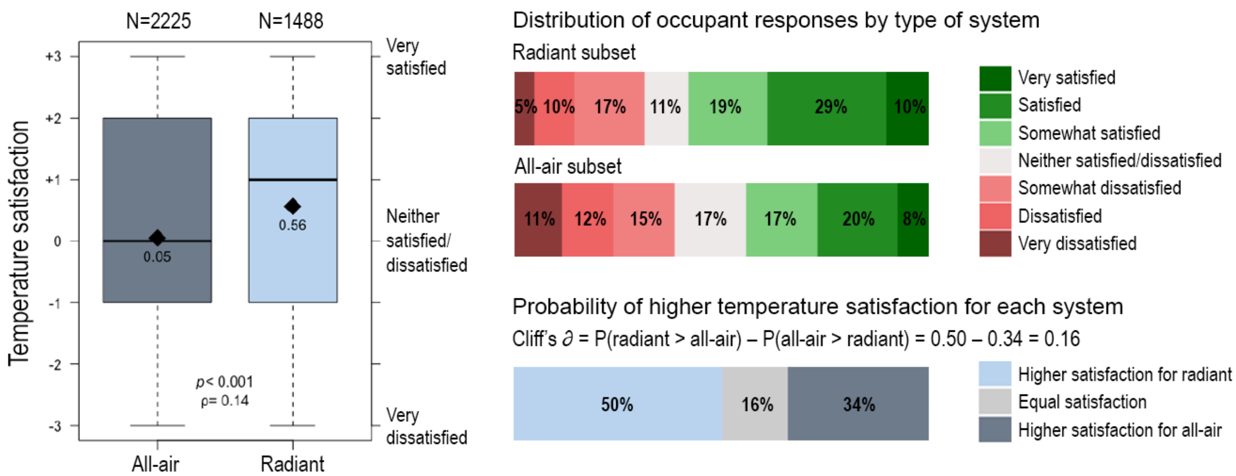


Figure 28: (Left) Boxplot of temperature satisfaction in which diamond dots represent mean values; (upper right) Bar chart showing the distribution of temperature satisfaction ; and (lower right) Probability of higher temperature satisfaction for the radiant and all-air conditioning subsets.

Effect size analysis

Spearman's rho (ρ) interpretation thresholds

We used Spearman's rho (ρ) effect size as a measure of association. We followed the method defined in section 5.2.3 to generate scenarios of 'best' vs. 'worst' groups, which we compared against one another using Spearman's rho effect size. We conducted this analysis using temperature satisfaction which, as noted previously, showed the highest effect size. Table 6 reports the results of this analysis and shows a difference in means between the extreme groups of 2.82, which is substantial (47% of the scale). The effect size is ρ=0.65, which represents a large effect according to Cohen (Cohen 1988), but only a moderate effect according to Ferguson (Ferguson 2009). The two authors propose the values of 0.5 and 0.8, respectively, as their thresholds for large effect. The Spearman's rho obtained for our extreme scenarios is exactly in-between these two cited thresholds for a large effect. This analysis aimed to contextualize the threshold proposed. Cohen's thresholds have been criticized for over-interpretation. On the other hand, according to Ferguson's thresholds, the difference between the best 3-5 and worst 3-5 buildings in a 60 building sample (i.e. an extreme scenario) would be defined as 'moderate', not 'large', which may be unreasonable. As noted above, both Cohen and Ferguson commonly warn about the challenge of interpretation of effect sizes, which vary from one field to another, and our field does not yet have accepted values for these thresholds. Defining appropriate effect size thresholds is difficult and beyond the scope of this paper. From this comparison, we can infer that there is a tendency of higher temperature satisfaction for radiant systems, but with either a negligible or small practical significance.

Table 6: Resulting effect sizes for extreme scenarios

Scenario	∆Mean	Effect size Spearman's ρ
3-5 best/worst buildings for temperature satisfaction ~ 106 vs. 132 votes (the two groups represent 6.1% of the full sample)	2.82	0.65

Comparison of effect size for other building characteristics

In this section, we used the total sample of this study (from the 26 radiant and 34 all-air buildings). We re-created two groups based on a series of variables: ventilation strategy (mixed-mode vs. mechanical), building size (≥ or < median size), building age (≥ or < median age), window to wall ratio (WWR) (> or ≤ 50%), distance to window (≤ or >4.6 m), type of office (open office vs. enclosed), and gender (male or female). Table 7 compares the two groups against each other using differences in means, medians, *p-values* and Spearman's rho effect sizes. The results provide a way to compare the outcome obtained for our radiant vs. all-air analysis to other variables. We note that, for all the variables tested, gender has the highest effect size for temperature satisfaction (ρ=0.2), followed by conditioning type (ρ=0.14). The other variables show low effect size comparatively. Karjalainen (Karjalainen 2012) conducted a meta-analysis to determine the impact of gender on thermal comfort. His results showed that females were more likely than males to express thermal dissatisfaction (odds ratio: 1.74, 95% confidence interval: 1.61–1.89). He concluded that there was a statistical difference based on p-value, but did not comment on effect size thresholds for practical significance. Applying Ferguson's proposed thresholds, where an effect size less than an odds ratio of 2 is a "negligible" effect, would suggest that the effect of gender within Karjalainen's analysis remains below the recommended minimum effect size for a practically significant effect. For our sample, gender just reaches the threshold of 'small' practical significance according the Ferguson's scale for Spearman's rho. The outcome for conditioning (radiant vs. all-air) is below gender. Therefore, as with gender, we can conclude that there is a tendency toward higher temperature satisfaction for radiant systems, but with either a negligible or small practical significance.

Table 7: Comparison of effect size for building characteristics

				Comparison			Effect size
Category	Variables	Group 1	Group 2	∆Mean	∆Mdn	p-value	Spearman's ρ
temperature	gender[(a)]	male	female	0.74	1	<0.001 ***	0.2
satisfaction	conditioning	radiant	all-air	0.51	1	<0.001 ***	0.14
	ventilation strategy	mixed-mode	mechanical	0.09	1	0.139	0.02
	building size	< median (6763 m^2)	≥ median (6763 m^2)	-0.07	-1	0.241	-0.02
	WWR	≤ 50%	> 50%	0.09	0	0.221	0.02
	distance to window	close (≤4.6 m)	far away (>4.6 m)	0.05	1	0.535	0.01
	type of office	enclosed office	open office	-0.02	0	0.898	0

Probability of higher, lower and equal temperature satisfaction

We looked at the Cliff's delta (∂) effect size that, in our case, measures the probability of higher temperature satisfaction. We also decomposed the Cliff's delta equation to determine the probability that a randomly selected observation from the radiant group has higher satisfaction than a randomly selected observation from the all-air group P(radiant > all-air), its reverse probability P(all-air < radiant), and the probability of equal satisfaction P(radiant = all-air). In a space using a radiant system, a person has a 50% chance of having a higher temperature satisfaction rating, a 16% chance of having an equivalent rating, and a 34% chance of having a lower temperature satisfaction rating than in an all-air building. Figure 28 (lower right) displays the distribution of these three probabilities. The Cliff's delta associated with this analysis is ∂=0.16 (or 50%-34%=16% probability of higher temperature satisfaction for occupants in buildings with radiant systems). We could not find references for interpretation for Cliff's delta values. Thus, this analysis should be viewed as a useful means of interpreting the results of the survey, and nothing more.

Analysis by building

In this section, we move from individual responses to building scores as the main unit of analysis. Figure 29 shows the distribution of temperature satisfaction for each building ordered by means. We represent each building equally, independently from the number of votes. Buildings with radiant conditioning systems dominate the upper third of the graph; they demonstrate higher means and medians than all-air buildings.

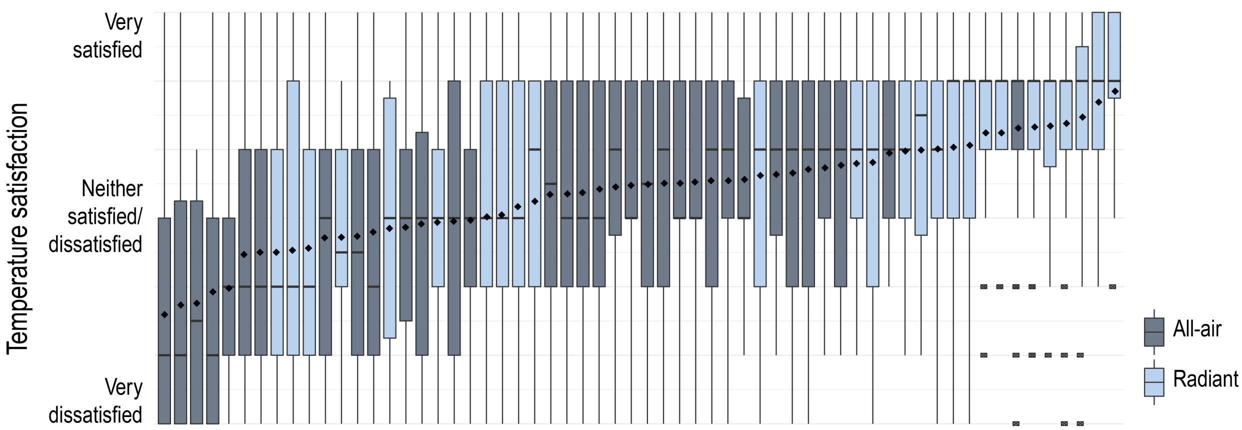


Figure 29: Boxplot of temperature satisfaction by buildings ordered by mean value (diamond dots). Colors indicate the type of conditioning system used (blue for radiant and gray for all-air)

ASHRAE Standard 55's objective of 80% satisfied occupants

Per ASHRAE Standard 55-2013 (ANSI/ASHRAE 2016) with approved addenda and errata as of 2016, buildings are intended to achieve 80% satisfaction with regards to thermal comfort. The standard defines a method of assessment based on occupant survey results: "the probability of occupants satisfied shall be predicted from seven-point satisfaction survey scores by dividing the number of votes falling between -1 and +3 inclusive, by the total number of votes". We assessed this objective with the buildings of our study based on temperature satisfaction for both the radiant and all-air subsets. We also assessed ASHRAE's 80% objective by looking at 'neutral to satisfied' and 'slightly satisfied to satisfied' votes, as the latter is what was prescribed in the original version of the standard (ANSI/ASHRAE 2013). Table 8 and Figure 30 reports the results. This analysis shows that 57% of the buildings of this study meet the requirements of the current standard (slightly better results for the all-air group). Reducing ASHRAE's requirement to the 'neutral to satisfied' votes brings a significant drop. Only 10% of the buildings would comply with this definition of satisfied. When accounting only for positive votes ('slightly satisfied' to 'very satisfied' rating), as in the original version of the standard, only 2 radiant buildings (out of 26) and none of the all-air buildings of this study provided a satisfactory thermal comfort level to at least 80% of their occupants. If we look more closely at the larger database from (Altomonte and Schiavon 2013) (144 buildings), 44% of the buildings meet the 'slightly dissatisfied' to 'very satisfied' interval, 10% meet the 'neutral' to 'very satisfied' interval, and only 1% meet the 'slightly satisfied' to 'very satisfied' interval. In other words, the buildings of this study are outperforming the larger sample. We note that this study did not comply with the response rate suggested by the standard (section 7.3.1 (ANSI/ASHRAE 2016)) for all the buildings in the analysis. Despite this point, we may not expect a large difference in the outcome. This analysis showed that this standard's objective for thermal comfort assessment based on occupant satisfaction surveys does not appear realistic in its practical application. The number of complying buildings remains surprisingly low, despite the fact that the current proposed metric includes 'slightly dissatisfied' votes among positive responses. This inclusion is further questionable as it brings a contradiction to the definition of thermal comfort according to the same standard ("the condition of mind that expresses satisfaction with the thermal environment").

Table 8: Percent of buildings that provide 80% occupant satisfaction with temperature

	% of building that shows a **slightly dissatisfied to very satisfied rating** (-1 to +3 votes) for at least ***80%*** of its occupants[(a)]	% of building that shows a **neutral to very satisfied rating** (0 to +3 votes) for at least ***80%*** of its occupants	% of building that shows a **slightly satisfied to very satisfied rating** (+1 to +3 votes) for at least ***80%*** of its occupants[(b)]
Radiant	54% (14/26 buildings)	15% (4/26 buildings)	8% (2/26 buildings)
All-air	59% (20/34 buildings)	6% (2/34 buildings)	0% (0/34 buildings)
Radiant & all-air	57% (34/60 buildings)	10% (6/60 buildings)	3% (2/60 buildings)
Database (Altomonte and Schiavon 2013)	44% (63/144 buildings)	10% (14/144 buildings)	1% (2/144 buildings)

[(a)] As in ASHRAE Standard 55-2013 (2016 update), section 7.4.1 (ANSI/ASHRAE 2016);
[(b)] As in ASHRAE Standard 55-2013 (original version), section 7.4.1 (ANSI/ASHRAE 2013)

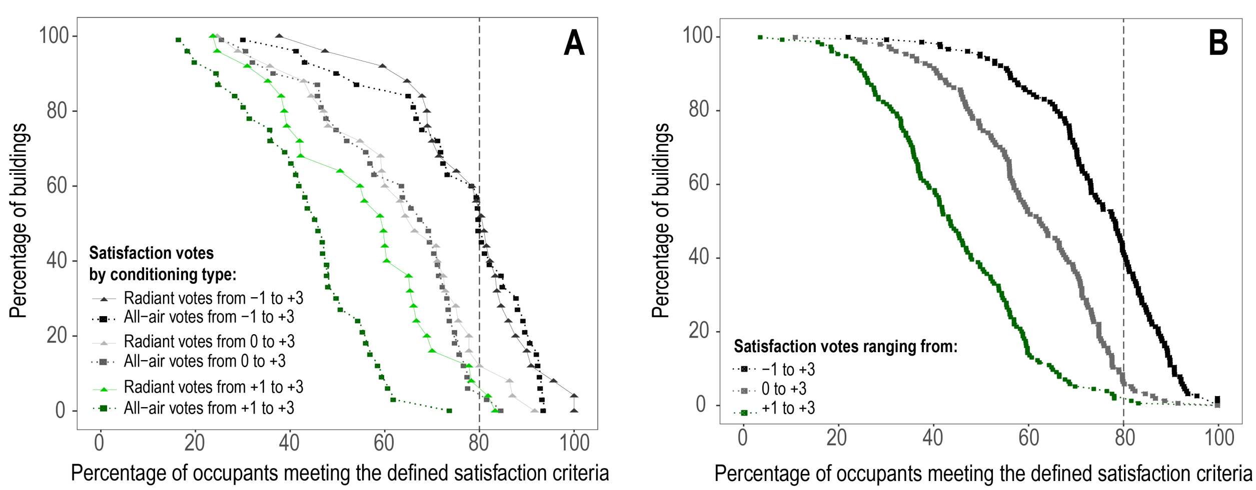


Figure 30: Percentage of building meeting the ASHRAE Standard 55-2013 objective of 80% satisfied occupants for (A) the radiant and all-air buildings of this study, and (B) the 144 buildings from (Altomonte and Schiavon 2013). The 80% satisfaction objective is represented with a vertical dashed line.

Mixed-effects models for temperature and acoustic satisfaction

Mixed-effects models for temperature satisfaction

Table 9 presents the results of testing a mixed-effect linear model to predict temperature satisfaction based on radiant and all-air conditioning types. We used 'building' as a random effect. We calculated the random effect (between groups variability) based on the intercept and residual standard deviations. The random effect reached 12%, which means that 'between-building' differences describes 12% of the overall variance in temperature satisfaction (as Figure 29 suggests). The regression coefficient (in this case equal to the difference in means) for 'radiant system' conditioning type is 0.52 suggesting a minor impact of the random effect on the outcome (compared to 0.51 as found without random effect).

Previous studies have emphasized the impact of type of ventilation (mechanical vs. mixed-mode ventilation) on occupant temperature satisfaction (Brager and Baker 2009). We used a mixed-effect linear model with interactions to predict the combined effect of conditioning types (radiant/all-air) and ventilation types. For this model, the fixed intercept describes the prediction of temperature with all-air and mechanical ventilation. The three regression coefficients to temperature satisfaction are associated with: 'radiant', 'mixed-mode ventilation', and for the interaction of 'radiant and mixed-mode ventilation'. Only the correlation for 'radiant' is statistically significant. Its regression coefficient (0.59) is higher than in the previously tested models, but does not account for the ventilation strategy (as it is not a statistically significant outcome). This analysis reveals that the type of ventilation cannot explain further differences in temperature satisfaction, thus we exclude this variable in subsequent models. We used a mixed-effect linear model to determine the correlation between conditioning types (radiant/all-air) and climates on temperature satisfaction (see Table 4 for the detail of types of climates). Only the regression coefficient for 'radiant' reached statistical significance (regression coefficient of 0.51, p=0.01). The difference between climates cannot explain the differences observed in temperature satisfaction, thus we also excluded this variable in subsequent models.

Table 9: Linear models -with and without mixed-effects- for temperature and acoustics categories

Prediction of	Variables	Population	Equation for random effect	Intercept (fixed)	Regression coefficient	Random effect	Difference rad / all-air
temperature	conditioning	Full sample	1 | bldg. ID	0.10 [(a)]	0.52 (p=0.005) cond. -Radiant	12%	+0.52 (radiant)
temperature	conditioning * ventil. type	Full sample	1 | bldg. ID	0.03 [(b)]	0.59 (p=0.03) cond. -Radiant 0.26 (p=0.28) ventil. type -MM -0.26 (p=0.48) interact.	12%	+0.59
temperature	conditioning + climate	Full sample	1 | bldg. ID	-0.29 [(c)]	0.51 (p=0.006) cond. -Radiant 0.40 (p=0.24) climate-Cool 0.48 (p=0.15) climate-Mixed 0.41 (p=0.25) climate-Warm	12%	+0.50 (radiant)
noise	conditioning (radiant by type)	Full sample	1 | office type	0.23 [(a)]	0.02 (p=0.78) cond. -Inslab 0.03 (p=0.72) cond. -Panels	8%	n.s.
sound privacy	conditioning (radiant by type)	Full sample	1 | office type	-0.61 [(a)]	0.17 (p=0.02) cond. -Inslab 0.05 (p=0.58) cond. -Panels	21%	n.s.

[(a)] intercept for all-air; [(b)] intercept for all-air and mechanical ventilation; [(c)] intercept for all-air and 'very cold' climate; [(d)] intercept for all-air and 'very dissatisfied' with building

Mixed-effects models for acoustic satisfaction
Buildings using radiant systems are often associated with lower acoustical quality; this is particularly the case for ESS and TABS types of radiant systems due to large, exposed, and acoustically reflective surfaces (Bauman et al. 2011). Based on Table 5, neither of the two acoustic categories (noise and sound privacy) showed statistically significant differences in satisfaction ratings between the radiant and all-air subsets. Previous occupant satisfaction studies have shown that the type of office has a major impact on acoustic satisfaction (Frontczak et al. 2012; Kim and de Dear 2013). We used a mixed-effect model to better describe acoustic satisfaction categories for radiant and all-air buildings with 'type of office' (cubicles with high partitions, cubicles with low partitions, enclosed private office, enclosed shared office, open office with no partitions) as the random effect. For this model, we distinguished in-slab (ESS & TABS) from panel (RCP) types of radiant systems. Table 9 presents the results of these models. The output for noise satisfaction was not statistically significant between the two groups. Satisfaction with sound privacy showed a weakly significant regression coefficient (+0.17, p=0.02) in favor of in-slab radiant systems compared to all-air systems. The random effect equaled 21% suggesting that the large spread in the variance can be described by 'between office type' differences. In Figure 31, we can clearly see that sound privacy is more of an issue for open space offices (with or without partitions). Overall these results reveal that acoustic satisfaction categories are comparable across the two conditioning types. This outcome is relevant because it provides evidence disproving common biases against radiant systems specifically. Acoustic satisfaction appears as most challenging aspect in regard to occupant satisfaction in buildings.

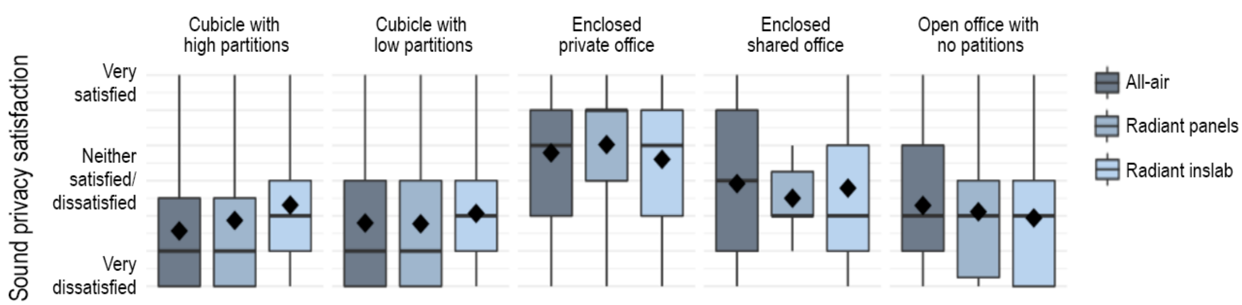


Figure 31: Boxplots of satisfaction with sound privacy for different types of conditioning systems and in offices of different types. Diamond dots represent mean values

5.3.4 Study limitations

We selected the buildings of this study following the methodology detailed in section 5.2.2. Collecting data for the radiant subset was difficult due to the general lack of buildings with radiant conditioning in North America. As described above, we sampled the all-air buildings data from a larger database based on characteristics that followed the radiant buildings demographical and physical characteristics (see section 0). Overall, the buildings of this study (both conditioning types) have a higher environmental quality compared to the average building of the CBE survey database. As a reference, the mean overall workspace and overall building satisfaction ratings considering the entire CBE database are 0.93 (N=76,598) and 1.06 (N=80,869), respectively, while they reach 1.22 (N=3,573) and 1.38 (N=3,574), respectively, for all the buildings of this study. This study involved 26 radiant surveys and 34 all-air buildings, with 1,645 and 2,247, occupants respectively. While this is a large sample size, it is not a randomized statistically-representative sample, which is a limitation of the study.

We sampled the all-air buildings by creating a subset of the CBE IEQ survey database from 2011 (Altomonte and Schiavon 2013). The radiant subset was mostly based on survey data collected within the framework of this study (2016 and 2017). This difference in database collection times resulted in a difference in age of buildings between the two subsets (a difference in median year of construction of 6 years, see Table 4 for more detail). Frontzcak et al. (Frontczak et al. 2012) qualitatively observed that building age affects building satisfaction, though the categories of building age differed by decades instead of years. In the context of the length of the lifetime of a typical building, and the pace of change in the industry regarding common design practice and material selection, we do not believe this age difference is a major confounding factor, but this unavoidable disparity between the two datasets may have had an effect of the overall study results.

In addition to comparing the 26 radiant buildings to the 34 all-air buildings of this study, we compared them to the dataset from (Altomonte and Schiavon 2013) (144 buildings). The highest difference between the two groups was observed for air quality satisfaction ($\Delta M=0.68$, $p<0.001$, $\Delta Mdn=1$. $\rho=0.12$, $\partial=0.25$), followed by temperature satisfaction ($\Delta M=0.68$, $p<0.001$, $\Delta Mdn=1$. $\rho=0.1$, $\partial=0.21$), both in favor of the radiant subset. The present study did not show such a large difference in air quality satisfaction, which can be related to the exclusion of larger and older buildings within our all-air sample (to match the building characteristics of the radiant subset). The difference observed for temperature satisfaction confirms the overall outcome of this study (negligible to small effect in favor of the radiant subset, depending on effect size interpretation thresholds).

While temperature satisfaction showed the largest difference between the radiant and all-air subsets, there was also a difference in means for overall building satisfaction between the two groups ($\Delta M=0.26$, $p<0.001$, $\Delta Mdn=0$. $\rho=0.08$, $\partial=0.09$). Though statistically significant, this has negligible practical significance based on the effect size thresholds discussed earlier. However, this difference might indicate that buildings with radiant systems offer a slightly more satisfactory building design, an interpretation that we believe may be realistic. We performed this study on buildings in the U.S. and Canada, where radiant systems are not common practice. Designers adopting radiant systems in these locations go beyond ‘business as usual’, which brings with it a certain ambition and motivation and may trigger a more thoughtful design process. This in turn, may be reflected in the overall results.

The satisfaction questions within the CBE survey are based on an ordinal scale. In the present analysis, we often treated these observations as though they were on an interval scale, which assumes an equal distance between answers (e.g., same interval between ‘slightly satisfied’, ‘satisfied’, ‘very satisfied’). This is a common simplification in this field as interval scales allow for analysis using descriptive statistics and modelling (analysis of means and linear models). Other types of analysis (rank based correlations, analysis of medians, probability of superiority, Spearman rho effect size) respected the ordinal structure of the data.

Our paper included an analysis on effect size interpretation thresholds as a precedent for discussion as we could not find thresholds that are representative of our field in existing publications. Further research in this area may yield a different overall conclusion as to the practical significance of the observed and reported effect sizes.

5.4 Conclusions

We used the CBE IEQ occupant survey to compare occupant satisfaction with their indoor environment in radiant and all-air conditioned buildings. This study involved the administration of new occupant satisfaction surveys to 1,284 people (20 buildings) exposed to radiant systems. We supplemented this dataset with responses from 361 occupants (6 buildings) previously collected. For the all-air sample, we used a subset of the CBE database that aligned with key building characteristics of the radiant subset. This comparison involved 1,779 respondents from 26 buildings with radiant systems and 1,978 respondents from 34 buildings with all-air systems. To our knowledge, this is the largest such dataset used in a comparison study of occupant satisfaction in radiant buildings. The main conclusions of this study are:

- The analysis shows that radiant and all-air buildings have equal indoor environmental quality, including acoustic satisfaction, with a tendency towards improved temperature satisfaction in radiant buildings.
- From this dataset, a person has a 50% chance of experiencing a higher temperature satisfaction in a space using a radiant system compared to an all-air system. The reverse probability reaches 34%. There is a 16% chance for the two systems to bring equal satisfaction.
- Acoustic satisfaction showed the lowest scores from all the categories surveyed. This result shows acoustical quality to be the most challenging aspect in regard to occupant satisfaction in buildings. It is important for designers to pay more attention to improve acoustical experience in buildings. We observed equal acoustic satisfaction (noise and sound privacy) in radiant and all-air systems, disproving some commonly held biases against radiant systems.
- Less than 60% of the buildings used in this analysis met the ASHRAE Standard 55 thermal comfort objective based on post-occupancy surveys. This result is surprisingly low, in particular as the current metric within the standard includes 'slightly dissatisfied' votes among the positive comfort responses. This observation raises questions regarding the practicality and applicability of the comfort metric as currently written.

6 Cooling capacity and acoustic performance of radiant slab systems with free-hanging acoustical clouds

This chapter focus on the change in cooling capacity and acoustic performance of an office room with a radiant cooled ceiling and free-hanging acoustical clouds. It follow-up on previous research detailed in the introduction (section 1.4). This section presents the testing conducted without air movement.

6.1 Background

Radiant slab systems have the potential to achieve significant energy savings primarily due to the use of lower temperature differences between the space and the heating or cooling source and the possibility to store energy in the slab (Babiak, Olesen, and Petras 2009). This allows for higher efficiency at the device that generates hot or cold water, such as a boiler or chiller. In some cases, the relatively lower chilled water temperatures may eliminate the need for a chiller altogether, thereby enabling the use of non-refrigerant cooling sources, such as ground water, cooling towers, or indirect evaporative coolers. The use of water as the circulating fluid instead of air also reduces the energy costs of transferring heat to or from the space. Lastly, the thermal inertia inherent in a radiant slab system also provides opportunity for peak demand reduction and load shifting (Meierhans 1993; Feustel and Stetiu 1995; B. W. Olesen et al. 2006; Lehmann, Dorer, and Koschenz 2007; Gwerder et al. 2008; Babiak, Olesen, and Petras 2009). However, the low temperature differences require large surfaces to ensure heat exchange with the space. As a result, designers typically implement radiant systems on the largest surfaces in the space (e.g., ceilings or floors).

By definition, a radiant heating and cooling system exchanges at least 50% of its heat through thermal radiation (ASHRAE 2012a). Radiant heat exchange between surfaces involves view factors, which represent the proportion of the radiation that leaves a first surface that strikes a second surface. To preserve heat transfer (and avoid compromising view factors), it is desirable to keep radiant systems uncovered. In the case of a commercial building, this is in direct conflict with the need to add acoustic absorption in the ceiling plane to improve the acoustic quality of the space. A suspended acoustical ceiling is the least expensive approach to adding sound absorption but it also requires a large surface area to be effective for reverberation control within the space. Practitioners have identified this conflict as a barrier for the implementation of radiant systems (Moore, Bauman, and Huizenga 2006). Occupant satisfaction studies conducted in buildings using radiant systems are commonly showing lower acoustic satisfaction compared to non-radiant buildings (Bauman et al. 2012) proving that acoustic quality in buildings using radiant systems needs to be addressed.

All materials have the ability to absorb and reflect sound. Acousticians refer to the sound absorption coefficient as an indicator of the fraction of sound energy absorbed by a surface. Sound absorption coefficients depend on frequency and their value varies between zero (a perfectly reflective surface) and one (a perfectly absorptive surface). Massive radiant types such as thermally active building systems (TABS) are made of exposed concrete, whose sound absorption coefficients are typically 0.02 over the normal frequencies (Egan 2007). This means that TABS cause excessive sound reflection in spaces, which yield too much reverberation. By contrast, porous sound absorbing materials have sound absorption coefficients typically ranging between 0.30 and 0.90 depending on frequencies and types (Egan 2007). Sound absorbing material reduce both the sound level and reverberation time, which are key factors in the design of offices for acoustic comfort.

In this study, we wanted to address the problem of poor sound absorption of TABS. We proposed to combine a radiant cooled ceiling with several configurations of free-hanging discontinuous acoustic clouds (sometimes called canopies –see illustration Figure 32). This type of sound absorber is known to have an increased acoustic performance compared to regular suspended ceilings due to the larger surface area exposed to sound, i.e., both the upper and lower surfaces, since sound has access to both sides of the cloud. For simplicity, when referring to ceiling coverage for these clouds, we report the perpendicular projection of the cloud on the total ceiling area. It has been reported that covering 60% of a ceiling with free-hanging cloud can have comparable performance to 100% coverage with a suspended ceiling (Chigot 2010). As they are free-hanging, these clouds have an open air space above them. Air can freely circulate between the cloud and the ceiling allowing heat exchange by convection from the radiant cooled ceiling.

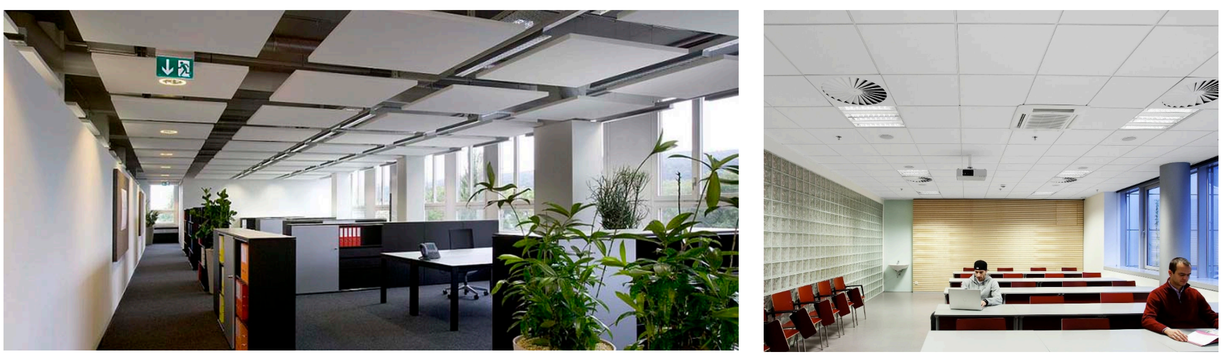


Figure 32: Free-hanging clouds and acoustic tiles (Images: Ecophon (left) and Armstrong World Industries (right))

The combination of radiant cooled ceiling with free-hanging acoustic clouds has raised research interest over the last decade, but we did not find experimental-based peer-reviewed journal publications on this topic. The conference papers and manufacturer white papers that we found consistently showed that the reduction of cooling capacity between slab and room was small and lower than expected (Weitzmann, Pittarello, and Olesen 2008; Peperkamp and Vercammen 2009; Chigot 2010; Le Muet, Peperkamp, and Machner 2013; Ecophon 2014). Overall, it appears that partial coverage of a radiant chilled ceiling with free-hanging clouds does not cause a proportionally equivalent reduction of the cooling capacity. Instead, the reduction in cooling capacity is 3-4 times lower than the percentage cloud coverage. This outcome opens doors to the potential combination of acoustical clouds with TABS.

The objectives of this study are to: (1) experimentally assess the effect on radiant ceiling system cooling capacity for various coverage areas of free-hanging acoustic clouds, and (2) determine the change in sound absorption for the same configurations. We conducted two laboratory studies: the first one in a certified controlled climatic chamber equipped with a radiant cooled ceiling and free-hanging clouds, and the second one in a certified reverberant chamber equipped with comparable configurations of acoustic clouds.

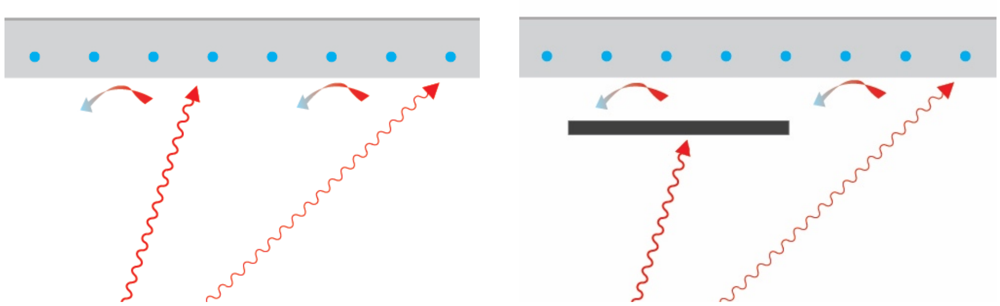


Figure 33: Section of a radiant TABS with no acoustical treatment (left) and with free-hanging acoustical clouds (right)

6.2 Method

6.2.1 Cooling capacity measurements

Experimental facilities and room description

We conducted the cooling capacity experiments in a climatic chamber (4.27 m × 4.27 m × 3.0 m) at Price Industries in Winnipeg, Manitoba in September 2015. The floor area of the chamber is 18.2 m^2, and the volume is 54.7 m^3. The chamber has no windows. The walls, the ceiling, and the floor have similar construction and thermal properties with an overall conductance of 0.135 W/m^2K. Starting from the exterior the detailed wall composition includes: 3.522 m^2K/W insulation, a stagnant 0.102 m air gap (0.352 m^2K/W), aluminum extruded walls, 0.102 m of polyurethane board (3.522 m^2K/W). This chamber is accredited by the EN 14240 (CEN 2004) for chilled ceiling testing. It is located inside a large laboratory facility maintained at 21.6º C ± 0.5º C (we note that the facility temperature was not measured for the test with highest coverage (71%), which was the first experiment conducted).

We installed graphite metal radiant panels in the chamber as part of a suspended ceiling placed at a height of 2.5 m above the floor. Each radiant panel is 1.83 m long and 0.61 m wide (area equal to 1.11 m^2). Cotton fiber insulation was placed on top of the panels (2.288 m^2K/W). In total, twelve radiant ceiling panels were centered on the ceiling, covering 73.5% (13.4 m^2) of the total area. The twelve panels were the maximum number that could be accommodated in the chamber. The intention was to represent TABS, the system of interest in this study, as near as possible by covering close to 100% of the ceiling area. We connected the panels in series in order to reduce overall measurement uncertainty (we reached lower uncertainty by increasing the temperature difference between supply and return water temperature). To ensure a more uniform distribution of surface temperatures at the ceiling level, the panels were connected in mixed order so that the average temperature of adjacent panels was similar. The chamber did not include an air system because we wanted to specifically focus on the change in cooling capacity of the radiant chilled ceiling, and adding an air system would increase overall uncertainty in the results.

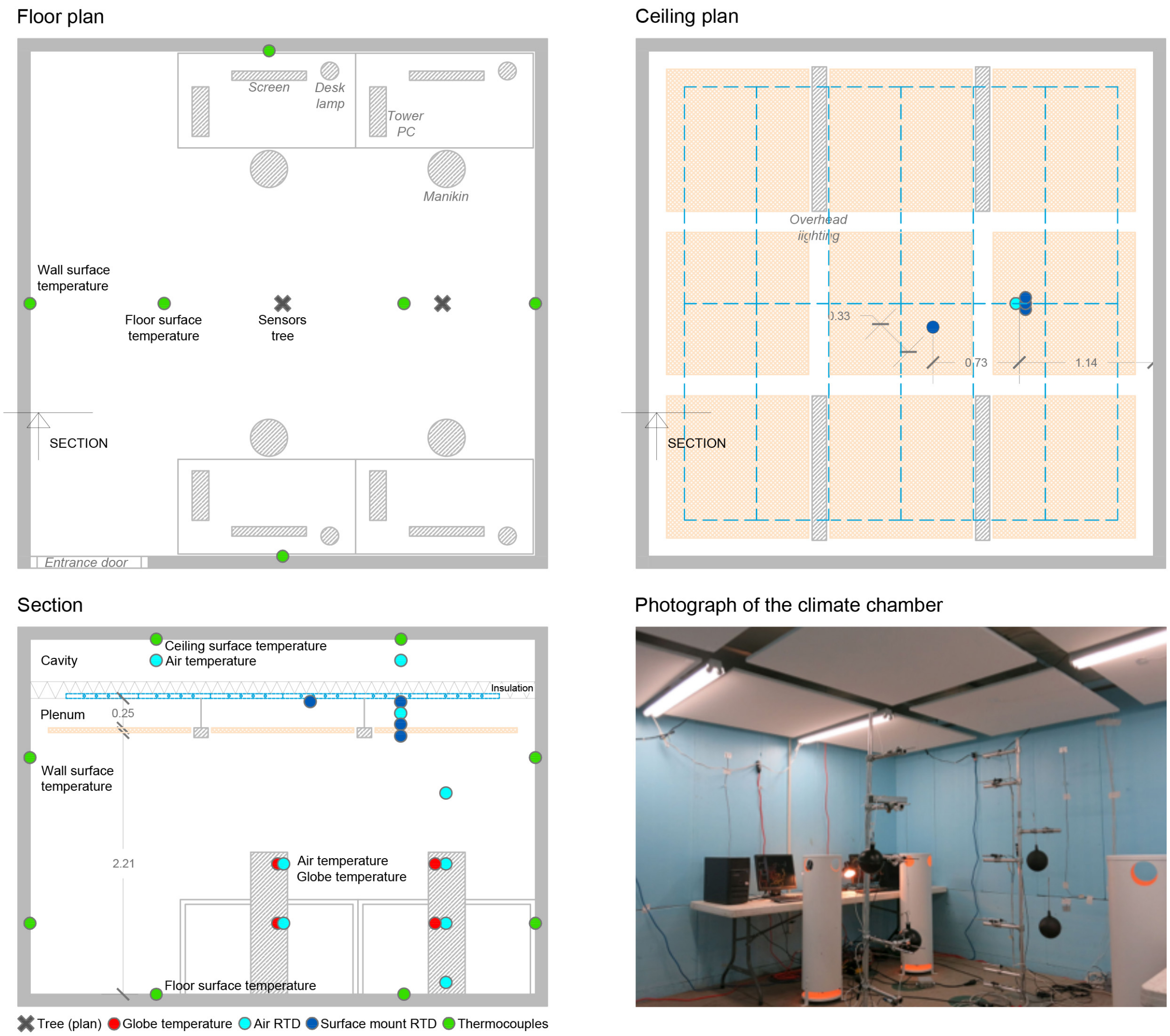


Figure 34: Plan, section and photograph of test chamber with sensors and cloud coverage. The acoustical clouds are in orange, the radiant ceiling in blue and the furniture/equipment in gray.

Table 10 summarizes the heat sources used in the experiment. We modeled office heat sources using computers (tower CPUs), flat screens and desk lamps on the desks, and overhead lighting (0.25 m below the radiant ceiling). We simulated occupants with power-adjustable dummies according to EN 14240 (CEN 2004). When fully installed, the test chamber represented a four-person office with a computer at each workstation; a relatively high occupant density of 4.55 m^2 per person. The data acquisition system was located outside the chamber, and therefore not listed here.

We also equipped the chamber with acoustical panels made out of fiberglass pre-formed in a 1.20 m square and 0.04 m thick cloud shape. These clouds were suspended at 0.25 m below the radiant ceiling (similar to the overhead lighting) centered on the ceiling in a 3 by 3 array, 1.37 m center to center. A maximum of 9 clouds could be positioned on the ceiling with a spacing of 0.17 m between them and 0.16 m between them and the walls. The overhead lighting fixtures could fit in the space in between two clouds. Figure 34 shows a plan and a section of the chamber with locations of the four simulated workstations and instruments. Blue lines represent the area of the radiant ceiling and orange squares represent the acoustic clouds. Figure 34 also includes a photograph of the lab set-up with 9 clouds (71% coverage) in the room.

Table 10: Heat load summary

Heat source	Number	Total power (W)	Power per floor area (W/m^2)	Percent of total (%)
People (dummies)	4	300	16.5	27%
Computers	4	300	16.5	27%
Desk lamp and screens	4	283	15.5	26%
Overhead lighting	4	214	11.8	20%
Total		1097	60.3	100%

Experimental conditions and procedures

We used the operative temperature measured in the center of the room at 1.1 m as our reference temperature to determine the cooling capacity between the ceiling and the room. The operative temperature accounts for both convective and radiant heat exchange mechanism and it is also a reference for thermal comfort analysis. It is often suggested as the most appropriate reference temperature for total heat transfer coefficient calculations (Bjarne W. Olesen et al. 2000). As this experiment did not include air movement, we approximated the operative temperature with the mean of the air and mean radiant temperatures. The mean radiant temperature was derived from the globe temperature with the following equation:

$$T_{mr} = \left[\left(T_{globe} + 273\right)^4 + 2.5 \cdot 10^8 \cdot v_a{}^{0.6} \cdot \left(T_{globe} - T_a\right)\right]^{1/4} - 273 \qquad (1)$$

The heat exchange between the radiant ceiling and the room is expressed in Equation 1, where U_{cc} represents the cooling capacity coefficient in ($W \cdot m^{-2} \cdot K^{-1}$).

$$q_{cc} = U_{cc}\, A_{panels}\left(t_{op} - t_{w,m}\right) \qquad (2)$$

$$\text{with } t_{op} = \frac{t_a + t_{mr}}{2}, \text{ and } t_{w,m} = \frac{t_{w,s} + t_{w,r}}{2}$$

On the water side, the radiant system cooling rate is expressed as:

$$q_w = \dot{m}_w\, c_{p_w}(t)\left(t_{w,r} - t_{w,s}\right) \qquad (3)$$

Under steady state conditions, the water in the radiant panels absorbs the electrical power of the heat sources, and thus:

$$P = q_{cc} = q_w \qquad (4)$$

By substituting (2) & (3) in (4), and rearranging:

$$U_{cc} = \frac{\dot{m}_w\, c_{p_w}\left(t_{w,r} - t_{w,s}\right)}{A_{panels}\left(t_{op} - t_{w,m}\right)} \qquad (5)$$

We conducted all tests with the same water mass flow rate (220 kg/h ± 0.02%) and water supply temperature (15º C ± 0.01º C). As the internal loads were kept constant (1097 W) for the different variants, the water return temperature also remained constant for all the tests (19.2 ± 0.1º C). From equation (5), the remaining variable able to influence the cooling capacity is the operative temperature, which is the equilibrium temperature at which the room settles for a given variant of ceiling coverage under steady state conditions.

We recorded the data once each test had reached steady state conditions. We defined it as a difference of less than 0.05 °C between the mean of the most recent 60 30-sec samples against the mean of the 60 samples immediately prior for every temperature sensor used in this experiment. After recording the data we calculated: (1) the average temperatures in the occupied zone, and (2) the cooling capacity coefficient U_{cc} according to Equation 5.

The experimental sequence was partially randomized. We started with the tests that included the largest amount of clouds (9, 8 and 6 clouds), then conducted the test without coverage and ended with the tests with lowest coverage (4 and then 2 clouds). We also included in the experiments two duplications (6 and 0 clouds) that we ran after all other variants were completed.

Comparison with EN 14240

European Standard EN 14240 (CEN 2004) provides a different methodology to determine the cooling capacity of radiant panels. The expression of the results is of the form: $k \cdot \Delta\theta^n$, where k is the characteristic constant, n the exponent and $\Delta\theta$ the temperature difference between mean cooling water temperature and the reference room temperature (which is the globe temperature, measured in the middle of the room at a height of 1.1 m above the floor). This formula is based on empirical observation and the n exponent is used to account for the non-linearity of radiant heat exchange.

We also used this method to determine the coefficients k and n for our tests. We also calculated a linear model for this equation (using the same $\Delta\theta$). We found that the two models were comparable at predicting the cooling capacity (R^2 of 0.98 for the power model and 0.97 for the linear model). This is due to the moderate range of temperatures in which radiant systems typically operate. As a result, we used a linear model in our analysis. Additionally, we note that the EN 14240 uses the globe temperature as the room reference. Globe temperature is an often used approximation to operative temperature, a relationship that breaks down under some conditions such as elevated air speed. Thus, we used the operative temperature as the room reference.

Instrumentation

Table 11 lists the instruments and equipment used in this experiment. The water supply and return temperatures of the radiant system, and the air temperatures on the trees in the chamber were monitored continuously with resistance temperature detector (RTD). We calibrated these prior to the experiment using a Fluke 1524 Reference Thermometer in a Fluke 9171 Metrology dry well calibrator (± 0.029 °C). We shielded air temperature sensors from radiant heat transfer using a reflective aluminium cylinder. At 0.6 m and 1.1 m height on both trees, the globe temperature was measured with a black-globe thermometer. The black-globe thermometer fulfills the requirements of ISO 7726 (ISO, EN 2001). We used RTDs to measure the temperatures of the surface of the radiant panels (above and below) and of the acoustical clouds for a few locations. We measured the inner surface temperatures of the ceiling, floor and vertical walls (12 in total, and two for each wall) using thermocouples. The accuracy obtained after calibration for each type of sensor is reported in Table 11. The electrical power was measured with a Fluke 41 power harmonic analyzer. Figure 34 shows the location of the sensors in plan and section view.

Table 11: List of sensors and specifications for the cooling capacity measurements

Instruments	Accuracy	Unit	Measured variables and sensor location
Air RTD Omega Air RTD PR-25AP-1/10 DIN	±0.07	°C	Air temperatures: - 2 trees at 0.1, 0.6, 1.1, 1.7 m - below the radiant panels and above the clouds
			Globe temperatures: - 2 trees at 0.6 and 1.1 m
Water RTD Omega 1/10 DIN	±0.07	°C	Supply and return water temperatures
Surface Mount RTD Omega Class A Platinum RTD	±0.15	°C	Upper and lower sides of the radiant panels
			Upper and lower sides of the acoustical clouds
Thermocouples Omega Type J 24 Gauge Thermocouple Wire	±0.15	°C	Wall surface temperatures
			Floor and ceiling surface temperature
Anemometer TSI 8475 Velocity Sensors	±3.0 of reading +1% of full scale	%	Central tree at 0.6 and 1.1 m
			Side tree at 0.1, 0.6, 1.1 and 1.7 m
Water Flow Meter Siemens Sitrans Massflo2100 Coriolis Meter	±0.02	%	Radiant cooling panel water flow rate

Uncertainty

We analysed the data in accordance with the ISO 13005 (ISO, EN 2000) and the JCGM 100 guidelines (JCGM 2008) for the expression of uncertainty. The uncertainty in a primary measurement is estimated as the root sum of the square of the uncertainties. The uncertainty due to an individual source of error can be instrument uncertainty, random error due to spatial variation (when averaging spatially distributed sensors), and error from data acquisition system, etc. For each type of measure, the global expanded uncertainty was evaluated according to ISO 13005 with a level of confidence of 95% (coverage factor at k = 2). Error bars represent the uncertainty when presented on the graphs.

The uncertainty of the mean radiant temperature is of the form:

$$u_{t_{mr}} = \sqrt{\left(\frac{\delta t_{mr}}{\delta t_{globe}}\right)^2 u_{t_{globe}}{}^2 + \left(\frac{\delta t_{mr}}{\delta v_a}\right)^2 u_{v_a}{}^2 + \left(\frac{\delta t_{mr}}{\delta t_a}\right)^2 u_{t_a}{}^2} \qquad (6)$$

with: $$\frac{\delta t_{mr}}{\delta t_{globe}} = \frac{2.5\cdot 10^8\cdot v_a{}^{0.6} + (t_{globe}+273)}{4\cdot\left[(t_{globe}+273)^4 + 2.5\cdot 10^8\cdot v_a{}^{0.6}\cdot(t_a - t_{globe})\right]^{3/4}}$$

$$\frac{\delta t_{mr}}{\delta v_a} = \frac{37.5\cdot 10^6\cdot(t_{globe} - t_a)}{v_a{}^{0.4}\cdot\left[(t_{globe}+273)^4 - 2.5\cdot 10^8\cdot v_a{}^{0.6}\cdot(t_a - t_{globe})\right]^{3/4}}$$

$$\frac{\delta t_{mr}}{\delta t_a} = \frac{62.5\cdot 10^6\cdot v_a{}^{0.6}}{\left[(t_{globe}+273)^4 + 2.5\cdot 10^8\cdot v_a{}^{0.6}\cdot(t_{globe} - t_a)\right]^{3/4}}$$

The uncertainty of the radiant system cooling rate can be approximated by:

$$u_{q_w} = c_{p_w}(t)\sqrt{\left(\dot{m}_w\, u_{\Delta t_w}\right)^2 + \left(\Delta T_w\, u_{\dot{m}_w}\right)^2} \qquad (7)$$

with: $\Delta T_w = t_{w,r} - t_{w,s}$, and $u_{\Delta T_w} = \sqrt{\left(u_{t_{w,s}}\right)^2 + \left(u_{t_{w,r}}\right)^2}$

The uncertainty in the cooling capacity coefficient U_{cc} is of the form:

$$u_{U_{cc}} = \sqrt{\left(\frac{\delta U_{CC}}{\delta \dot{m}_w}\right)^2 u_{\dot{m}_w}{}^2 + \left(\frac{\delta U_{CC}}{\delta t_{w,s}}\right)^2 u_{t_{w,s}}{}^2 + \left(\frac{\delta U_{CC}}{\delta t_{w,r}}\right)^2 u_{t_{w,r}}{}^2 + \left(\frac{\delta U_{CC}}{\delta T_{op}}\right)^2 u_{t_{op}}{}^2} \quad (8)$$

with: $$\frac{\delta U_{CC}}{\delta \dot{m}_w} = \frac{c_{p_w}(t_{w,r} - t_{w,s})}{A_{panels}\left(t_{op} - \frac{t_{w,r} + t_{w,s}}{2}\right)}$$

$$\frac{\delta U_{CC}}{\delta t_{w,s}} = \frac{\dot{m}_w\, c_{p_w}\,(t_{op} - t_{w,s})}{A_{panels}\left(t_{op} - \frac{t_{w,r} + t_{w,s}}{2}\right)}$$

$$\frac{\delta U_{CC}}{\delta t_{w,r}} = \frac{\dot{m}_w\, c_{p_w}\,(t_{w,r} - t_{op})}{A_{panels}\left(t_{op} - \frac{t_{w,r} + t_{w,s}}{2}\right)}$$

$$\frac{\delta U_{CC}}{\delta t_{op}} = -\frac{\dot{m}_w\, c_{p_w}\,(t_{w,r} - t_{w,s})}{A_{panels}\left(t_{op} - \frac{t_{w,r} + t_{w,s}}{2}\right)}$$

The sample uncertainties of the derived quantities (water temperature differences, cooling load removed by the panels, and the cooling capacity coefficient U_{cc}) have been evaluated based on the equations above. The derived uncertainties for the total cooling load and for U_{cc} were respectively 4.9% and 5.1% (maximum values).

Variant description

We tested 6 different levels of acoustical coverage: 0% (0 clouds), 16% (2 clouds), 32% (4 clouds), 47% (6 clouds), 63% (8 clouds) and 71% (9 clouds). Figure 35 (left) shows the location of the clouds for each level. The reference case for the study is the variant without acoustical coverage. For cooling capacity testing of the variant with highest acoustical coverage (71%), by mistake additional heat load of 43 W (4% of the total internal load) occurred at the wall level due to the operation of guard heaters in the wall. These heaters were not used in any other tests. This additional load did not affect the cooling capacity results but it did modify wall mean surface temperature. Thus, not all results are represented for this variant.

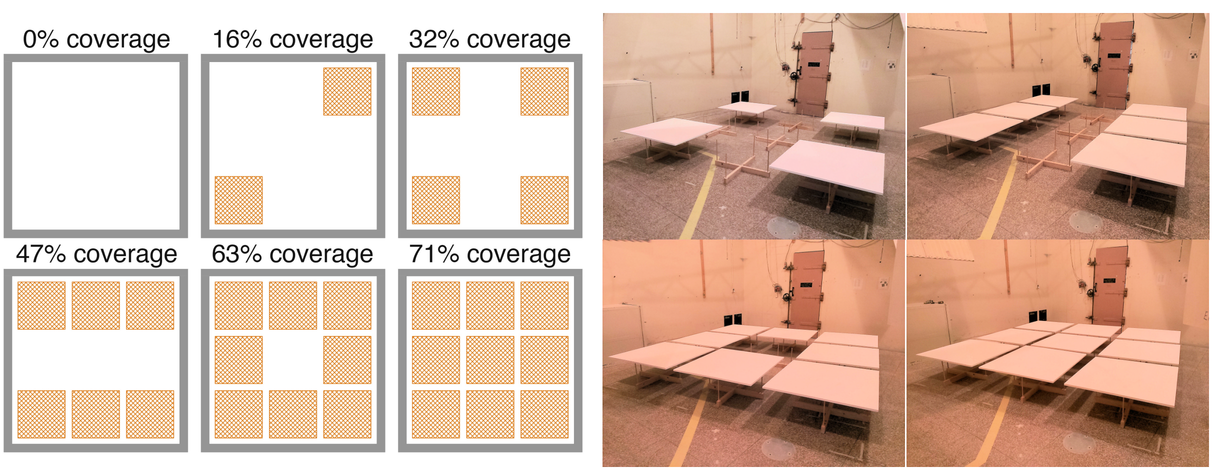


Figure 35: (Left) Schematics of coverage going from 0% to71% (as tested in the climatic chamber) (Right) Series of 4 photographs taken in the reverberation chamber with 4, 6, 8 and 9 clouds.

6.2.2 Sound absorption measurements

Experimental facilities and test room description

We conducted the sound absorption experiments in a reverberation chamber (6.22 m × 8.17 m × 5.22 m) at Armstrong World Industries in Lancaster, PA in August 2015. This chamber is accredited by the NVLAP (National Voluntary Laboratory Accreditation Program (USA)) for sound absorption testing following ASTM Standard C423 - 09a (ASTM 2009). The walls are constructed of 0.2 m concrete block with filled cavities, and the ceiling is reinforced concrete. Both the wall and ceiling surfaces are painted with an epoxy paint. The floor is reinforced concrete topped with terrazzo tile. The test room sits on springs on independent foundations from the surrounding building.

Experimental conditions and procedure

The sound absorption of a test sample is based on the measurement of a decay rate within the reverberation chamber. We conducted these measurements according to ASTM Standard C423 (ASTM 2009). For this testing, a broadband random noise signal is turned on long enough for the sound pressure level to reach steady state. When the signal is turned off, the sound pressure level decreases and the decay rate in each frequency band is determined by measuring the slope of a straight line fitted to the sound pressure level of the average decay curve. Typically 50 to 60 measured decays are averaged to meet the precision requirements of the test code. This decay curve is adjusted by subtracting the sound absorption of the empty room itself including that due to air. The sound absorption is calculated based on the Sabin formula:

$$A_{absorption} = 0.9210 \, \frac{V\,d}{c} \qquad (9)$$

where A is the equivalent sound absorption area in [Sabin, m^2], V the volume of the reverberation room in [m^3], c the speed of sound in [m/s], and d the decay rate in [dB/s].

Instrumentation

To comply with the methodology prescribed in the ASTM C423, we used the following measuring instruments: microphones, pre-amplifiers for those microphones, temperature/humidity sensor, barometric pressure sensor, speakers, and data collection hardware. Table 12 summarizes the sensors and electronic equipment.

Uncertainty

The reverberation chamber is certified by the ASTM C423. The uncertainty is based on the methodology given in Section 13 of ASTM C423 listing the 95% probability limits for repeatability of the sound absorption measurements in this test chamber.

Table 12: List of sensors and specifications for the acoustical measurements

Instruments	Unit	Measured variables and sensor location
Microphones Bruel & Kjaer Model 4176	6	microphones randomly located in the reverberant room
Preamplifiers Bruel & Kjaer Model 2669,	6	preamplifiers connected to the microphones
Temperature/RH sensor Vaisala HMP 231	1	Sensor located on the North wall of reverberant room

Barometric Pressure Sensor Setra Model 370	1	Sensor physically located in the control room (input connected by 6.35 mm tubing to the chamber)
Speakers Soundsphere Q-15 speakers	1	Speakers located in the upper NW and SE corners of the room
Data Collection hardware Bruel & Kjaer Multi-Analyzer modules - Model 3050-A-040	2	Data Collection hardware located in the control room
Data Collection hardware Bruel & Kjaer Multi-Analyzer modules - Model 3160-A-022	1	Data Collection hardware located in the control room

Variant description

We used the same acoustic cloud for the cooling capacity and acoustic testing. These acoustical panels are fabricated of a fiberglass pre-formed into a 1.20 m square by 0.04 m thick cloud shape (Armstrong Soundscapes Shapes). Figure 35 (right) shows the four acoustical cloud configurations tested in the reverberation chamber (using 4, 6, 8 and 9 clouds). As with the cooling capacity testing, up to 9 clouds were positioned into a 3 by 3 array, 1.37 m center to center. The tests were conducted with the panels raised above the floor (as opposed to suspended), and the distance to the floor is equivalent to plenum depth if suspended. In these acoustic tests the height was 0.40 m (note that a suspension distance of 0.25 m was used in the cooling capacity testing).

Complementary analysis

The output of the acoustic testing per ASTM C 423 is the sound absorption as reported in metric Sabin (Sabin, m^2). In order to understand the effect of the clouds in a more realistic setting, we used this output to model the reverberation time (T_{60}) of a typical closed office space. The reverberation time is defined as the time it takes for a sound, such as a loud hand clap, to dissipate after being made by a level reduction of 60 dB. It is commonly abbreviated T_{60} in seconds (s), and it can be calculated for each frequency using the Sabine formula:

$$T_{60} = 0.161 \frac{V}{A_{absorption}} \qquad (10)$$

with *A,* equivalent sound absorption area in [Sabin, m^2], and *V*, volume of the room in [m^3].

We modelled an office with comparable characteristics as for the cooling capacity testing. We assumed an 18.2 m^2 room with a floor to ceiling height of 2.65 meters (the cooling capacity test room has 2.50 m floor to ceiling height, and so, the plenum depth in the acoustic testing was 0.15 m deeper). We further assumed a concrete finish ceiling (for the base case with no clouds), plaster on the walls, and a light carpet on the floor. We assumed a furnished but unoccupied space. We assumed wooden panels for the entrance door and the desks and three lightly upholstered chairs. The sound absorption coefficient of these materials and finishes are reported in Table 13.

Reverberation time criteria are published by ASHRAE in the Performance Measurement Protocols (PMP) Best practices Guide (ASHRAE 2012b) and can be used to determine what acoustical cloud coverage may be needed. Reverberation time relates to speech clarity, and low reverberation (on the order of 0.6 or 0.8 seconds depending on office type) is usually desirable for speech intelligibility.

Table 13: Sound absorption coefficient used for reverberation time calculation

Surface	Materials	Qu.	Unit	Octave Band (Hz)					
				125	250	500	1000	2000	4000
Ceiling	Concrete[(1) (4)]	18.1	m^2	0.01	0.01	0.02	0.02	0.02	0.02
Walls (door subtracted)	Plaster (gypsum on wood lath) [(1)]	43.4	m^2	0.12	0.09	0.07	0.05	0.05	0.04
Floor	Carpet (up to 0.006m pile height)[(2)]	18.1	m^2	0.02	0.04	0.06	0.20	0.30	0.35
Door and desks	Wood [(2)]	6.4	m^2	0.10	0.08	0.06	0.05	0.05	0.05
Chair	Lightly upholstered chairs[(3)]	3	items	0.35	0.45	0.55	0.60	0.60	0.60

[(1)] Source: (Egan 2007), [(2)] Source: (DIN 2004), [(3)] Source: (Ermann 2015), [(4)] We used concrete ceiling as the reference case because this is the typical finish for TABS

6.3 Results

We present the results separately below for the cooling capacity and the sound absorption testing. For the cooling capacity testing, we include further results on thermal comfort. The sound absorption results also include T_{60} derived from the measurements and calculated for a typical (furnished but unoccupied) closed office.

6.3.1 Cooling capacity testing

Table 19 summarizes the main results of the testing. The chamber reached steady-state equilibrium at an operative temperature between 26.5º C (no coverage) and 29.3º C (71% coverage).

Table 14: Experimental results for parameters

Coverage	t_a[(1)] (º C)	t_{mr}[(1)] (º C)	t_{op}[(1)] (º C)	$t_{w,r}$ - $t_{w,s}$ (º C)	$t_{w,m}$ (º C)	q (W)	q'' [(2)] (W/m²)	q'' [(3)] (W/m²)	U_{cc}[(3)] (W/(m².K))	Change in U_{cc}[(3)] (%)
0%[(4)]	27.1	26.0	26.5	4.2	17.1	1069	79.8	58.7	8.46	100%
16%	27.2	26.4	26.8	4.2	17.1	1067	79.7	58.6	8.19	97%
32%	27.2	26.7	27.0	4.1	17.1	1054	78.7	57.9	7.93	94%
47%[(4)]	27.9	27.3	27.6	4.1	17.1	1061	79.2	58.3	7.54	89%
63%	28.0	28.1	28.0	4.1	17.0	1041	77.7	57.2	7.06	83%
71%	29.1	29.5	29.3	4.3	17.2	1112	83.0	61.1	6.85	81%

[(1)] Measured at the central tree at 1.1 m. high; [(2)] Per unit of panel area; [(3)] Per unit of floor area; [(4)] Replicated experiment

Cooling capacity coefficient

Figure 36 shows the change in cooling capacity coefficient (U_{cc}) at each level of cloud coverage. We observe that adding coverage reduces this coefficient. Yet, this reduction is small compared to the coverage: the cooling capacity coefficient only decreases by 3.2% for 16% coverage and by 19% for 71% coverage. The reduction in cooling capacity coefficient is on average 4 to 5 times lower than the percentage coverage. This result generally confirms the trend observed in the literature but with a slightly larger difference between cooling capacity coefficient and coverage, depending on the reference. The difference in cooling capacity between the replicated tests, performed non-sequentially, was 0.25% and 0.04% for the 0% and 47% coverage cases, respectively. This indicated that the lab is very reliable and provides reproducible results.

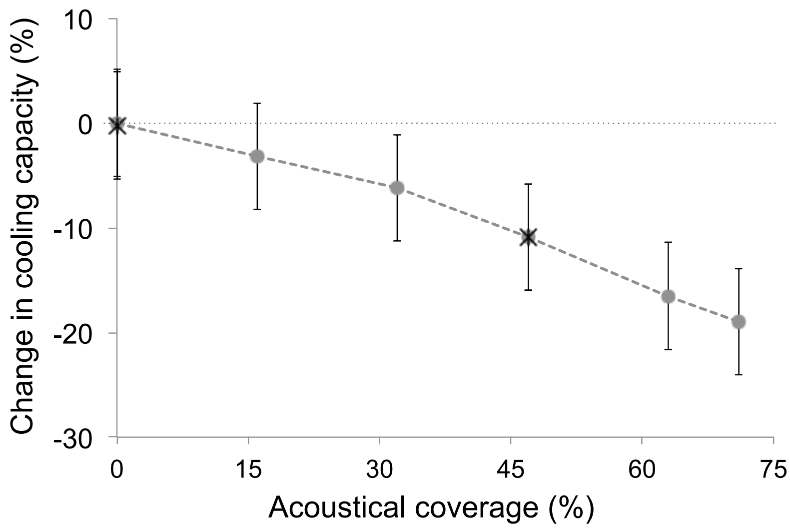


Figure 36: Change in cooling capacity coefficient (U_{cc}) as a function of acoustical coverage. The grey dots show the main results of the testing. The black "ex" represents replications

Thermal comfort

Figure 37A shows the difference between air and mean radiant temperatures measured at 1.1 m high as a function of acoustical coverage. Both the central and the side measurement trees are represented. For both locations and with lower coverage, the air temperature is slightly higher than the mean radiant temperature. We note that in the case of air systems, this difference is commonly greater (Feng, Schiavon, and Bauman 2013). As the coverage increases, the difference between both temperatures diminishes and the mean radiant temperature eventually gets slightly higher than the air temperature. We can hypothesize that the share of heat transfer exchange between the panels and the sensors becomes more convective and less radiative. Between 16% and 71% coverage, the air and mean radiant temperatures stay within 1º C of each other. For the tree close to the wall, the difference between the two temperatures is always less than 1º C. This is an important observation. Differently from air systems, in radiant systems the air and mean radiant temperature are quite similar, therefore, if these results are confirmed in field tests, we may infer that an operative temperature sensor is not needed for control purposes, an an air temperature sensor will be sufficient. With no obstruction, the difference between air and mean radiant temperatures is 0.6º C lower for the side tree compared to the central tree, even though the two trees are only 1.35 m apart from each other. At neutral comfort condition, a change of 0.6º C equals 0.2 PMV or 0.11 clo. This change is comparable to the effect of a shirt (slightly more than a T-shirt). This mean radiant temperature reflects the influence of the wall temperature (warmer than the ceiling) and it shows the importance of view factor. This point also brings the question of the location of sensors in rooms. Close to the wall, where the thermostat is usually located, the two temperatures are even closer, supporting the hypothesis that an operative sensor may not be needed. With higher coverage, we note that air and mean radiant temperature are overall getting closer to each other. Using the operative temperature (or even the air temperature) to control a radiant system with coverage is going to be more accurate compared to a radiant system without coverage. Figure 37B shows the difference in average surfaces temperature between walls and radiant ceiling as a function of acoustical coverage. As the coverage increases, this difference increases: the active ceiling temperature stays constant at around 21.6º C ± 0.3º C while the wall temperatures rise continuously from 26.8º C to 28.3º C. Thus, the difference in average surface temperatures between wall and ceiling is mainly due to the effect of the wall temperatures. As coverage increases, the view factor from the ceiling to the walls decreases, which impacts the wall temperatures and further the mean radiant temperature. Considering a larger office and in the case of no coverage, the cooler ceiling temperature will have a bigger influence on the mean radiant temperature in the center of the room. The difference between the air and the mean radiant temperature will increase compared to what we measured and reported in Figure 37A. In this regard adding coverage may bring a steeper slope than the one observed here in Figure 37A.

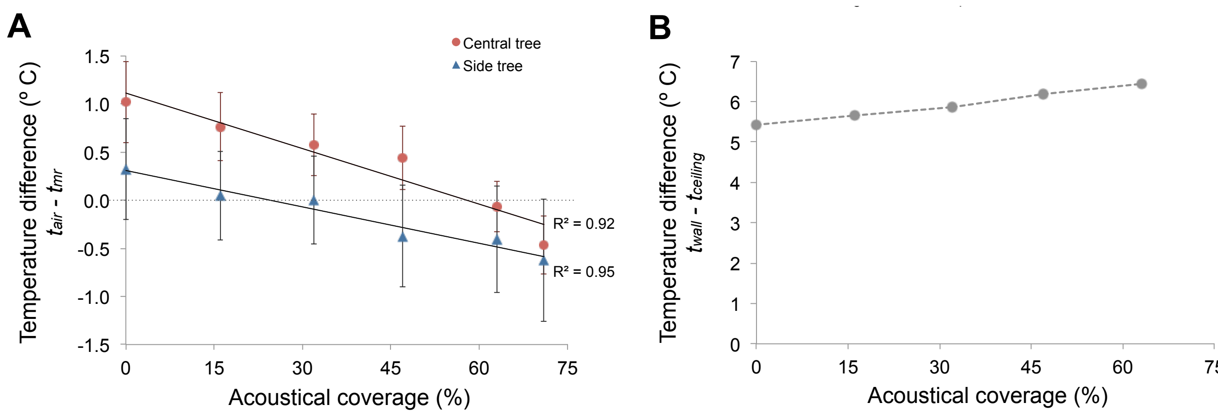


Figure 37: (A) Difference between air and mean radiant temperature (central and side trees at 1.1 m height) as a function of acoustical coverage, (B) difference in average surface temperature between wall and ceiling as a function of acoustical coverage

6.3.2 Acoustic absorption

Sound absorption

Table 15 summarizes the main results of the acoustic testing from the reverberation chamber measurements presented as sound absorption for each cloud configuration by normalized octave band frequency. This absorption is obtained by subtracting the absorption of the bare room from the treated room.

Table 15: Sound absorption of each cloud configuration in Sabin, m²

Number of clouds	Octave Band (Hz)							
	63	125	250	500	1000	2000	4000	8000
4	0.3	2.1	5.1	5.7	8.2	10.0	10.1	9.2
6	0.3	2.3	7.2	8.1	11.5	13.8	13.7	12.8
8	0.7	3.8	9.5	10.2	14.5	17.5	17.3	16.0
9	1.0	4.8	9.9	10.8	15.8	18.3	18.5	16.7

Reverberation time

The reverberation time, T_{60} is defined as the length of time required for the sound level in a room to fall by 60 dB after the sound was made. The reverberation time is directly proportional to the room volume, and inversely proportional to the sound absorption properties of its walls, floor, ceiling, and its furnishings, as well as the air absorption. To complete our acoustical assessment, we calculated T_{60} for a closed office based on the sound absorption of our variants and on additional assumptions reported in section 6.2.2. The ceiling surface area is the same as for the cooling capacity section. Therefore, the number of clouds aligns with the percent cloud coverage reported previously and we will report the results based on coverage. The total absorption of the ceiling for the reference case (concrete ceiling) was calculated using the sound absorption coefficient (reported in Table 13) multiplied by the actual ceiling surface (18.1 m^2). The values for the 32%, 47%, 63% and 71% coverage were directly taken from Table 15 (4, 6, 8 and 9 cloud variants). For the 16% coverage, we did an extrapolation based on the experimental results obtained for the 4, 6 and 8 clouds variants and on coverage. Additionally, we looked for standards and guidelines to find recommended values for the T_{60} in offices. We found maximum recommended values for open-space office (0.6 s.) and private office (0.8 s.) in the ASHRAE PMP Best practices Guide (ASHRAE 2012b). These values are reported on the graphs for comparison.

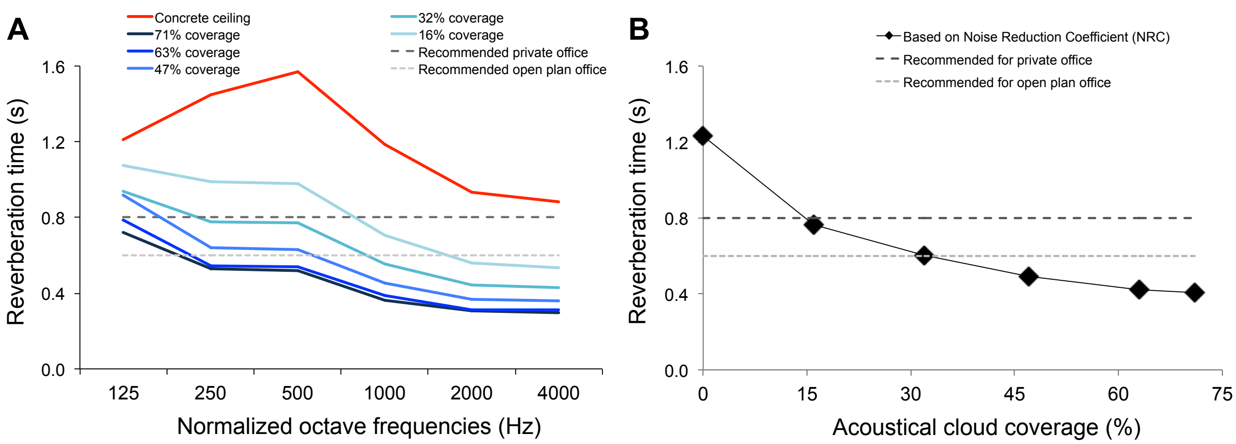


Figure 38: (A) Reverberation time as function of frequencies for different acoustical cloud coverage, and (B) reverberation time as a function of coverage based on the Noise Reduction Coefficient (NRC)

Figure 38 (A) shows the result of this analysis. T_{60} is shown as a function of frequency for different acoustic coverage. We observe that adding acoustical clouds in the ceiling plane strongly reduces the reverberation time. The peak of 1.6 s at 500 Hz for the concrete variant (base case) drops to 1 s for 16% coverage, 0.8 s for 32% coverage, 0.6 s for 47% coverage and down to 0.5 s for 71% coverage. With 32% coverage, the room meets the recommended values for private offices for most of the frequencies. Figure 38 (B) shows the T_{60} as function of acoustical cloud coverage. For this figure, we used the single-value Noise Reduction Coefficient (NRC) that is determined by calculating the mean value from the four octave values of the sound absorption coefficient between 250 Hz and 2000 Hz, and rounded the result to the nearest 0.05. One limitation of this average is that frequencies below 250 Hz (that include the human voice for instance) are not taken into account. Nevertheless, this single-value allows us to visualize the effect of acoustical coverage for an average sound absorption of the material. Based on these recommended values, we can conclude that: (1) even low cloud coverage (around 15%) can considerably improve acoustic absorption compared to a concrete ceiling; (2) covering 30% of the ceiling with an acoustical cloud would provide an acceptable absorption in the private offices, and (3) covering 50% of the ceiling with an acoustical cloud would provide an acceptable absorption in open plan offices. This study was focused on specific cloud configurations for the ceiling. The office simulation was built with sound reflective surfaces. Different plenum heights and different spacing between the clouds or such configuration in a different space may yield different results.

6.4 Discussion

The purpose of this study was to conduct laboratory experiments for a closed office space with different percent coverage of free-hanging acoustical clouds below a thermally activated building system to look at the interactions between thermal and acoustical factors. We used a hydronic test chamber to investigate the cooling capacity of the TABS, and a reverberation chamber to determine the sound absorption of our acoustical cloud variants. We wanted to simulate comparable variants for both experiments, and for the cooling capacity testing we could directly simulate an office room. For the acoustical testing, we tested the sound absorption of the different ceiling variants, and used this output to calculate the reverberation time of an office space. We assumed a similar space for the thermal and acoustical testing.

We conducted the cooling capacity experiment in a chamber equipped with radiant metal panels. Yet, with our experiment, we were trying to address the case of TABS. Unlike radiant panels, TABS typically cover a larger area leaving less space for acoustical treatment. In our experiments, most of the ceiling (73.5%) was covered with radiant panels, which is reasonably representative of the TABS system of interest in this study (we note that TABS do not typically cover the entire ceiling due to structural constraints). Additionally, the composition of radiant panel systems can offer the possibility to integrate acoustical treatment within the panel (e.g., using a perforated metal panel and adding insulation on the back). We did not consider these options and stayed with a plain non-perforated metal panel. We used a chamber equipped with radiant panels because we only had this option and this configuration allowed us to reach steady state condition faster. The emissivity of the two finished surfaces was comparable. Thus, the cooling capacity experiment applies to both radiant panels and TABS. Embedded surface systems (ESS) are another type of radiant system. ESS are comparable to TABS and therefore we can extend our cooling capacity results for this type of system under steady-state conditions. We performed our acoustical analysis with a simulated concrete ceiling because we wanted to address the case of TABS. As reported in Table 13, the acoustical absorption of concrete is extremely low. Using different materials or finishes on a radiant system yields comparable or better results.

With this experiment, we found that the cooling capacity coefficient only decreases by 3.2%, 11% and 19% for a ceiling cloud coverage of 16%, 47% and 71% respectively. We compared this outcome with the results found in the literature. With free-hanging acoustical clouds and a plenum of 0.2 m, Peperkamp and Vercammen (Peperkamp and Vercammen 2009) observed a reduction of the cooling capacity of ~20% for 50% coverage, Chigot (Chigot 2010) reported a reduction of ~16% for 45% coverage and Ecophon (acoustic material manufacturers) (Ecophon 2014) reported a reduction of ~18% for 45% and of ~27% for 60% coverage. With a larger plenum of 0.6 m, Weitzmann et al. [12] observed a reduction of the cooling capacity of ~30% for 83% coverage. Overall, though coverage and plenum height was not exactly the same as in previous experiments, we can state that the results of our experiment show less reduction in cooling capacity compared to what was previously reported.

It is important to underline that there is a difference between geometrical percentage of covered ceiling and occupant perception of ceiling coverage. It is unlikely that occupants will visually perceive the relatively narrow gaps between the acoustic clouds as open space. Therefore, the ceiling appears more covered than what it is. For example, from occupants and designers' perspective, 47% coverage is likely to be perceived as closer to two thirds of the ceiling and the 71% variant closer to the full ceiling being covered (see schematics Figure 35 (left)). What we describe in this paper as low coverage (16% and 32%) still represent a meaningful portion of the ceiling from a visual perception perspective.

We conducted this experiment for a cooling application. As the heat released by the internal loads rises, this configuration is also best to ensure higher natural convection and mixing near the ceiling. This creates an airflow cell in the room, counteracting the effect of stratification. It is unlikely that a heating application will show similar results as there would be stagnant, warmer air between the ceiling and the acoustic clouds. The share of convection compared to radiation would be lower than in the cooling application (Awbi 2003). Thus, we would expect a stronger reduction in heating capacity with increasing coverage by the clouds. This heating application warrants further testing.

Cooling capacity experiments were conducted for a room with no ventilation system. We decided to focus on the change in cooling capacity due to acoustic coverage. Adding an air distribution system would bring multiple questions, including the type of system, location and type of diffusers, volumetric air flow and supply temperature. The design of the ventilation system may also impact the ratio of convective and radiative heat transfer close to the cooled ceiling surface. Air systems (particularly overhead mixing) will also likely increase surface convention (Novoselac, Burley, and Srebric 2006) . The question of the interaction between the chilled ceiling and both coverage and ventilation system would be worth investigating.

The size of the laboratory chamber for the cooling capacity testing has an impact on the view factors between the ceiling and the sensors. Compared to a larger room (typical for open plan offices), our chamber has relatively small dimensions, leading to lower view factors to the ceiling (and floors) and higher view factors to the walls. In the case of a larger office, the ceiling temperature will have more effect on the mean radiant temperature than is the case for our chamber. Based on Figure 37(B), the average ceiling temperature was about 6ºC lower than the average wall temperature. This difference is going to impact the mean radiant temperature taken in the center of the space for larger rooms. In the case of no coverage, the mean radiant temperature is going to be cooler than in the small office tested. Thus, we may expect the cooling capacity to be higher for the reference case (no coverage) in a large office. Due to the increased impact of the ceiling, it is likely that the percent reduction in cooling capacity would be slightly more important if tested in a larger office space. Overall we would expect this difference to be moderate. Also, this phenomenon can be counteracted by the presence of furniture, cubicles or other forms of partitions, depending on office layout and type.

Based on recommended values, we estimated that covering 30% of the ceiling with acoustical clouds would provide an acceptable absorption for private offices. In shared offices we would need to cover 50% of the ceiling. On the energy side, our experiment showed that 47% coverage would reduce the cooling capacity by about 11% in the case of the small office. This reduction may increase in the case of larger office but this change should not exceed a few percent. The application of radiant slab systems with free-hanging acoustical clouds in both small and large shared offices would mainly depend on the cooling requirements and on how much a 10-15% reduction of the cooling capacity may impact thermal comfort.

While the cooling capacity testing was mainly dependent on the coverage, the acoustic testing was also highly dependent on the type of product tested. Thus, it is important to note that the results obtained for sound absorption and reverberation time are valid for these clouds (Armstrong SoundScapes) and similar types of absorbers. The plenum depth for the acoustic testing was 0.15 m greater than the plenum used in the cooling capacity testing. Also, the reverberant chamber was larger than the hydronic chamber and therefore, the walls were further away from the canopies being tested in the center of the space. Both the plenum depth and the distance to the walls may have slightly influenced the sound absorption values obtained at low frequency (a higher plenum and larger distance to the walls would help the sound to access and be absorbed on the back of the canopy). Yet, the acoustical analysis based on T_{60} calculation was rather conservative in its assumptions (lower sound absorption coefficient (see Table 13)), and therefore, the results are reasonable. We may also point out here that we tested horizontal clouds and not vertical baffles that may bring a different set of constraints between cooling capacity, sound absorption, ceiling appearance and perceived ceiling height.

The acoustic section of this article was focused on the absorption of the different panel configurations and on the reverberation time. Absorption at the ceiling is likely to provide only a partial answer to room acoustical problems. It addresses background noise level and reverberation time, but does not by itself solve sound privacy issues, which are perceived as one of the biggest constraints in open plan offices (Frontczak et al. 2012; Kim and de Dear 2013). Our experiment was able to show that combining a radiant chilled ceiling with free-hanging acoustical clouds can reduce reverberation issues while having a relatively modest impact on heat transfer. However, it is clear that additional modifications would be required for increased acoustical quality in shared office spaces.

6.5 Conclusions

We conducted laboratory experiments to investigate the change in cooling capacity and the sound absorption of an office room with different coverage areas of free-hanging acoustical clouds below a radiant cooled ceiling. The main conclusions of this study are:

- The cooling capacity coefficient only decreases by 3.2%, 11% and 19% for a ceiling cloud coverage of 16%, 47% and 71% respectively. This reduction is on average 4 to 5 times less than the percentage coverage.
- The difference observed between air temperature and mean radiant temperature is less than 1º C. This implies that measuring operative temperature or mean radiant temperature may not be needed in radiantly conditioned buildings. With lower coverage, the air

temperature is slightly higher than the mean radiant temperature. As the coverage increases, the difference between both temperatures diminishes and the mean radiant temperature eventually gets slightly higher than the air temperature.

- The acoustical clouds exhibited the greatest sound absorption between 200 and 2500 Hz. The sound absorption per cloud slightly decreases with increasing number of clouds.
- Compared to exposed concrete, our tested cloud variants resulted in a substantial reduction of reverberation time. Even low cloud coverage (15-35%) can considerably improve sound absorption at the ceiling. The acoustic results showed that if the clouds covered 30% of the ceiling in a private office or 50% in an open plan space, acceptable sound absorption at the ceiling was achieved.

Overall, this study addresses one limitation often associated with radiant systems using exposed concrete ceilings, which is the question of can you achieve acceptable acoustical quality without overly compromising the cooling performance of the radiant ceiling. The results demonstrated a practical solution in which free-hanging acoustical clouds are positioned below a radiant chilled ceiling. The observed reduction in cooling capacity at 47% coverage was only 11%, while this condition created an acceptable acoustic solution (reverberation time) for both private and open plan offices.

7 Effect of acoustical clouds and fans on thermal comfort and cooling capacity of a radiant chilled ceiling

This chapter follow-up on the previous one. It focuses on the change in cooling capacity and thermal comfort for an office room with a radiant cooled ceiling, free-hanging acoustical clouds and fans. Multiple fan configurations are being tested (a ceiling fan and small fans hidden above the acoustical clouds). We wanted to investigate the effect of air movement on radiant ceiling performance as it potentially provides two key advantages: increasing convective heat exchange at the cooled ceiling surface, and providing the option of having elevated air movement in the occupied zone for thermal comfort.

7.1 Background

We conducted a review on studies that quantify the effect of air movement on cooling capacity. We did not find any studies that investigated the combination of fans below a radiant ceiling. Yet, we found studies on the effect of ventilation (type of air diffusers) on radiant ceiling capacity. Awbi and Hatton (Awbi and Hatton 2000) tested the effect of convection for a heated floor partially covered by an air jet (the convective performance of a cooled ceiling has often been compared to that of a heated floor and therefore we decided to report this study). They modelled the total convective heat transfer coefficient based on natural and forced components. For the natural component, the correlation was based on the temperature difference between the mean panel temperature and the air, and for the forced component, the correlation was based on the width of the nozzle and the diffuser discharge air speed. Jeong and Mumma (Jeong and Mumma 2003) modelled the effect of mixed convection on radiant ceiling cooling panels due to a nozzle diffuser at the ceiling level. They developed equations for the total heat transfer and radiant heat transfer coefficient based on surfaces and air temperatures. They found that, under normal operating temperatures, the total capacity of the panels can be enhanced by 5% to 35% depending on the air speed. Novoselac et al. (Novoselac, Burley, and Srebric 2006) tested the effect of mixed convection due to the presence of a high aspiration diffuser. For natural convection, they found a convection correlation that was based on the temperature difference between the cooled ceiling surface and the air. For forced convection, a correlation between the total heat exchange coefficient and the air change was established. Their results showed an increase of the total convection coefficient at cooled ceiling surfaces by 4% to 17% with high aspiration diffusers. Venko et al. (Venko et al. 2014) studied the effect of natural and mixed convection along a thermally activated cooled wall (vertical configuration). Forced convection was generated from an air jet flowing downward through a longitudinal ceiling slot parallel to the cooled wall. The forced convection coefficient was a function of the height on the wall (distance to diffusor). It was correlated with the temperature difference between the cooled wall surface and the air, the height and the air speed. The increase in cooling capacity was not quantified. Overall, these studies show promising opportunities for the use of increased air movement on radiant cooled ceiling heat transfer. We could not find studies directly involving fans, which shows a gap in the literature. Moreover, we did not find papers on the interaction among acoustical panels, fans and radiant systems.

Using fans to bring air movement in the occupied zone presents multiple advantages in terms of thermal comfort and associated energy use. Elevated air speed is accepted as an effective strategy to cool people in moderately warm environments (McIntyre 1978; S. Tanabe 1989; Shinichi Tanabe and Kimura 1994; Chow and Fung 1994; Fountain et al. 1994; Kitagawa et al. 1999; Ho, Rosario, and Rahman 2009; Zhai et al. 2013; Schiavon, Yang, et al. 2016). It allows equivalent thermal comfort conditions to be achieved at higher indoor air temperatures (ANSI/ASHRAE 2013; Arens et al. 2009). This offset of warm indoor temperature has been identified as a relevant potential for energy savings as it can be used to reduce the need for compressor-based conditioning systems (Sekhar 1995; Aynsley 2005; Schiavon and Melikov 2008; Yang et al. 2015; Hoyt, Arens, and Zhang 2015). Additionally, if the fan is controlled by occupants, it provides the possibility to resort to air movement when needed which allows for more flexibility in the approach to thermal comfort and personal preferences.

The combination of fans and radiant systems may provide numerous advantages. The objective of this study is to determine the cooling capacity one can reach when adding both free-hanging acoustical clouds and increased air movement at the ceiling level. This paper focuses on air movement as a strategy to offset and increase the cooling capacity. We investigated the effect of two different strategies to provided increased air movement close to the radiant panels: (1) using a ceiling fan located in the center of the chamber that can operate in either upward or downward directions and (2) using a series of small fans located above the clouds (see schematics in Figure 39). We conducted laboratory experiments in a controlled climatic chamber equipped with a radiant cooled ceiling, fans and free-hanging clouds. This testing follows a first study with no air movement and detailed acoustical analysis (Karmann et al. 2017). The earlier series of tests with no air movement served as the reference case for our testing.

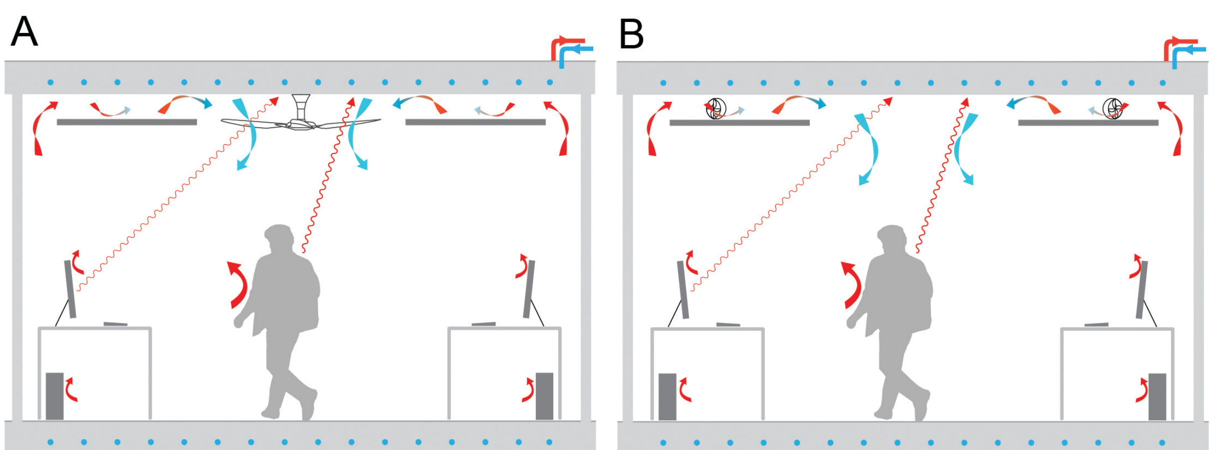


Figure 39: A: Schematic with ceiling fan blowing downwards from the ceiling – B: Schematic with small fans blowing along the ceiling (towards the center)

7.2 Method

7.2.1 Experimental facilities and room description

The chamber used for this experiment has a floor area of 18.2 m^2 and a volume of 54.7 m^3 (4.27 m × 4.27 m × 3.0 m). It is nearly adiabatic, has no windows and is located inside a large laboratory facility that was maintained at 21.6 ºC ± 0.5 ºC during our testing. The chamber's construction, thermal properties, radiant panels and acoustical clouds used for this testing are the same as described in our previous study (Karmann et al. 2017). We conducted our experiment in September 2015. The chamber is accredited by the EN 14240 (CEN 2004) for cooled ceiling testing. Figure 34 shows a plan, sections and photographs of the chamber.

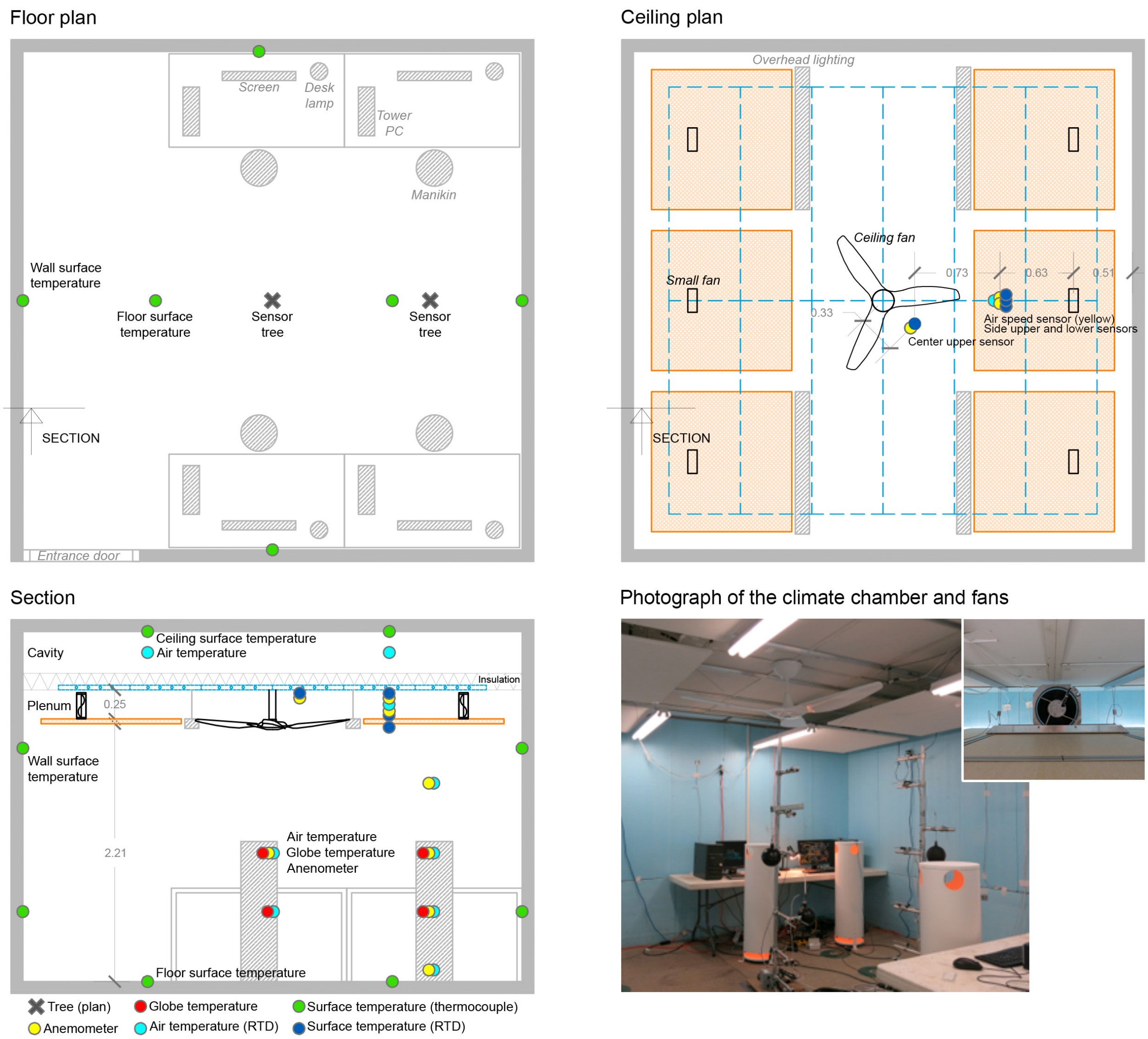


Figure 40: Plan, section and photographs of test chamber with sensors and cloud coverage. The acoustical clouds are in orange, the radiant ceiling in blue and the furniture/equipment in gray. The fans are represented in black: ceiling fan in the center and small fans above the clouds. This figure represents the variants with 47% ceiling coverage (6 clouds). The two photographs of the testing show the test chamber and one small fan mounted above a cloud.

We used exactly the same radiant panels and the same acoustical clouds (type, size, location, thermal properties and set-up) for this testing as for the testing reported in (Karmann et al. 2017). Twelve radiant ceiling panels were centered on the ceiling to cover 73.5% (13.4 m^2) of the total ceiling area. This represented the maximum area that could be covered by the panels given the chamber ceiling design. The intention was to represent TABS, the system of interest in this study, as closely as possible by covering the largest possible ceiling area. For this study, a maximum of 8 clouds could be positioned on the ceiling (as represented in Figure 41). Acoustical coverage was one of the variables in this study and therefore the number of clouds is not constant across the tests. Figure 41 represents the various coverage ratios tested. The chamber did not include an air system because we wanted to specifically focus on the change in cooling capacity of the radiant cooled ceiling, and adding an air system would increase overall uncertainty in the results.

Table 16 summarizes the heat sources used in the experiment. We modeled office heat sources using computers (tower CPUs), flat screens and desk lamps on the desks, and suspended overhead lighting (0.25 m below the radiant ceiling). We simulated occupants with power-adjustable dummies according to EN 14240 (CEN 2004). When fully installed, the test chamber represented a four-person office with a computer at each workstation; a relatively high occupant density of 4.55 m^2 per person. The data acquisition system was outside the chamber, and therefore not listed here as a heat load.

Table 16: Heat load summary

Heat source	Number	Total power (W)	Power per floor area (W/m^2)
People	4	300	16.5
Computers	4	300	16.5
Desk lamps and screens(1)	4	283	15.5
Overhead lighting	4	214	11.8
Total		1097	60.3

(1) For some of the small fan variants, the power of the desk lamps and screens was sometimes used to compensate for the added power generated by the small fans. This power adjustment was done by switching off devices.

Table 17: Fan test conditions and power

Fan	Configuration	Fan rotational speed (rpm) (6)	Fan setting	Number of fans	Total fan power measured(1) (W)	Added load in the room (%)	Adjustment(2) (W)
Ceiling	Blowing up	146	speed 4	1	6	0.6%	0
	Blowing down	146	speed 4	1	6	0.6%	0
Small	Low speed	648	35%(3)	2(4)	22(5)	2%	0
	Medium speed	832	40%(3)	2(4)	26(5)	2.4%	-21
	Low speed	648	35%(3)	4(4)	43(5)	4%	-30
	Medium speed	832	40%(3)	4(4)	57(5)	5.2%	-30
	Low speed	648	35%(3)	6(4)	65(5)	6%	-30
	Medium speed	832	40%(3)	6(4)	87(5)	7.7%	-58
	Low speed	648	35%(3)	8(4)	87(5)	7.7%	-58
	Medium speed	832	40%(3)	8(4)	115	9.3%	-58

(1) Power measurements reported for the small fan testing were corrected based on a calibration conducted after the testing; (2) Adjustments were made to compensate the fan output power; they were based on the power measurements conducted during the testing (prior to more accurate calibration of the power meter); (3) Based on the variable transformer input; (4) Based on the number of clouds; (5) The uncertainty of the power meter increases at low values; (6) Rotational speed for the ceiling fan was given by the manufacturer; rotational speed for the small fan was measured using a strobe (Nova-Strobe Basic Battery LED Stroboscope from Monarch Instruments) and averaged.

We equipped the climatic chamber with fans. Figure 34 shows a ceiling plan of the chamber (with 6 clouds) and photographs of the two types of fan. The type and configuration of fans was the second variable of this study. We used the two following models in our experiment:
(1) DC motor ceiling fan (Haiku 52” - 1.32 m diameter from Big Ass Fans) located in the center

of the room in between the clouds. This fan can blow air either towards the room (downward direction) or towards the ceiling (upward direction). We conducted preliminary testing and decided to conduct all our testing at 146 rpm that corresponded to speed 4 (for both directions) because: (a) it provided acceptable average air velocities for comfort in the occupied zone (< 0.8 m/s as prescribed by ASHRAE Standard 55 (ANSI/ASHRAE 2013)); (b) it represented the mid-speed between 1 and 7; (c) it was the maximum upward speed and we wanted to be able to directly compare it with the same downward speed; and (d) we ran the experiment at fixed speed as we had limited experimental time, which was insufficient to investigate a fourth factor thoroughly.

(2) Small/compact AC axial fans (W2E200-HK86-01 - 0.20 m diameter from Ebm-Papst) mounted above the clouds and blowing air parallel to the ceiling. We controlled the fan speed with a variable transformer (1032, Control Concepts, USA). During preliminary testing, we discovered that the maximum speed of this fan exceeded our needs and we decided to run our experiment at 648 and 832 rpm (35% and 40% of the maximum setting, respectively), which we defined as 'low' and 'medium' speed, respectively.

Table 17 reports the fan test conditions. While the ceiling fan at the selected speed was extremely efficient in terms of energy, the small fans were less efficient, and brought an additional source of power within the chamber. We chose the small fan model based on their compatibility with the variable transformer used for this chamber and we realized much later that they were not energy efficient. There are far more efficient small (DC) fans available in the market that would be used in a final product, once the desired airflow is known, and the constraint to operate at a variable AC voltage is no longer necessary. Thus, to maintain a similar total heat load level during all experiments, we turned off selected office equipment (e.g., desk lamps, computer screens) that was approximately equal to the amount of extra heat added by the additional inefficient small fans used in this experiment. This adjustment was based on original power measurements conducted during the testing. Yet, these original measurements have proved to be inconsistent and the power meter was recalibrated after the testing. The value reported in Table 17 are the corrected values. We do not have the uncertainty of the power meter used to control the small fan.

7.2.2 Variant description

Figure 41 represents the location of the clouds and fans for different levels of acoustical cloud coverage: 0% (0 clouds), 16% (2 clouds), 32% (4 clouds), 47% (6 clouds) and 63% (8 clouds). We tested four different fan configurations: (1) ceiling fan blowing upward, (2) ceiling fan blowing downward, (3) small fans at low speed and (4) small fans at medium speed. For the test with small fans and 8 clouds, we oriented the fans toward the center of the room to avoid the corner fans blowing perpendicular to other fans. Table 18 summarizes all the experiments.

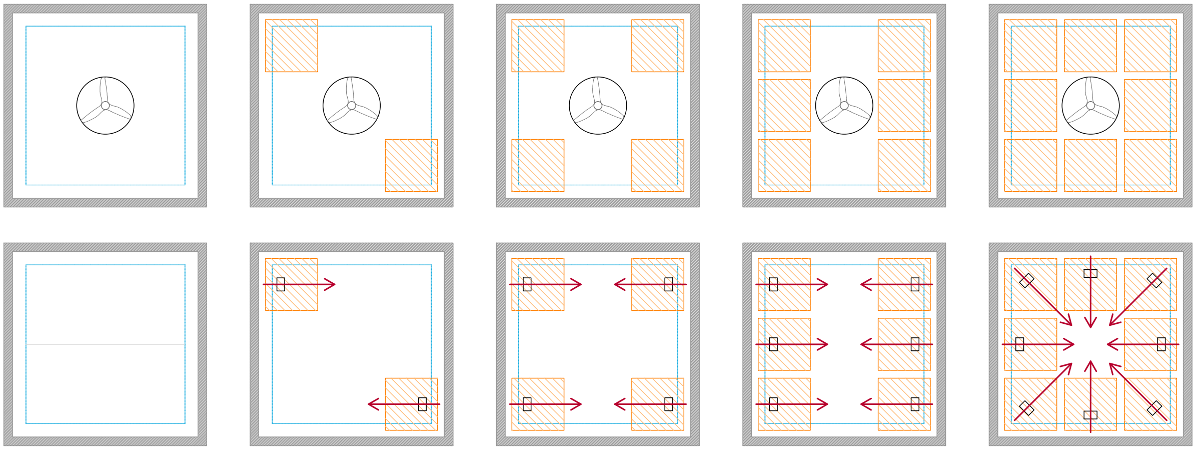


Figure 41: Acoustical coverage and fan configurations. The grey squares represent ceiling views. The orange hatched surfaces are the acoustical clouds (0%, 16%, 32%, 47%, 63% coverage). The upper row shows the variants with ceiling fan and the lower row the variants with small fans (the arrows on the small fan represent the blowing direction)

Table 18: Sequence of experiments and testing conditions

Test sequence	Coverage	Number of fans	Type of fan	Fan setting
1	63% coverage	8	Small fan	Medium speed (832 rpm/fan)
2	63% coverage	1	Ceiling fan	Upward blowing (146 rpm)
3	63% coverage	-	No fan	-
4	63% coverage	1	Ceiling fan	Downward blowing (146 rpm)
5	47% coverage	6	Small fan	Medium speed (832 rpm/fan)
6	47% coverage	6	Small fan	Low speed (648 rpm/fan)
7	47% coverage	1	Ceiling fan	Downward blowing (146 rpm)
8	47% coverage	1	Ceiling fan	Upward blowing (146 rpm)
9	47% coverage	6	No fan	-
10	0% coverage	1	Ceiling fan	Downward blowing (146 rpm)
11	0% coverage	-	No fan	-
12	0% coverage	1	Ceiling fan	Upward blowing (146 rpm)
13	32% coverage	4	Small	Medium speed (832 rpm/fan)
14	32% coverage	4	No fan	-
15	32% coverage	1	Ceiling fan(3)	Upward blowing (146 rpm)
16	32% coverage	4	Small fan	Low speed (648 rpm/fan)
17	16% coverage	2	Small fan	Low speed (648 rpm/fan)
18	16% coverage	-	No fan	-
19	16% coverage	1	Ceiling fan(3)	Upward blowing (146 rpm)
20	16% coverage	2	Small fan	Medium speed (832 rpm/fan)
21	47% coverage	6	Small fan	Low speed (648 rpm/fan)
22	47% coverage	6	No fan(1)	-
23	47% coverage	6	Small fan	Medium speed (832 rpm/fan)
24	0% coverage	-	No fan(1)	-
25*	0% coverage	-	No fan(2)	-
26*	0% coverage	1	Ceiling fan(2)	Upward blowing (146 rpm)

(1) Duplicated experiment; (2) EN14240 test; (3) Ceiling fan downward blowing for 16% and 32% coverage tests were not conducted

7.2.3 Experimental conditions and procedures

The experimental conditions and procedures were similar to the ones used for our previous study (Karmann et al. 2017). This section therefore mainly focuses on the differences in the testing conditions and equations used.

7.2.3.1 Procedure

As in our first study, we used the operative temperature measured in the center of the room at 1.1 m as our reference temperature to determine the cooling capacity between the ceiling and the room. In order to account for the effect of elevated air movement, we calculated the operative temperature based on the air and mean radiant temperatures and local air speed (ISO, EN 2001):

$$t_{op} = \frac{t_{mr} + (t_a \cdot \sqrt{10\, v_a})}{1 + \sqrt{10\, v_a}} \qquad (1)$$

The mean radiant temperature was derived from the globe temperature with the following equation (ISO, EN 2001):

$$T_{mr} = \left[\left(T_{globe} + 273\right)^4 + 2.5 \cdot 10^8 \cdot {v_a}^{0.6} \cdot \left(T_{globe} - T_a\right)\right]^{1/4} - 273 \qquad (2)$$

The heat exchange between the radiant ceiling and the room is expressed in Equation 3, where U_{cc} represents the cooling capacity coefficient in ($W \cdot m^{-2} \cdot K^{-1}$).

$$q_{cc} = U_{cc}\, A_{panels} \left(t_{op} - t_{w,m}\right) \qquad (3)$$

$$\text{with } t_{w,m} = \frac{t_{w,s} + t_{w,r}}{2}$$

On the water side, the radiant system cooling rate is expressed as:

$$q_w = \dot{m}_w\, c_{p_w}(t) \left(t_{w,r} - t_{w,s}\right) \qquad (4)$$

Under steady state conditions, the water in the radiant panels absorbs the electrical power of the heat sources, and thus:

$$P = q_{cc} = q_w \qquad (5)$$

By substituting (2) & (3) in (4), and rearranging:

$$U_{cc} = \frac{\dot{m}_w\, c_{p_w} \left(t_{w,r} - t_{w,s}\right)}{A_{panels} \left(t_{op} - t_{w,m}\right)} \qquad (6)$$

We conducted all tests with the same water mass flow rate (220 kg/h ± 0.02%) and water supply temperature (15 ºC ± 0.01 ºC). The return water temperature stayed between 19.1 ºC and 19.9 ºC. From the cooling capacity coefficient equation, the remaining variable that can influence the cooling capacity is the operative temperature, which is the equilibrium temperature at which the room settles for a given variant of acoustical cloud coverage under steady state conditions.

Steady state conditions were defined as a difference of less than 0.05 °C between the mean of the most recent 60 samples (most recent 30 minutes) against the mean of the 60 samples immediately prior (30 minutes immediately prior), for every temperature sensor used in this experiment. We recorded all monitored data once the test had reached steady state conditions, which typically took 300 minutes from the start of each experiment. After recording the data, we calculated the average temperatures in the occupied zone and the cooling capacity coefficient U_{cc}. This procedure was the same as in our previous study.

7.2.3.2 Comparison with EN 14240

We compared our approach to determine the cooling capacity of radiant panels to the methodology provided by the European Standard EN 14240 (CEN 2004). When maintained at the same temperature, we expect the surface heat transfer from the cooled surface to the room to be the same in the case of TABS as for radiant panels (we are not modelling conduction). Our comparison showed that for the range of temperatures tested, using a linear model was comparable to a power model in predicting the cooling capacity: R^2 of 0.980 for the power model and 0.974 for the linear model for the test without air speed, and R^2 of 0.986 for the power model and 0.985 for the linear model for the test with air speed. This comparison confirmed our approach of using a linear model.

7.2.3.3 Experimental sequence

The experimental sequence was partially randomized. Yet, to ease the laboratory set-up, we grouped the tests for each cloud coverage level. We started with the tests that included the largest amount of clouds (9, 8 and 6 clouds), then conducted the test without coverage and ended with the tests with lowest coverage (4 and then 2 clouds). Within each coverage, the tests were randomized. We included in the experiments two replications (no fans for 0 and 6 clouds) that we ran at the end of the testing. Table 18 reports the testing sequence.

7.2.4 Measuring instruments and uncertainty

The measuring instruments and equipment used in this experiment and their accuracy were the same as in our previous study. This testing included additional air velocity sensors that were the same models as the ones previously used. Figure 34 shows the location of all the sensors in plan and section view. We analysed the data in accordance with the ISO 13005 (ISO, EN 2000) and the JCGM 100 guidelines (JCGM 2008) for the expression of uncertainty. The equation used to calculate the operative temperature included the air speed. Thus, the expression of the uncertainty of the operative temperature is of the form:

$$u_{t_{op}} = \sqrt{\left(\frac{\delta t_{op}}{\delta t_a}\right)^2 {u_{t_a}}^2 + \left(\frac{\delta t_{op}}{\delta t_{mr}}\right)^2 {u_{t_{mr}}}^2 + \left(\frac{\delta t_{op}}{\delta v_a}\right)^2 {u_{v_a}}^2} \qquad (7)$$

with: $\frac{\delta t_{op}}{\delta t_a} = \frac{\sqrt{10 \cdot v_a}}{1+\sqrt{10 \cdot v_a}}$

$$\frac{\delta t_{op}}{\delta t_{mr}} = \frac{1}{1+\sqrt{10 \cdot v_a}}$$

$$\frac{\delta t_{op}}{\delta v_a} = \frac{\sqrt{2.5} \cdot t_a}{(1+\sqrt{10 \cdot v_a}) \cdot \sqrt{v_a}} - \frac{\sqrt{2.5} \cdot (t_{mr} + t_a \cdot \sqrt{10 \cdot v_a})}{(1+\sqrt{10 \cdot v_a})^2 \cdot \sqrt{v_a}}$$

All other uncertainty equations stayed the same as those reported in our previous study. The uncertainty in the cooling capacity coefficient U_{cc} is of the form:

$$u_{U_{cc}} = \sqrt{\left(\frac{\delta U_{CC}}{\delta \dot{m}_w}\right)^2 {u_{\dot{m}_w}}^2 + \left(\frac{\delta U_{CC}}{\delta t_{w,s}}\right)^2 {u_{t_{w,s}}}^2 + \left(\frac{\delta U_{CC}}{\delta t_{w,r}}\right)^2 {u_{t_{w,r}}}^2 + \left(\frac{\delta U_{CC}}{\delta T_{op}}\right)^2 {u_{t_{op}}}^2}$$

(8)

with: $\frac{\delta U_{CC}}{\delta \dot{m}_w} = \frac{c_{p_w}(t_{w,r} - t_{w,s})}{A_{panels}\left(t_{op} - \frac{t_{w,r} + t_{w,s}}{2}\right)}$

$$\frac{\delta U_{CC}}{\delta t_{w,s}} = \frac{\dot{m}_w \, c_{p_w} \left(t_{op} - t_{w,s}\right)}{A_{panels}\left(t_{op} - \frac{t_{w,r} + t_{w,s}}{2}\right)}$$

$$\frac{\delta U_{CC}}{\delta t_{w,r}} = \frac{\dot{m}_w \, c_{p_w} \left(t_{w,r} - t_{op}\right)}{A_{panels}\left(t_{op} - \frac{t_{w,r} + t_{w,s}}{2}\right)}$$

$$\frac{\delta U_{CC}}{\delta t_{op}} = -\frac{\dot{m}_w \, c_{p_w} \left(t_{w,r} - t_{w,s}\right)}{A_{panels}\left(t_{op} - \frac{t_{w,r} + t_{w,s}}{2}\right)}$$

The derived uncertainties for the operative temperature, the total cooling load and U_{cc} were respectively 0.8%, 4.9% and 5.7% (maximum values).

7.3 Results

Table 19 summarizes the main results of the testing. The chamber reached steady-state equilibrium at an operative temperature between 25.2 ºC (no coverage and ceiling fan up) and 28.0 ºC (63% coverage and no fan).

Table 19: Experimental results

Cov	Fan(5)	ta(1) (ºC)	tmr(1) (ºC)	top(1) (ºC)	v(1) (m/s)	tw,r-tw,s (ºC)	tw,m (ºC)	q (W)	q"(2) (W/m2)	q"(3) (W/m2)	Ucc(3) (W/(m2·K))	Change(3) (%)
0%	No fan	27.1	26.0	26.5	0.10	4.2	17.1	1069	79.8	58.7	8.46	0.0%
	No fan(4)	26.7	25.7	26.2	0.11	4.1	17.0	1038	77.5	57.0	8.44	-0.3%
	Clg-Dn	25.4	26.1	25.6	1.21	4.2	17.1	1076	80.3	59.1	9.51	12.4%
	Clg-Up	25.3	25.1	25.2	0.40	4.4	17.2	1113	83.1	61.2	10.36	22.4%
16%	No fan	27.2	26.4	26.8	0.12	4.2	17.1	1067	79.7	58.6	8.19	-3.2%
	Clg-Up	25.3	25.3	25.3	0.37	4.4	17.2	1120	83.6	61.5	10.35	22.3%
	Small-L	26.3	25.9	26.1	0.05	4.2	17.1	1075	80.3	59.1	8.93	5.5%
	Small-M	26.3	25.9	26.1	0.06	4.2	17.1	1079	80.5	59.3	9.00	6.4%
32%	No fan	27.2	26.7	27.0	0.12	4.1	17.1	1054	78.7	57.9	7.93	-6.2%
	Clg-Up	25.5	25.7	25.6	0.41	4.4	17.2	1129	84.3	62.0	10.04	18.7%
	Small-L	26.0	25.9	25.9	0.09	4.2	17.1	1079	80.5	59.3	9.12	7.8%
	Small-M	25.5	25.7	25.6	0.15	4.3	17.1	1097	81.9	60.2	9.69	14.6%
47%	No fan	27.9	27.3	27.6	0.07	4.1	17.1	1061	79.2	58.3	7.54	-10.9%
	No fan(4)	27.5	27.2	27.3	0.07	4.1	17.0	1040	77.6	57.1	7.54	-10.9%
	Clg-Dn	25.9	26.7	26.1	0.90	4.2	17.1	1085	81.0	59.6	9.06	7.1%
	Clg-Up	25.7	26.1	25.8	0.36	4.4	17.2	1124	83.9	61.8	9.72	14.9%
	Small-L	25.7	26.0	25.8	0.24	4.3	17.1	1090	81.4	59.9	9.36	10.7%
	Small-M	25.4	25.9	25.6	0.33	4.4	17.2	1125	84.0	61.8	10.05	18.8%
63%	No fan	28.0	28.1	28.0	0.08	4.1	17.0	1041	77.7	57.2	7.06	-16.5%
	Clg-Dn	25.7	26.9	26.0	0.85	4.1	17.0	1048	78.2	57.6	8.75	3.4%
	Clg-Up	25.6	26.2	25.8	0.35	4.2	17.1	1077	80.4	59.2	9.25	9.3%
	Small-L(6)	26.3	26.3	26.3	0.39	4.4	17.2	1137	84.9	62.5	9.35	10.5%
	Small-M(6)	26.1	26.4	26.1	0.72	4.8	17.4	1239	92.5	68.1	10.67	25.5%

(1) Measured at the central tree at 1.1 m; (2) Per unit of panel area; (3) Per unit of floor area; (4) Replicated experiment; (5) Fan acronyms: 'Small-L' and 'Small-M' stands for small fan low and medium speeds, 'Clg-Up' and 'Clg-Dn' stands for ceiling fan blowing up and down; (6) Small fans blowing towards the center of the room (not set parallel to each other)

7.3.1 Cooling capacity

Figure 42 shows the change in cooling capacity coefficient, U_{cc}, for all tests as a function of the acoustical coverage. For the variants with no fan, adding coverage reduces the cooling capacity.

Yet, the reduction in cooling capacity is on average a factor of 4 to 5 times lower than the percentage increase in cloud coverage. A detailed description of this result and a comparison with the literature is reported in (Karmann et al. 2017). The variants with ceiling fan have a similar slope as the reference case. However, the ceiling fan (for the speed and type tested) brings an increase in the cooling capacity of 22-26% when blowing up and 12-20% when blowing down compared to the reference case over the same range of coverage. When the ceiling fan is used, the increase in cooling capacity is substantial. For both blowing directions, even with the highest coverage of 63%, the change in overall cooling capacity is positive compared to the reference case (no fan and no coverage). The variants with small fans also show a significant increase in cooling capacity compared to the reference case (up to 26% increase). We note that the slope here goes in the opposite direction: the cooling capacity increases with increasing coverage. We can conclude that the combination of a small fan on top of a cloud brings an increase of cooling capacity compared to the bare ceiling. Increasing the air speed of the small fans brings an additional increase in cooling capacity. Yet, the two curves (for small fan low and medium speeds) do not show equal slopes: at 63% coverage, the slope for the medium speed increases while the slope for low speed flattens. For this last coverage, we oriented the fans toward the center of the room (see Figure 41) and a possible explanation for this may be a stronger increase in convection occurring at this speed. Overall the results of the testing (for both fans) demonstrate that we can increase the acoustical coverage to provide the needed sound absorption while simultaneously increasing the cooling capacity of a radiant ceiling. Section 7.3.4 compares the increased cooling capacity and increased fan power.

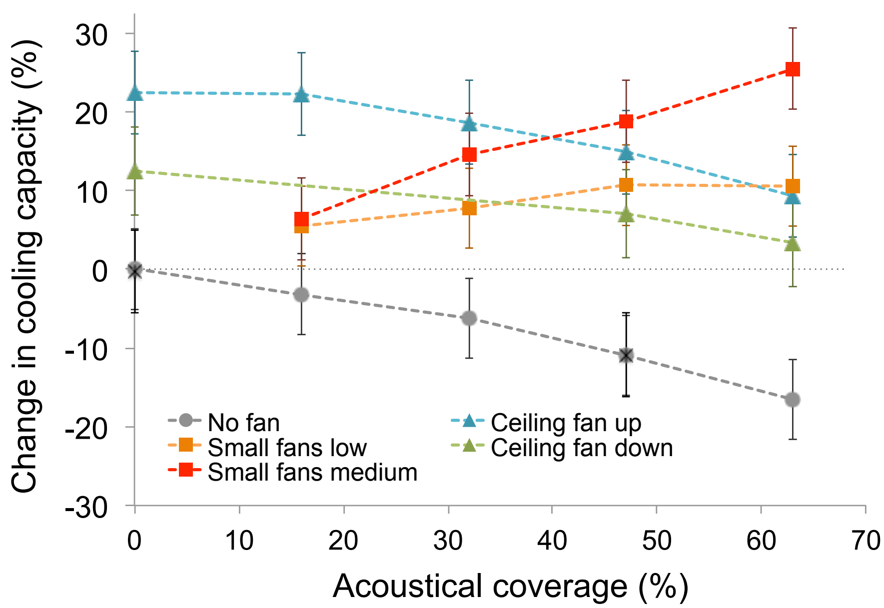


Figure 42: Change in cooling capacity coefficient as a function of acoustical cloud ceiling coverage. Three configurations are reported: (1) no fans, (2) small fans, and (3) ceiling fan. The black "x" represent the replications

Our testing conditions did not allow for a detailed analysis of the two components (convection and radiation) of the heat transfer coefficient. To do so, our set-up would have involved heat flux sensors and more surface temperature and air speed sensors close to the ceiling. We decided not to investigate this point due to budget and sensor constraints and due to its limited value on practical applications. Its relevance is mainly from the theoretical point of view, and this was not a priority in this work. This question is further addressed in section 7.5 (study limitations and future directions).

7.3.2 Air speed in the ceiling plenum

Figure 43 shows the air speed in the ceiling plenum for a selected set of variants. We measured air speed with three sensors whose locations are reported in Figure 34. Due to the limited number of speed measurements, we have only used these results to help interpret the overall magnitude of air movement generated by the different variants and fan types. Our observations assume that air speeds measured by the upper sensors (closer to the radiant panels) will more closely correlate with changes in the cooling capacity from the ceiling. For the variants with no fan, all measurements in the plenum stayed within 0.10 - 0.19 m/s. For the ceiling fan variants, the measured air speed on the upper center sensor (above the blades) stayed at around 0.75 m/s for both blowing directions and coverage. The air speed above the upper side sensor (above one lateral cloud) measured air speeds of 0.60 and 1.13 m/s for the ceiling fan in downward and upward directions respectively. We found that for all ceiling fan tests, the coverage has only a small effect on air speed. If we look at the small fans, the lower side sensor shows a higher air speed than the upper sensors. However, the opposite would be more beneficial to increase cooling capacity. For both the low and medium speeds, the 47% coverage shows overall higher air speed than the 63% coverage, especially for the lower sensor. At 63% coverage, the small fans with low speed have an average air speed of 0.38 m/s for the two upper sensors while the small fans with medium speed average 0.65 m/s. We can conclude that the medium fan speed brings substantially more air flow close to the ceiling. With a denser distribution of small fans at the ceiling level, the medium speed can bring a large increase in cooling capacity.

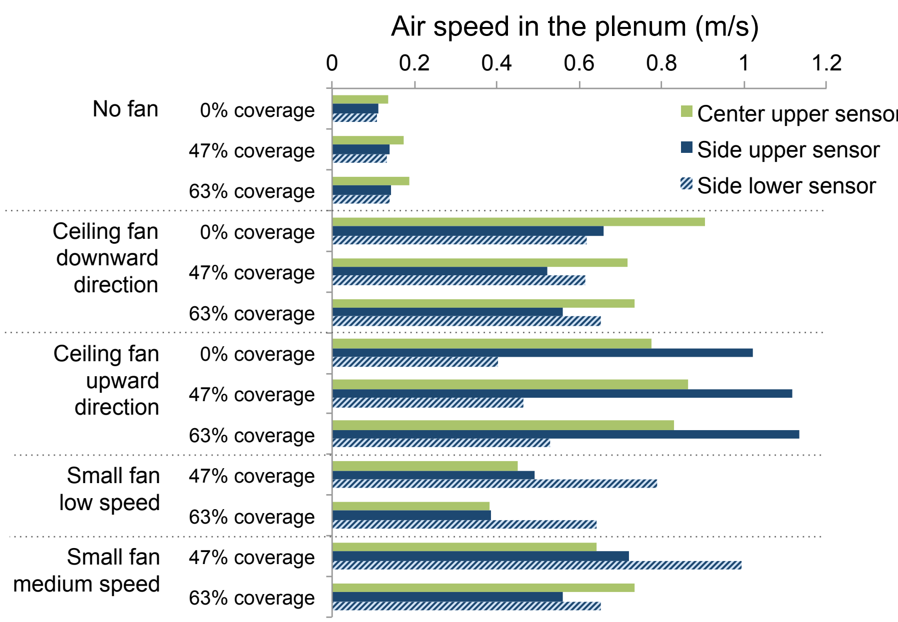


Figure 43: Air speed in the plenum for a selected set of variants. The locations of the sensors are detailed in Figure 34. Lower and upper sensors are within the plenum at 0.17 m and 0.08 m from ceiling respectively (or 0.08 m and 0.17 m from upper side of the acoustical cloud).

7.3.3 Air speed in the occupied space and its impact on thermal comfort

We measured the thermal comfort parameters at two locations (see Figure 34, center and side trees). The side tree is more representative of a typical occupant location underneath an acoustical cloud, so we conducted our analysis for this location. We measured the air speeds at the different heights typically used in thermal comfort assessment (0.1, 0.6, 1.1 and 1.7 m). Figure 44 presents the results of this analysis for the different fan variants and coverages. The air speed profile for the ceiling fan variants confirms what we would expect from these fans: in the downward direction, the fan pushes the air towards the room and the highest speed is observed nearby the floor (when it meets an obstacle). The ceiling fan in the upward direction is commonly used to counteract stratification. For this variant, the air speed is between 0.10 and 0.33 m/s throughout the measurement locations and the highest speed is observed at 1.1 m height. Figure 44 (C), shows the readings for the small fan at low speed. All the measurements between 0.6 and 1.7 m height stay below 0.12 m/s and the highest air speed is observed nearby the floor for the highest coverage and it reaches 0.29 m/s. In ASHRAE 55, an air speed of 0.2 m/s is the minimum speed to be allowed to cause a cooling effect due to elevated air speed (based on (ANSI/ASHRAE 2013)). We can conclude the air speeds for the small fan at low speed have a negligible effect on thermal comfort in the occupied zone. Similar conclusions apply for the small fan at medium speed with 16%, 32% and 47% coverage (see Figure 44 (D)). For 63% coverage, the velocities at 0.1 and 1.1 show an increase of about 0.35 m/s compared to the velocities at 0.6 and 1.7. The heights of 0.1 and 1.1 m can be associated with ground and desk height near the tree. As this variant brought the highest air speed at the plenum level (compared to other small fan variants), we can hypothesize that this irregular profile may be related to more complex air flow patterns and obstacles.

Figure 45 shows how air speed affects the thermal comfort zone. For this assessment, we assumed a seated occupant performing office work (1.1 met) with a total clothing insulation of 0.7 clo (wearing a short sleeve shirt, long trousers, socks, and business shoes (0.55 clo) and sitting in an office chair with a cushion seat and mesh back (0.15 clo)). We used a relative humidity of 50%. For each variant, we calculated the average air speed of 3 heights (0.1, 0.6, 1.1m) representing the occupied zone of a seated occupant using the measurements recorded on the side tree. We used this average air speed as input to the CBE Thermal Comfort Tool (Hoyt et al. 2013) to determine the center of the comfort zone based on the following: (1) the PMV model when air speed is below 0.2 m/s, and (2) the PMV+SET models when the air speed is above 0.2 m/s (ANSI/ASHRAE 2013; Arens et al. 2009). For simplicity, we used the operative temperature output as the center of the thermal comfort zone. Figure 45 reports acceptable ranges of operative temperature as a function of air speed for each variant. This graph shows that the ceiling fan blowing down allows an increase in the operative temperature of 2.9 ºC compared to the base case with no fan and no coverage. This increase is due to elevated air speed in the occupied zone. The literature has identified this as a relevant potential for energy savings (Schiavon and Melikov 2008; Hoyt, Arens, and Zhang 2015). Increasing the cooling setpoint temperature will likely have a larger savings effect for surface-conditioning systems as they operate over a far smaller temperature differential than traditional air-conditioning systems, and thus an increase of 1 ºC has a proportionally larger effect. Additionally, in many climates, increasing the cooling set point temperature will enable low energy cooling technologies, such as an evaporative cooling tower only approach instead of compressor based cooling, to be used where they would otherwise not be feasible.

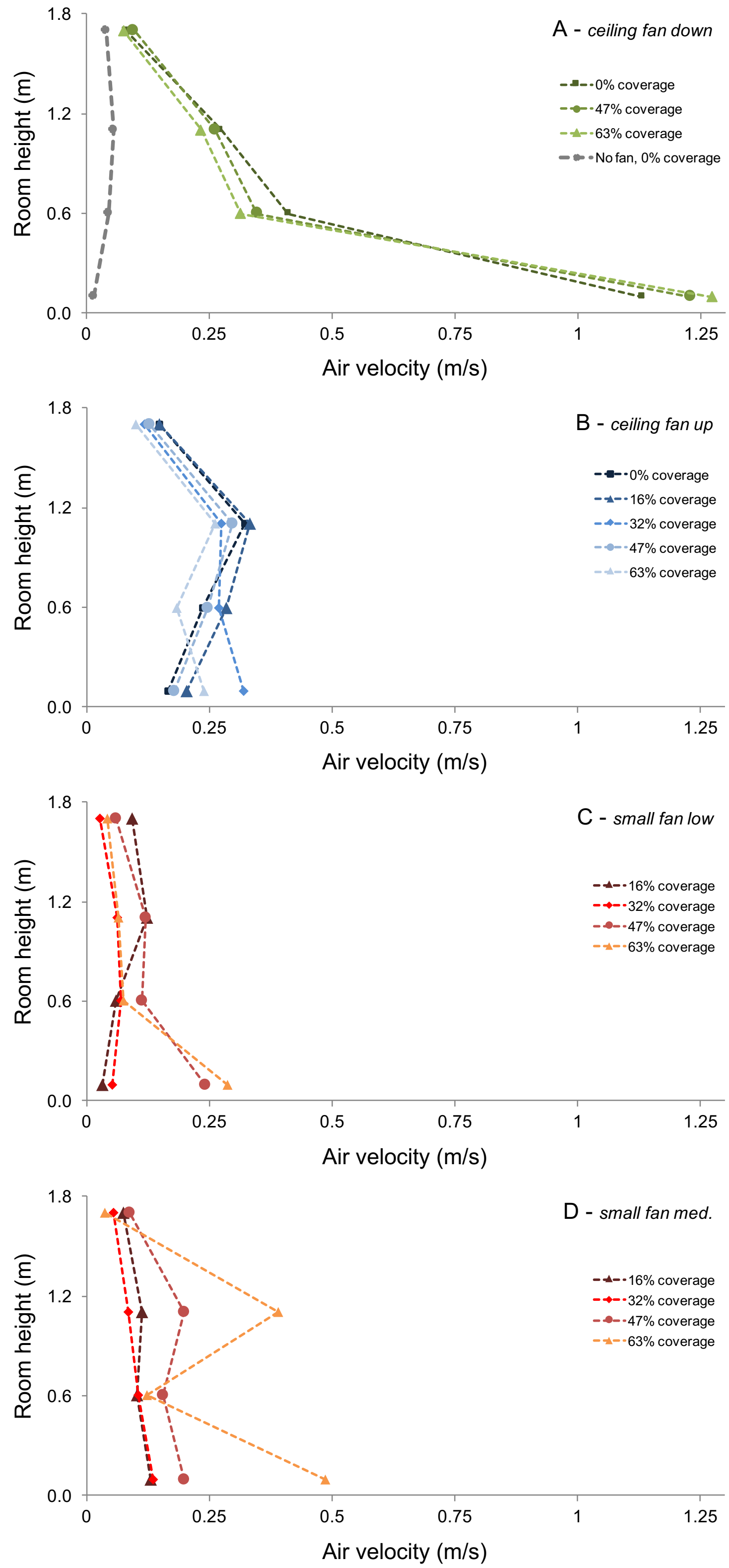


Figure 44: Speed profiles for the different variants and the different height measured at the side tree. This graph comprises 4 parts: (A) base case and ceiling fan blowing down for different coverage, (B) ceiling fan blowing down for different coverage, (C) small fans low speed for different coverage, (D) small fans medium speed for different coverage.

Based on Figure 44 (A), we note that the highest speeds for ceiling fan blowing down occur for the velocities at the ankle level, which may skew the results to a higher value while bringing new concerns in terms of thermal comfort (Schiavon, Rim, et al. 2016). Depending on the coverage, the ceiling fan blowing up allows an increase in operative temperature of 1.3-1.6 ºC to keep comparable thermal comfort as the base case. It also provides a lower and more uniform air speed at each of the measurement heights, avoiding potential concerns of high velocities at ankle level, or at the head when an occupant is directly below the fan jet. Additionally, having a ceiling fan that has a user selectable direction allows the user to have control over the air speed in the occupied zone, while still maintaining an increase in radiant system cooling capacity. The small fans at medium speed and 63% coverage show the highest air speed of all small fan tests. Yet, based on Figure 44 (D), the air speed profile for this variant is irregular and difficult to explain. It is possible that the walls of the test chamber and the positioning of the desks may have an impact on air speeds in the occupied zone. The other variants using small fan (low and medium speed, all coverages considered) are all below 0.2 m/s and therefore, the effect of air movement in the case of small fan is nearly negligible from a thermal comfort perspective. ASHRAE Standard 55 removed the draft risk model present in EN ISO 7730 (2005) (ISO, EN 2005), therefore we did not use it. The conditions reported here do not violate the draft local discomfort requirements specified by ASHRAE Standard 55.

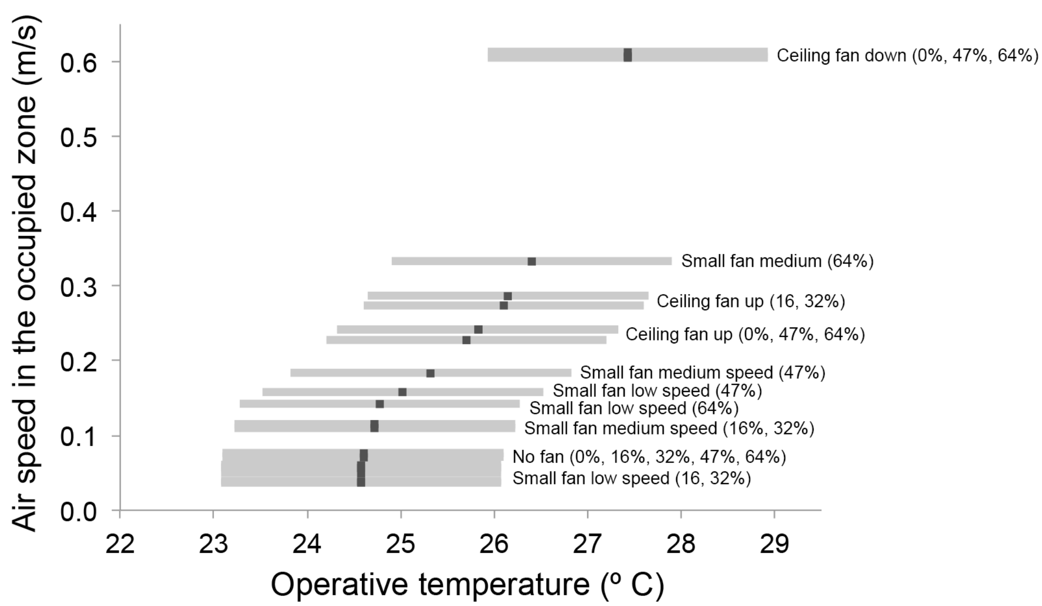


Figure 45: Acceptable ranges of operative temperature for different variants based on the average air speed for a seated person measured at the side tree at 0.1, 0.6 and 1.1 m. Assumptions for calculation: 0.7 clo, 1.1 met, RH 50%

7.3.4 Fan energy implications

Using fans to increase the cooling capacity and potentially expand the comfort zone also has an energy penalty related to the energy consumption of the fan. We looked at this parameter in the results output of our measurements. The ceiling fan used a DC motor and was particularly efficient in terms of energy usage (6 W or 0.3 W/m^2 when normalized by the room area). If we compare the change in cooling capacity to the change in heat losses from the ceiling fan, we are still benefiting from this strategy for all cases. By contrast, the small fans (these were chosen based on their maximum volumetric air flow and operability with the variable transformer of the chamber) used an AC motor and were not energy efficient models. We measured up to 115 W (or 6.3 W/m^2) heat added to the room for the highest fan usage (with 8 small fans at medium speed). This change in heat loads brought a higher penalty to the room conditions than the cooling benefit of increased cooling capacity. Observing this difference in fan power usage between models leads to one of the key conclusion of this study: the use of low-energy (DC) model fans is fundamental to provide energy benefits from increased air movement.

7.4 Discussion

In this study, we found the cooling capacity coefficient increases for all variants with fans compared to the reference case with no acoustical clouds and fans. The increase reached up to 22% for ceiling fan variants and up to 26% for small fan variants. We did not find prior studies that assessed the effect of fans nearby a radiant cooled ceiling but we found multiple studies have investigated the effect of ventilation systems located nearby a radiant ceiling. Depending on the diffuser and ventilation setting, these configurations brought an increase of the ceiling heat transfer coefficient from 17% (high aspiration diffusor) up to 35% (nozzle diffuser). Although these ventilation set-ups differ from our test configuration, we note a comparable order of magnitude in the results.

We conducted this experiment in a room without a ventilation system to reduce measurement uncertainty. The overall goal was to provide a proof-of-concept for the combined solution of adding acoustical clouds (reducing cooling capacity) and adding fans (increasing cooling capacity). The results are encouraging and warrant further testing and applications that will necessarily include a secondary air distribution system. For this experiment, we decided to focus on cooling capacity and thermal comfort by varying coverage and testing air movement strategies. Adding an air distribution system to our configuration introduces multiple questions, including the type of system (stratified or mixing), the type of diffusers and their location compared to positions of the fan and clouds. As seen in the literature, an overhead system will bring one additional source of air movement that may be nearby the plenum, which may increase the convective heat transfer even more. Combining the different systems at the ceiling will require design integration, both to address air flow dynamics and aesthetic aspects.

The combination of radiant ceiling with stratified ventilation systems (underfloor air distribution or displacement ventilation) have shown to be a reliable solution for energy and comfort aspects (Novoselac and Srebric 2002; 2012, 2015). Yet, as fans at the ceiling will disturb stratification, it is likely that the tested configuration may not work adequately with stratified systems, unless the mixing is confined to a region close to the ceiling. In this regard, small fans at lower speeds or reduced coverage could be compatible as they do not affect the air flow patterns in the occupied zone as much as the ceiling fan variants.

The use of fans has an impact on energy use and fan selection requires careful attention. DC fans can provide a low-energy solution (Yang et al. 2015), as shown with the ceiling fan used for this experiment. For the small fan variant, there are far more efficient products available than used in this experiment. Depending on their type and design, 0.20 m diameter computer fan models have the potential to offer a low-energy, but also low-cost and silent option (e.g., Antec Big Boy 200 -

200mm Tricool Computer Case Fan). Such types may be worth investigating in the context of practical application or future testing.

The use of fans will bring an additional source of noise in the space. We did not conduct sound pressure level measurement to determine its impact. Yet, for the types of fans and ranges of speed tested, current low-energy DC fans are much quieter than conventional AC models. Further, radiant systems are sometimes criticized for the lack of 'background noise' due to reduced mechanical systems in the space. In some cases, white noise generators (e.g., active sound masking) are installed to improve sound privacy of spaces with radiant systems. These points lead us to believe in minimal disturbance (if not a benefit) of a moderate fan noise added to the space.

This experiment was conducted for a medium sized office room. In larger spaces (such as open plan offices), we can expect that additional fans would be installed and that the absence of walls would lead to different air flow patterns. Yet, we can also hypothesize that the elevated air speed near the ceiling would bring comparable increases in cooling capacity as the ones observed in our testing. Acoustical clouds and fans could also be a good option for other type of environments (such as conference rooms, patient rooms, etc.).

7.5 Study limitations and future directions

For the small fan variants, the fans brought an additional source of heat within the chamber located above the acoustical panels and right below the radiant ceiling. The location of this heat source may be another explanation for the higher increase in cooling capacity observed. While we may assume that the convective component of the heat released from the small fans got spread into the full volume of the chamber (because of the well mixed conditions), we may also hypothesize that the radiative component of the heat transfer would mainly stay within the upper part of the room, and be absorbed directly into the radiant panels. The results presented for the small fans may therefore partially overestimate the effect of the small fan at increasing the cooling capacity based on increased air movement. This hypothesis could be verified through additional testing completed with low-energy small fans.

This testing did not focus on the changes in the relative strengths of radiant and convective heat transfer from the radiant ceiling panels. The limited information collected does not allow us to draw conclusive statements on this aspect. Yet, we conducted simplified calculations based on the data available to estimate the relative contributions from convection and radiation. Assuming a constant radiant heat transfer coefficient of 5.5 $W/m^2{\cdot}K$ (as observed in the literature (Babiak, Olesen, and Petras 2009; Causone et al. 2009)), we derived the values for the convective heat transfer coefficient, which remained below the radiant coefficient for all variants investigated. This simplified calculation deserves confirmation. Nevertheless, it supports the assessment that these combined ceiling systems can still be considered as 'radiant' (i.e., with a minimum of 50% of the load treated by radiation). In practical applications of these design solutions, the share of radiant and convective heat transfer has limited consequences.

We conducted this experiment in cooling mode. For heating applications, we expect that the reference cases with no air movement would bring a stronger decrease of heating capacity with added coverage. Convective heat transfer coefficients are lower for a warm vs. a cool ceiling surface due to buoyancy effects on natural convection. Thus, for a heating scenario, adding air movement nearby the ceiling will produce an increase in heating capacity that may be larger than the changes observed for a cooling scenario. The air speed strategy (fan type and mode) will need be chosen to avoid increased air movement in the occupied zone as it is counterproductive to cool occupants with air movement while trying to heat the space. Heating applications would be worth exploring and warrant further testing.

Another aspect of using fans to increase cooling capacity is that it offers a new approach for controlling cooling capacity and thermal comfort in spaces with exposed concrete radiant systems (TABS). Activating fans will increase the heat exchange between the room and the ceiling, and therefore, it offers additional control strategies to compensate for higher loads occurring in the space. Ceiling fans used for increased air motion in the occupied zone can also be used independently from the radiant system. Combining these strategies allows them to be used separately or simultaneously, bringing new ways to improve the control of the systems and occupant comfort, which is worth exploring in future work.

7.6 Conclusions

We conducted laboratory experiments to investigate the change in cooling capacity of an office room with a radiant cooled ceiling, and a combination of free-hanging acoustical clouds and fans. We tested two types of fans independently: a ceiling fan and a series of small fans located above the acoustical clouds. Compared to the reference case with no acoustical clouds and fans, the cooling capacity increases for all variants that use fans and for all levels of ceiling coverage, therefore, using elevated air movement is an effective strategy to compensate for the small reduction in cooling capacity due to the use of acoustical panels.

For the testing conducted with the ceiling fan the highest increase in cooling capacity happened with no coverage. It reached 22.4% for the upward blowing direction and 12.4% for the downward direction. The coverage has only a small effect on air speed in the occupied space. Elevated air motion in the occupied space can help expand the thermal comfort zone for the ceiling fan variants by about 1.4º C for the fans blowing upward and by 2.9º C for the fans blowing downwards. The increase in cooling capacity caused by the air movement generated by the fan far exceeds the heat gain due to the energy consumed by the fan.

For the testing conducted with small fans the highest increase in cooling capacity reached 26% at highest coverage (63%) for the small fans but elevated air movement in the occupied space was not relevant for thermal comfort purposes. In contrast to the ceiling fan variants, the increase in cooling capacity caused by the air movement generated by the fans does not exceed the heat gain due to the energy consumed by the fans. This is due to the particular model selected for this experiment, and a low-energy DC fan would likely yield a similar outcome as the ceiling fan variant, which used a highly efficient ceiling fan. The small fans provide comparable cooling capacity benefits as the ceiling fan, but without bringing a visible addition to the ceiling layout which may be more aesthetically desirable depending on design contexts.

In this study, we showed that combining acoustical clouds and fans not only offsets the modest reduction in cooling capacity from a radiant cooled ceiling caused by the presence of the clouds, but also provides an overall increase in cooling capacity compared to the reference case with no clouds and fan. This study offers a very promising and practical design solution regarding implementation of radiant slab ceiling systems.

8 Conclusions

Researchers and building professionals seek design strategies to simultaneously address the challenge of indoor environmental quality and energy use. Radiant heating and cooling systems have the potential to achieve significant energy savings primarily due to the use of lower temperature differences between the space and the heating or cooling source. These systems are seen as a market ready alternative to conventional all-air systems. Compared to buildings with all-air systems, buildings with radiant systems have been commonly associated with increased thermal comfort but decreased acoustic quality. This dissertation proposed to focus on these assumptions and investigated two major questions. First I wanted to know how buildings using radiant systems compare to buildings using all-air systems in terms of thermal comfort and acoustical quality. Second, I wanted test a practical solution designed to address cooling capacity, thermal comfort and sound absorption of massive radiant types.

As part of a larger research team I utilized multiple research methods to address these questions. I answered my first question using two methods: critical literature review on thermal comfort conditions for radiant vs. all-air systems; and statistical analysis of IEQ surveys to compare occupant satisfaction in buildings using radiant and all-air systems. For the second question, I tested a design solution involving free-hanging acoustical clouds and fans below a radiant chilled ceiling. I defined multiple variants of ceiling coverage and fans and conducted full-scale laboratory experiments to determine how this combination impacts the cooling capacity of the radiant chilled ceiling, and the thermal comfort and acoustic performance of the space. The key conclusions of this work are summarized below.

I performed a literature review to assess if radiant systems provide better, equal or lower thermal comfort than all-air systems. I reviewed a total of 73 papers from which eight would adequately answer our question and be judged conclusive. Five studies could not establish a thermal comfort preference between all-air and radiant systems and three studies showed a preference for radiant systems. Researchers used multiple methods to demonstrate their findings and, in addition, several types of all-air and radiant systems were tested. Overall, I found that a limited number of studies are available and therefore a solid answer cannot be given. There is suggestive evidence that radiant systems may provide equal or better comfort than all-air systems but additional studies are needed to confirm this statement. Both systems are able to provide acceptable thermal comfort, depending on several factors, including operation and control.

I used the CBE IEQ occupant survey to compare occupant satisfaction with their indoor environment in radiant and all-air conditioned buildings. This study involved the administration of new occupant satisfaction surveys to 1284 persons exposed to radiant systems. This dataset was supplemented with responses from 361 occupants previously collected. I used an existing database to extract a subset of occupant responses from all-air buildings (2247 persons) whose key characteristics match those radiant buildings. Radiant and all-air buildings have equal indoor environmental quality, including acoustical satisfaction (assessed for noise and sound privacy), with a tendency towards improved thermal comfort in radiant buildings. This disproves common biases against the acoustical performance of radiant systems. Less than 60% of the buildings used in this analysis were able to meet ASHRAE Standard 55 thermal comfort objective based on post-occupancy survey. Looking at a larger database, we found that only 44% of the buildings were able to comply with the standard's comfort objective. This result is surprisingly low, in particular as the current metric within the standards includes 'slightly dissatisfied' votes among the positive comfort responses. This observation raises questions regarding the practicality and applicability of the comfort metric as currently written.

The laboratory experiments investigated the change in cooling capacity, thermal comfort and sound absorption of an office room with different coverage areas of free-hanging acoustical clouds and fans below a radiant cooled ceiling. I tested 2 types of fans: a ceiling fan in-between the clouds (upward / downward blowing directions) and small fans above the clouds (two speed). For the testing conducted without fan, the observed reduction in cooling capacity at 47% coverage was only 11%, while this condition created an acceptable acoustic solution (reverberation time) for both private and open plan offices. Compared to the reference case with no clouds and fans, the cooling capacity increased for all variants that used fans for all levels of ceiling coverage. It increased by 22% for the ceiling fan blowing upward and by 12% for the ceiling fan blowing downward. Elevated air motion in the occupied space can help expand the thermal comfort zone for the ceiling fan variants by up to 2.9º C for the fans blowing downwards. For the variants with small fans, the cooling capacity increased with coverage proving that the combination of a cloud and a small fan has a positive effect on cooling capacity. The cooling capacity increased by 26% for the small fan running at medium speed and 63% coverage. The selection of a low-energy DC fan is critical to limit the heat gain due to the operation of the fan. Overall, this study showed that combining acoustical clouds and fans not only offsets the modest reduction in cooling capacity from a radiant cooled ceiling caused by the presence of the clouds, but also provides an overall increase in cooling capacity compared to the reference case with no clouds and fan.

This dissertation brought new insights and understanding on perceived thermal comfort and acoustic quality in building using radiant systems compared to all-air systems, and showed that the combination of free-hanging acoustic clouds and fans below a radiant chilled ceiling can offer a promising practical design solution to address the limited sound absorption of massive radiant ceilings.

Suggestion for future work

This dissertation was able to address practical questions on thermal comfort and acoustic quality in spaces using radiant systems.

This dissertation showed a few limitations and introduced a numbers of opportunities for future research:

- The occupant satisfaction study presented a few limitations related to: (1) the availability and number of buildings with radiant systems we could survey; (2) the dataset of all-air buildings that brought an unavoidable shift in building age between the two subsets; (3) the higher quality of buildings of this study compared to the average building of the CBE survey database; (4) the statistical methodology applied to the dataset (e.g., observer reliability) and to the matching of the two subset. For these reasons, it would be relevant to iterate our radiant vs. all-air comparison with an improved quantitative approach, larger subset and more representative buildings.

- We interpreted effect size based on the thresholds scales found in the statistical literature. Effect size is an important metric as it informs on practical significance. Yet, these scales were not validated for building science occupant satisfaction studies and therefore their appropriateness for our study is unknown. Future research shall address this lack of scales and thresholds for our field.
- Per ASHRAE Standard 55, buildings are intended to achieve 80% satisfaction with thermal comfort. However, our survey study showed that only 2 out of 60 buildings (3%) were able to achieve 80% satisfaction with thermal comfort. Enlarging ASHRAE's objective to 'neutral' and 'slightly dissatisfied' votes brought to 10% and 56% the number of buildings complying, respectively. Considering the higher quality of buildings gathered in this study, ASHRAE's objective seems impractical to achieve in buildings that use traditional systems. This observation opens the question of the definition of thermal comfort, the translation of sensation to satisfaction scales, the objectives according to different applications (laboratory studies, design and prediction, post-occupancy) and the means of assessment.
- We used the CBE IEQ occupant survey which consists of satisfaction questions. Thermal comfort is mainly captured by the question: "How satisfied are you with the temperature of your workspace?" The appropriateness of this metric to capture perceived thermal comfort in buildings deserves attention. We may compare this question to other (such as acceptability, preference, sensation that could be tailored for long-term evaluation), or to the use of proxies as alternative ways to capture the information sought. Further, satisfaction questions are based on a 7-point Likert scale. The distribution of votes for all IEQ categories is skewed, with a much larger representation of 'satisfied' votes compared to 'slightly satisfied' and 'very satisfied'. This observation leads us to question the level of detail in the scale used. As we want to know how occupants perceive their environment, it is fundamental to verify the appropriateness of the metrics and answer scales used, and to improve the ways to capture information.
- The question of the appropriate metric used for thermal comfort assessment based on occupant feedback become all the more relevant in the context of the recommendations stated in ASHRAE Standard 55. We noticed that the majority of the buildings surveyed (sample of 144 buildings) did not comply with the standard, despite the inclusion of 'slightly dissatisfied' votes among the positive comfort responses. This observation raises questions in regard to the practicality of the current metric. Additional means to assess thermal comfort based on post-occupancy occupant feedback would be worth studying.
- We investigated a practical solution of acoustical clouds with fans below a radiant chilled ceiling. Two prototypes were tested: a ceiling fan in-between the clouds and small fans above the clouds. The small fan solution appeared very promising. Yet, additional testing with DC powered fan would be needed to make it an appropriate solution on the energy side. For both variants, we conducted the testing in a chamber representing a small to medium office room. In larger spaces (such as open plan offices), we can expect that additional fans would be installed and that the absence of walls would lead to different air flow patterns. Testing the prototypes in a larger laboratory setting or in an actual open-space would be logical next steps to this study.
- For our testing without air movement, we only observed a small reduction of the cooling capacity (3.2%, 11% and 19% reduction for 16%, 47% and 71% coverage respectively). Yet, it is unlikely that a heating application would show similar results as there would be

stagnant, warmer air between the ceiling and the acoustic clouds, and we may expect a stronger reduction in heating capacity with increasing coverage by the clouds. Adding air movement nearby the ceiling will produce an increase in heating capacity that may be larger than the changes observed for a cooling scenario. The air speed strategy (fan type and mode) will need be chosen to avoid increased air movement in the occupied zone as it is counterproductive to cool occupants with air movement while trying to heat the space. Heating applications would be interesting to explore and warrant further testing.

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(to be submitted)	Karmann C, Schiavon S, Graham L, Bauman F, Raftery P, 2017. Comparing temperature and acoustic satisfaction in 60 radiant and all-air buildings. To be submitted to Building and Environment in August 2017
Conference papers	Karmann C, Schiavon S, Bauman F. 2015. Do radiant systems provide better thermal comfort than air systems? A brief critical literature review. Proceedings of the 9th International Symposium on Heating, Ventilation and Air Conditioning (ISHVAC) and the 3rd International Conference on Building Energy and Environment (COBEE). Tianjin, China. July 12-15.
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11 Appendices

11.1 Indoor Environmental Quality Occupant Satisfaction Survey

Name of the survey: EPIC Core Radiant Occupant Survey Master File

Default Question Block

Welcome!

Thank you for your participation in this building evaluation study. This study is part of a large research effort by the Center for the Built Environment at the University of California, Berkeley.

This study is being conducted in multiple buildings that have been designed with higher attention to energy efficiency. Your feedback will provide valuable data that will be used to identify how successful your building is in meeting its design goals. The results gathered will also help increase awareness of how real world energy-efficient buildings affect perceived occupant experience.

Questions/Feedback. If you have questions about the study, or for help using the survey or this website, please email (cbesurvey@berkeley.edu) the Center for the Built Environment at the University of California, Berkeley. A research specialist will respond to your question promptly. Your Rights. If you have questions about your rights as a participant in this research project, please email (subjects@uclink4.berkeley.edu) the University of California at Berkeley's Committee for Protection of Human Subjects (CPHS). Please do not contact CPHS regarding technical support issues.

Background

First, we'd like to ask you some questions about yourself.

How many years have you worked in this building?
- Less than 1 year (1)
- 1-2 years (2)
- 3-5 years (3)
- More than 5 years (4)

How long have you been working at your present workspace?
- Less than 3 months (1)
- 4-6 months (2)
- 7-12 months (3)
- More than 1 year (4)

In a typical week, how many hours do you spend in your workspace?
- 10 or less (1)
- 11-30 (2)
- More than 30 (3)

How would you describe the work that you do?
- Administrative support (1)
- Technical (2)
- Professional (3)
- Managerial/ supervisory (4)
- Other (5)

What is your age?
- 30 or under (1)
- 31-50 (2)
- Over 50 (3)

What is your gender?
- Female (1)
- Male (2)
- Transgender (3)

Personal Workspace Location

On which floor is your workspace located?
- ❍ 1 (2)

In which area of the building is your workspace located?
- ❍ North (facing XXX) (2)
- ❍ East (facing XXX) (3)
- ❍ South (facing XXX) (4)
- ❍ West (facing XXX) (5)
- ❍ Core (6)
- ❍ I don't know (7)

In which direction do the windows closest to your workspace face?
- ❍ North (facing XXX) (2)
- ❍ East (facing XXX) (3)
- ❍ South (facing XXX) (4)
- ❍ West (facing XXXl) (5)
- ❍ No windows (6)
- ❍ I don't know (7)

Are you near (i.e., within 15 feet)...

	Yes (1)	No (2)
An exterior wall (1)	❍	❍
A window (2)	❍	❍

Which of the following best describes your personal workspace?
- ❍ Enclosed office, private (1)
- ❍ Enclosed office, shared with other people (2)
- ❍ Cubicles with high partitions (about 5 or more feet high) (3)
- ❍ Cubicle with low partitions (lower than 5 feet high) (4)
- ❍ Workspace in open office with no partitions (just desks) (5)
- ❍ Other (6) ____

Office Layout

How satisfied are you with the...

	Very satisfied (1)	Satisfied (2)	Somewhat satisfied (3)	Neither satisfied nor dissatisfied (4)	Somewhat dissatisfied (5)	Dissatisfied (6)	Very dissatisfied (7)
Amount of space available for individual work and storage (1)	❍	❍	❍	❍	❍	❍	❍
Level of visual privacy (2)	❍	❍	❍	❍	❍	❍	❍
Ease of interaction with co-workers (3)	❍	❍	❍	❍	❍	❍	❍

Overall, does the office layout enhance or interfere with your ability to get your job done?

	Significantly enhances (1)	Enhances (2)	Somewhat enhances (3)	Neither enhances nor interferes (4)	Somewhat interferes (5)	Interferes (6)	Significantly interferes (7)
(1)	❍	❍	❍	❍	❍	❍	❍

Please describe any other issues related to the office layout that are important to you.

Display This Question:
If How satisfied are you with the... Amount of space available for individual work and storage - Very dissatisfied Is Selected
Or How satisfied are you with the... Amount of space available for individual work and storage - Dissatisfied Is Selected
Or How satisfied are you with the... Amount of space available for individual work and storage - Somewhat dissatisfied Is Selected

You have said that you are dissatisfied with the amount of space available for individual work and storage. Which of the following contribute to your dissatisfaction? (Check all that apply.)

- ❑ Amount of work surface area (1)
- ❑ Total area of work station (2)
- ❑ Available filing and storage space (3)
- ❑ Available space for personal items (4)
- ❑ Space for meeting with other people (5)
- ❑ Other (6) ____________________

Display This Question:
If How satisfied are you with the... Level of visual privacy - Very dissatisfied Is Selected
Or How satisfied are you with the... Level of visual privacy - Dissatisfied Is Selected
Or How satisfied are you with the... Level of visual privacy - Somewhat dissatisfied Is Selected

Q16 You have said that you are dissatisfied with the level of visual privacy. Which of the following contributes to your dissatisfaction? (Check all that apply)

- ❑ High density-- too little space separating people (1)
- ❑ Partitions or walls are too low or transparent (2)
- ❑ People can easily see in through exterior windows (3)
- ❑ Too many people walking in my work area (4)
- ❑ Other (5) ____________________

Display This Question:
If How satisfied are you with the... Ease of interaction with co-workers - Very dissatisfied Is Selected
Or How satisfied are you with the... Ease of interaction with co-workers - Dissatisfied Is Selected
Or How satisfied are you with the... Ease of interaction with co-workers - Somewhat dissatisfied Is Selected

Q17 You have said that you are dissatisfied with the ease of interaction with co-workers. Which of the following contribute to your dissatisfaction? (Check all that apply).

- ❑ My work station is not near my co-workers (1)
- ❑ My work station is difficult to find or out of the way (2)
- ❑ Conversations are discouraged because the noise is distracting to others (3)
- ❑ There are no spaces (i.e., break rooms) to casually interact with co-workers (4)
- ❑ There are few organized opportunities to interact with co-workers (5)
- ❑ Other (6) ____________________

Office Furnishings

How satisfied are you with...

	Very satisfied (1)	Satisfied (2)	Somewhat satisfied (3)	Neither satisfied nor dissatisfied (4)	Somewhat dissatisfied (5)	Dissatisfied (6)	Very dissatisfied (7)
The comfort of your office furnishings (chair, desk, computer, equipment, etc.) (1)	❍	❍	❍	❍	❍	❍	❍
Your ability to adjust your furniture to meet your needs (2)	❍	❍	❍	❍	❍	❍	❍
The colors and textures of flooring, furniture, and surface finishes (3)	❍	❍	❍	❍	❍	❍	❍

Do your office furnishings enhance or interfere with your ability to get your job done?

	Significantly enhance (1)	Enhance (2)	Somewhat enhance (3)	Neither enhance nor interfere (4)	Somewhat interfere (5)	Interfere (6)	Significantly interfere (7)
(1)	❍	❍	❍	❍	❍	❍	❍

Please describe any other issues related to office furnishings that are important to you.

Thermal Comfort

Which of the following do you personally adjust or control in your workspace? (Check all that apply)

- ❑ Window blinds or shades (1)
- ❑ Operable window (2)
- ❑ Thermostat (3)
- ❑ Portable heater (4)
- ❑ Permanent heater (5)
- ❑ Room air-conditioning unit (6)
- ❑ Portable fan (7)
- ❑ Ceiling fan (8)
- ❑ Adjustable air vent in wall or ceiling (9)
- ❑ Adjustable floor air vent (diffuser) (10)
- ❑ Door to interior space (11)
- ❑ Door to exterior space (12)
- ❑ None of the above (13)
- ❑ Other (14) ____________________

How satisfied are you with the temperature of your workspace?

	Very satisfied (1)	Satisfied (2)	Somewhat satisfied (3)	Neither satisfied nor dissatisfied (4)	Somewhat dissatisfied (5)	Dissatisfied (6)	Very dissatisfied (7)
(1)	❍	❍	❍	❍	❍	❍	❍

Overall, does your thermal comfort in your workspace enhance or interfere with your ability to get your job done?

	Significantly enhances (1)	Enhances (2)	Somewhat enhances (3)	Neither enhances nor interferes (4)	Somewhat interferes (5)	Interferes (6)	Significantly interferes (7)
(1)	❍	❍	❍	❍	❍	❍	❍

Display This Question:

If How satisfied are you with the temperature of your workspace?   - Very dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Somewhat dissatisfied Is Selected

You have said that you are dissatisfied with the temperature in your workspace. Which of the following contribute to your dissatisfaction?

Display This Question:

If How satisfied are you with the temperature of your workspace?   - Very dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Somewhat dissatisfied Is Selected

In warm/hot weather, the temperature in my workspace is: (Check all that apply)

- ❑ Often too hot (1)
- ❑ Often too cold (2)

Display This Question:

If How satisfied are you with the temperature of your workspace?   - Very dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Somewhat dissatisfied Is Selected

In warm/hot weather... (Check all that apply)

- ❑ My hands are too cold (1)
- ❑ My feet are too cold (2)
- ❑ Other (3) ____________________

Display This Question:

If How satisfied are you with the temperature of your workspace?   - Very dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Somewhat dissatisfied Is Selected

In cool/cold weather, the temperature in my workspace is: (Check all that apply)

- ❑ Often too hot (1)
- ❑ Often too cold (2)

Display This Question:

If How satisfied are you with the temperature of your workspace?   - Very dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Somewhat dissatisfied Is Selected

In cool/cold weather... (Check all that apply)

- ❑ My hands are too cold (1)
- ❑ My feet are too cold (2)
- ❑ Other (3) ____________________

Display This Question:

If How satisfied are you with the temperature of your workspace?   - Very dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Somewhat dissatisfied Is Selected

When is this most often a problem? (Check all that apply)

- ❑ Morning (before 11am) (1)
- ❑ Mid- day (11am - 2pm) (2)
- ❑ Afternoon (2pm - 5pm) (3)
- ❑ Evening (after 5pm) (4)
- ❑ Weekends/ holidays (5)
- ❑ Monday mornings (6)
- ❑ No particular time (7)
- ❑ Other (8) ____

Display This Question:

If How satisfied are you with the temperature of your workspace?   - Very dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Somewhat dissatisfied Is Selected

Q30 How would you best describe the source of this discomfort? (Check all that apply)

- ❑ Humidity too high (damp) (1)
- ❑ Humidity too low (dry) (2)
- ❑ Air movement too high (3)
- ❑ Air movement too low (4)
- ❑ Incoming sun (5)
- ❑ Hot/ cold floor surfaces (6)
- ❑ Hot/ cold ceiling surfaces (7)
- ❑ Hot/ cold wall surfaces (8)
- ❑ Hot/ cold window surfaces (9)
- ❑ Heat from office equipment (10)
- ❑ Drafts from windows (11)
- ❑ Drafts from vents (12)
- ❑ Drafts falling from the ceiling (13)
- ❑ My area is hotter than other areas (14)
- ❑ My area is colder than other areas (15)
- ❑ Thermostat is inaccessible (16)
- ❑ Thermostat is adjusted by other people (17)
- ❑ Heating/ cooling system does not respond quickly enough to the thermostat (18)
- ❑ Clothing policy is not flexible (19)
- ❑ Other (20) ____

Display This Question:

If How satisfied are you with the temperature of your workspace?   - Very dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Dissatisfied Is Selected
Or How satisfied are you with the temperature of your workspace?   - Somewhat dissatisfied Is Selected

Please describe any other issues related to being too hot or too cold in your workspace.

____

Air Quality

How satisfied are you with the air quality in your workspace? (i.e., stuffy/ stale air, cleanliness, odors)

	Very satisfied (1)	Satisfied (2)	Somewhat satisfied (3)	Neither satisfied nor dissatisfied (4)	Somewhat dissatisfied (5)	Dissatisfied (6)	Very dissatisfied (7)
(1)	❍	❍	❍	❍	❍	❍	❍

Overall, does the air quality in your workspace enhance or interfere with your ability to get your job done?

	Significantly enhances (1)	Enhances (2)	Somewhat enhances (3)	Neither enhances nor interferes (4)	Somewhat interferes (5)	Interferes (6)	Significantly interferes (7)
(1)	❍	❍	❍	❍	❍	❍	❍

Display This Question:

If How satisfied are you with the air quality in your workspace? (i.e., stuffy/ stale air, cleanline...   - Very dissatisfied Is Selected

Or How satisfied are you with the air quality in your workspace? (i.e., stuffy/ stale air, cleanline...   - Dissatisfied Is Selected

Or How satisfied are you with the air quality in your workspace? (i.e., stuffy/ stale air, cleanline...   - Somewhat dissatisfied Is Selected

You have said that you are dissatisfied with the air quality in your workspace. Please rate the level of each of the following problems.

	Not a problem (1)	Minor problem (2)	A problem (3)	Major problem (4)
Air is stuffy/ stale (1)	❍	❍	❍	❍
Air is not clean (2)	❍	❍	❍	❍
Air smells bad (odors) (3)	❍	❍	❍	❍

If there is an odor problem, which of the following contribute to the problem? (Check all that apply)

- ❑ Tobacco smoke (1)
- ❑ Photocopiers (2)
- ❑ Printers (3)
- ❑ Food (4)
- ❑ Carpet or furniture (5)
- ❑ Other people (6)
- ❑ Perfume (7)
- ❑ Cleaning products (8)
- ❑ Outdoor scents (car exhaust, smog) (9)
- ❑ Other (10) ____

Display This Question:

If How satisfied are you with the air quality in your workspace? (i.e., stuffy/ stale air, cleanline... - Very dissatisfied Is Selected

Or How satisfied are you with the air quality in your workspace? (i.e., stuffy/ stale air, cleanline... - Dissatisfied Is Selected

Or How satisfied are you with the air quality in your workspace? (i.e., stuffy/ stale air, cleanline... - Somewhat dissatisfied Is Selected

Please describe any other issues related to the air quality in your workspace that are important to you.

____

Lighting

Which of the following controls do you have over the lighting in your workspace? (Check all that apply)

- ❑ Light switch (1)
- ❑ Light dimmer (2)
- ❑ Window blinds or shades (3)
- ❑ Desk (task) light (4)
- ❑ None of the above (5)
- ❑ Other (6) ____

How satisfied are you with...

	Very satisfied (1)	Satisfied (2)	Somewhat satisfied (3)	Neither satisfied nor dissatisfied (4)	Somewhat dissatisfied (5)	Dissatisfied (6)	Very dissatisfied (7)
The amount of light in your workspace (1)	❍	❍	❍	❍	❍	❍	❍
The visual comfort of the lighting (e.g., glare, reflections, contrast) (2)	❍	❍	❍	❍	❍	❍	❍

Overall, does the lighting quality enhance or interfere with your ability to get your job done?

	Significantly enhance (1)	Enhance (2)	Somewhat enhance (3)	Neither enhance nor interfere (4)	Somewhat interfere (5)	Interfere (6)	Significantly interfere (7)
(1)	❍	❍	❍	❍	❍	❍	❍

Display This Question:
If How satisfied are you with... The amount of light in your workspace - Very dissatisfied Is Selected
Or How satisfied are you with... The amount of light in your workspace - Dissatisfied Is Selected
Or How satisfied are you with... The amount of light in your workspace - Somewhat dissatisfied Is Selected
Or How satisfied are you with... The visual comfort of the lighting (e.g., glare, reflections, contrast) - Very dissatisfied Is Selected
Or How satisfied are you with... The visual comfort of the lighting (e.g., glare, reflections, contrast) - Dissatisfied Is Selected
Or How satisfied are you with... The visual comfort of the lighting (e.g., glare, reflections, contrast) - Somewhat dissatisfied Is Selected

You have said that you are dissatisfied with the lighting in your workspace. Which of the following contribute to your dissatisfaction? (Check all that apply)

- ❑ Too dark (1)
- ❑ Too bright (2)
- ❑ Not enough daylight (3)
- ❑ Too much daylight (4)
- ❑ Not enough electric lighting (5)
- ❑ Too much electric lighting (6)
- ❑ Electric lighting flickers (7)
- ❑ Electric lighting is an undesirable color (8)
- ❑ No task lighting (9)
- ❑ Reflections in the computer screen (10)
- ❑ Shadows on the workspace (11)
- ❑ Other (12) ____

Display This Question:
If How satisfied are you with... The amount of light in your workspace - Very dissatisfied Is Selected
Or How satisfied are you with... The amount of light in your workspace - Dissatisfied Is Selected
Or How satisfied are you with... The amount of light in your workspace - Somewhat dissatisfied Is Selected
Or How satisfied are you with... The visual comfort of the lighting (e.g., glare, reflections, contrast) - Very dissatisfied Is Selected
Or How satisfied are you with... The visual comfort of the lighting (e.g., glare, reflections, contrast) - Dissatisfied Is Selected
Or How satisfied are you with... The visual comfort of the lighting (e.g., glare, reflections, contrast) - Somewhat dissatisfied Is Selected

Please describe any other issues related to lighting that are important to you.

____

Acoustic Quality

How satisfied are you with the...

	Very satisfied (1)	Satisfied (2)	Somewhat satisfied (3)	Neither satisfied nor dissatisfied (4)	Somewhat dissatisfied (5)	Dissatisfied (6)	Very dissatisfied (7)
Noise level in your workspace (1)	❍	❍	❍	❍	❍	❍	❍
Sound privacy in your workspace (ability to have communications without your neighbors overhearing and vice versa) (2)	❍	❍	❍	❍	❍	❍	❍

Overall, does the acoustic quality in your workspace enhance or interfere with your ability to get your job done?

	Significantly enhances (1)	Enhances (2)	Somewhat enhances (3)	Neither enhances nor interferes (4)	Somewhat interferes (5)	Interferes (6)	Significantly interferes (7)
(1)	❍	❍	❍	❍	❍	❍	❍

Display This Question:

If How satisfied are you with the... Noise level in your workspace - Very dissatisfied Is Selected

Or How satisfied are you with the... Noise level in your workspace - Dissatisfied Is Selected

Or How satisfied are you with the... Noise level in your workspace - Somewhat dissatisfied Is Selected

Or How satisfied are you with the... Sound privacy in your workspace (ability to have communications without your neighbors overhearing and vice versa) - Very dissatisfied Is Selected

Or How satisfied are you with the... Sound privacy in your workspace (ability to have communications without your neighbors overhearing and vice versa) - Dissatisfied Is Selected

Or How satisfied are you with the... Sound privacy in your workspace (ability to have communications without your neighbors overhearing and vice versa) - Somewhat dissatisfied Is Selected

You have said you are dissatisfied with the acoustics your workspace. Which of the following contributes to this problem? (Check all that apply)

- ❑ People talking on the phone (1)
- ❑ People talking in neighboring areas (2)
- ❑ People overhearing my private conversations (3)
- ❑ Office equipment noise (4)
- ❑ Office lighting noise (5)
- ❑ Telephones ringing (6)
- ❑ Mechanical (heating, cooling and ventilation system) noise (7)
- ❑ Excessive echoing of voices or other sounds (8)
- ❑ Outdoor traffic noise (9)
- ❑ Other outdoor noise (10)
- ❑ Other (11) ____________________

Display This Question:

If How satisfied are you with the... Noise level in your workspace - Very dissatisfied Is Selected

Or How satisfied are you with the... Noise level in your workspace - Dissatisfied Is Selected

Or How satisfied are you with the... Noise level in your workspace - Somewhat dissatisfied Is Selected

Or How satisfied are you with the... Sound privacy in your workspace (ability to have communications without your neighbors overhearing and vice versa) - Very dissatisfied Is Selected

Or How satisfied are you with the... Sound privacy in your workspace (ability to have communications without your neighbors overhearing and vice versa) - Dissatisfied Is Selected

Or How satisfied are you with the... Sound privacy in your workspace (ability to have communications without your neighbors overhearing and vice versa) - Somewhat dissatisfied Is Selected

Please describe any other issues related to acoustics hat are important to you.

____

Cleanliness and Maintenance

How satisfied are you with...

	Very satisfied (1)	Satisfied (2)	Somewhat satisfied (3)	Neither satisfied nor dissatisfied (4)	Somewhat dissatisfied (5)	Dissatisfied (6)	Very dissatisfied (7)
General cleanliness of the overall building (1)	❍	❍	❍	❍	❍	❍	❍
Cleaning service provided for your workspace (2)	❍	❍	❍	❍	❍	❍	❍
General maintenance of the building (3)	❍	❍	❍	❍	❍	❍	❍

Does the cleanliness and maintenance of this building enhance or interfere with your ability to get your job done?

	Significantly enhances (1)	Enhances (2)	Somewhat enhances (3)	Neither enhances nor interferes (4)	Somewhat interferes (5)	Interferes (6)	Significantly interferes (7)
(1)	❍	❍	❍	❍	❍	❍	❍

Display This Question:

If How satisfied are you with... General cleanliness of the overall building - Very dissatisfied Is Selected
Or How satisfied are you with... General cleanliness of the overall building - Dissatisfied Is Selected
Or How satisfied are you with... General cleanliness of the overall building - Somewhat dissatisfied Is Selected
Or How satisfied are you with... Cleaning service provided for your workspace - Very dissatisfied Is Selected
Or How satisfied are you with... Cleaning service provided for your workspace - Dissatisfied Is Selected
Or How satisfied are you with... Cleaning service provided for your workspace - Somewhat dissatisfied Is Selected
Or How satisfied are you with... General maintenance of the building - Very dissatisfied Is Selected
Or How satisfied are you with... General maintenance of the building - Dissatisfied Is Selected
Or How satisfied are you with... General maintenance of the building - Somewhat dissatisfied Is Selected

You have told us that you are dissatisfied with the cleaning service provided for your workspace. How often do you have significant problems?

- ❍ Always (1)
- ❍ Often (2)
- ❍ Sometimes (3)
- ❍ Rarely (4)
- ❍ Never (5)
- ❍ Don't know/ no opinion (6)

Display This Question:

If How satisfied are you with... General cleanliness of the overall building - Very dissatisfied Is Selected
Or How satisfied are you with... General cleanliness of the overall building - Dissatisfied Is Selected
Or How satisfied are you with... General cleanliness of the overall building - Somewhat dissatisfied Is Selected
Or How satisfied are you with... Cleaning service provided for your workspace - Very dissatisfied Is Selected
Or How satisfied are you with... Cleaning service provided for your workspace - Dissatisfied Is Selected
Or How satisfied are you with... Cleaning service provided for your workspace - Somewhat dissatisfied Is Selected
Or How satisfied are you with... General maintenance of the building - Very dissatisfied Is Selected
Or How satisfied are you with... General maintenance of the building - Dissatisfied Is Selected
Or How satisfied are you with... General maintenance of the building - Somewhat dissatisfied Is Selected

Which of the following contribute to this dissatisfaction? (Check all that apply)

- ❑ Sureface dust on work surfaces close to you (1)
- ❑ Surface dust on other surfaces you might touch (2)
- ❑ Surface dust on surfaces difficult to reach (3)
- ❑ Spills and debris (4)
- ❑ Dirty floors (5)
- ❑ Trash cans are not emptied overnight (6)
- ❑ Trash cans get too full during the day (7)
- ❑ Trash cans are a significant source of odor (8)
- ❑ Other (9) ____________________

Display This Question:

If How satisfied are you with... General cleanliness of the overall building - Very dissatisfied Is Selected
Or How satisfied are you with... General cleanliness of the overall building - Dissatisfied Is Selected
Or How satisfied are you with... General cleanliness of the overall building - Somewhat dissatisfied Is Selected
Or How satisfied are you with... Cleaning service provided for your workspace - Very dissatisfied Is Selected
Or How satisfied are you with... Cleaning service provided for your workspace - Dissatisfied Is Selected
Or How satisfied are you with... Cleaning service provided for your workspace - Somewhat dissatisfied Is Selected
Or How satisfied are you with... General maintenance of the building - Very dissatisfied Is Selected
Or How satisfied are you with... General maintenance of the building - Dissatisfied Is Selected
Or How satisfied are you with... General maintenance of the building - Somewhat dissatisfied Is Selected

Q50 Please describe any other issues related to cleaning and maintenance that are important to you.

Building Features

For each of the building features listed below, please indicate how satisfied you are with the effectiveness of that feature

	Very satisfied (1)	Satisfied (2)	Somewhat Satisfied (3)	Neither satisfied nor dissatisfied (4)	Somewhat Dissatisfied (5)	Dissatisfied (6)	Very Dissatisfied (7)
Thermostats (1)	❍	❍	❍	❍	❍	❍	❍
Automatic daylight controls (2)	❍	❍	❍	❍	❍	❍	❍
Window blinds (3)	❍	❍	❍	❍	❍	❍	❍
Adjustable furniture (4)	❍	❍	❍	❍	❍	❍	❍

How well informed do you feel about using the above mentioned features in this building?

	Very well informed (1)	Informed (2)	Somewhat informed (3)	Neither informed nor uninformed (4)	Somewhat uninformed (5)	Uninformed (6)	Very uninformed (7)
(1)	❍	❍	❍	❍	❍	❍	❍

Please describe any other issues related to the design and operation of the above mentioned features that are important to you.

General Comments

All things considered, how satisfied are you with your personal workspace?

	Very satisfied (1)	Satisfied (2)	Somewhat satisfied (3)	Neither satisfied nor dissatisfied (4)	Somewhat dissatisfied (5)	Dissatisfied (6)	Very dissatisfied (7)
(1)	❍	❍	❍	❍	❍	❍	❍

Please estimate how your productivity is increased or decreased by the environmental conditions in the building (e.g., thermal, lighting, acoustics, cleanliness)

	Increased 20% (1)	Increased 10% (2)	Increased 5% (3)	Neither increased nor decreased (4)	Decreased 5% (5)	Decreased 10% (6)	Decreased 20% (7)
(1)	❍	❍	❍	❍	❍	❍	❍

How satisfied are you with the building overall?

	Very satisfied (1)	Satisfied (2)	Somewhat satisfied (3)	Neither satisfied nor dissatisfied (4)	Somewhat dissatisfied (5)	Dissatisfied (6)	Very dissatisfied (7)
(1)	❍	❍	❍	❍	❍	❍	❍

Any additional comments or recommendations about your personal workspace or building overall?

Thank you for participation! Please click "submit" to complete this survey. Upon submission you will be redirected to another page where you can have the option to be entered into a raffle to win a $50 Amazon gift card.

End of Block

11.2 Building Characteristics Form

Name of the survey: Building Characteristics Master File

Default Question Block

Insert building name ____

General Information

	(1)
Building name (1)	
Address (2)	
City (3)	
State/Province (4)	
Zip code (5)	
Country (6)	
Year building opened (7)	
Year of last major renovation (if applicable) (8)	
Owner (9)	
Number of occupants (10)	
Maximum occupancy (11)	

	(1)	(2)	(3)	(4)
Owner occupied? (1)	○ Yes (1)	○ Partially (2)	○ No (3)	○ - (4)
Tenancy (2)	○ Single tenant (1)	○ Multi-tenant (2)	○ - (3)	○ - (4)
Location description (3)	○ Central business (1)	○ Other urban (2)	○ Suburban (3)	○ Rural (4)

Design Team

`

	(1)
Primary architect (1)	
Mechanical engineer (2)	
Energy/sustainability consultant (3)	

General Building Features Please fill in the units you work with regularly

	Sq. meters (1)	Sq. feet (2)	Other comments (3)
Gross building area (non-parking) (1)			

	(1)
total # of floors in building (2)	
# of floors surveyed (with occupant survey) (3)	

Primary activity

- ○ Office (Professional services, Governmental offices, Non - profit organizations) (1)
- ○ Commercial (Retail, Pharmacies, TV & radio stations) (2)
- ○ Educational (Child care facilities, Schools, Colleges, Libraries, Museums) (4)
- ○ Public order and safety (Courthouses, Correctional facilities, Police stations, Fire stations) (3)
- ○ Health care (Hospitals, Medical offices, Medical laboratories, Rehabilitation facilities, Nursing homes) (5)
- ○ Public assembly (Religious bldgs, Rec & sports facilities, Performing arts & Entertainment venues) (6)
- ○ Laboratory (Research laboratories) (7)
- ○ Hospitality (Hotels, Motels, Restaurants, Commercial kitchens) (8)
- ○ Residential (Single family units, Condominiums, Apartments) (9)
- ○ Transportation (Parking garages, Airports & Terminals, Train and Bus stations) (10)
- ○ Industrial & storage (Factories, Warehouses, Agricultural buildings) (11)
- ○ Other (12)

Design Features

Building Certifications

- ▢ LEED (1)
- ▢ WELL (2)
- ▢ Energy Star Rating (12)
- ▢ Site EIU (if applicable) (13) ____
- ▢ BREAM (5)
- ▢ GreenMark (8)
- ▢ Livable Building Challenge (3)
- ▢ CHPS (4)
- ▢ Reach Code Program (9)
- ▢ Energy Efficiency Incentive Program (e.g., utility rebate) (7)
- ▢ Other (6) ____

Display This Question:
If Building Certifications LEED Is Selected

LEED version

- ○ choose one (1)
- ○ Pilot (2)
- ○ 1.0 (3)
- ○ 2.0 (4)
- ○ 2.1 (5)
- ○ 2.2 (6)
- ○ V3/ 2009 (7)
- ○ V4 (8)

Display This Question:
If Building Certifications LEED Is Selected

LEED Rating

- ○ choose one (1)
- ○ None (2)
- ○ Pending (3)
- ○ Certified (4)
- ○ Silver (5)
- ○ Gold (6)
- ○ Platinum (7)

Sources for Cooling
Note % refers to the percentage of the building that use this feature

	Not in this building (1)	1-25% (3)	26-50% (4)	51-75% (5)	76-100% (6)	I don't know (7)
Mechanical (e.g. air conditioning) (2)	○	○	○	○	○	○
Passive/natural cooling with mechanical cooling (i.e., "mixed-mode") (7)	○	○	○	○	○	○
Passive/natural (8)	○	○	○	○	○	○

HVAC Distribution Systems

Note % refers to the percentage of the building that use this feature

	Not in this building (1)	1-25% (3)	26-50% (4)	51-75% (5)	76-100% (6)	I don't know (7)
Variable Airflow Volume (VAV) (24)	○	○	○	○	○	○
Overhead air distribution system (38)	○	○	○	○	○	○
Underfloor air distribution (1)	○	○	○	○	○	○
Displacement ventilation (14)	○	○	○	○	○	○
Air conditioner (e.g., packaged split or window-air conditioner) (2)	○	○	○	○	○	○
Variable Refrigerant Flow (VRF) air conditioning (10)	○	○	○	○	○	○
Radiant heating or cooling (11)	○	○	○	○	○	○
Chilled beams (active or passive) (12)	○	○	○	○	○	○
Zone fan coils (13)	○	○	○	○	○	○
Dedicated outdoor air system (DOAS) (3)	○	○	○	○	○	○
Perimeter heating unit (4)	○	○	○	○	○	○
Ceiling fans (6)	○	○	○	○	○	○
Other space conditioning system (9)	○	○	○	○	○	○

Display This Question:
If HVAC Distribution Systems Overhead air distribution system - 0-25% Is Selected
Or HVAC Distribution Systems Overhead air distribution system - 26-50% Is Selected
Or HVAC Distribution Systems Overhead air distribution system - 51-75% Is Selected
Or HVAC Distribution Systems Overhead air distribution system - 76-100% Is Selected

Overhead air distribution system features (check all that apply)

	Yes (1)	No (2)	I don't know (3)
Overhead variable air volume multi-zone system (1)	○	○	○
Overhead variable air volume single zone units (2)	○	○	○
Overhead constant volume system (3)	○	○	○
Overhead thermally powered diffusers (4)	○	○	○

Display This Question:
If HVAC Distribution Systems Radiant heating or cooling - 0-25% Is Selected
Or HVAC Distribution Systems Radiant heating or cooling - 26-50% Is Selected
Or HVAC Distribution Systems Radiant heating or cooling - 51-75% Is Selected
Or HVAC Distribution Systems Radiant heating or cooling - 76-100% Is Selected

Radiant heating or cooling system features (check all that apply)

	Yes (1)	No (2)	I don't know (3)
Radiant heating (1)	○	○	○
Radiant cooling (2)	○	○	○
Radiant in-slab system (e.g., thermally activated building system (TABS), topping slab, embedded surface system (ESS) (6)	○	○	○
Radiant panels (e.g., metal panels) (4)	○	○	○

Display This Question:
If HVAC Distribution Systems Passive/natural cooling - 0-25% Is Selected
Or HVAC Distribution Systems Passive/natural cooling - 26-50% Is Selected
Or HVAC Distribution Systems Passive/natural cooling - 51-75% Is Selected
Or HVAC Distribution Systems Passive/natural cooling - 76-100% Is Selected

Passive/natural cooling system features (check all that apply)

	Yes (1)	No (2)	I don't know (3)
Manual openings (1)	○	○	○
Automated openings (2)	○	○	○
Passive and active cooling (e.g., mixed-mode) (3)	○	○	○
Other (4)	○	○	○

Windows and Shading Note % refers to the percentage of the window area of the building.

	Not in this building (1)	1-25% (2)	26-50% (3)	51-75% (4)	76-100% (5)	I don't know (6)
Automated operable windows (1)	○	○	○	○	○	○
Electrochromic glass (2)	○	○	○	○	○	○
High performance glass (e.g., tinted, low-e) (3)	○	○	○	○	○	○
Manually operable windows (5)	○	○	○	○	○	○
Automated exterior shading (11)	○	○	○	○	○	○
Automated interior blinds/shades (12)	○	○	○	○	○	○
Fixed exterior shading (13)	○	○	○	○	○	○
Manual exterior shading (14)	○	○	○	○	○	○
Manual interior blinds (15)	○	○	○	○	○	○

Estimation of window-to-wall ratio

- ○ 0-25% (1)
- ○ 26-50% (2)
- ○ 51-75% (3)
- ○ 76-100% (4)
- ○ I don't know (5)

HVAC controls Note % refers to the percentage of the building that use this feature

	Not in this building (1)	1-25% (2)	26-50% (4)	51-75% (5)	76-100% (6)	I don't know (3)
Demand controlled ventilation (1)	o	o	o	o	o	o
Energy management system / building monitoring system (2)	o	o	o	o	o	o
Temperature (write-in average # of workstations per control) (3)	o	o	o	o	o	o

Additional occupant controls Note % refers to the percentage of the building that use this feature

	Not in this building (1)	1-25% (2)	26-50% (3)	51-75% (4)	76-100% (5)	I don't know (6)
Manual light switch (9)	○	○	○	○	○	○
Occupant lighting controls (1)	○	○	○	○	○	○
Occupancy sensors (2)	○	○	○	○	○	○
Occupant temperature control (thermostat) (3)	○	○	○	○	○	○
Web/mobile device lighting control (6)	○	○	○	○	○	○
Web/mobile device temperature control (5)	○	○	○	○	○	○
Other (4)	○	○	○	○	○	○

Any additional information we should know about the building?

May we publish your building information?

- ○ Yes, by name (1)
- ○ Yes, but please make building name anonymous (3)
- ○ No (2)

Please include your name, phone number and email address so that we may contact you for clarification. **Your personal information will not be used for any other purpose.**

	(1)
Name (1)	
Phone number (2)	
Email (3)	

End of Block