Title

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Abstract
Unlike the case of air systems where the cooling load is purely convective, the cooling load for radiant systems consists of both convective and radiant components. The main objectives of this energy simulation study were to investigate whether the same design cooling load calculation methods can be used for radiant and air systems by studying the magnitude of the cooling load differences between radiant and air systems over a range of configurations and to suggest potential improvements in current design guidelines. Simulation results show that 1) zone level 24-hour total cooling energy of radiant systems can be 5-15% higher than air systems due to differences in conduction load through the building envelope; 2) peak cooling load at the radiant system hydronic level can be 7-31% higher than air system for zones without solar load. The differences can increase up to 93% at the hydronic level for floor system in zones with solar load; 3) the cooling load differences between the two systems originate from: a) radiant cooling surface(s) directly remove part of the radiant heat gain and reduce heat accumulation in the building mass; b) only part of the convective heat gain becomes instantaneous cooling load. This indicates that simplified methods such as Radiant Time Series Method is not appropriate for cooling load calculation in radiant system design. Radiant systems should be modeled using a dynamic simulation tool that is capable of capturing radiant heat transfer for cooling load calculation.

Keywords: Radiant cooling; Cooling load; Heat gain; Air system; Radiant cooling panel (RCP), Embedded surface cooling systems (ESCS), Thermally activated building systems (TABS).

1. Introduction

Water-based radiant cooling systems are gaining popularity as an energy efficient approach for conditioning buildings [1-3]. The design of radiant systems is complicated because of the coupling between thermal load, building structure and the hydronic system and because of the important impact of both radiation and convection on thermal comfort. Dedicated radiant system design and testing standards have been developed to address issues like system sizing, installation, operation and control [4-9]. However, radiant cooling systems are still considered as an innovative approach, and their application in North America is still limited [10, 11]. In this study, we investigated the impacts of the presence of activated cooled surface on zone cooling loads.

Cooling load calculations are a crucial step in designing any HVAC system. Compared to air systems, the presence of an actively cooled surface changes the heat transfer dynamics in the room, and two potential impacts on zone cooling loads studied here are: 1) cooled surfaces may create different inside surface temperatures of the non-active exterior building walls, causing different heat gain through the building envelope, and in turn different zone level total energy, and 2) changes the effect of thermal mass on cooling loads, and therefore creating different peak cooling load.

Two research studies were identified that looked at heating load calculations in terms of the impact of the radiant system on wall surface temperatures and the resultant room load [12, 13]. However, both studies focused on heating load calculation under steady-state conditions. In another study, Chen [14] suggested that the total heating load of a ceiling radiant heating system was 17% higher than that of the air heating system because of the role of thermal mass and higher heat loss through the building envelope due to slightly higher inside surface temperatures. For cooling applications, no studies were found on this topic, and in current radiant system design guidelines [4, 8], such impacts are not considered or evaluated.

Secondly, the interaction of building mass with heat source is influenced by the presence of activated radiant cooling surface(s). One phenomenon mentioned in the literature was radiant surface(s) as part of the building mass, instead of storing themal energy as in the case of air systems, removes radiant heat gain (e.g. solar, radiative internal load and radiative envelope load) that is directly impinging on it. This phonomenon fundamentally changes the cooling load dynamics in a room. Niu [15] pointed out that this direct radiation may create high peak cooling loads. He modified the thermal analysis program ACCURACY [16] to account for the direct radiant heat gain as instantaneous cooling load for radiant systems. However, no information can be found on how he implemented the modification and the software is not accessible for the public. In an effort to develop a new cooling load calculation approach for radiant systems, Corgnati [17] also tackled the direct radiant heat gain effect using a similar strategy to Niu. Based on Corgnati’s work, Causone et al. [18] focused on the cases with the presence of direct solar gain. However, the methods proposed in these research studies only looked at the effect of direct radiant heat gain on cooling load, and the rest of the radiant heat gain and the convective heat gain are still considered to interact with building mass as if the radiant system does not exist. In addition, no research can be found that fundamentally studies the differences of the heat transfer process in zones conditioned by an air and a radiant system, and how these differences are going to impact the cooling load calculation and what could be the magnitude of the differences.

Although research has demonstrated that cooling loads for radiant systems need to be considered differently than for air systems, current radiant design standards do not explicitly acknowledge these differences. Several standards and handbooks were reviewed, including: chapter 6 of *ASHRAE Equipment and HVAC systems* [19], radiant heating and cooling handbook (2002) [9], chapter 18 of *ASHRAE Fundamental* (2012), ISO 11855 (2012) [4], and European standard EN 15377 (2008) [8]. The first three do not offer any guidance on the selection of the calculation methods when radiant systems are involved. In chapter 18 of *ASHRAE Fundamental* (2012) handbook, the description of the cooling load calculation process is based on the implicit assumption that an air system is used for conditioning the space. Some simplified cooling load calculation methods, such as Transfer Function method (TF)[20] and Radiant Time Series method (RTS)[21], have also been developed for air system. These algorithms are widely implemented in building thermal simulation or load calculation tools, including HAP (TF), TRANE TRACE (RTS), BLAST, and DOE-2 (TF) based tools such as eQuest, Energy-pro, Green Building Studio and VisualDOE. These tools are often used for cooling load estimates during initial design stage and for detailed energy and comfort analysis even when radiant systems are involved [22]. The European standards reviewed indirecly reference EN 15255 [23] for cooling load calculation procedure. EN 15255 classified all cooling load calculation methods into different catogories according to their capability to model different types of cooling system and control method. Methods that are able to simulate radiant systems controlled by operative temperature are in Class 4b. This implies that cooling load calculation method for radiant systems should be properly distinguished from air systems. However, this standard does not explicitly provide cooling load calculation mehtods for radiant system.

A recent survey conducted by the authors of radiant cooling design practitioners revealed that the differences in cooling load between radiant and air systems are not fully understood. Some of the most experienced professionals acknowledge the complications and lack of guidance in the standards and developed rule-of-thumb methods for initial system design calculation. Among those methods, either heat gain is directly used as cooling load for system sizing [24, 25], or a portion of the heat gain is considered as direct heat removal by the active radiant surface. The percentages of the direct removal depend on load type (lighting/people/equipment), and are obtained based on experience [26]. In design practice, it is not often that dynamic simulation tools that can properly model radiation heat transfer are used at the cooling load estimation stage. Radiant system manufacturers have developed some tools for system sizing [27], but they are mainly used for heating applications, where steady-state heat transfer is adequate to capture the thermal behavior.

The objectives of this simulation study are to 1) assess the cooling load differences between the two systems by comparing the zone level peak zone cooling load and 24-hour total cooling energy for a radiant cooling system (with activated chilled surface) vs. an air system; and 2) suggest potential improvements in current design guidelines for radiant cooling system.

2. Background and theory

In this section, we give a brief introduction to the three types of radiant cooling systems investigated in this paper and explain how their thermal characteristics affect the design approach. Since radiant and air systems are different in many ways, the simulation study had to be designed carefully to provide a fair comparison.

2.1. Radiant cooling systems

The REHVA guidebook on radiant systems [7] has roughly categorized these systems into three types: radiant cooling panels (RCP), water-based embedded surface cooling systems (ESCS), and thermally activated building systems (TABS). As shown in Figure 1, RCP are metal panels with integrated pipes usually suspended under the ceiling with heat carrier temperature relatively close to room temperature. ESCS have pipes embedded in plaster or gypsum board or cement screed, and they are thermally decoupled from the main building structure (floor, wall and ceiling) by the use of thermal insulation. They are used in all types of buildings and work with heat carriers at relatively high temperatures for cooling. Finally, “systems with pipes embedded in the building structure (slab, walls), TABS, which are operated at heat carrier temperatures very close to room temperature and take advantage of the thermal storage capacity of the building structure.” These systems usually have different applications due to their thermal and control characteristics, and therefore, the design and dimensioning strategies for these systems vary.

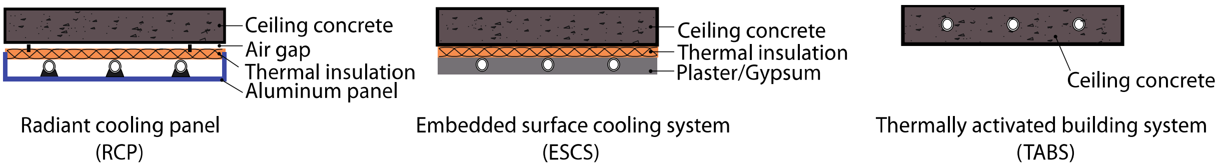

Figure 1: Schematic of the three types of radiant surface ceiling systems (not to scale)

2.2. Radiant vs. air systems

A comparison between radiant and air systems is challenging. In this section, we discuss the differences between the two systems that dictate the modeling approach used in this study. Besides those mentioned in the literature [28], the main difficulties include:

- Types of load (sensible/latent) and the expected amount of load to be handled by the two systems are different. Air systems are usually designed to be the only system to handle both latent and sensible loads, while radiant systems must operate in hybrid mode with a reduced-sized air system (for ventilation and latent loads). Radiant cooling systems are always sized to handle a portion (as much as possible) of the sensible-only cooling load. To address this issue, neither the latent load nor ventilation system was simulated. This was to simplify our analysis.
- The design cooling load concept is different for the two systems. According to ASHRAE Handbook [29], the sensible cooling load for an air system is calculated in terms of

maintaining a constant zone air temperature, while radiant systems, particularly TABS, are not capable of maintaining a constant zone air temperature due to large thermal inertia of the active surfaces. For this reason, in this comparison study, we sized and controlled the simulated radiant systems to maintain an acceptable thermal comfort range during the simulation period. Operative temperature was used as the control temperature for both systems [28, 30]. To ensure equivalent comfort conditions between the two systems for fair comparison, all simulations of the air system were subsequently controlled to closely track the hourly operative temperature profile derived from the radiant system simulation for the identical input conditions.

- For an air system the zone cooling load is equal to the heat extraction rate by the mechanical system when the room air temperature and humidity are constant. But this is not always the case in a radiant system. Other than panel systems, radiant cooling systems (ESCS and TABS) are integrated with the building structure with hydronic pipes embedded in the mass. As a result, heat removed from the zone at the chilled surface can be quite different from the heat removed by the hydronic loop. Sizing of the radiant system cooling equipment is highly dependent on specifications of the cooling surface (slab material/thickness, tube spacing, and surface finishing). This indicated that we needed to investigate heat transfer of the radiant system at both the surface and hydronic levels, which is discussed in detail below.

2.3. Heat transfer at radiant surface and hydronic level

Radiant systems remove the sensible heat in a room at the cooling surface. We define this cooling rate as surface cooling rate. Define the control volume as the inside face of the cooling slab, with positive sign means heat being transferred into the control volume and negative indicates heat leaving the control volume, the heat balance for the cooling surface can be written as follows (1) [31]:

$$q^{"}_{surf} = q^{"}_{surf,conv} + q^{"}_{surf,rad} = -q^{"}_{surf,cond} \qquad (1)$$

Surface cooling rate serves as one key design parameter for determining required radiant system area and selection of system type.

Hydronic cooling rate is the heat extraction rate based on an energy balance on the hydronic circuit. The hydronic cooling rate is important for sizing of waterside equipment, such as pumps, chillers and cooling tower. Hydronic cooling rate can be calculated by equation (2) [31]:

$$q^{"}_{hyd} = \left(\dot{m}c_p\right)_{water}(T_{wi} - T_{wo}) \qquad (2)$$

Both RCP and most ESCS operate during occupied hours to maintain a relatively constant comfort condition in the space, so the difference between the surface and hydronic rate is only a function of thermal properties of the panel/slab. For RCP systems, if insulation is installed on the backside of the panel, hydronic cooling rate can be assumed to be the same as surface cooling output due to high conductivity of the surface material [6], which is usually desired. TABS are usually designed and operated to take advantage of the thermal storage effect of the slab, so the difference between the surface and hydronic rate is also a function of the operational strategies, which will be discussed later.

3. Methodology and modelling approach

To investigate the impacts of the presence of activated cooled surface on zone cooling load, we adopted the following methodology:

- Two single zone models, one conditioned by an air system and one by radiant system were developed in EnergyPlus v7.1 for comparison. All three radiant systems (RCP/ESCS/TABS) were studied. Because the construction of each radiant system type is different and is highly influential on overall building response, the comparison air models were configured to match the construction of the radiant systems.
- The models were parameterized for studying the influences of envelope thermal insulation, thermal mass, type of internal gain, solar heat gain with different shading options, and radiant surface orientation (ceiling, floor).

EnergyPlus v7.1 was used for the simulation study because it performs a fundamental heat balance on all surfaces in the zone. The heat balance model ensures that all energy flows in each zone are balanced and involve the solution of a set of energy balance equations for zone air and the interior and exterior surfaces of each wall, roof, and floor. It captures both longwave and shortwave radiation heat transfer and has been extensively validated [32, 33]. In addition, EnergyPlus is able to integrate the heat transfer calculation in the radiant cooling systems with changing zone conditions; therefore it is able to capture the transient behavior of the systems [31].

3.1. Simulation Runs

In total, seventy-four simulation cases were configured, including 13 (11 for RCP) variations for the three types of radiant systems and their equivalent air systems. The different combinations and ranges of parameters are listed in Table 1.

Table 1: Simulation runs summary

Group	**Case**	**Building**	**Int. heat gain[1]**	**Window**	**Radiant surface**	**Boundary conditions[3]**
G1: insulation	hw_r2	heavyweight	no	no	ceiling	environment
	hw_r1	hW_smallR[2]	no	no	ceiling	environment
G2: thermal mass	hw_r2	heavyweight	no	no	ceiling	environment
	lw_r2	lightweight	no	no	ceiling	environment
G3: Int. heat gain[1]	rad0	heavyweight	RadFrac[1] = 0	no	ceiling	adiabatic
	rad0.3	heavyweight	RadFrac=0.3	no	ceiling	adiabatic
	rad0.6	heavyweight	RadFrac=0.6	no	ceiling	adiabatic
	rad1	heavyweight	RadFrac=1	no	ceiling	adiabatic
G4: ceiling with solar	cl_ noshade	heavyweight	no	yes	ceiling	environment
	cl_shade	heavyweight	no	yes+shade	ceiling	environment
G5: floor with solar[4]	flr_noshade	heavyweight	no	yes	floor	environment
	flr_shade	heavyweight	no	yes+shade	floor	environment
G6: typical ceiling	cl_shade_rad 0.6	heavyweight	RadFrac = 0.6	yes+shade	ceiling	environment

Note:

1. Int. heat gain= Internal heat gain; RadFrac = Radiative fraction of internal heat gain;
2. HW_smallR=Heavy weight construction with half thermal insulation at exterior walls;
3. Both roof and floor have boundary conditions set to adiabatic for simplicity, and the boundary conditions specified in this column are for exterior walls;
4. These cases are not simulated for radiant panel systems.

Cases hw_r2 and hw_r1 in Group 1 are designed for studies of the impact of thermal insulation, and hw_r2 and lw_r1 in Group 2 are for studies of thermal mass. These represent perimeter zones without windows, only subjected to building envelope conductive heat gains. Cases in G3, rad0 to rad1, are to evaluate the impacts of internal load with different radiant fractions, defined as the portion of radiative heat gain to total heat gain given off by a heat source. Radiant fraction of lighting ranges from 0.48 to 1.0 depending on luminaire type [34]; for people the radiant fraction can be from 0.2 to 0.6 depending on surrounding air velocity and people's activity (e.g., walking, running, etc.) [29]; and for office equipment, the range is usually between 0.1 to 0.4 depending on equipment type [35]. For these cases, the building envelope was set to be adiabatic to represent an interior zone and isolate the influences from outside environment. Two windows were modelled on the south wall in the next groups, G4-G6, in order to study the impact of solar gains in perimeter zones. Radiant ceiling and floor systems were both simulated. Case cl_shade_rad0.6 was configured to represent a zone with real internal load and windows with exterior shading that is conditioned by a radiant ceiling system. All three types of radiant systems were modelled for all cases, except that the RCP systems were not simulated for the radiant floor case because it is not a common practice.

3.2. Model Specifications

Since the objective of the study was to understand the heat transfer and the resultant cooling load differences between a radiant and an air system, a representative single zone model is adequate. The model was developed primarily based on ASHRAE Standard 140 [36]. Weather file provided in the standard was used. System and design parameters for the radiant system were adopted fromRADTEST [37]. Additional details are summarized below.

The test case (Figure 2) was a rectangular, heavy weight construction single zone building (8 m wide × 6 m long × 2.7 m high) with no interior partitions. Both the floor and roof boundary conditions were set to be adiabatic to simplify the analysis. Only cases in G4-G6 have 12 m^2 of south-facing windows. The overall U-Factor was 2.721 W/(m^2·K) with Glass SHGC at 0.788. The baseline construction was based on case 900 (ASHRAE 140 2007 Table 11), except that the ceiling/floor constructions were modified so that radiant ceiling/floor systems can be simulated. Exterior walls for Case hw_r2 had U-value of 0.454 W/(m^2·K). Case hw_r1 was modified to have U-value of 0.83 W/(m^2·K), and Case lw_r2 was modified with lightweight construction based on case 600 (ASHRAE 140 2007 Table 1). Floor and ceilings were configured separately for each case depending on location of the activated cooling surface and radiant system types. Table 2 is a summary of the radiant ceiling/floor construction specifications. For cases in G3, the internal gain was 720 W from 6:00 to 18:00. The radiant fraction was different for each run as specified in Table 1. There was zero air infiltration for all runs because we did not want to have an additional confounding factor. Table 3 lists the radiant system design specifications that are developed based on RADTEST case 2800. When windows were simulated, tube spacing changed from 0.3 m to 0.15 m in order to maintain similar thermal comfort level. Design flow rates for RCP were reduced for cases in Group 1 and 2, since these systems have higher cooling capacity as compared to the other two radiant systems. As for control, the goal was to maintain operative temperature setpoint at 23°C for 24 hours with a 2°C deadband [31]. For the air system models, the EnergyPlus object "IdealLoadsAirSystem" was used for simplicity to ensure the same operative temperature as the corresponding radiant systems.

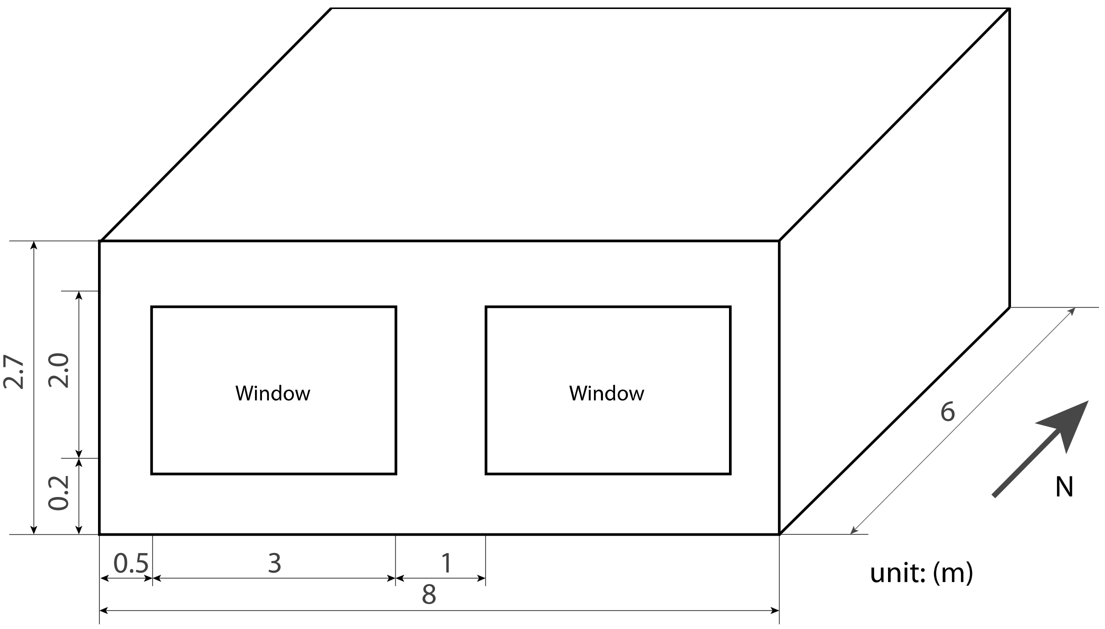

Figure 2: Isometric Base Case (Only G4-G6 have windows)

Table 2: Radiant surface constructions specifications (inside to outside)

	Thickness (m)	Specific Heat (J/kg·K)	Density (kg/m^3)	Conductivity (W/m·K)
		RCP ceiling		
Aluminum panel	0.001	910	2800	273.0
		Water Tube		
Insulation	0.05	1210	56	0.02
Concrete slab	0.08	1000	1400	1.13
Insulations	0.1118	840	12	0.04
Roof deck	0.019	900	530	0.14
		ECS ceiling		
Lime plaster	0.012	840	1050	0.7
		Water Tube		
Lime plaster	0.014	840	1050	0.7
Insulation	0.05	1210	56	0.02
Concrete	0.08	1000	1400	1.13
Insulations	0.1118	840	12	0.04
Roof deck	0.019	900	530	0.14
		ECS floor		
Floor finish	0.0016	1250	1922	0.17
Cement Screed	0.04	988	1842	1.2
		Water Tube		
Cement Screed	0.01	988	1842	1.2
Insulation	0.05	1210	56	0.02
Concrete	0.08	1000	1400	1.13
Insulation	1.007	n/a	n/a	0.04
		TABS ceiling		
Concrete	0.04	1000	1400	1.13
		Water Tube		
Concrete	0.04	1000	1400	1.13
Insulations	0.1118	840	12	0.04
Roof deck	0.019	900	530	0.14
		TABS floor		
Concrete	0.04	1000	1400	1.13
Water Tube				
Concrete	0.04	1000	1400	1.13
Insulations	1.007	n/a	n/a	0.04

Table 3: Hydronic loop specifications

Inner diameter (m)	0.015
Total pipe length (m)	139.2
Inlet water temp (°C)	15
Tube spacing (m)	0.3 (0.15 for cases with windows)
Design mass flow rate (kg/s)	0.167 (0.06 for RCP system in cases without window)

4. Parameters investigated

Table 4 lists the parameters that were evaluated during the simulations. Peak cooling rate is commonly used for equipment sizing in the case of air system and the fast responsive RCP and lightweight ESCS. 24-hour total cooling energy is studied for all radiant systems because it reflects the consequence of the impact of radiant cooling system on exterior wall surface temperature. Comparisons were made at both the surface and hydronic levels for the radiant systems. Percentage differences between the radiant and air systems were reported, and are defined in the last two rows in Table 4.

Table 4: Parameters analysed

	24 hour-total cooling energy	Peak cooling rate
Air system	24-hour total sensible cooling energy, *kJ/m²* ($\dot{q}_{air,tot}$)	Specific peak sensible cooling rate, *(W/m²)*))($q^{"}_{air,\mathrm{pk}}$)
Radiant system	24-hour total surface cooling energy, *kJ/m²* ($\dot{q}_{surf,tot}$)	Specific peak surface cooling rate, *W/m²* ($q^{"}_{surf,pk}$)
	24-hour total hydronic cooling energy, *kJ/m²* ($\dot{q}_{hyd,tot}$)	Specific peak hydronic cooling rate, *(W/m²)* ($q^{"}_{hyd,pk}$)
Percentage difference	$P_{surf\ ,tot}\ = \frac{(\dot{q}_{surf\ ,tot}\ - \dot{q}_{air\ ,tot}\ )}{\dot{q}_{air\ ,tot}} \times 100\ \%$	$P_{surf,pk}\ = \frac{(q^{"}_{surf,pk} - q^{"}_{air,pk})}{q^{"}_{air,pk}} \times 100\ \%$
	$P_{hyd\ ,tot}\ = \frac{(\dot{q}_{hyd\ ,tot}\ - \dot{q}_{air\ ,tot}\ )}{\dot{q}_{air\ ,tot}}\ \times 100\ \%$	$P_{hyd,pk}\ = \frac{(q^{"}_{hyd,pk} - q^{"}_{air,pk})}{q^{"}_{air,pk}} \times 100\ \%$

5. Results

Results from the 99.6% cooling design day simulations are reported and compared for surface cooling rate, hydronic cooling rate and air system cooling rate in this section. To evaluate the influence of each investigated parameter, the ranges of the $P_{surf,pk}$, $P_{hyd,pk}$, $P_{surf,tot}$, and $P_{hyd,tot}$ are reported graphically.

5.1. 24-hour total cooling energy

The expected impact of the radiant cooling system is to cause lower surface temperatures at the inside of building envelope, resulting in higher envelope heat gain and total cooling energy. This hypothesis was tested by a comparison of the 24-hour total envelope heat gain for a zone conditioned by a radiant vs. air system, as shown in Table 5. For cases in G1 and G2, the heat gains were merely heat conduction through exterior walls, and for the other cases, the heat gains also included solar radiation through windows. G3 cases were not reported because they were modeled to have adiabatic boundary conditions for all exterior surfaces that resulted in near zero heat gain through the building envelope. Table 5 shows higher conductive heat transfer through the building envelope for the radiant system. The reason for this finding was the lower surface temperature (at an average of 0.5°C) at the inside face of the exterior walls caused by the radiant system, as is proved by Figure 3. Table 5 presents the summer design day 24-hour total cooling energy for both radiant and air systems. Comparing heat gain differences between the two systems reported in Table 5 and the 24-hour total energy differences reported in Table 5, we can confirm that heat gain through the building envelope caused higher 24-hour total cooling energy for the radiant systems.

Figure 4 plots the range of $P_{surf,tot}$ (left) and $P_{hyd,tot}$ (right) for each group investigated for RCP, ESCS, and TABS. For example, in the left plot, the first black bar in “G1: insulation” represents the range of $P_{surf,tot}$ for cases in the first group, with the lower end representing $P_{surf,tot}$ for case hw_r2, and the high end representing $P_{surf,tot}$ for case hw_r1. $P_{surf,tot}$ and $P_{hyd,tot}$ are defined in Table 4, and can be calculated using data from table 5. . Note that since there is only one case in G6 for each type of radiant system, the single lines represent $P_{surf,tot}$ for the cases cl_shade_rad0.6.

From Figure 4, we can see that the differences in surface/hydronic level 24-hour total energy between the two conditioning systems were influenced by the thermal insulation in exterior walls but only slightly influenced by thermal mass of the building. Compared to Case hw_r2, Case hw_r1 had half the thermal insulation in exterior walls and the percentage difference in hydronic total cooling energy increased from 6% to 8% for the RCPs, 7-9% for ESCS, and 8-10% for the TABS; similar ranges were seen at the surface level. G3 cases have adiabatic boundary conditions, and therefore, have negligible differences in total cooling energy. For G4 and G5, the total surface energy was 6-14% higher, and hydronic energy was 6-15% higher. The difference in total energy was not sensitive to the type of radiant surface (ceiling or floor), but was sensitive to the amount of direct solar radiation. When exterior shadings were modeled, $P_{surf,tot}$ and $P_{hyd,tot}$ decreased about 5%. This means higher window surface temperature (caused by direct solar) enhanced the radiation heat transfer between the window surfaces and radiant cooling surface, and resulted in larger heat gain through the window for radiant system.

The three types of radiant systems displayed similar trends. For RCP systems, zone hydronic level cooling energy was almost the same as surface level, while for the ESCS and TABS, hydronic level energy was always slightly higher than surface level total energy. The difference was the energy used to cool the mass of the slab itself.

Table 5: Comparison of 24-hour total heat gain through building envelope

Group	Cases	RCP (kJ/m^2)	Air (kJ/m^2)	% diff (%)	ESCS (kJ/m^2)	Air (kJ/m^2)	% diff (%)	TABS (kJ/m^2)	Air (kJ/m^2)	% diff (%)
G1	hw_r2	391	368	6.2	401	376	6.6	403	377	6.9
	hw_r1	630	582	8.2	651	600	8.5	652	600	8.8
G2	hw_r2	391	368	6.2	401	376	6.6	403	377	6.9
	lw_r2	440	424	3.9	443	425	4.3	445	422	5.5
G4	cl_ noshade	1,956	1,735	12.7	1,898	1,678	13.1	1,902	1,679	13.3
	cl_shade	1,245	1,155	7.8	1,226	1,137	7.8	1,230	1,139	8.0
G5	flr_noshade	NA	NA	NA	1,946	1,710	13.8	1,909	1,674	14.0
	flr_shade	NA	NA	NA	1,249	1,147	8.9	1,239	1,137	9.0
G6	cl_shade_rad0.6	1,244	1,132	9.9	1,195	1,086	10.1	1,200	1,088	10.3

Note: Group 2 cases have adiabatic boundary conditions, therefore, no heat transmission through building envelope

Table 6: 24-hour total cooling energy comparison for summer design day

Group	Cases	RCP vs. Air (kJ/m^2)			ESCS vs. Air (kJ/m^2)			TABS vs. Air (kJ/m^2)		
		$\dot{q}_{surf,tot}$	$\dot{q}_{hyd,tot}$	$\dot{q}_{air,tot}$	$\dot{q}_{surf,tot}$	$\dot{q}_{hyd,tot}$	$\dot{q}_{air,tot}$	$\dot{q}_{surf,tot}$	$\dot{q}_{hyd,tot}$	$\dot{q}_{air,tot}$
G1	hw_r2	391	391	368	401	403	376	403	406	377
	hw_r1	630	630	582	651	654	600	654	659	600
G2	hw_r2	391	391	368	401	403	376	403	406	377
	lw_r2	441	441	421	444	445	419	446	445	420
G3	rad0	647	646	647	644	636	647	647	642	649
	rad0.3	648	647	647	650	647	647	648	647	646
	rad0.6	648	649	648	648	651	648	646	647	648
	rad1	648	648	648	648	652	649	648	656	648
G4	cl_ noshade	1,949	1,948	1,730	1,892	1,903	1,676	1,897	1,920	1,679
	cl_shade	1,236	1,234	1,153	1,221	1,229	1,136	1,226	1,244	1,143
G5	flr_noshade	NA	NA	NA	1,936	1,954	1,699	1,899	1,906	1,674
	flr_shade	NA	NA	NA	1,244	1,259	1,140	1,234	1,241	1,141
G6	cl_shade_rad0.6	1,861	1,858	1,754	1,816	1,827	1,717	1,823	1,848	1,722

In general, even if the total zone level cooling energy may be 5-15% higher for radiant systems compared to air systems, there are many potential advantages of using hydronic-based radiant systems such as, improved plant-side equipment efficiency with warmer chilled water temperatures [38], possibility of nighttime pre-cooling to reduce peak demand [39], utilization of natural cooling resources, and energy efficiency in transporting energy with water compared to air [3]. The combination of all these factors has the potential to produce lower energy consumption for radiant cooling vs. air systems.

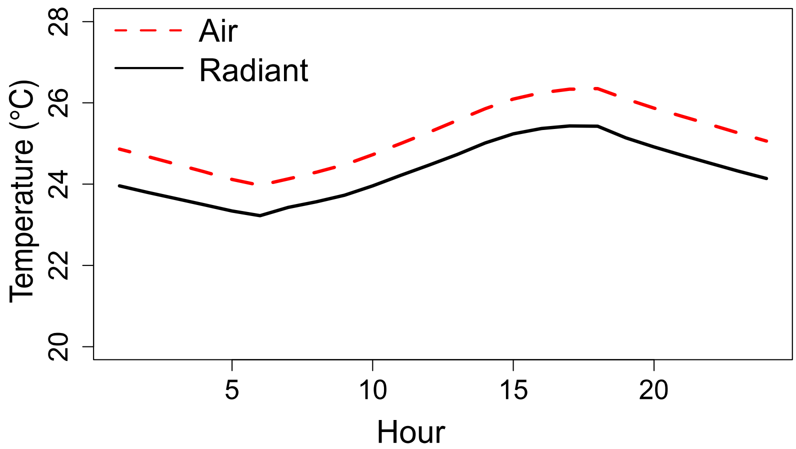

Figure 3: Comparison of temperatures at the inside surface of exterior wall between radiant and air systems. (G6 typical ceiling: cl_shade_rad0.6)

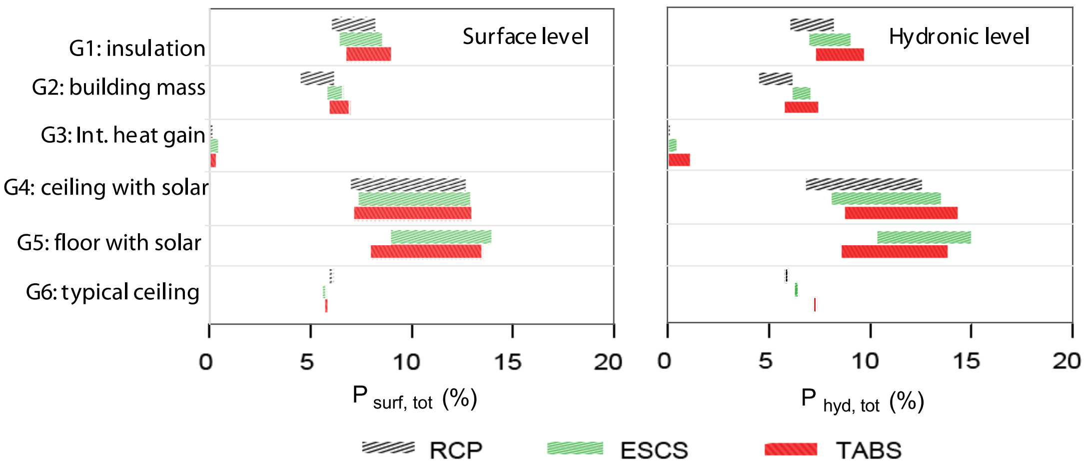

Figure 4: Range of 24-hour total energy percentage difference between air system and radiant system at surface level (left) and hydronic level (right)

5.2. Peak cooling rate

Figure 5 gives an example (G6: typical ceiling) of the cooling rate profiles for the radiant systems and their equivalent air systems. It can be seen that radiant system cooling rate profiles were different from the case of an air system. In general, a large portion of the heat was removed during the occupied period for the radiant case, and the radiant systems peak cooling rates were higher than the air system. Table 6 reports the values of the specific peak cooling rate for the radiant (both hydronic and surface) and the air systems.

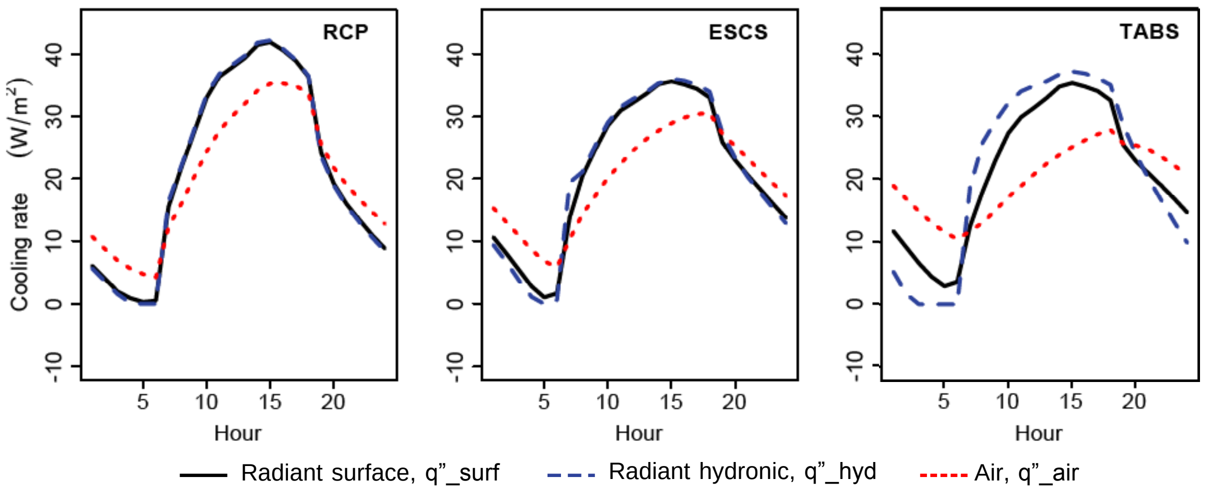

Figure 5: Comparison of design day cooling rate profiles between radiant and air systems. (G6 typical ceiling: cl_shade_rad0.6)

Table 7: Peak cooling rate comparison for summer design day

Group	Cases	RCP vs. Air (*W/m²*)			ESCS vs. Air (*W/m²*)			TABS vs. Air (*W/m²*)		
		$q''_{surf,pk}$	$q''_{hyd,pk}$	$q''_{air,pk}$	$q''_{surf,pk}$	$q''_{hyd,pk}$	$q''_{air,pk}$	$q''_{surf,pk}$	$q''_{hyd,pk}$	$q''_{air,pk}$
G1	hw_r2	7.7	7.8	6.2	8.5	8.7	6.7	8.5	9.7	6.3
	hw_r1	12.9	13.0	10.4	13.9	14.2	11.0	13.6	15.1	10.1
G2	hw_r2	7.7	7.8	6.2	8.5	8.7	6.7	8.5	9.7	6.3
	lw_r2	14.1	14.0	12.6	14.4	14.6	12.4	14.4	16.6	11.0
G3	rad0	14.5	14.5	13.6	14.6	14.7	13.6	14.0	15.0	12.8
	rad0.3	13.9	13.9	12.6	14.5	14.6	12.7	14.0	15.1	11.4
	rad0.6	13.2	13.2	11.7	13.8	13.9	12.0	13.8	14.9	11.2
	rad1	12.5	12.6	10.9	13.1	13.3	11.3	13.0	13.7	10.3
G4	cl_ noshade	51.7	52.2	37.9	39.8	39.5	29.4	39.9	40.6	26.8
	cl_shade	29.4	29.7	23.5	26.0	26.7	21.0	25.6	29.0	19.3
G5	flr_noshade	NA	NA	NA	54.6	62.1	32.2	48.4	44.7	26.2
	flr_shade	NA	NA	NA	28.8	33.0	20.6	25.1	30.7	18.3
G6	cl_shade_rad0.6	41.9	42.2	35.2	35.6	36.0	30.6	35.5	37.3	28.0

Figure 6 plots the ranges of $P_{surf,pk}$ and $P_{hyd,pk}$ for RCP, ESCS, and TABS. Results show that the radiant system peak surface/hydronic cooling rates exceed that of the air system by a wide range depending on radiant system type and zone load conditions.

- For cases in G1 and G2, representing perimeter zones that are only subjected to building envelope load, $P_{surf,pk}$ ranged from 12-25% for the RCPs, and 16 - 27% for the ESCS. For RCP and ESCS, $P_{hyd,pk}$ was in a similar range as $P_{surf,,pk}$,While little variation in both $P_{surf,pk}$ and $P_{hyd,pk}$ can be noted for changes in thermal insulation conditions, reduction of thermal mass resulted in much less peak load differences between the radiant and air systems.
- For G3, the total internal load was the same for all cases but with different radiant and convective splits for each case. The peak cooling rate differences ranged from 7- 27% at the surface level and from 7- 33% at the hydronic level. Higher radiant fraction in heat gain produces larger differences in peak loads between the two systems at the surface level. This was further demonstrated in G4-G6.
- For G4, solar gain contributed to a pronounced increase in the radiation heat transfer at the radiant surface(s). When exterior shading was not modeled, RCP ceiling surface peak cooling

rate is 36% higher than the air system, and for ESCS ceiling system it is 35%. When exterior shading was modeled, the transmitted solar gain was mostly diffuse allowing it to be evenly distributed among all surfaces. Exterior shading reduced the direct solar impact, but the surface peak cooling rates were still 24-33% higher for the ceiling system.

- When the floor was used as the radiant cooling surface and when it was illuminated by direct solar, both $P_{surf,pk}$ and $P_{hyd,pk}$ increased dramatically compared to the ceiling cases. The ESCS surface peak cooling rate was 69% higher and for TABS it was 85% higher. Exterior shadings greatly reduced the absolute values of the peak load in all systems and the difference between radiant and air systems at the surface level for both radiant systems.

While the high peak-cooling rate shown maybe regarded as an enhancement of cooling capacity of the radiant cooling system [40, 41], the sizing of the associated waterside equipment must take this increase into account.

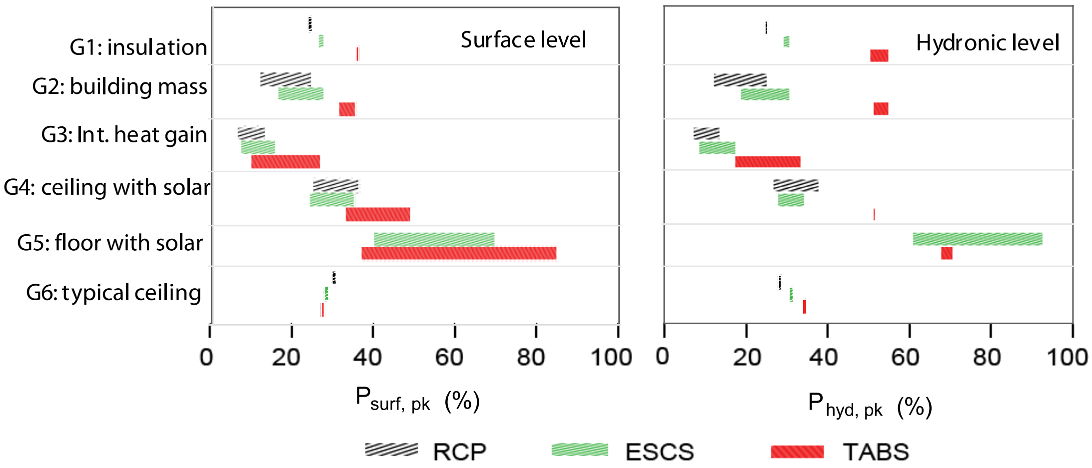

Figure 6: Peak cooling rate percentage difference between radiant and air systems

6. Discussion

6.1. Cooling load dynamic for radiant system

In order to explain why the radiant system peak cooling rate is higher than the equivalent air systems, Figure 7 investigates zone cooling load dynamics for the two systems. Using case rad0.6 (RCP) as an example, the figure compares the processes of how radiative and convective heat gains are converted into zone cooling load for the two systems. To assist the explanation, Figure 8 plots the operative temperature, air temperatures, and active and non-active wall surface temperatures for the two systems. Radiant cooling surface temperature is also plotted. For Case rad0.6, the total internal heat gain (15 W/m^2 during occupied hours) was divided into convective heat gain (6 W/m^2) and radiative heat gain (9 W/m^2). As shown, the cooling load for both systems was composed of two components, one that originated as convective heat gain from internal loads, and one that originated as radiative heat gain from internal loads. The instantaneous cooling load depends both on the magnitude and on the nature of the heat gains acting at the same instant. In a zone conditioned by an air system, the cooling load is 100% convective, while for the radiant systems the cooling load represents the total heat removed at the activated ceiling surface, which includes incident radiant loads, longwave radiation with non-activated zone surfaces and convective heat exchange with the warmer room air. In the case of air system (left plots), convective heat gain becomes cooling load instantaneously, and radiative gains are absorbed by zone thermal mass and re-released as convective load. The fact that building mass delays and dampens the instantaneous heat gain is well recognized by cooling load calculation methods. For the radiant cooling system (right plots), a large portion of the radiative heat gain converts to cooling load directly during the occupied period due to the presence of the cooling surface(s). Not all convective gains instantaneously contribute to cooling load, a smaller amount compared to the air system, during the occupied hours because a higher zone air temperature is reached to balance the cooler ceiling surface temperature, thereby maintaining an equivalent operative temperature, as is shown in Figure 8. And because of the higher zone air temperature, a small part of the convective heat gain is absorbed by non-activated building mass and removed by the radiant surface via longwave radiation. The bottom plots stack up the two cooling load components, and the solid black lines in the bottom plots are hourly cooling loads, which reach their peak value at the end of the occupied period for both systems. These predicted cooling loads represent the total amount of heat being removed by each system to maintain the same operative temperature profile. Note that the peak cooling rate for the radiant system is predicted to be 13.0% greater than that for the air system.

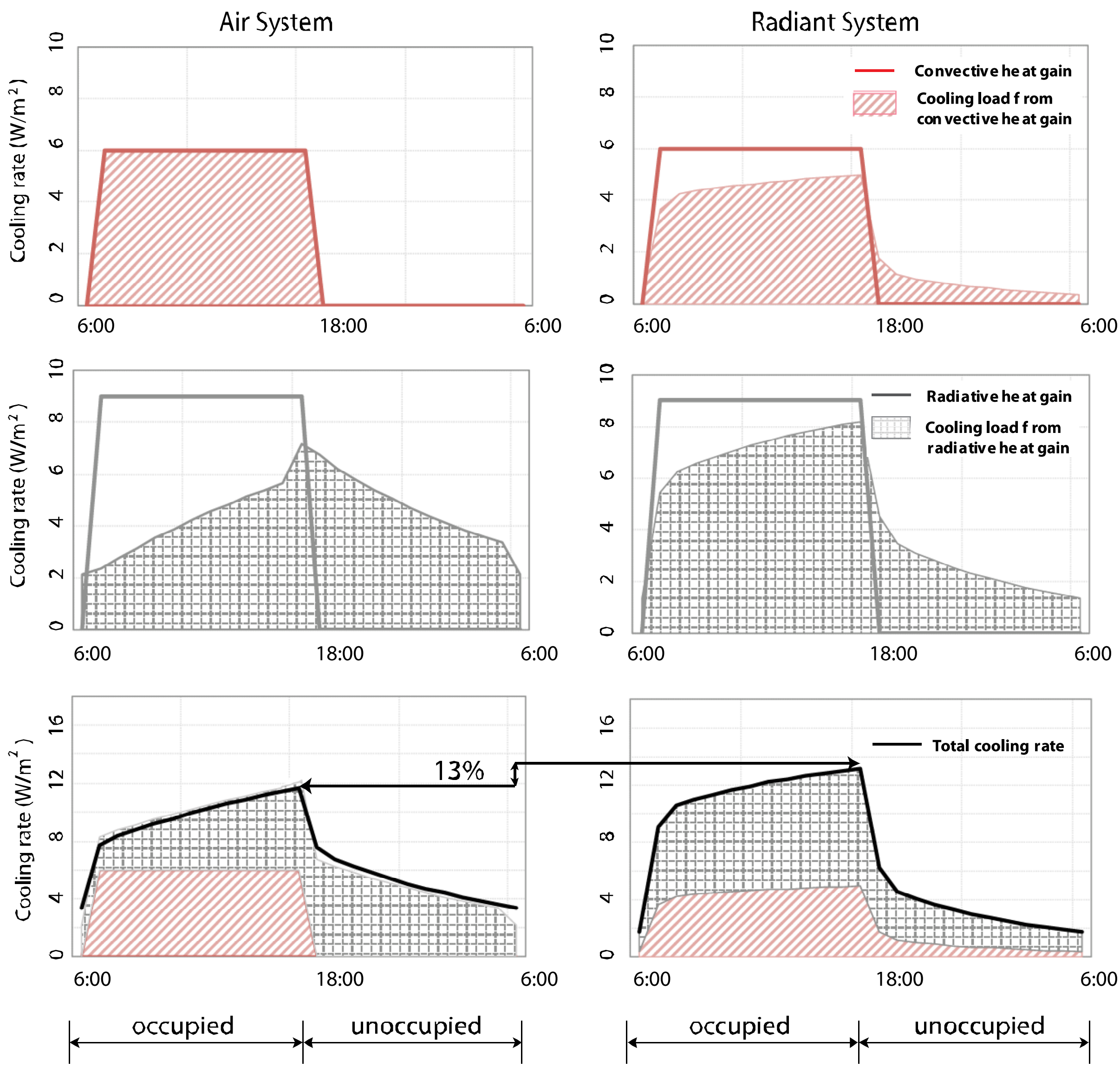

Figure 7: Comparison of surface cooling breakdown (convective and radiative part) for G3 Case rad0.6: air system (left) and radiant cooling panel (RCP) system (right)

Based on the discussion above, the author modified the cooling load generation diagram presented in chapter 18 of ASHRAE Fundamentals (2009) to represent the cooling load generation process when the zone is cooled by a radiant system (Figure 9). The original diagram was used to explain the cooling load generation process for an air system, and based on which most of the simplified cooling load calculation methods have been developed. The modifications are highlighted in red lines. This modified diagram illustrates that the cooling load differences between the two systems originate from two aspects: 1) radiant cooling surface(s) directly remove part of the radiant heat gain and reduce heat accumulation in the building mass; 2) only part of the convective heat gain becomes instantaneous cooling load, and the remainder partly contributes to increased air temperature and partly is stored in building mass and removed by the radiant surface as surface cooling load.

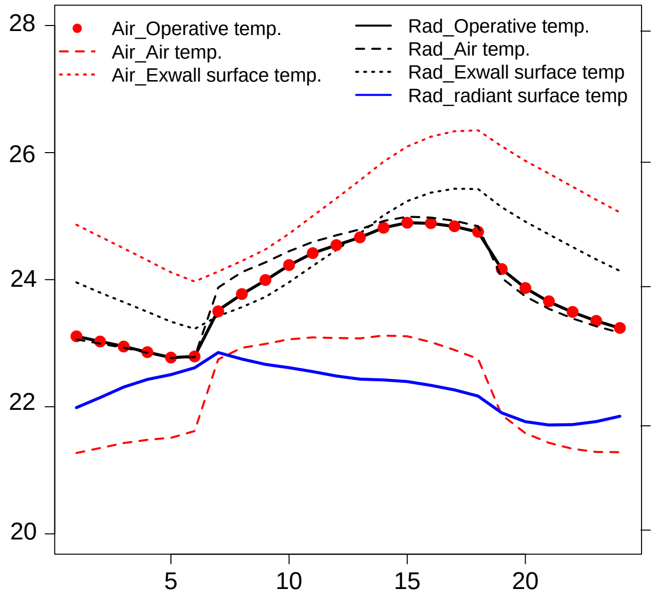

Figure 8: Comparison of zone air temperatures, operative temperatures, active and non-active surface temperatures between radiant and air systems (G6 typical ceiling: cl_shade_rad0.6)

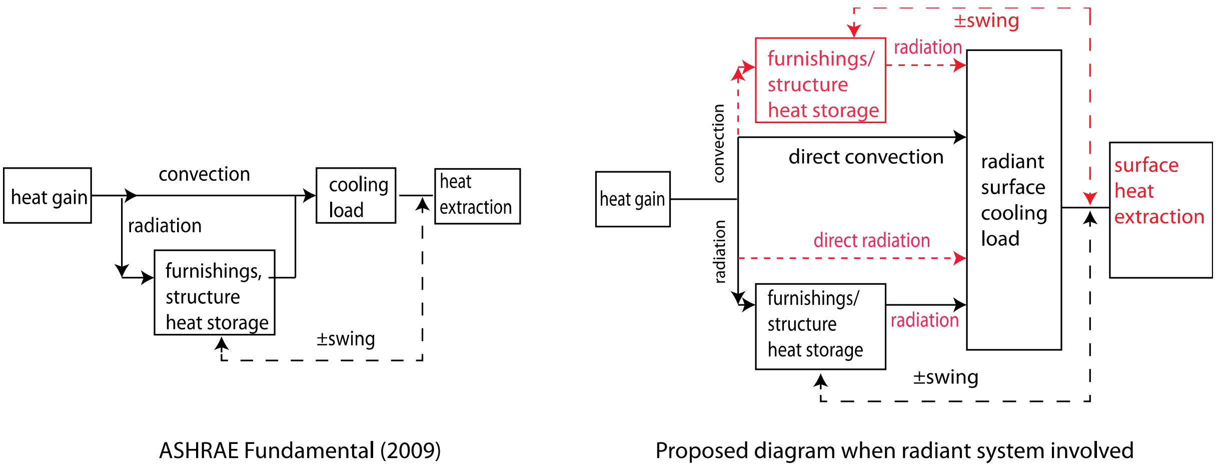

Figure 9: The cooling load generation scheme for air system adapted from ASHRAE Fundamentals (2009) and proposed modifications for radiant system

6.2. Definition of cooling load for different radiant system types

Throughout this study, we felt there is a need to clarify the definitions of design cooling load for sizing radiant systems and to distinguish between the three types of systems for the following reasons:

1. There is no clear definition of design cooling load for sizing radiant systems. According to ASHRAE Handbook [29], cooling load is defined as: "*the rate at which sensible and latent heat must be removed from the zone to maintain a constant zone air temperature and humidity*". However, zone air temperature is not recommended as the control temperature when radiant systems are involved [4]. In addition, in ISO 11855 (2012), design sensible cooling load is defined as: "*required sensible thermal output necessary to achieve the specified design conditions at the outside summer conditions*." It is not clear from this definition what the "specified design conditions" is.

2. Differences in thermal and control characteristics of the three radiant system types are usually not accounted for when determining design cooling loads. Peak instantaneous cooling load is normally used for sizing air system equipment, but it is not the most relevant for sizing all types of radiant systems. One example is the TABS. The reasons are: 1) intermittent or night-time operation is often implemented in order to take advantage of the storage capability of the active surface for load shifting, 2) time constants for these systems are large so it is not feasible to control the hydronic system in response to short-term environmental changes (load, setpoint changes) [30, 42].
3. As mentioned before, radiant cooling systems (ESCS and TABS) are integrated with the building mass. As a result, cooling rates at the surface and at the hydronic level are different due to the mass (thermal storage and delay). In cases of air systems, zone cooling load is directly used for sizing the HVAC systems, while in the case of a radiant system, the cooling load imposed on the hydronic loop is a better reference for sizing of cooling plant equipment.

Based on the discussion above, we propose to: 1) distinguish the design cooling load definition for sizing the quicker-response RCP/ESCS from the slower-response TABS; 2) define surface cooling load for the determination of required cooling surface area, and define hydronic cooling load for sizing hydronic equipment (pumps, cooling plant, etc.).

For RCP and lightweight ESCS, the cooling load definitions are:
Surface cooling load: the rate at which sensible heat must be removed by the actively cooled surface(s) from the zone to maintain a constant zone operative temperature during cooling design day. Peak surface cooling load should be used for determining total required cooling surface area.
Hydronic cooling load: the rate at which heat must be removed by the hydronic loop to maintain a constant zone operative temperature during cooling design day. Peak hydronic cooling load should be used for sizing cooling plant equipment.

The specific surface cooling load can be theoretically calculated by Eq. (1) at design conditions. If we further breakdown the surface radiation term into different radiation components, Eq. (1) can be expanded as following,

$$q''_{surf} = q''_{conv} + q''_{lw\_surf} + q''_{lw\_int} + q''_{sw\_sol} + q''_{sw\_int} \tag{3}$$

The last three terms, longwave radiant exchange flux from internal loads, transmitted solar radiation flux absorbed at surface and net shortwave radiation flux to surface from internal loads (lights), are the incident radiation that we discussed in the previous sections. During the sizing process, these three terms can be considered as an enhancement of cooling capacity [41], therefore, even if the peak cooling load of a radiant system may be higher than the cooling load calculated using traditional tools without capability to capture radiation heat transfer, the total area required may not need to be increased. Future research is needed to quantify how the three incident radiation terms mentioned above will affect sizing of cooling surface area.

For the RCP, hydronic cooling load is the surface cooling load plus heat loss from the backside of the panels, if any. For ESCS, the correlation between surface cooling rate and hydronic cooling rate is complicated by the heat conduction through the slab. Part 2 of ISO 11855 [4] recommends three methods for estimating surface cooling output and correlating the output with hydronic side operating conditions.

Design cooling load calculation for TABS has to take into account the control and operation strategy. For example, Part 4 of ISO 11855 (2012) provides guidance on calculating cooling capacity and cooling power demand on waterside to be used to select the cooling system, and it proposed to size the cooling equipment based on the sum of the heat gain values acting during the whole design day, internal load pattern, hydronic loop operation schedule, as well as radiant system specifications. Therefore, cooling load used for sizing TABS is not a unique value.

6.3. Proposed improvements in the design standards

As mentioned before, current radiant cooling design standards do not explicitly identify the differences in cooling load calculations between radiant and air systems. This results in the misapplication of tools in design practice not just for cooling load calculations but sometimes for detailed energy and comfort analysis. Currently, there are three classes of zone thermal models used in energy simulation tools: Heat Balance (HB), Thermal Network (TN), and Transfer Function (TF) models [43]. The HB and TN methods require relatively extensive computation time and effort from their users, and therefore are not widely adopted in tools used by design practitioners. The tools that use these two zone models, however, have the capability to capture detailed heat transfer processes in the zones and are recommended for use when radiant cooling systems are involved in the design. Modifications to the TF method, in particular the RTS method for radiant system cooling load calculations could be a good solution, but would require future research and is not an easy job due to the coupling of the radiant slab with the building structure.

Radiant systems should be modeled to ensure that the cooled surfaces are participating at the zone level heat transfer during the design calculation. A review of design tool showed that even though dynamic simulation tools are used for energy analysis, the cooling equipment sizing is often based on cooling loads calculated for an ideal air system. For example, the authors observed this in the EnergyPlus "autosizing" algorithm for radiant systems. In EnergyPlus, the HB method is used as the zone heat transfer model so it has the capability to calculate cooling load accurately when radiant systems are involved. However, it also assumes that the cooling load for a radiant system is the same as for an air system. Therefore, if "autosizing" function is used, an "ideal air system" is simulated first for load calculation, and this cooling load is used for sizing all associated cooling equipment, including radiant system design mass flow rate, total tube length, and radiant system plant equipment. The users can manually adjust the design parameters if necessary, but failure to realize the cooling load differences between radiant and air systems can produce significant errors.

In summary, the following text and recommendations could be included in radiant cooling design guidelines to improve understanding of radiant cooling system and to facilitate better design solution:

- The cooling load for zones conditioned by a radiant system is different from the cooling load for zones conditioned by an air system. The differences between the two systems originate from two aspects: 1) active radiant cooling surface(s) directly remove part of the radiant heat gain and reduce heat accumulation in the building mass; 2) only part of the convective heat gains becomes instantaneous zone cooling load (as is the case in an air system), and the other portion partly contributes to increased air temperature and partly is stored in building mass and subsequently removed by the active radiant surface as surface cooling load.
- For RCP and lightweight ESCS, peak surface cooling load shall be used for dimensioning total required cooling surface area, and peak hydronic cooling load shall be used for sizing associated cooling equipment. For TABS, equipment sizing depends on total heat gain energy, heat pattern, operational strategy, etc.
- Simulation tools that use either heat balance or thermal network methods for zone level thermal modelling are recommended for design cooling load and system sizing calculations for radiant systems. Examples of the recommended tools are: EnergyPlus, IES Virtual Environment, IDA ICE, Esp-r, TRNSYS.
- The following design procedure is recommended for load calculation and system equipment sizing: 1) conduct a basic cooling load calculation as if an ideal air system with unlimited cooling capacity is used for conditioning the space. This basic cooling load value can be used for comparing different design options. If a radiant system is choosen, the basic cooling load value can be used as a starting point for dimensioning the radiant cooling system; 2) recalculate design surface cooling load and hyrdonic cooling load for the radiant system. During this process, the radiant cooling system should be modeled with a computer program that meets the prescribed requirements mentioned above. Size the radiant system properly to satify prescribed thermal comfort requirements. 3) Size the cooling plant equipment based on design hydronic cooling load calculated from step 2.

7. Conclusions

This simulation study investigated the impacts of the presence of an activated cooled surface on zone cooling loads by comparing the peak cooling rate and the 24-hour total cooling energy for radiant and air systems. Three radiant system types (RCP/ESCS/TABS) and single zone models were developed for comparison between each radiant system type and their equivalent air system. The models were configured to study the impacts of the following parameters: envelope thermal insulation, thermal mass, type of internal gain, solar heat gain with different shading options, and radiant surface orientation (ceiling, floor). The simulation results are summarized below:

- For the simulated cases, when compared to an air system for equivalent comfort conditions (operative temperature), 24-hour total cooling energy removed by the RCP system hydronic loop was 5-13 % higher, 6-15% higher for the ESCS, and 6-15% higher for the TABS. This was caused by lower surface temperatures at the inside surfaces of the building envelope created by the active (cooled) radiant surface.
- For perimeter zones that were only subjected to building envelope heat gain, $P_{surf,pk}$ ranged from 12% to 25% for the RCPs, 16% to 27% for the ESCS, and 31% to 35% for the TABS. For the RCP and the ESCS, $P_{hyd,pk}$ were in the similar range as $P_{surf,,pk}$,but for the TABS, $P_{hyd,pk}$ increased to 50-54%.
- For interior zones with longwave radiant heat gain, the peak cooling rate differences ranged from 7% to 27% at the surface level and from 7% to 33% at the hydronic level. This implies that higher radiant fraction in heat gain produces larger differences in peak cooling rates between the two systems at the surface level. This was further demonstrated in cases with solar load.
- For perimeter zones and atrium where direct solar heat gain constitutes a large portion of the cooling load, the peak cooling load difference is pronounced. When exterior shading was not installed, RCP ceiling surface peak cooling rate is 36% higher than the air system, and for ESCS ceiling system it is 35%, and 49% for TABS ceiling systems. Exterior shading reduced the direct solar impact, but the surface peak cooling rate were still 24-33% higher for the ceiling system.
- When the floor was used as the radiant cooling surface and when it was illuminated by direct solar, both $P_{surf,,pk}$ and $P_{hyd,,pk}$ increased dramatically compared to the ceiling cases. The ESCS surface peak cooling rate was 69% higher and for TABS, 85% higher. Exterior shading greatly reduced the difference between radiant and air systems at the surface level for both radiant systems. However, $P_{hyd,,pk}$ for TABS was not much affected by the installation of shading system.
- Cooling rate differences between the two systems originate from two effects: 1) radiant cooling surface(s) directly remove part of the radiant heat gain and reduce heat accumulation in the building mass; 2) only part of the convective heat gain becomes instantaneous cooling load, the remainder partly contributes to increased air temperature and partly is stored in the building mass and removed by the radiant surface as surface cooling load.

In conclusion, zones conditioned by a radiant system have different peak cooling loads than those conditioned by an air system. While the increase in 24-hour total cooling energy is relatively small and may be offset by other energy savings benefits associated with radiant cooling systems, the differences in peak cooling load both in terms of magnitude and time compared to the air systems require special attention in system and control design. In the radiant design standards, these differences in cooling load should be clearly stated and translated into requirements for tools and methods. Transfer Function (TF) based methods are not appropriate for cooling load calculation when radiant cooling systems are involved. Radiant systems should be modeled using a dynamic simulation tool that employs either Heat Balance (HB) model or Thermal Network (TN) models during the design process for accurate cooling load calculation.

8. Nomenclature

RCP	Radiant cooling panels
ESCS	Embedded surface cooling systems (lightweight)
TABS	Thermally activated building systems
G1 - G6	Simulation group index
q''	Heat flux, W/m^2
q''_{surf}	Heat flux at the exposed face of the cooling surface(s) , W/m^2
$q''_{surf,cond}$	Conduction heat transfer at the exposed face of the cooling surface(s) , W/m^2
$q''_{surf,conv}$	Convection heat transfer at the exposed face of the cooling surface(s) , W/m^2
$q''_{surf,rad}$	Radiation heat transfer at the exposed face of the cooling surface(s) , W/m^2
q''_{lw_surf}	Net longwave radiation flux to radiant active surface from other surfaces, W/m^2
q''_{lw_int}	Longwave radiant exchange flux from internal load, W/m^2
q''_{sw_sol}	Transmitted solar radiation flux absorbed at surface, W/m^2
q''_{sw_int}	Net shortwave radiation flux to surface from internal load (lights). , W/m^2
$q''_{surf,pk}$	Specific peak radiant system surface cooling load, W/m^2
$q''_{hyd,pk}$	Specific peak radiant system hydronic cooling load, W/m^2
$q''_{air,pk}$	Specific peak sensible cooling load for air system, W/m^2
$\dot{q}_{surf,tot}$	Specific 24-hour total surface cooling energy, kJ/m^2
$\dot{q}_{hyd,tot}$	Specific 24-hour total hydronic cooling energy, kJ/m^2
$\dot{q}_{air,tot}$	Specific 24-hour total sensible cooling energy, kJ/m^2
$P_{surf,pk}$	Percentage difference of surface peak cooling rate between radiant and air system, %
$P_{hyd,pk}$	Percentage difference of hydronic peak cooling rate between radiant and air system, %
$P_{surf,tot}$	Percentage difference of surface level 24-hour total cooling energy between radiant and air system, %
$P_{hyd,tot}$	Percentage difference of hydronic level level 24-hour total cooling between radiant and air system, %
Subscript	
surf	Variable measured at radiant surface level
hyd	Variable measured at radiant cooling water loop
pk	Peak cooling load
tot	24 hour total cooling energy

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