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Experimental comparison of zone cooling load between radiant and air systems

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Highlights

- We experimentally compared cooling loads between radiant and air systems
- Radiant system has on average 18%-21% higher cooling rates compared to air system
- A new definition must be used for radiant system cooling load
- HB approach should be used for radiant system cooling load calculation
- RTS or weighting factor methods may lead to incorrect results for radiant systems

Abstract

Radiant cooling systems work fundamentally differently from air systems by taking advantage of both radiant and convective heat transfer to remove space heat. This paper presents an experiment investigating how the dynamic heat transfer in rooms conditioned by a radiant system is different from an air system, and how such differences affect the sensible cooling load and cooling load calculation methods for radiant systems. Four tests with two heat gain profiles were carried out in a standard climatic chamber. For each profile, two separate tests were carried out to maintain a constant operative temperature: one with radiant chilled ceiling panels; and a second with an overhead mixing air distribution system. Concrete blocks were used to create a thermal mass effect. The experiments show that, during the periods the heat gain was on, the radiant system has on average 18%-21% higher instantaneous cooling rates compared to the air system, and 75-82% of total heat gains were removed, while for the air system only 61-63% were removed. Based on the study, we conclude that a new definition must be used for radiant system cooling load, which should be characterized as the combined radiant and convective heat removal at the cooled surface. Calibrated dynamic energy simulation based on a fundamental heat balance approach showed good accuracy. Simplified cooling load calculation methods, such as RTS or weighting factor method, may lead to incorrect results for radiant systems.

Keyword

Radiant Cooling System, Cooling load, Air system, Experimental Testing, Dynamic Response, Modelling Method

1 Introduction

Interest and growth in radiant cooling and heating systems have increased in recent years because they have been shown to be energy efficient in comparison to all-air distribution systems[1, 2]. Olesen and others have discussed the principles of designing radiant slab cooling systems, including load shifting, the use of operative temperature for comfort control, and cooling capacity[3, 4]. Several case study examples with design information have been reported for an airport[5], large retail store with floor cooling[6], and

other thermally active floor systems[7]. However, it is difficult to find detailed standardized guidelines for calculating cooling loads for radiant cooling systems, which is the subject of this paper.

Cooling load calculations are a crucial step in designing and sizing any HVAC system. Compared to air systems, the presence of an actively cooled surface changes the heat transfer dynamics in a zone of a building. The chilled surface is able to instantaneously remove radiant heat (long and short wave) from any external (solar) or internal heat source, as well as interior surface (almost all will be warmer than the active surface) within its line-of-sight view. This means that radiant cooling systems may impact zone cooling loads in several ways: (1) heat is removed from the zone through an additional heat transfer pathway (radiant heat transfer) compared to air systems, which rely on convective heat transfer only; (2) by cooling the inside surface temperatures of non-active exterior building walls, higher heat gain through the building envelope may result; and (3) radiant heat exchange with non-active surfaces also reduces heat accumulation in building mass, thereby affecting peak cooling loads. Using simulations we previously demonstrated that dynamic responses of rooms when conditioned by radiant cooled surface(s) are significantly different from the case of air systems and consequently the cooling loads for system sizing are also drastically different (in fact, often higher for the studied cases)[8]. Thus, current cooling load calculation and modeling methods may not be applicable for radiant systems.

The objectives of the study reported below are: 1) experimentally compare sensible zone cooling loads between a radiant and well-mixed air system; and 2) provide guidance on radiant system cooling load prediction and energy modeling methods.

2 Review of current zone cooling load prediction methods for radiant systems

Based on the standard cooling load calculation methods described in ASHRAE Handbook – Fundamentals [9], the zone sensible cooling load is the rate at which sensible heat must be removed from the zone air to maintain a constant air temperature. Currently, there are two recommended cooling load calculation procedures, the Heat Balance (HB) method and the Radiant Time Series (RTS) method. There are, however, important limitations when these methods are applied to radiant systems.

The procedure based on HB method is considered the most scientifically rigorous method [10]. The heat balance model ensures that all energy flows in each zone are balanced by iteratively solving for a set of energy balance equations in the following loops: outside surface and the environment, conduction through building envelope, inside surface heat balance, and finally the air heat balance. This procedure calculates the cooling load by balancing the air loop heat transfer convectively according to the ASHRAE cooling load definition. However, for radiant systems, heat is removed at the actively cooled surface both convectively and radiatively. Therefore, although the heat transfer fundamentals are properly accounted for, it is questionable whether the current cooling load calculation procedure based on HB method and ASHRAE definition of cooling load is appropriate for radiant applications.

The RTS method is a simplified calculation procedure [11], originally developed to provide an approximation to the HB Method. According to this procedure, each heat gain (conduction portions along with lights, occupants, and equipment) is split into radiative and convective portions. The convective portion is assumed to instantly become cooling load and, therefore, only needs to be summed to find its contribution to the hourly cooling load. Radiant heat gain, on the other hand, must first be absorbed by the non-active surfaces that enclose the zone (floor, walls, ceiling) and objects in the zone (e.g., furniture). These surfaces will eventually increase their temperature above the air temperature to allow heat to be

transferred by convection to the air, thereby contributing to the convective zone cooling load. So for all-air systems, it is always assumed that radiant heat gains become cooling load only over a delayed period of time. This process is graphically presented in Figure 1. The method for converting the radiative components to cooling loads involve calculations of a series of radiant time factors, which were generated with the assumption of a well-mixed all-air system with no active radiant cooling surface(s) [11].

In addition to these two methods, there are several other simplified methods (e.g. cooling load temperature difference/cooling load factor/solar cooling load factor (CLTD/CLF/SCL) method[12], weighting factor method[13], etc.) that are widely used in modeling software for cooling load prediction purposes. All these methods are developed with an underlying assumption that convective heat transfer by air is the only mechanism to remove heat from a zone.

Due to the mismatch between how radiant heat transfer is handled in traditional cooling load calculation methods compared to its central role in radiant cooling systems, this research examined the fundamentals of cooling load calculations for radiant cooling systems.

Figure 1: Cooling load diagram from ASHRAE Handbook – Fundamentals

3 Experimental comparison of cooling load between radiant and air systems

Current methods for testing radiant system performance are based on steady state conditions [4, 14, 15], which is not adequate for cooling load prediction. A testing method was established in this study to investigate the dynamic behavior of radiant systems and the resultant zone cooling load.

3.1 Experimental facilities and setup

The experiments were carried out in a climatic chamber (4.27 m x 4.27 m x 3.0 m). This chamber has been used for standard radiant cooling panel testing and meets the requirements stated in DIN EN 14240 [15]. The climatic chamber is located within a large conditioned laboratory space. The room has no windows. The walls, ceiling and floor have similar construction and thermal properties. Starting from the exterior, the chamber wall is comprised of 3.522 m²K/W insulation, a stagnant 0.102 m air gap (0.352 (m²K)/W), aluminum extruded walls with water tubes attached, and another layer of 0.102 m of polyurethane board (3.522 (m²K)/W). By adding up this assembly, the overall resistance is 7.396 (m²K)/W).

For the radiant cooling test, 12 aluminum radiant panels were connected in parallel and installed in the suspended ceiling placed at a height of 2.5 m above the floor, and each was 1.83 m long and 0.61 m wide (73.5% of the ceiling area was covered by panels). Copper pipes are thermally connected to aluminum channels in panels with a spacing of 0.15 m. Cotton fiber insulation was placed on the topside of the panels (1.76 (m²K)/W). The same chamber was used for the air system test, during which one radiant

panel was replaced with an insulation board with opening cut to accommodate one air diffuser for conditioning the zone. See Figure 2 for test chamber setup.

Thermal mass was a crucial element in this experiment. In the test, 64 pieces of concrete pavers ($0.46 \text{ m} \times 0.46 \text{ m} \times 0.04 \text{ m}$) with a total weight of 1350 kg were placed on the floor. The vertical walls in the test chamber were lightweight rigid thermal insulation board construction. With this configuration, the distribution of thermal mass in the chamber was intended to represent an interior zone where most building mass is located either in the floor or ceiling structure.

Heat gain was simulated with a thin electric resistance heating mat, laid on top of the concrete blocks. The loose mesh design of the heating mats allowed the radiant cooling ceiling panels to interact directly with both the heater and the concrete pavers.

Figure 2: The test chamber setup

3.2 Measuring instruments and uncertainty

Table 1 lists the measurement instruments and equipment used in this experiment. Thermocouples were used to measure inner surface temperatures of the vertical walls (eight in total, and two for each wall) and Resistance Temperature Detectors (RTDs) were placed on the other side of the polyurethane board layer of the walls for temperature measurements. Panel surface temperatures were also measured with thermocouples (five on each side of the panel surface). Measurement accuracy of thermocouples after calibration was $\pm 0.15^\circ\text{C}$. RTDs were used for other temperature measurements, and they were calibrated prior to the tests. The sensor accuracy was at $\pm(0.03+0.0005 \cdot T)^\circ\text{C}$. For measuring surface temperatures, heat transfer paste was used to ensure good thermal contact between sensors and measuring surfaces. For the calculations of thermal storage in concrete pavers, ten RTDs were evenly distributed (five on top and five on the bottom) on the pavers, and the average temperature was used. Air temperatures, globe temperatures, and air velocities at five different heights (0.1 m, 0.6 m, 1.1 m, 1.7 m, 2.4 m) were measured. Since globe temperature was part of the equation for calculating operative temperature, used for zone temperature control during the tests, a fast-response globe temperature sensor was desired. The response time of a black globe thermometer specified in ISO 7726 [16] is about 20-30 minutes, which was too long for this study. The globe thermometer was constructed by inserting an RTD sensor into a table tennis ball coated with gray paint. According to Benton et al. (1990) [17], this type of globe thermometer has a response time of 5.8 minute to reach 90% of its final value, which was considered acceptable for this experiment. Globe temperature and air velocity were used for calculation of mean

radiant temperature, and occupied zone operative temperature was calculated as an average of the mean radiant temperatures and air temperatures measured at heights 0.6 m, 1.1 m, and 1.7 m [16]. Emissivity was measured with an emissometer. Emissivity was 0.20 for the internal wall surfaces, 0.95 for the concrete floor, and 0.92 for the radiant panels, with uncertainties of ± 0.02 . The airflow rate during the air test was measured with a calibrated plate orifice having an accuracy of better than $\pm 3\%$ of the reading. The cooled water mass flow rate was measured with a high quality Coriolis mass flow meter with an accuracy of $\pm 0.02\%$ of the reading.

The data are analyzed in accordance with the ISO and JCGM guideline [18, 19] for the expression of uncertainty. The error sources for evaluating uncertainties in supply and return air temperature measurements include sensor accuracy and spatial variation, and the derived uncertainty of the temperature difference for tested conditions was within $\pm 0.11^\circ\text{C}$. Uncertainties in primary measurements are propagated to calculate radiant and air system cooling rate uncertainties. For this study, during the periods the heaters were on, the relative uncertainties in air system cooling rate are within $\pm 10\%$, and radiant system cooling rate uncertainties are within $\pm 8.5\%$. The level of confidence is 95% (coverage factor 2).

Table 1: List of sensors, accuracy and specifications

Instruments	Sensor accuracy	Unit	Number	Measured variables
Temperature RTD	$\pm(0.03+0.0005 \cdot T)$	$^\circ\text{C}$	38	Temperature of water into and out of the radiant panels; Supply and return air temperature; Chamber air temperatures; Chamber globe temperatures; Top and bottom temperatures of concrete pavers ; External side wall insulation surface temperatures;
Water flow meter	$\pm 0.02\%$	Kg/h	1	Radiant cooling panel water flow rate
Air flow meter	$\pm 3\%$	m^3/s	1	Supply air volumetric air flow rate
Thermocouples	± 0.15	$^\circ\text{C}$	16	Vertical wall surface temperatures; Surface and backside temperatures of radiant panels
Anemometer	$\pm 3\%$	m/s	5	Air velocity tree
Emissometer	± 0.01	-	1	Surface emissivity

3.3 Test procedure

Two series of tests were conducted, one at a heat input level of 1080 W (59.3 W/m^2), and one at 1500 W (82.4 W/m^2). The heater was turned on for 6.1 hours in the 1080-W test, and 6.9 hours in the 1500-W test.

For each set of test conditions, two separate experiments were conducted. First, radiant chilled ceiling panels were used to condition the chamber with controlled heat gain. No air system was operated during the radiant system test. Second, an overhead mixing air distribution system was used to remove the same heat gain profile. The approximated 12-hour test procedure was as follows. Prior to beginning the test, the

chamber and concrete thermal mass were allowed to reach a uniform, steady state initial temperature of 24°C. The test was started when the heater was turned on and maintained at a constant value for approximately six hours. The heater was then turned off and the experiment continued for approximately another six hours. For the entire duration of each test, the radiant or air system was controlled to maintain a constant operative temperature in the chamber. Operative temperature was chosen for two reasons: 1) it can better represent thermal comfort when compared to air temperature especially in the cases with non-uniform temperatures in interior surfaces; 2) it was prescribed as the control temperature in radiant system design standards [4]. Cooling rates were continuously monitored by measuring supply and return temperatures and flow rates for the hydronic radiant panel system and the overhead air distribution system.

3.4 Results

For the radiant system, we calculated the sensible cooling load as the combination of radiant and convective heat exchange at the actively cooled surface during that same time step. We focused on the surface heat transfer because it directly impacts thermal balance and comfort in the zone.

Figure 3 shows occupied zone operative temperatures over the course of all experiments. As can be seen, the differences in operative temperature between the radiant and air system tests were maintained less than $\pm 0.5^{\circ}\text{C}$ during all testing periods except for short periods (less than 0.5 hours) immediately after the heaters were turned on or off. The operative temperature for the air system test tended to run slightly warmer than that for the radiant system test due to the difficulties in control under transient conditions. Calibrated energy simulations were conducted to check the influence of operative temperature differences on resultant system cooling rates. For both radiant and air systems, 0.5°C difference in operative temperature setpoint caused less than 1.5% difference in system cooling rates during the periods when heaters were turned on, and less than 5% after heaters were turned off. Thus the impacts of the temperature differences between radiant and air system tests can be considered as negligible.

To characterize the dynamic behavior of thermal mass, Figure 4 shows the temperatures (average of top and bottom surfaces) of the concrete pavers and vertical wall surface temperatures. The thick lines are the average of the measurements and the shaded area represent the variation at different locations. The average temperature of the concrete in the radiant system 1080-W test reaches 28.0°C at the peak, and 29.0°C for the air system test. In the 1500-W tests, the peak temperatures were 30.3°C and 31.4°C for the radiant and air system tests, respectively. During the period the heater was on, the surface temperatures of the vertical walls in the chamber during the air system tests were also on average 0.7°C (1080-W test) and 1.2°C (1500-W test) higher compared to radiant system tests. Higher temperatures in the concrete and the chamber walls indicate a greater amount of energy storage when the heater was activated during the air test compared to the radiant test. Given that the same amount of heat is added to the chamber, there is more heat removal by the radiant panels (and less heat accumulation in the thermal mass) compared to the air system when the heaters are turned on.

Figure 3: Comparison of operative temperatures between radiant and air system tests. Grey area shows $\pm 0.5^{\circ}\text{C}$ range around the operative temperature for the radiant system. (A) 1080-W test and (B) 1500-W test

Figure 4: Comparison of concrete and wall temperatures between radiant and air system tests. Shaded areas represent the range of the different sensors measuring surface temperatures. (A) 1080-W test and (B) 1500-W test

Figure 5 compares the instantaneous radiant and air system cooling rates. The waviness of the lines are the result of the dynamic response of the HVAC control system when subjected to the transient heat transfer process. The shaded area in the cooling rate plots indicates the 95% uncertainty in the measurements. As mentioned earlier, the cooling rate uncertainties for both systems are lower than 10% during the heater-on periods. Even though there are fluctuations from the control system, radiant system cooling rates are clearly higher than air system cooling rates. Fifteen-minute moving averages of the percentage differences in instantaneous cooling rates between the two systems are plotted in Figure 6. For the 1080-W test, the cooling rate of the radiant panel reached an average of 807 W between hours 1-2 and was about 48% higher than the air system case. The cooling rate slowly ramped up over the next 4 hours until it reached an average of 969 W in the last hour before the heater was turned off, and this was about 18% higher than the air system case. On average between hours 2-6 (after the control was stabilized), the radiant system cooling rate was 21% higher than that of the air system. For the 1500-W test, a similar trend can be observed and the average radiant cooling rate for the last hour before the heater was turned

off was 1335 W, and this was 11% higher than the air system case. On average between hours 2-6.5, the radiant system cooling rate was 18% higher than that of the air system. Thus, we can conclude that radiant systems have on average 18%-21% higher instantaneous cooling rates compared to air systems for the tested conditions.

Figure 5: Comparison of measured instantaneous cooling rates for radiant and air systems. Shaded area represent the 95% uncertainty of the cooling rate. (A) 1080-W test and (B) 1500-W test

Figure 6: Percentage differences of measured instantaneous cooling rates for radiant and air systems, i.e. $[(\text{radiant cooling rate} - \text{air cooling rate}) / (\text{air cooling rate})]\%$. (A) 1080-W test and (B) 1500-W test

Figure 7 shows the cumulative energy input from the heater and corresponding energy removal by the radiant panels and air system. The differences between heater input and HVAC system heat removal are approximately equal to the energy storage in the concrete pavers and chamber walls. It can be seen that a greater amount of energy was stored in the concrete as the concrete temperatures increased for the period the heater was on. After the heater turned off, the stored energy was released back into the zone and was removed by the radiant panel/air systems. There was less energy storage during the radiant system tests,

as indicated by the smaller difference between the heater energy input lines and radiant system energy removal lines.

Figure 8 compares the percentage of total energy removed by the radiant panels and by the air system during the period with heater turned on. For the radiant system 82.0% of total heat gain was removed for the 1080-W test and 74.8% for the 1500-W test, while it was only 63.3% and 61.4% for the two air system tests, respectively. Energy balance was also checked during the heater-on periods. Energy was considered balanced if the total heater energy input was equal to the sum of the total energy removed by the HVAC system and the amount of thermal storage. Thermal storage consisted of storage in the concrete and other chamber surface layers, which was calculated with measured average temperatures of the surfaces and their estimated thermal properties. The discrepancies between energy input and output for the 1080-W tests were $\leq 4.2\%$, and for the 1500-W test, the discrepancies were $\leq 2.9\%$.

Figure 7: Profiles of cumulative heater energy input to chamber and cumulative energy removal by radiant and air systems. (A) 1080-W test and (B) 1500-W test

Figure 8: Comparison of percentage of total heat gain being removed and percentage of energy storage for radiant and air systems during heater-on periods: (A) 1080-W test and (B) 1500-W test

3.5 Discussion

For the tested conditions, radiant system peak cooling rates were 18% and 11% higher for the 1080-W and 1500-W tests, respectively. These numbers agree well with previous simulation results, which showed differences from 6-15% for interior zones depending on the radiation factor of the heat source [8].

It is important to note that for radiant slab systems, cooling rates at the room side (surface level) and at the hydronic level are different due to thermal mass effects. The hydronic level cooling load is a better reference for sizing of cooling plant equipment. A fast responsive radiant panel system was used in the experiment because of its better controllability. The resultant dynamic interactions between actively cooled surface and its thermal environment and the implications for cooling load differences can well represent the situations with any other types of systems that utilize radiation as a heat transfer path to the conditioned zone.

TABS or other thermally massive radiant systems are known to respond slowly to control signals. However, as shown in Figure 7-8, the radiant surface is able to respond quickly to changes in heat gains in the zone. Therefore, we may conclude that radiant systems are both quick and slow depending on the context. This feature of radiant systems may impact design and operation in many ways. For example, radiant floor systems have wide applications in conditioning areas with high solar load, such as atriums and lobbies, because they are considered to be especially effective at removing solar load with enhanced cooling capacity reaching 100 W/m^2 [3, 20]. The fast responsive feature is considered advantageous for maintaining thermal comfort when compared to air systems. On the other hand, designers need to realize that air and radiant systems interact differently with building non-active mass. For air systems, heat gain can be significantly delayed before becoming cooling load by non-active building mass, and this may be beneficial in many applications. For radiant systems, as demonstrated by the above-described experiments, there is less heat accumulation in the non-active thermal mass and therefore less time delay

effect. There are also differences between lightweight radiant panel systems and thermally massive radiant systems. For TABS and other massive radiant systems, cooling requirements may be completely shifted to unoccupied periods if the radiant layers can be preconditioned. In fact, because these systems are slow to respond to control signals, they are not designed to maintain a constant zone temperature.

Cooling load is not the same as energy consumption. Hydronic-based radiant systems have verified advantages over air systems, such as the improved transport efficiency of using water instead of air as the thermal distribution fluid[21], improved plant side equipment efficiency with warmer cold water temperatures[22], and, particularly with TABS, the possibility of night pre-cooling using cooling towers[23].

4 Cooling load calculation methods for radiant systems

To assess the accuracy of current cooling load calculation methods when applied to radiant systems, we compared the measured results with predicted instantaneous cooling rates using the fundamental HB method and simplified RTS method.

4.1 Methodology

A single-zone EnergyPlus (v8.0) model was developed to closely match the test chamber construction, boundary conditions, and system operating schedule during the experiments. EnergyPlus employs the HB method for zone thermal modeling and has been validated and widely used as a whole-building energy simulation tool [6, 24, 25]. The heater on the floor was modeled using module *ZoneHVAC:HighTemperatureRadiant*.

The default zone cooling load calculation algorithm implemented in EnergyPlus uses the ASHRAE recommended procedure, which defines the cooling load as the heat convectively removed from the zone air volume to maintain the air temperature setpoint. Cooling load for a radiant system is assumed to be the same as for an air system in EnergyPlus. Therefore, the current algorithm does not require modeling of radiant system to obtain cooling load, instead an "ideal air system" is to be simulated. However, as mentioned before, this definition cannot directly apply to radiant systems. A new definition of cooling load must be used, defined as the combined radiative and convective heat removal rate at the actively cooled surface to maintain a constant zone operative temperature setpoint. Therefore, to obtain the actual radiant cooling load, the radiant ceiling panels need to be modeled in EnergyPlus. The built-in EnergyPlus radiant model is able to integrate the heat transfer calculation in the systems with changing zone conditions, and therefore is able to capture the transient system behavior [26-28]. For calculation of cooling load for air system, an idealized air system was modeled.

As mentioned above, the RTS method is a simplified calculation procedure[11] [11], originally developed to provide an approximation to the HB Method. To avoid complex heat transfer calculations, the RTS method converts heat gain into cooling load by applying periodic response factors (PRF) and conduction time factors (CTF).

The CTFs and PRFs used to calculate cooling load for the experimental cases were generated by CTF/PRF Generator [29], where the climatic chamber geometry and construction specifications were used as inputs. These conversion factors were then used to calculate resultant cooling load in a spreadsheet where heat gain intensity, schedule and radiant/convective split were specified closely to match testing conditions. Based on calculations of radiation and convective heat transfer at the heated

floor, the radiation factor of heat sources was roughly estimated to be 0.9 for the radiant system and 0.7 for the air system.

4.2 Results

Figure 9 presents the results for the 1080-W test, similar trends were observed for 1500-W test. With the revised definition of radiant cooling load, Figure 9A shows good agreement between measured and predicted cooling rates for both radiant and air systems, and the differences are expressed as normalized mean bias error (NMBE) at 8.3% for the radiant case, and 9.4% for the air case [30]. Figure 9B, however, demonstrates the limitations of applying the RTS method to the test chamber configuration. Due to the underlying assumption that radiant heat gains are only released by non-active thermal mass as convective loads after a time delay, the RTS method under-predicts the measured radiant system cooling load. The RTS method also assumes that radiant heat gains are uniformly distributed on all zone surfaces. In the case of the chamber experiment, the location of the heater on top of the concrete transferred a higher percentage of heat gain into the thermal mass, resulting in an over-prediction of the air system cooling load by the RTS method.

Figure 9: Comparison of measured and predicted instantaneous cooling rates using heat balance (HB) method (A) and using radiant time series (RTS) method (B) for radiant and air systems. The 1080-W test was used.

4.3 Discussion

In summary, a new definition for radiant system cooling load based on the heat removal by both radiation and convection at the cooled surfaces must be used. The main differences are: a) radiant heat transfer allows chilled surfaces to directly remove part of radiant heat gains from a zone, thereby bypassing the assumed time-delay effect caused by the interaction of radiant heat gain with non-active thermal mass for air systems; and b) operative temperature is a better reference for thermal comfort and should be used for calculating cooling load for radiant systems. Note that for thermally massive systems, cooling load is also dependent on operational strategy.

Sensible cooling load calculations for radiant systems should utilize dynamic energy simulation programs or design tools based on a fundamental heat balance approach that properly takes into account how heat gains are removed from a zone by an actively cooled surface. Some examples of whole building

simulation tools with such capability are EnergyPlus, IES-VE, TRNSYS, IDA-ICE, and ESP-r. Those tools are widely used in the design industry for code compliance and LEED certification.

Simplified cooling load calculation methods, such as RTS or weighting factor methods, may lead to incorrect results for radiant systems.

5 Conclusion

This research demonstrated how the dynamic heat transfer in rooms conditioned by radiant systems differs from that of air systems, and how such differences impact design cooling load calculation and energy modeling methods. Key conclusions are summarized below.

1. The experimental results confirm that radiant system cooling rates are different from air systems even when similar thermal comfort conditions are maintained:
 - The instantaneous cooling rate for the radiant system was on average 18%-21% higher than the air system for the tested conditions.
 - The radiant system removed 75-82% of total heat gains during the period when the heater was on, while the air system removed 61-63%. The differences were caused by the amount of energy stored in the non-active thermal mass. The temperatures of non-active mass (concrete blocks in this study) were approximately 1°C lower for the radiant tests compared to the air tests under peak cooling conditions.
 - During the radiant system tests there was less heat accumulation in the thermal mass because of the direct coupling between the radiant panels and the concrete pavers on the floor.
2. A new definition for radiant system cooling load (different from air system cooling load) must be used. Radiant heat transfer allows chilled surfaces to directly remove part of radiant heat gains from a zone, thereby bypassing the assumed time-delay effect caused by the interaction of radiant heat gain with non-active thermal mass for air systems.
3. Sensible cooling load calculations for radiant systems should utilize dynamic energy simulation programs or design tools based on a fundamental heat balance approach that properly takes into account how heat gains are removed from a zone by an actively cooled surface.
4. Simplified cooling load calculation methods, such as RTS or weighting factor methods, may lead to incorrect results for radiant systems.
5. All radiant systems are quick to respond to changes in zone heat gains but thermally massive systems (e.g., TABS) are slow to respond to control signal changes.

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