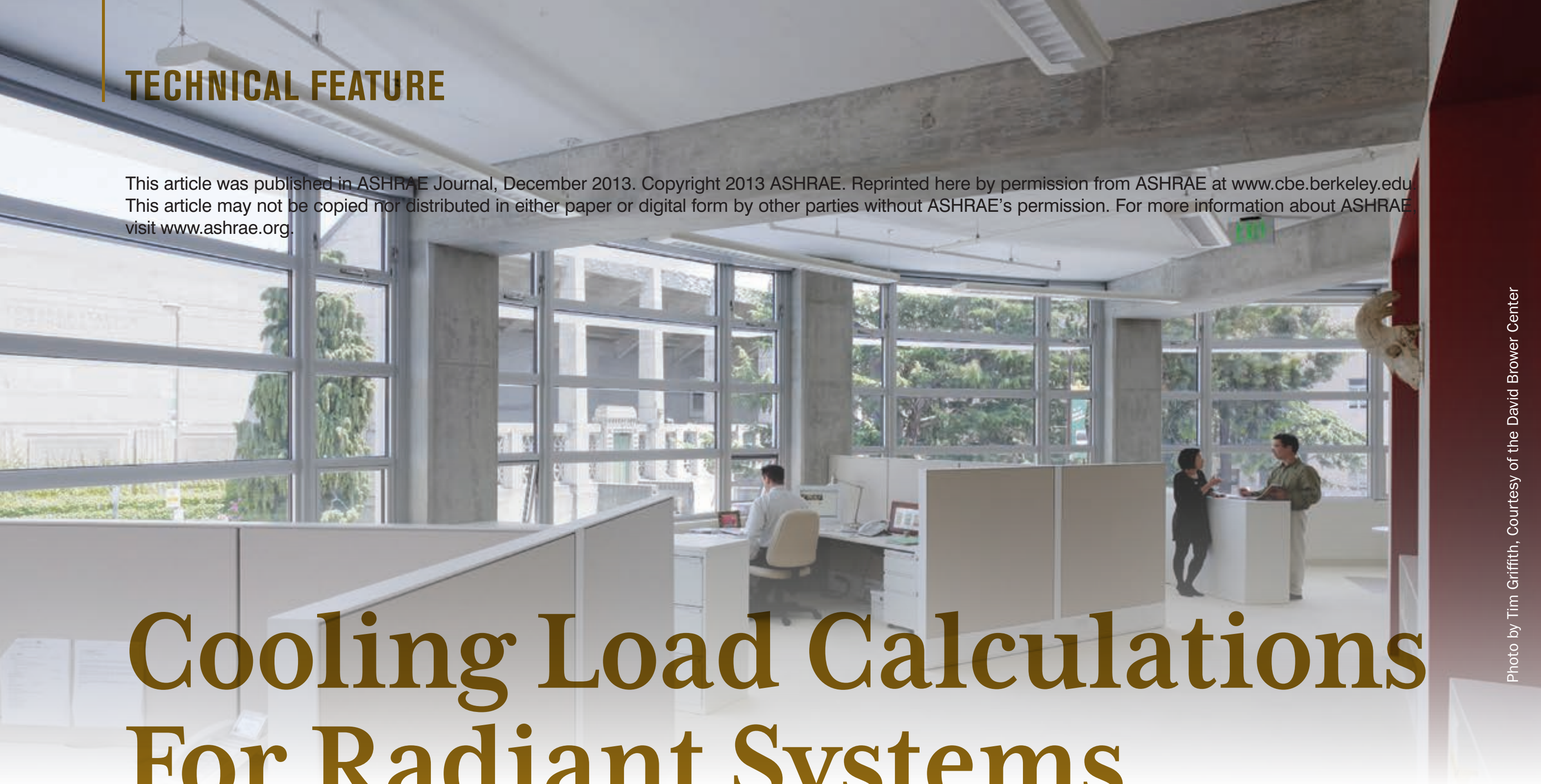

Photo by Tim Griffith, Courtesy of the David Brower Center

Cooling Load Calculations For Radiant Systems

Are They the Same as Traditional Methods?

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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] A database of representative buildings with radiant systems can be found at http://bit.ly/RadiantBuildingsCBE. However, it is difficult to find detailed standardized guidelines for calculating cooling loads for radiant cooling systems, which is the subject of this article.

A radiant system is a sensible cooling and heating system that provides more than 50% of the total heat flux by thermal radiation. There are three primary types of water-based radiant systems: (1) for new construction: plastic tubing (e.g., PEX) embedded in the structural slabs, often referred to as thermally activated building system (TABS); (2) for retrofit or new construction: suspended metal ceiling panels with copper tubing attached to the top surface (radiant ceiling panel, RCP); and (3) for retrofit or new construction: prefabricated or installed-in-place systems consisting of embedded tubing (e.g., PEX, or small, closely spaced plastic tubing "mats") in thinner layers (e.g., topping slab, gypsum board, or plaster) that are isolated (insulated) from the building structure (embedded surface system, ESS).

In this article, we present recent research evidence that sensible zone cooling loads for radiant systems are different (in fact, are often higher) than cooling loads for traditional air systems. This finding has important implications for the proper design and sizing of radiant systems along with the required reduced-sized air distribution system (for ventilation, control of latent loads, and supplemental cooling). Higher peak design cooling loads, however, is not the same as higher overall energy

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LEFT The David Brower Center, Berkeley, Calif., features an exposed radiant ceiling slab (TABS) for both cooling and heating.

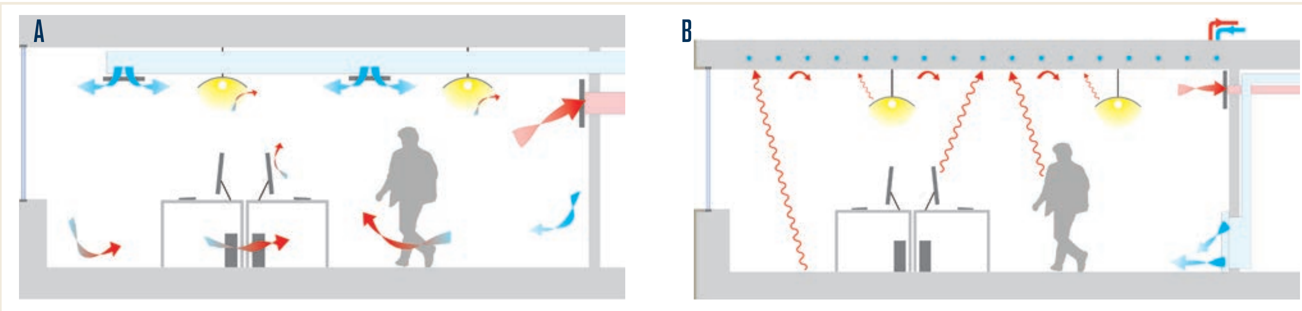

FIGURE 1A (LEFT) Overhead air distribution system. FIGURE 1B (RIGHT) Radiant ceiling cooling system.

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, improved plant side equipment efficiency with warmer cold water temperatures, and, particularly with TABS, the possibility of night precooling using cooling towers.

We begin by reviewing current cooling load calculation methods and then describe the results of simulation and experimental studies addressing the sensible zone cooling load differences between radiant and air systems.

Review of Cooling Load Calculations

Compared to air systems, the presence of an actively cooled surface changes the heat transfer dynamics in a zone of a building (*Figure 1*). 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. Additional details of these differences are discussed later.

There are important limitations in the definition of cooling load for a mixing air system described in Chapter 18 of *ASHRAE Handbook—Fundamentals*[8] when applied to radiant systems. While all internal sensible heat gains are composed of both radiant and convective components, it is only the rate at which convective heat energy is removed from the zone air at a given point in time that contributes to the cooling load. In an all-air system, radiant energy must first be absorbed by the non-active surfaces that enclose the space (floor, walls, ceiling) and objects in the space (e.g., furniture). These surfaces will eventually increase in temperature enough 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. Another time delay phenomenon is conductive heat transfer through opaque massive exterior building surfaces. The two primary methods described by ASHRAE for calculating cooling loads—heat balance (HB), and radiant time series (RTS)—were developed in part to take into account these time delay effects for air systems.

Due to the obvious mismatch between how radiant heat transfer is handled in traditional cooling load calculation methods compared to its central role in radiant cooling systems, ongoing research is examining the fundamentals of energy performance modeling and, in particular, cooling load calculations for radiant cooling systems.

Comparison of Radiant vs. Air System Cooling Loads

Recent simulation and experimental results addressing the differences between radiant and air systems follow.

SIMULATION STUDY

A simulation study was conducted to investigate the impact of the presence of an activated cooled surface on zone cooling loads. Two identical single zone models, one conditioned by an air system and one by a radiant system, were developed in EnergyPlus v7.1[9] for comparison. The test case was a rectangular, heavyweight construction single zone building (26.2 ft [8 m] wide × 19.7 ft [6 m] long × 9 ft [2.7 m] high) with no interior partitions. Both the floor and roof boundary conditions were highly insulated (adiabatic) to simplify the analysis. 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 approach ensures that all energy flows (radiation, convection, and conduction) in each zone are properly modelled, allowing hourly cooling loads for both the air and radiant systems to be calculated. Hourly design day simulations were used for the study. For full details, see Feng, et al.[10]

The simulation study investigated all three radiant system types (RCP, ESS, TABS), and 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). Due to differences in the design and operation of radiant and air systems, the following challenges needed to be addressed to ensure that the simulations produced a fair comparison.

- 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. Although including an air system in the radiant model would be more realistic, for example, increased convective heat transfer along the radiant ceiling due to higher diffuser air velocities, to simplify the analysis, neither the latent load nor ventilation system was simulated.
- The sensible cooling load for an air system is calculated in terms of maintaining a constant zone air temperature at each time step, 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, the simulated radiant system was sized to maintain an acceptable thermal comfort range (not setpoint) during the simulation period. Due to the importance of radiation on occupant comfort, operative temperature (not air temperature) was used as the control temperature for both systems. To ensure equivalent comfort conditions between the two systems, 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.[10]
- The key difference was the zone cooling load definition used for each system model. For the air system, the amount of convective heat removed from the zone air was the cooling load for that time step. For the radiant system, we defined the 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. 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 total of 74 simulations were completed. Two example results for the model configuration with south-facing windows and no external shading are shown for a floor TABS (*Figure 2a*) and ceiling TABS (*Figure 2b*). These results emphasize the impact of solar load, which, compared to other parameters studied, was identified as the factor that had the most significant influence on cooling loads. Compared to the air system, the radiant system cooling

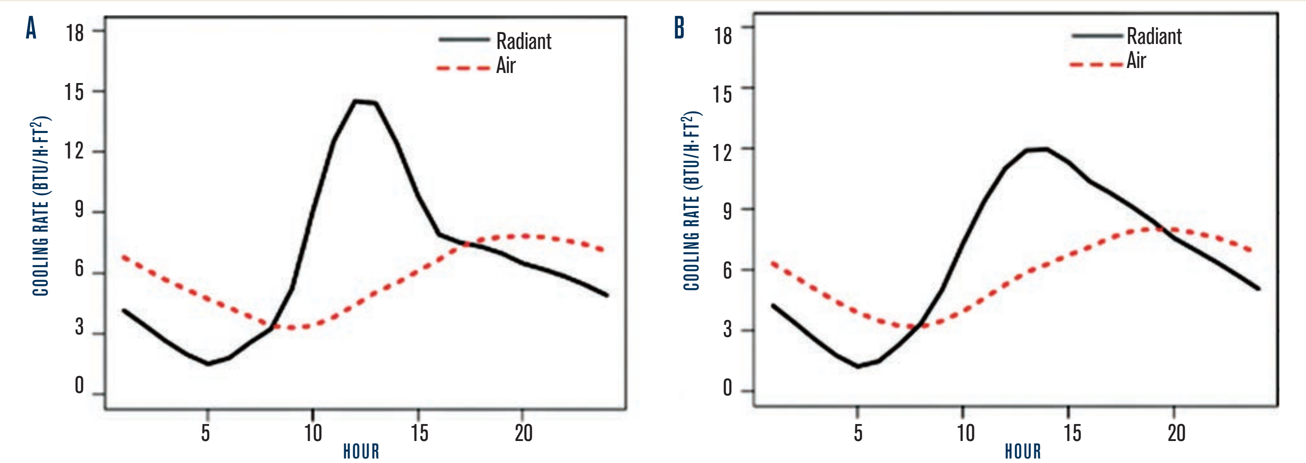

FIGURE 2 Comparison of design day cooling rate profiles between air system and floor TABS (Figure 2A) and ceiling TABS (Figure 2B).

rate, representing the hourly sensible cooling load, reaches a higher peak value for both the floor TABS (85% higher) and ceiling TABS (49% higher). The higher peak cooling load for the floor TABS is indicative of the acknowledged increased cooling capacity of radiant floor cooling when directly illuminated by solar radiation.[3,11,12] Note that the air system results, although similar, are slightly different due to the location of the thermal mass for the two cases.

Summarizing other simulation results when solar was not present, peak cooling rate differences between chilled radiant ceiling systems of all three types and air systems ranged from 12% to 35% higher for perimeter zones and 7% to 27% higher for interior zones.[10] Although the results from this limited and simplified simulation study indicate that peak cooling loads for radiant systems are higher than those for air systems, it is important to recognize that control and operational differences between the two systems may lead to lower cooling loads, energy use and costs for radiant systems. For example, a common control strategy for the thermally massive TABS is to use nighttime pre-cooling to reduce or eliminate active cooling during daytime hours.

To explain why radiant system peak zone cooling rates are higher than those for the equivalent air systems, *Figure 3* compares the heat transfer fundamentals for the two systems for a typical case from the above simulation study. The simulated case shown represents an interior zone with adiabatic walls, floor, and ceiling, so that the only cooling loads are the result of interior heat sources. The figure shows how convective and radiative heat gains are converted into zone cooling load for the air system (on the left) and a radiant cooling panel system (on the right).

In this example, the total internal heat gain (4.8 Btu/h·ft^2 [15 W/m^2]) during occupied hours, 6 a.m. to 6 p.m.) was divided into convective heat gain (1.9 Btu/h·ft^2 [6 W/m^2]) and radiative heat gain (2.9 Btu/h·ft^2 [9 W/m^2]), representing a typical 60% radiation factor. For the air system, 100% of the convective heat gain instantaneously becomes cooling load, while a large portion of radiative gains are absorbed by zone thermal mass and released after a time delay 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 system, not all convective gains instantaneously become cooling load. During occupied hours, part of the convective heat gain contributes to a higher zone air temperature which is reached to balance the cooler ceiling surface temperature, thereby maintaining an equivalent operative temperature in the zone. 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.

As shown, a significantly larger portion of the radiative heat gain converts directly to cooling load during the occupied period due to the presence of the actively cooled surface. The bottom plots add up the two cooling load components, and the solid black lines in the bottom plots represent hourly cooling loads, which reach their peak value at the end of the occupied period for both systems. These predicted cooling load profiles display the total amount of heat being removed by each system to maintain the same operative temperature profile. Note that for this example, the peak cooling rate for the radiant system is predicted to be 13% higher than that for the air system.

EXPERIMENTAL STUDY

An experimental study was undertaken to verify the observation that sensible zone cooling loads for radiant systems are different from air systems. The tests were carried out in a climatic chamber (14 ft [4.27 m] × 14 ft [4.27 m] × 9.8 ft [3.0 m]) that has been used for standard radiant cooling panel testing. The high insulation level of walls, floor and ceiling makes the room almost adiabatic. The test chamber (*Figure 4*) was configured to include a 1.5 in. (0.04 m) layer of concrete pavers covering most of the floor to provide thermal mass (3,000 lbs [1350 kg]). 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 blocks. The radiant ceiling panels were well insulated on the top surface.

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 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 75.2°F (24°C). The test was started when the heater was turned on and maintained at a constant value for six hours. After six hours, the heater was turned off and the experiment continued for another six hours. The heater schedule is shown in *Figure 5*. For the entire duration of each test, the radiant or air system was controlled to maintain a 75.2±0.9°F (24 ±0.5°C) operative temperature in the chamber. 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. Full details of the experiment are described by Feng, et al.[13]

Two series of tests were conducted, one at a heat input level of 18.8 Btu/h·ft^2 (59.3 W/m^2), and one at 26.1 Btu/h·ft^2 (82.4 W/m^2). *Figure 5a* compares the instantaneous cooling rates for the radiant and air systems during the 18.8 Btu/h·ft^2 (59.3 W/m^2) test. The shaded area around each measurement line indicates the experimental uncertainty. The cooling rate of the radiant panel system reached an average of 14 Btu/h·ft^2(44.3 W/m^2) between hours 1 and 2, and at that time it was about 48% higher than the air system case. The radiant cooling rate slowly ramped up over the next four hours until it reached an average of 16.9 Btu/h·ft^2 (53.2 W/m^2) in the last hour before the heater was turned off. This value was about 18% higher than the air system case. Similar trends were observed for the 26.1 Btu/h·ft^2 (82.4 W/m^2) test.

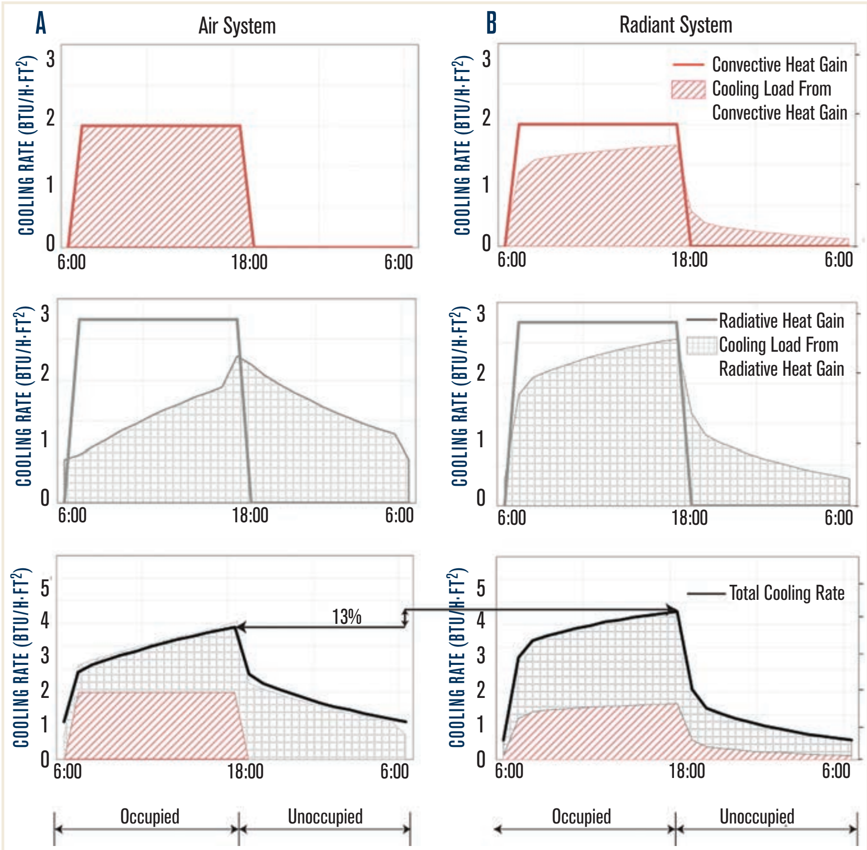

FIGURE 3 Conversion of convective and radiative heat gain into cooling load for air system (Figure 3A, left) and radiant cooling panel system (Figure 3B, right).

Key findings from the experiment include:

- The radiant system has a higher cooling rate than the air system, meaning that it is faster to remove heat gains while maintaining equivalent comfort conditions.
- For the tested cases, 75% to 82% of the total heat gain was removed by the radiant system during the period when the heater was on, while for the air system, 61% to 63% was removed.
- For the radiant system, because of the direct radiant exchange between radiant cooling panels and the

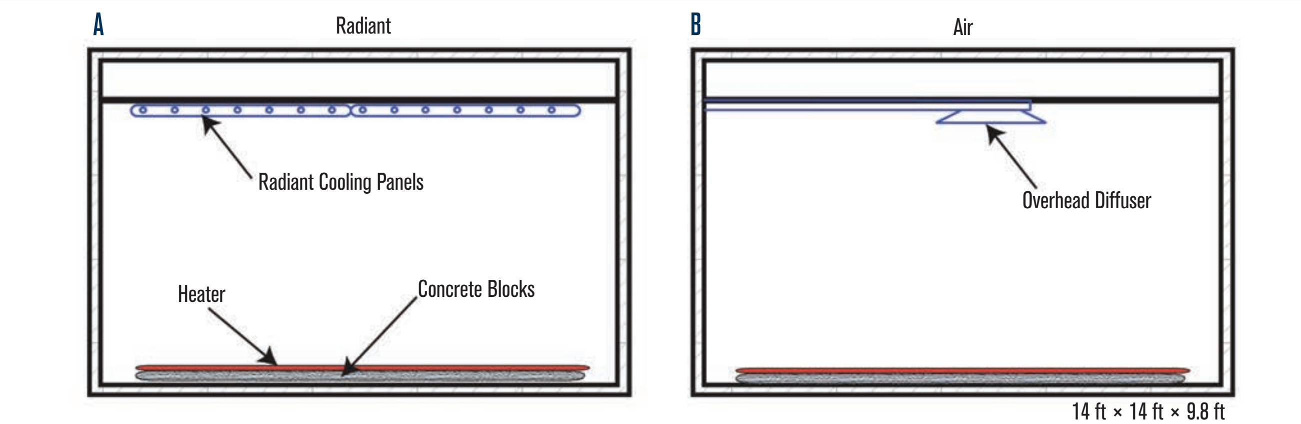

FIGURE 4 Laboratory test configurations for radiant panel (Figure 4A) and overhead air-distribution systems (Figure 4B).

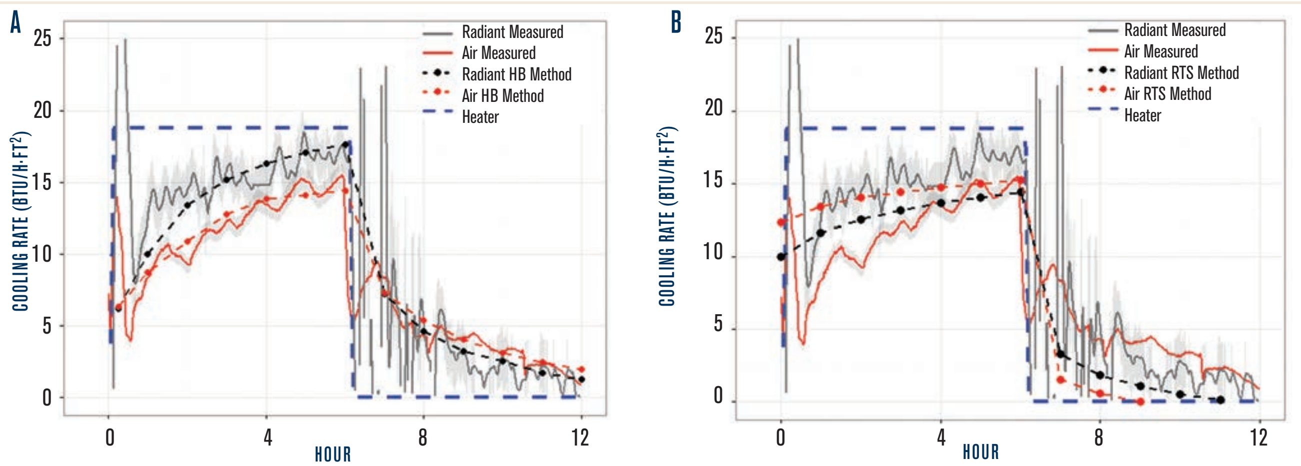

FIGURE 5A (LEFT) Comparison of measured and predicted instantaneous cooling rates using heat balance (HB) method for radiant and air systems: 18.8 Btu/h·ft^2 test.
FIGURE 5B (RIGHT) Comparison of measured and predicted instantaneous cooling rates using radiant time series (RTS) method for radiant and air systems 18.8 Btu/h·ft^2 test.

concrete blocks, there is less heat accumulation in the thermal mass.

TABS or other thermally massive radiant systems are known to respond slowly to control signals. However, as shown above, 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.

To assess the accuracy of current cooling load calculation methods when applied to radiant systems, we compared the same measured results with predicted instantaneous cooling rates using the fundamental HB method (*Figure 5a*) and simplified RTS method (*Figure 5b*). An EnergyPlus v8.0 model matching the test chamber was developed to apply the HB method. As discussed above, the cooling load for the radiant system was defined as the heat removed by the radiant ceiling panels. With this revised definition of radiant cooling load, *Figure 5a* shows good agreement between measured and predicted cooling rates for both radiant and air systems.

However, *Figure 5b* 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 as convective loads after a time delay, the RTS method underpredicts 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 overprediction of the air system cooling load by the RTS method.

Conclusions

The simulation and experimental research results presented in this article have demonstrated that sensible zone cooling loads for radiant systems are not the same as those for traditional air systems. 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 signals. A new definition for radiant system zone cooling load must be developed and used. Cooling load calculations for radiant systems should use the ASHRAE heat balance method. The RTS or weighting factor methods may lead to incorrect results for radiant systems.

Acknowledgments

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