Radiant Slab Cooling for Retail

By **Ian Doebber,** Associate Member ASHRAE; **Mike Moore, P.E.,** Associate Member ASHRAE; **Michael Deru, Ph.D.,** Member ASHRAE

Most large retail spaces are ventilated, cooled, and heated by all-air systems via constant air volume (CAV) packaged rooftop units (RTUs). CAV RTUs are low first cost, easy to install, and straightforward to maintain. However, designers have begun to decouple ventilation from occupant comfort conditioning. One strategy pairs a dedicated outdoor air system (DOAS) for ventilation and dehumidification with a radiant floor system for sensible cooling.

This combination provides greater design and control flexibility to reduce energy consumption, shift peak loads, increase thermal comfort, and improve HVAC acoustics.[1,2,3] The following case study summarizes the design, field installation, control, and anticipated performance of a radiant-cooled floor and DOAS in a big box retail store.

Project Background

Over the past seven years, Walmart has built a line of high-efficiency stores. A sixth-generation high-efficiency store opened in Sacramento, Calif., in June 2009. The store combined established best practices from previous iterations with an HVAC system designed with the following performance objectives:

- Maximize evaporative cooling;
- Decouple ventilation from occupant comfort conditioning;
- Shift peak cooling; and
- Improve thermal comfort.

About the Authors

Ian Doebber, National Renewable Energy Laboratory (NREL), served as consulting engineer on the project while working for ARUP. **Mike Moore, P.E.,** is with Newport Ventures in Schenectady, N.Y. He is a member of ASHRAE SSPCs 62.2 and 90.2. **Michael Deru, Ph.D.,** NREL, worked with Walmart to monitor the performance of the installation. He is chair of TC 7.6, Building Energy Performance.

The main construction objective was affordable scalability, necessary to implement the design in a fleet of new stores in hot, dry climates.

Why Radiant Floor Cooling?

The project engineers designed a radiant-cooled floor with a DOAS because it satisfied the performance and construction objectives.

Maximize Evaporative Cooling

All-air systems typically supply 55°F (12.8°C) air that requires 45°F (7.2°C) refrigerant or chilled water. Radiant-cooled floor systems typically maintain a 68°F to 72°F (20°C to 22.2°C) floor surface temperature[2] that requires chilled water supply of 55°F to 60°F (12.8°C to 15.6°C). The warmer supply temperature increases system efficiency by (1) extending the operating range for water-side economizing; and (2) enabling a warmer refrigerant suction temperature under chiller operation.

Decouple Ventilation from Occupant Conditioning

Radiant floor cooling can meet most or all of the sensible cooling demand in a retail store. A separately controlled DOAS can supply ventilation air at space-neutral conditions or cooled to meet any latent and/or additional sensible load.

Shift Peak Cooling

Demand charges and time-of-use tariffs can significantly increase electricity bills. Radiant floor cooling can leverage thermal mass to offset peak demand without compromising thermal comfort. A morning charge at lower dry-bulb and wet-bulb conditions further capitalizes on water-side economizing or more efficient chiller operation with lower condensing temperatures.

Improve Thermal Comfort

Thermal comfort models such as the Fanger Method[4] account for the role radiative heat exchange plays in thermal comfort.[5] The Fanger Method calculates the *predicted percentage dissatisfied* (PPD) metric; the percentage of occupants who will be uncomfortable under given conditions.[6] Standard design practice attempts to maintain an overall PPD for general comfort at less than 10% (greater than 90% satisfaction).

	All-Air System	Radiant-Cooled Floor and DOAS	Radiant-Cooled Floor and DOAS
PPD	7%	5%	8%
MRT	76°F	72°F	72°F
Dry-Bulb Temperature	76°F	76°F	78°F
Relative Humidity	50%		
Metabolic Rate	1.7 met (Walking Around)		
Clothing Insulation	0.5 clo (ASHRAE Standard 55-2004[7] Summer)		
Air Speed	70 fpm (Walking Around)		

Table 1: Thermal comfort comparison.[5]

	Standard Radiant-Cooled Floor	Installed Radiant-Cooled Floor
Tube Spacing	6 in. to 9 in. O.C.	6 in. O.C.
Tube Diameter	5/8 in.	1/2 in.
Loop Length	300 ft	260 ft
Tube Depth	1.5 in. to 2 in. Below Slab Surface	Bottom of Slab
Slab Thickness	6 in.	4 in.
Edge and/or Sub-Slab Insulation	0 in. to 1 in. Foam Board	None
Supply Water Temperature	55°F to 60°F	58°F
Water Temperature Rise	5°F to 9°F	5°F
Maximum Flow Rate	1.2 gpm/Loop	0.79 gpm/Loop

Table 2: Standard versus installed radiant-cooled floor design.

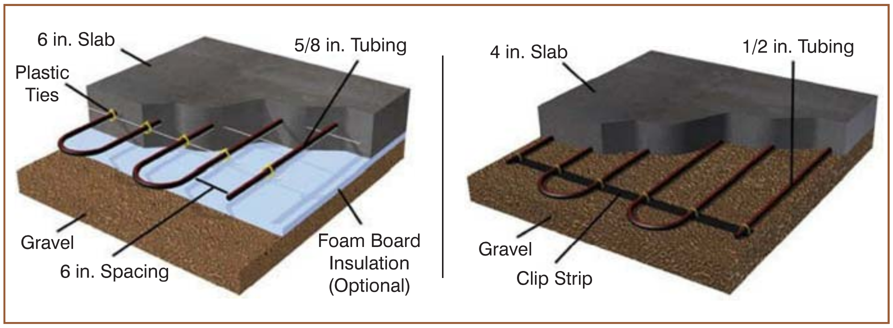

Figure 1: Standard radiant-cooled floor (left) and installed radiant-cooled floor (right).

The radiant floor and DOAS actively regulate both dry-bulb temperature and mean radiant temperature (MRT); an all-air system actively regulates dry-bulb temperature only. *Table 1* shows how a lower MRT of 72°F (20°C) can meet the comfort criteria at a higher dry-bulb temperature of 78°F (26°C).

Scalability

Over the past few years, much effort has been expended to make radiant floors scalable and economical. The focus has been on reducing the labor required to lay and fasten the tubing in place. A radiant floor OEM worked with the project engineers to develop a preconfigured module to reduce installation time.

Condensation Considerations

Condensation was not considered an issue for three reasons: (1) Sacramento is in a dry climate (ASHRAE Climate Zone 3B); (2) the internal latent loads are minimal (occupants are the main source of moisture); and (3) the grocery section demands the DOAS maintain a 58°F (14.4°C) or less store dew point based on maintaining 75°F (23.9°C) dry bulb and 55% relative humidity for refrigerated cases.

Radiant Floor Design

The project engineers used standard design practices, whole-building energy simulation, and finite element analysis in designing the system. The final design, labeled *installed radiant floor* (*Table 2*, Page 29), met a sensible cooling demand of 15.8 Btu/h·ft^2 (49.8 W/m^2) with a 66°F (18.9°C) floor surface temperature.

Figure 1 (Page 29) shows how the installed radiant floor reduced first costs over a standard radiant floor by reducing slab thickness, eliminating under-slab insulation, and specifying smaller diameter tubing resting directly on the compacted gravel base. The following subsections detail the process of designing a cooling only radiant floor for a hot, dry climate.

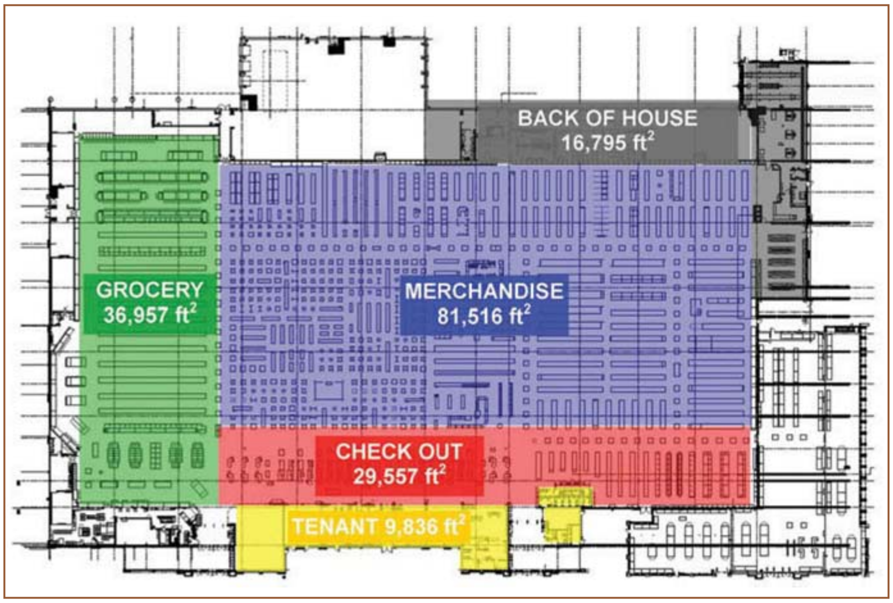

Figure 2: Retail store thermal zoning.

Zoning

The design began by dividing the retail store into five thermal zones (*Figure 2*), two of which incorporated radiant cooling.

Merchandise Zone

- Initial load calculations determined the peak sensible cooling load would be less than 16.0 Btu/h·ft^2 (50.5 W/m^2), within the range of radiant floor cooling.[2]
- The cooling demand was uniform across the entire area, allowing for a single radiant floor controlled zone.
- The rectangular geometry permitted specification of preconfigured, scalable radiant tubing modules to reduce installation and balancing time.

Checkout Zone

- This zone had the same operating schedule and simple geometry as the merchandise zone, but needed additional sensible cooling capacity provided by its own DOAS to offset the infiltration loads from the entryways.

The radiant floor was not selected for the back of house and tenant zones because of highly variable loads and challenging floor geometries. The grocery zone needed minimal sensible cooling because of the refrigerated cases and could maintain occupant comfort conditioning with its DOAS.

Meeting Most or All of the Sensible Cooling Load

The design intent in the checkout and merchandise zones was to meet as much of the sensible cooling load as possible with the radiant floor. The average floor surface temperature drives the cooling capacity through convective heat transfer with the air and long wave radiative heat exchange with the other surfaces, especially the ceiling. National and international standards recommend a minimum average floor surface temperature of 64°F to 66°F (17.8°C to 18.9°C).[2]

Cursory hand calculations based on established floor heat transfer coefficients,[7] 0.97 Btu/h·ft^2·°F (5.5 W/m^2·K) long wave radiation and 0.26 Btu/h·ft^2·°F (1.5 W/m^2·K) convection, provided initial floor cooling capacity estimates. Equation 1 presents an example calculation for a 66°F (18.9°C) average floor surface temperature assuming 76°F (24.4°C) space dry bulb and 78°F (25.6°C) roof/wall surface temperatures. Using the same calculation, *Table 3* provides the peak cooling capacity at different average floor surface temperatures.

$$\underbrace{0.97\frac{\text{Btu}}{\text{h}-\text{ft}^2-{}^\circ\text{F}}\left(78^\circ\text{F}-66^\circ\text{F}\right)}_{\textbf{Long Wave Radiation}}+\underbrace{0.26\frac{\text{Btu}}{\text{h}-\text{ft}^2-{}^\circ\text{F}}\left(76^\circ\text{F}-66^\circ\text{F}\right)}_{\textbf{Convection}}=14.2\frac{\text{Btu}}{\text{h}-\text{ft}^2-{}^\circ\text{F}} \quad (1)$$

Average Floor Surface Temperature	Peak Cooling Capacity
64°F	16.7 Btu/h·ft^2
66°F	14.2 Btu/h·ft^2
68°F	11.8 Btu/h·ft^2
70°F	9.3 Btu/h·ft^2
72°F	6.9 Btu/h·ft^2

Table 3: Peak radiant floor cooling capacity for different average floor surface temperatures, based on established floor heat transfer coefficients[7] and assuming 78°F (25.6°C) roof/wall surface temperatures and 76°F (24.4°C) space dry bulb.

Estimates in *Table 3* are conservative because they neglect the significant short-wave solar gain through the skylights. A radiant floor's cooling capacity increases by absorbing incident short-wave radiation before that solar load can convect into the space (total cooling capacity can exceed 32 Btu/h·ft^2 [101 W/m^2] with significant direct solar radiation on the floor).[2] The team used a whole-building energy modeling program[8] to capture the convective, long-wave radiative, and short-wave radiative thermal interactions between the floor surface and internal/envelope loads. Additionally, understanding the significant long-wave radiative interaction between the floor and ceiling was of critical importance. The goal was to determine the minimum average floor surface temperature that would ensure the radiant floor would meet most or all of the sensible cooling load throughout the 1.0% wet-bulb design day shown in *Figure 3*. *Table 4* summarizes the model inputs.

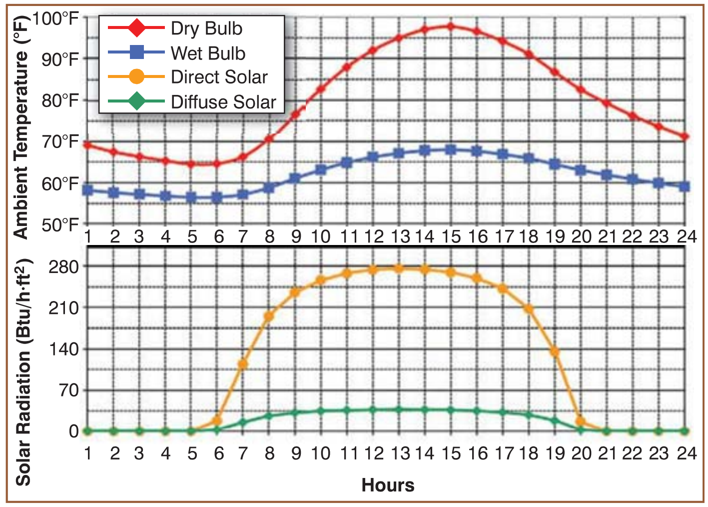

Figure 3: Sacramento, Calif., 1% wet-bulb design day.

The design team did not account for the transient behavior of a radiant-cooled floor when sizing the system, conservatively disregarding slab and ground thermal mass benefits.[9] Instead, the radiant-cooled floor was artificially modeled to have an accelerated response time, much like an all-air system. For each hour of the design day, the floor surface temperature was artificially modeled to instantly react, achieving a floor surface temperature to exactly offset the sensible load and maintain the space dry-bulb temperature between 76°F and 78°F (24.4°C and 25.6°C).

Component	Model Inputs
Lighting	1.0 W/ft^2 (42% Long Wave/18% Visible/40% Convective) Dimmable to 30% Light Output to Meet 50 Horizontal Footcandles at 2.5 ft Above Floor Surface
Plug loads	0.65 W/ft^2 (50% Long Wave/50% Convective)
Occupancy	350 Peak Occupancy at 550 Btu/h·occupant (13% Long Wave/31% Convective/59% Latent)
Infiltration	0.11 cfm/ft^2 of Exterior Wall Surface Area (Supermarket "Good Practice" at 0.05 in. w.c.)[10]
Roof	R-30 Continuous Insulation With 50% Reflective Exterior Surface
Wall	Adiabatic (Merchandise and Checkout Zones are Surrounded by Conditioned Spaces)
Skylight	4.2% Skylight-to-Floor Fraction SHGC = 0.49 and U-Factor = 0.82 Btu/h·ft^2·°F
Internal mass	2.0 lb/ft^2 Across Entire Floor Area

Table 4: Model inputs.

Figure 4 (Page 33) shows the merchandise zone sensible loads on a design day. The radiant floor columns indicate heat absorbed by the floor based on artificially set hourly floor temperatures as shown in *Figure 7*, Page 34 (disregarding mass effects). Using this methodology, simulation results indicated that the radiant-cooled floor design needed to maintain a 66°F (18.9°C) surface temperature during peak hours while providing 15.8 Btu/h·ft^2 (49.8 W/m^2) of cooling. This conservative sizing strategy was used because the design team believed that there were too many unknowns (e.g., slab-to-ground thermal interaction and ground thermal properties) to leverage thermal mass to downsize the system.

Designing to Meet the Peak Cooling Capacity

The following parameters were configured to maintain a 66°F (18.9°C) average floor surface temperature while providing a 15.8 Btu/h·ft^2 (49.8 W/m^2) peak cooling rate.

- **Tube spacing.** Typical spacing is 6 in. or 9 in. (152 mm or 229 mm) on center (O.C.). The team chose 6 in. (152 mm) O.C. to use the warmest fluid temperature to meet the cooling load.
- **Tube diameter.** Typical outside diameters are 1/2 in. and 5/8 in. (13 mm and 16 mm). Although 5/8 in. (16 mm) would have provided a slightly greater cooling capacity, the marginal performance benefit was not sufficient to override the incremental cost.
- **Tube length.** Standard practice provides that the loop lengths should be within 300 ft (91 m) to keep hydraulics within reasonable circulator selections. Loop length was specified to be 260 ft (79 m).
- **Tube depth.** Standard practice is a 1.5 in. to 2 in. (38 mm to 51 mm) depth to be shallow enough for dynamic response while avoiding cracked concrete. These depths were not possible because (1) the tubing had to be at least 3 in. (76 mm) deep to avoid being punctured by 2 in. (51 mm) bolts used for shelving; and (2) suspending the tubing required either using chairs (cost prohibitive) or connecting to steel reinforcement (which the slab did not have). The tubing was placed at the bottom of the slab to minimize first costs.
- **Slab thickness and insulation.** The design team questioned whether increasing the slab thickness from 4 in. to 6 in. (102 mm to 152 mm) would augment the thermal mass benefits of

precharging the slab in the mornings and floating during peak hours. Yet, could the ground under a 4 in. (102 mm) slab provide thermal mass benefits similar to an additional 2 in. (51 mm) of concrete, especially with the tubing located at the bottom of the slab? If so, sub-slab insulation would be counterproductive because it would resist the "free" thermal mass of the ground.

The transient behavior of the slab was modeled[9] with a whole-building energy simulation program[8] to evaluate the thermal mass effects of a 4 in. (102 mm) slab without insulation versus a 6 in. (152 mm) slab with insulation. The sub-slab was modeled as a 3.3 ft (1 m) layer of clay/silt: 0.51 Btu/h·ft·°F (0.88 W/m·K), 75 lb/ft^3 (1201 kg/m^3), and 0.52 Btu/lb·°F (2176 J/kg·K). The bottom of the ground layer was assumed to be adiabatic. Results showed that the uninsulated 4 in. (102 mm) slab and ground had at least the same available thermal mass benefits as the insulated 6 in. (152 mm) slab. For this reason, the 4 in. (102 mm) uninsulated slab was specified.

- **Flow rate.** Standard practice in fluid flow provides for a minimum velocity to establish turbulent flow and ensure sufficient convective heat transfer.[2] Based on the typical temperature rise of 5°F to 9°F (2.8°C to 5.0°C),[2] 5°F (2.8°C) was specified to maintain turbulence and enable a warmer supply temperature to maximize waterside economizing at the expense of increased pumping energy.
- **Supply-return temperatures.** With the aforementioned design parameters fixed, a *steady-state* finite element analysis calculated that a 58°F (14.4°C) supply and 63°F (17.2°C) return would maintain a 66°F (18.9°C) floor surface temperature while providing a 15.8 Btu/h·ft^2 (49.8 W/m^2) cooling rate.

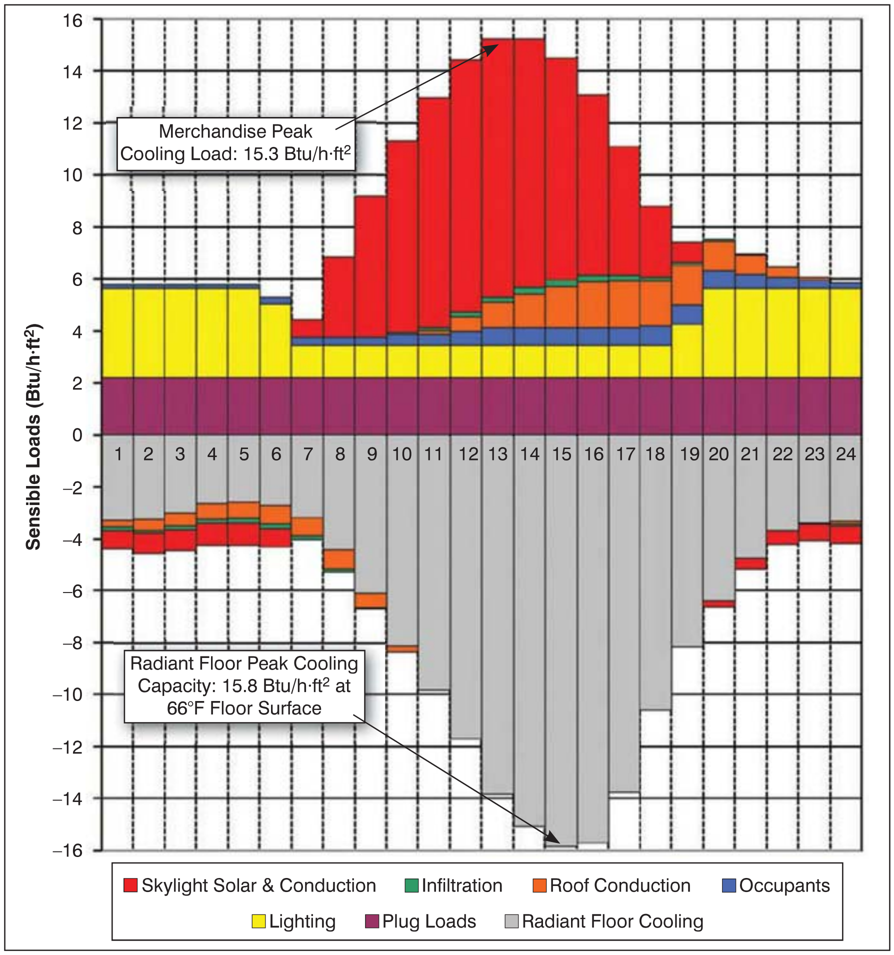

Figure 4: Whole-building simulation of the merchandise zone across the 1.0% wet-bulb design day with the floor surface temperature controlled each hour to offset each hourly sensible load. *Figure 7* shows the hourly floor surface temperatures.

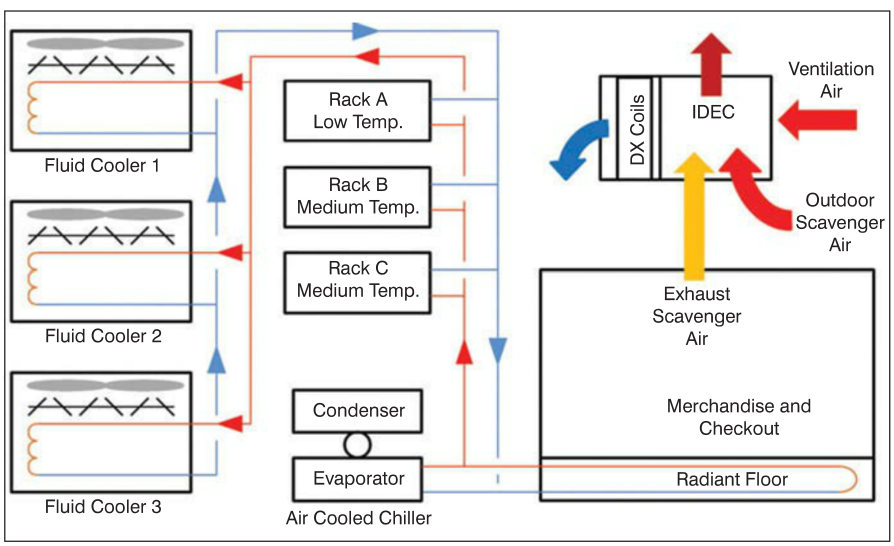

Figure 5: Mechanical system schematic.

Mechanical System

Figure 5 shows how the radiant floor integrated into the mechanical and refrigeration systems. Three fluid coolers rejected heat from the refrigeration racks and provided waterside economizing for the radiant floor. An air-cooled chiller conditioned the radiant floor when not water-side economizing. Merchandise and checkout zones were ventilated by four DOAS units that incorporated indirect evaporative cooling (IDEC) and supplemental direct exchange (DX) cooling. The IDEC used evaporatively cooled scavenger air (mixture of exhaust air and outdoor air). Each DOAS could air-side economize by increasing the outdoor air supply beyond the minimum ventilation requirements.

Radiant Floor Control

Although the radiant-cooled floor was sized independent of thermal mass benefits, the control strategy could still leverage thermal mass. The control strategy had to balance water-side economizing and warmer supply temperatures against excessive pumping, fluid cooler operation, and overcooling. Due to the complexity of this issue, a whole-building energy model[11] was used to explore various control strategies. All findings within this section are based on building energy model outputs, and not actual performance data. Data are being collected to verify modeling results.

The design team initially modeled a constant flow-constant temperature strategy. When the radiant floor went from the "off" position to supplying a constant 613 gpm (2.32 m^3/min) at 58°F (14.4°C), the slab—and eventually space dry-bulb and MRT—cooled too quickly. By the time the dry-bulb thermostat was satisfied, turning the pump off, the inertia of the thermal mass within the slab overcooled the space.

Constant flow-variable temperature strategies were explored using various relationships between supply temperature and space dry-bulb. The results showed either overcooling and short cycling or excessive pumping and fluid cooler operation.

After many iterations, building simulations predicted that a variable flow-variable temperature strategy would provide the best performance. *Figure 6* shows the variable-flow part of the strategy. The radiant floor would initiate at 20% of the design flow when the space experienced minimal sensible loads. As the sensible loads increased (represented by an increasing space dry-bulb temperature), the flow rate would increase linearly. The simulation showed that this control strategy would mitigate the peak cooling demand so the flow rate would never have to exceed 60% of design flow as shown in red in *Figure 6*. To be on the safe side, the design team implemented a more conservative control shown in green in *Figure 6* that achieves the maximum design flow rate when the space dry bulb reaches 78°F.

The variable temperature was based on staging the fluid coolers and chiller. When the space dry-bulb temperature was lower than 78°F (26°C), the fluid coolers conditioned the radiant floor, maintaining a 63°F (17.2°C) supply temperature. When the space dry-bulb reached 78°F (26°C) or the fluid coolers could not maintain 63°F (17.2°C), the chiller conditioned the radiant floor based on the reset schedule in *Table 5* until the space dry-bulb temperature dropped below 76°F (24.4°C).

Figure 7 shows how the floor surface and cooling provided to the space changes based on a control strategy accounting for or disregarding thermal mass. Red lines represent the artificial control of the floor surface temperature that exactly offsets the

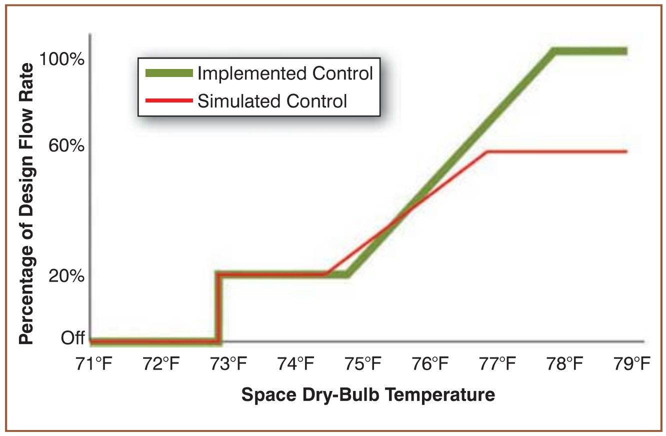

Figure 6: Radiant floor variable flow control strategy.

Radiant Floor Return	Chilled Water Setpoint
≥63°F	54°F
62°F	55°F
≤61°F	56°F

Table 5: Chiller supply water reset schedule.

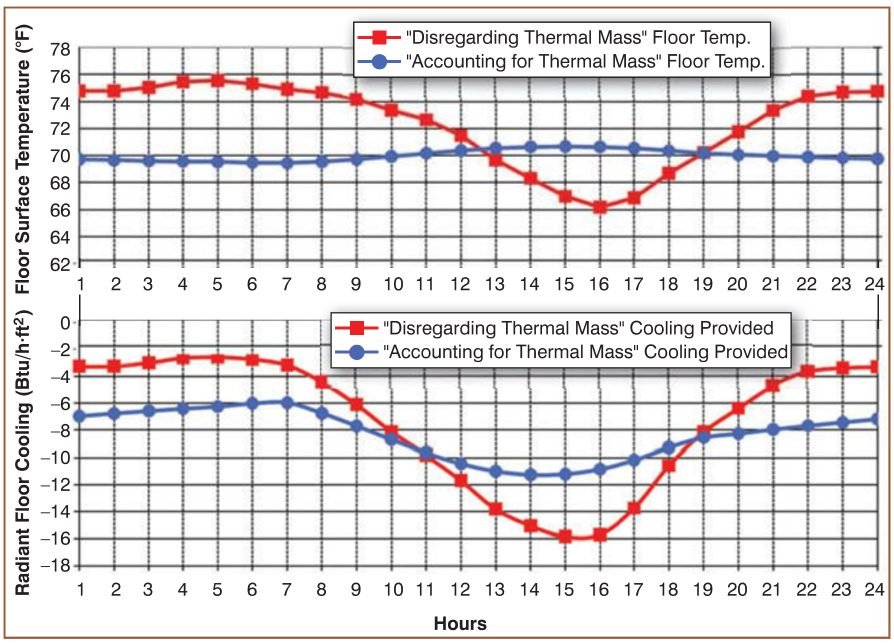

Figure 7: Comparison of radiant floor cooling using a control strategy that accounts for or disregards thermal mass across the design day.

	Standard Efficiency CAV DX RTUs	High Efficiency VAV DX RTUs	Radiant Floor DOAS: Constant Flow–Variable Supply Temperature	Radiant Floor DOAS: Variable Flow–Variable Supply Temperature
DX and Chiller	189,855	125,866	41,365	32,916
Pumps	–	–	22,728	16,163
Fluid Coolers	–	–	121,302	61,810
Fans	247,914	217,964	78,838	73,240
Total HVAC	437,769	343,830	264,233	184,130
Savings Over CAV Baseline (%)	0%	21%	40%	58%

Table 6: HVAC annual electrical energy consumption (kWh).

cooling load each hour of the design day (method used to size the radiant floor system, disregarding thermal mass). Blue lines represent the implemented variable flow-variable temperature control strategy that accounts for thermal mass benefits. Note how accounting for thermal mass maintains the floor surface temperature between 69°F and 71°F (20°C and 22°C) compared to varying between 66°F and 76°F (19°C and 24°C) when disregarding thermal mass. For the radiant floor cooling, accounting for thermal mass requires only 11.2 Btu/h·ft^2 (35 W/m^2) of peak cooling since additional cooling is provided in the morning and late at night to augment peak hours.

Performance

Four HVAC system configurations were simulated to quantify energy performance implications. All the non-HVAC parameters such as building envelope, lighting, and refrigeration were kept constant.

- Baseline 1: Standard efficiency CAV DX RTUs
- Baseline 2: High Efficiency VAV DX RTUs
- Radiant floor, DOAS: constant flow–variable supply temperature
- Radiant floor, DOAS: variable flow–variable supply temperature

Table 6 shows the projected radiant floor/DOAS system energy savings over all-air systems and the importance of proper control to maximize those savings.

Field Installation

Simulations supported proof of concept of a 4 in. (102 mm) uninsulated radiant-cooled slab that would conserve energy while meeting thermal comfort requirements. Walmart still had valid concerns, however, about the first costs and the impact of the installation on construction costs. For example, one laborer can typically install 1,000 to 1,250 linear ft (305 to 381 m) of tubing during an 8 hour shift, so installing 200,000 linear ft (60 960 m) of tubing would have been cost prohibitive.

Project engineers worked with a radiant floor original equipment manufacturer to develop a scalable, preconfigured tubing module that could be installed quickly. The module came in 5 ft to 6 ft wide sections of 0.5 in. outside diameter (13 mm) tubing spaced at 6 in. (152 mm) outer diameter (OD) in customizable lengths. Before the 4 in. (100 mm) slab was poured, modules were rolled onto the compacted gravel


Photo courtesy of Viega

Photo 1: Scalable, preconfigured tubing modules reduced first costs of the radiant-cooled slab.

base, configured in parallel, staked at regular intervals, and stubbed to manifolds. About 188 labor hours per 10,000 ft^2 (929 m^2) of floor area were saved. Balancing requirements were reduced because the modules were uniform. Overall, Walmart realized 60% to 75% in labor first-cost savings over a traditional radiant-cooled slab distribution system.

Conclusion

Design

- The design of a radiant-cooled slab for large retail buildings can be optimized for occupant comfort, energy conservation, peak load shifting, and constructability using commercially available analysis tools and systems. The approach used in this project can serve as an example for similar projects where radiant-cooled slabs are specified.
- For spaces with refrigerated cases, design dew-point temperatures are typically dictated by concern for frost accumulation on the cases' evaporator coils and not by radiant slab temperature.
- Control strategy of the radiant floor and accompanying DOAS is central to realizing improved occupant comfort, energy conservation, and peak load shifting

Performance

- Energy conservation for the radiant floor and DOAS versus a standard efficiency CAV system is predicted to exceed 50% when controlled properly.
- The radiant floor cooling system is expected to oper-

ate at a higher dry-bulb cooling setpoint than an all-air system (e.g., 78°F vs. 76°F [25°C vs. 24°C]) at a comparable comfort level.

Affordability

- Specifying a modular radiant slab tubing system reduces installation time and can save 60% to 75% on radiant distribution system labor costs.

Data Analysis

Data is being collected on energy use, peak power, slab surface temperature, sub-slab soil temperatures, slab water supply and return temperatures, ambient temperature, and merchandise and checkout zone dry-bulb temperatures to compare actual building performance versus projections. The data has not yet been released for analysis; however, Walmart has noted that the radiant slab is performing “better than expected.” The whole-building simulation provided a first-pass comparison between the control strategies, and retrocommissioning is planned to evaluate different control strategies on the building and HVAC system.

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