PG&E's Emerging Technologies Program ET14PGE8581

Experimentally–Determined Characteristics of Radiant Systems for Commercial Buildings

ET Project Number: ET14PGE8581

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Issued: **February 20, 2017**

Pacific Gas and Electric Company®

ACKNOWLEDGEMENTS

Pacific Gas and Electric Company's Emerging Technologies Program is responsible for this project. It was developed as part of Pacific Gas and Electric Company's Emerging Technology - Technology Introductory Support program under internal project number ET14PGE8581. The Lawrence Berkeley National Laboratory conducted this technology evaluation for Pacific Gas and Electric Company with overall guidance and management from Jeff Beresini. For more information on this project, contact ETInquiries@pge.com.

The authors wish to thank Fred Bauman (UC Berkeley), Jason Kirkpatrick (Interface Engineering) and Cindy Regnier for helpful review comments.

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ABBREVIATIONS AND ACRONYMS

ASHRAE	American Society of Heating, Refrigeration and Air-Conditioning Engineers
AUST	Averaged Uncontrolled Surface Temperature
CEC	California Energy Commission
CBECS	DOE's Commercial Building Energy Consumption Survey
CCD	Cooling Degree Days
DOE	US Department of Energy
EPIC	Electric Program Investment Charge (CEC)
gpm	Gallons Per Minute
HVAC	Heating, Ventilation and Air-Conditioning
IOU	Investor-Owned Utility
ISO	International Organization for Standardization
LBNL	Lawrence Berkeley National Laboratory
PG&E	Pacific Gas and Electric Company
RCP	Radiant Cooling Panel System
RCS	Radiant Cooling Floor Slab System
RFS	Radiant Floor Slab System
RP	Radiant Panel System
RTD	Resistance Temperature Detector
VAV	Variable-Air-Volume

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EXECUTIVE SUMMARY

PG&E operates a wide range of building energy efficiency programs designed to meet its energy and demand goals, both for the program cycles and in the longer term, for the big bold CA Strategic Energy Efficiency goals leading to Net Zero Energy building by 2030. Increasingly, utilities would like to explore shifting their programs from component-based solutions to more innovative integrated system solutions. However, the field performance of these integrated systems is more difficult to evaluate and quantify, which is one of the key motivations for the use of the FLEXLAB building test facility at Lawrence Berkeley National Laboratory (LBNL). FLEXLAB enables repeatable experiments in 600 ft^2 test chambers that are representative of a perimeter zone in a commercial building.

PROJECT GOAL

The main goal of this Project was to characterize the behavior of radiant heating and cooling systems in order to provide information for designers that would reduce both the cost and the perceived risk associated with radiant systems. The aim is to improve energy and comfort performance by encouraging the appropriate adoption of radiant systems in both new construction and retrofit of commercial buildings in California.

A secondary goal was to assess the benefits of integrated control of HVAC, active facades and daylight harvesting.

PROJECT DESCRIPTION

The systems studied, and particular aspects of their design and operation relating to performance and cost, were selected in conjunction with an Industrial Advisory Group consisting largely of California practitioners – see Appendix 1. Two systems were studied in FLEXLAB:

- Radiant floor slab heating and cooling systems
- Radiant ceiling panel heating and cooling systems

Radiant slab systems have the ability to smooth and shift peak HVAC loads but are often difficult to retrofit and so are applicable primarily to new construction, although pipes embedded in a relatively thin topping slab can be installed on the structural slab in some cases. Radiant panel systems can be installed in suspended ceilings, and therefore are relatively easy to retrofit, but have no thermal storage capability. They have the disadvantage of being more expensive than radiant slab systems for new construction and so are more applicable to retrofit situations.

To motivate the study of radiant system characteristics of relevance to designers, a simulation-based assessment of the savings potential of radiant ceiling panel systems relative to a conventional variable-air-volume (VAV) system was performed for a medium-sized office building in four California climates. Radiant slab systems respond relatively slowly and so their control, and their simulation, is not as straightforward as for the fast-acting radiant panel systems. However, a field study comparing the performance of a radiant slab system to that of a conventional air-based system was been identified and its results are summarized in Section 4.6.

Integrated control of HVAC, active facades and daylight harvesting was assessed using simulation for the same four climates, with a view to evaluating the technical potential of off-the-shelf integrated control products.

The ability of radiant systems to address perimeter zone effects using simple controls was investigated, including the effect of ceiling fans, as was the ability of radiant floor slab systems to shift and smooth peak HVAC loads. Priorities for study topics were chosen in consultation with the Industry Advisory Group.

PROJECT FINDINGS/RESULTS

The overall findings of the project are:

- The simulation-based estimate of the energy savings potential of a radiant ceiling panel systems relative to a conventional, Title-24-compliant, variable-air-volume (VAV) system was performed for a medium-sized office building in San Jose (climate zone 4), Los Angeles International Airport (climate zone 6), Los Angeles Civic Center (climate zone 9) and Stockton (climate zone 12). The HVAC energy savings are in the range 14-26%. In a previously published side-by-side comparison of the measured performance of a radiant slab system and a variable-air-volume (VAV) office building in India, the radiant slab system consistently used 1/3 less energy than the VAV system while providing greater occupant satisfaction with the indoor environment.

- Both radiant panel and radiant slab cooling systems can provide thermal comfort at the modest zone cooling loads typical of well-designed office buildings, enabling low energy design based on water-side free cooling in suitable climates such as California coastal climates.

- Modest variations in comfort conditions were observed near unshaded south-facing windows. These variations could be mitigated by interior or exterior shading or by ceiling fans, without needing separate hydronic loops and controls, thus avoiding increased design, installation or maintenance costs.

- Radiant slab systems can provide significant peak load shifting and smoothing, although they are primarily suited to new construction and have limited retrofit potential.

- Radiant panel systems, which can be retrofitted into conventional T-bar ceiling systems, can provide good thermal comfort conditions with good dynamic response to varying internal and solar gains.

- Simulations in four California climate zones indicate negligible benefit from the use of integrated control of active facades, HVAC and daylight harvesting systems in medium-sized office buildings with radiant panel systems. Benefits are limited to perimeter zones and times when there is both a heating load and available solar radiation.

PROJECT RECOMMENDATIONS

1. Expand current knowledge and information regarding radiant systems: Verify and extend the preliminary findings of this study through case studies of radiant systems in California commercial buildings that span design, construction and operation and address both technical and financial issues
2. Promote the appropriate deployment of radiant systems in commercial buildings, based on current knowledge and information gained from new case studies, through:
   - Design guides and related training classes, based on results from this project, formal case studies and the expertise of the relatively limited numbers of designers and contractors working productively with radiant systems
   - Targeted incentives, including Savings by Design, based on demonstrated benefits; target radiant slab systems for new construction to gain load shifting benefits and target radiant panel systems for relative ease of retrofit
3. Control methods for radiant slabs need to be further developed and tools provided. Three current R&D projects address this in different ways:
   - Optimizing Radiant Systems for Energy Efficiency and Comfort – UC Berkeley Center for the Built Environment, funded by CEC EPIC
   - Model Predictive Control of Radiant Slabs – UC Berkeley, Department of Mechanical Engineering, funded by DOE
   - OpenBuildingControl: Performance Evaluation, Specification and Verification of Building Control Sequences – Lawrence Berkeley National Laboratory, funded by DOE and CEC EPIC

See Section 6 – *Recommendations* for further details.

These projects can be expected to reduce the current difficulties in the design of the controls for radiant slabs. PG&E / State-wide programs can then help to disseminate these control strategies and tools and promote the adoption of radiant slabs, in particular, based on proven design methods and control strategies.

1 INTRODUCTION

The main goal of the Project was to characterize the behavior of radiant heating and cooling systems for commercial buildings in order to provide information for designers that would limit both the cost and the perceived risk associated with such systems. The aim is to improve energy and comfort performance by encouraging the appropriate adoption of radiant systems in both new construction and retrofit of commercial buildings in California. The focus was on cooling rather than heating because cooling loads tend to be larger in California commercial buildings.

A secondary goal was to assess the benefits of integrated control of HVAC, active facades and daylight harvesting.

Tests were conducted in side-by-side FLEXLAB test rooms in March and April, 2016. Different control configurations were tested at different times with different HVAC configurations in order to separate the effects of the different controls.

The performance issues addressed in the tests were selected in consultation with an Industry Advisory Group consisting primarily of design practitioners. Appendix 1 lists the members of the Group and summarizes their responses to a detailed web-based questionnaire designed to help establish priorities for the project. A number of research questions were identified; those that could be fully or partially addressed in the project are:

- Managing Perimeter Zone Heating and Cooling: the greatest variability occurs at the building perimeter. Some potentially efficient HVAC systems are never selected because they are unable to handle peak cooling under worst-case solar loads. FLEXLAB is designed to explore these interactions and compare the ability of different HVAC systems to handle a diversity of load profiles and magnitudes. How do measured loads vary with distance from a window wall under different conditions, and do these effects differ with different types of HVAC system? Can perimeter heating be eliminated for a high performance façade?
- Low Energy cooling strategies – how effective can these be for typical CA coastal climates? Options include radiant panel cooling and radiant slab cooling that use waterside free cooling.
- Effectiveness of radiant slabs to smooth and shift peak cooling loads.

One aim of the project was to generate design advice that relates to the degree of design flexibility for radiant systems. Testing was conducted in unoccupied spaces with artificial standard internal loads (people, equipment, and lighting) characteristic of the range expected in office buildings.

2 BACKGROUND

The State of California has set aggressive goals for efficiency improvements in commercial buildings, both for retrofit and for new construction. The utilities, in particular, the investor-owned utilities (IOU's) have a substantial role to play in enabling and incentivizing these efficiency improvements. Identifying the applicability and the benefits of new technologies, both singularly and in combination, is a key precursor to their appropriate deployment and is the main goal of the project reported on here.

California has a diverse set of climates but all of these climates, to a degree, are characterized by relatively low humidity levels year-round, resulting from the interaction of the prevailing westerly winds and the cold California ocean current that flows southwards from Alaska along the California coast. This low humidity results in relatively large diurnal temperature swings and favors evaporative cooling systems. Radiant cooling systems use higher supply temperatures than mixing ventilation forced air systems and so can make more use of water-side free cooling. Hydronic systems also reduce fan power by more than their increased pump power, which also reduces system energy consumption.

However, hydronic systems suffer from two barriers to wider adoption.

- Their maximum heat removal capacity can be less than that of air systems, requiring measures to reduce envelope cooling loads in conventional buildings and making them unsuitable for buildings with high cooling load intensities.
- There has been a lack of familiarity with radiant systems on the part of both designers and contractors, resulting in a reluctance to specify such systems and a tendency for bid prices to include significantly higher allowances for contingencies than for the more familiar air systems.

There is, therefore, a need for better design guidance for radiant system sizing and for zoning and the associated level of controls complexity. The project described here has attempted to address these needs in order to reduce barriers to adoption and enable designers and owners to make more informed choices regarding the selection of radiant systems for commercial buildings.

3 TECHNICAL APPROACH/TEST METHODOLOGY

3.1 LABORATORY FACILITY

The experiments were carried out in one of the four test beds that constitute the FLEXLAB test facility at LBNL [1], shown in Figure 1**.** The test bed used consists of a pair of matched cells (3A and 3B), each 30 ft x 20 ft x 16 ft (9.14 m x 6.09 m x 4.88 m), equipped with radiant panels located in a suspended ceiling placed at a height of 9 ft (2.74 m) above the floor. The floor area of each cell is 600 ft2 (59.3 m^2). The south facade is reconfigurable and the other surfaces are relatively highly insulated. In these experiments, the south façade was configured to comply with the prescriptive requirements of the California Title 24 building energy code.

FIGURE 1. GENERAL VIEW OF THE CLIMATE TEST ROOM

In order to simulate more realistic office conditions, six simulated work spaces consisting of thermal manikins, computers, desks and partitions were set up, together with ceiling fans and artificial light, as shown in Figure 2. The manikins were wound with heating tape and the supply voltage adjusted to produce a sensible heat dissipation rate of ~80 watts each, corresponding to the metabolic rate of sedentary office works in a low humidity environment. The advantage of the thermal manikins is that their radiative/convective splits, and the characteristics of their thermal plumes, are more realistic than the heated cylinders specified in the European Standard E 14240.

The computers, LCD screens, lights and manikins were activated on a typical office schedule and the power consumption of the computers was managed using a script running on each computer that varied the computational load to follow a predefined profile in time. The internal sources are summarized in Table 1. Three ceiling fans were installed in order to investigate their influence on air speed and stratification and on convective heat transfer at the surfaces of the slab and the radiant panels.

FIGURE 2. THE TEST ROOM CONFIGURATION

TABLE 1. HEAT LOAD SUMMARY

Heat source	Number	Schedule	Peak power (W)*	Power per unit floor area (Btu/hr.ft^2 [W/m^2])
Personal computers	6	08:00-18:00	532	2.84 (8.96)
Thermal manikins	6	07:00-19:00	499	2.66 (8.40)
Overhead lights	6	00:00-24:00 (Mar. 25-28) 07:00-19:00 (Since Mar. 29)	279	1.49 (4.70)
Measuring instruments and data acquisition	N/A	00:00-24:00	65	0.35 (1.09)
Circulation pump	1	00:00-24:00	215	1.15 (3.62)
Total	N/A	N/A	1590	5.65 (18.28)

* At 122 V supply voltage

Two different radiant heating and cooling system were studied: a radiant ceiling panel system and a radiant cooling slab system, each installed in cell 3B. Cell 3A served as a reference cell, with cooling and heating supplied by a dedicated air-handling unit served by hot and chilled water. Hot and chilled water for both the radiant systems and the air-handling unit were supplied by a dedicated boiler and a dedicated chiller, each equipped with a buffer tank to reduce the supply temperature fluctuations due to boiler or chiller cycling. Since the focus was on sensible cooling performance, there was no secondary air system in cell 3B and no economizer operation in cell 3A.

3.1.1 RADIANT PANEL SYSTEM

The radiant panel system consisted of 72 panels, each 2 ft x 2 ft (609.6 mm x 609.6 mm), manufactured by Twa Panel Systems. Each panel consists of ½ in (12.7 mm) diameter copper pipe clipped to back of a sheet of aluminum and covered on the top side by foil-faced fiberglass insulation, as shown in Figure 3. The panels are mounted in the existing T-bar ceiling and are connected together at the back by push-on flexible hoses. Areas of the ceiling not fitted with thermally active panels have conventional ceiling tiles. The radiant panels were painted to match the optical reflectivity of the ceiling tiles, so as not to affect the daylighting performance. The radiant panels occupied 51% of the ceiling area and were divided into three sub-zones, with each subzone have two circuits of panels connected in series, as shown in Figure 4.

FIGURE 3. RADIANT CEILING PANELS: FRONT (UPPER LEFT), BACK (UPPER RIGHT), PIPING (LOWER LEFT), CONNECTIONS (LOWER RIGHT)

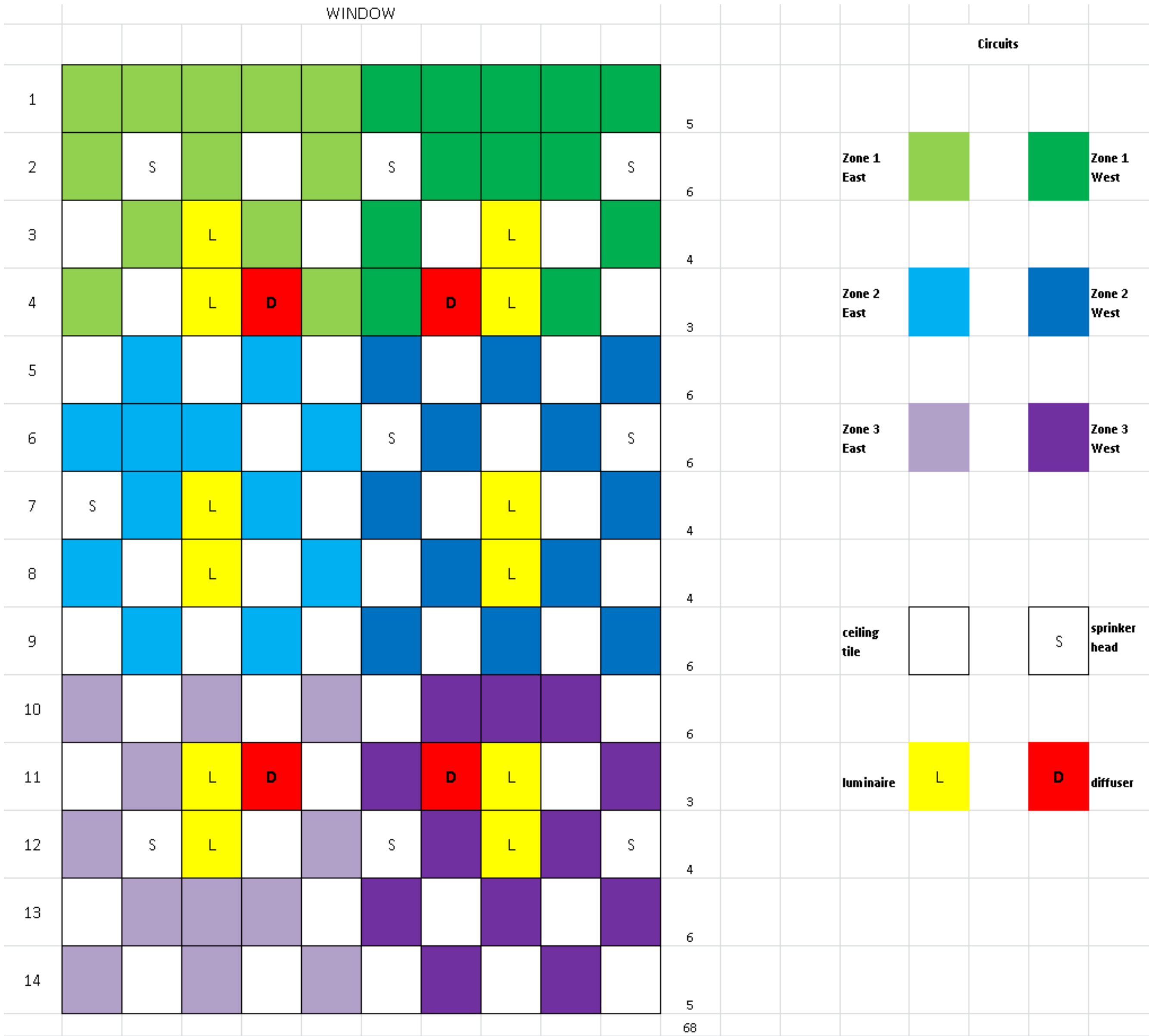


FIGURE 4. THE RADIANT CEILING PANELS AND POSITIONS. THE DIFFERENT COLORS INDICATE THE DIFFERENT CIRCUITS.

3.1.2 RADIANT SLAB SYSTEM

In the radiant slab system, chilled water is seperately supplied to three sub-zones, each containing multiple circuits of PEX tubing, and with floor areas of 19.5 m^2 (210 ft^2), 15.3 m^2 (165 ft^2), and 23.0 m^2 (248 ft^2), as shown in Figure 5. The nominal diameter of the tubing is 5/8 in (15.9 mm) and the internal diameter is 1/2 in (12.7 mm).

3.1.3 RADIANT SYSTEM PLUMBING

Hot water or chilled water is selected using automated valves, which also have manual overides. The inlet water temperature to the radiant system is varied by mixing supply water from the appropriate HVAC secondary loop with return water from the radiant system. The flow rate in each subzone may be controlled independently by modulating a control valve but each subzone has the same inlet water temperature. The plumbing, with the associated valves, flow sensors and temperatures, are shown in Figure 6**.**

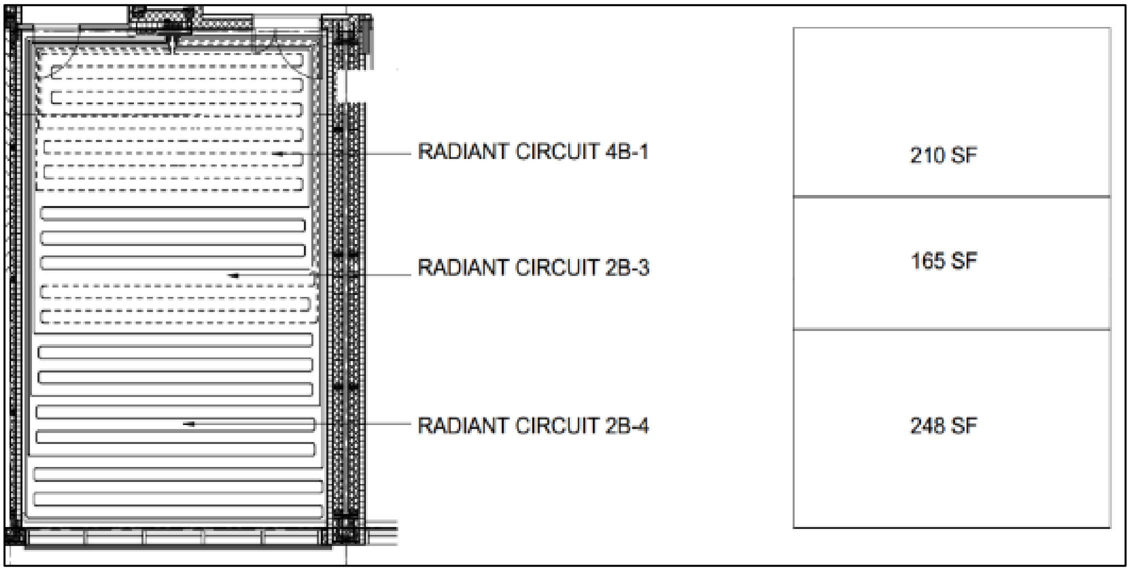


FIGURE 5. THE CIRCUITS OF EMBEDDED RADIANT CEILING SLAB SYSTEM

FIGURE 6. THE RADIANT SYSTEM CONTROL PLUMBING

3.2 INSTRUMENTATION PLAN

61 temperature sensors were used to measure the surface temperatures of the walls, windows, floor and ceiling with every surface having at least six temperature sensors, as shown in Table 2.

TABLE 2. THE CONDITIONS OF THE SURFACE TEMPERATURE SENSORS

Location	Number of temperature sensors
South wall	2 thermistors on the wall, 9 thermocouples on the window and window frame
North wall	9 thermistors
West wall	12 thermistors
East wall	12 thermistors
Drop ceiling	6 thermistors
Floor (under the carpet tiles)	9 thermistors

Three vertical arrays of temperature sensors, fixed at seven different heights, were used to observe air temperature stratification, as shown in Figure 7. The heights of the temperature sensors on the vertical 'trees' were taken from ASHRAE Standard 55-2013[2], as shown in Table 3.

TABLE 3. THE HEIGHTS OF AIR TEMPERATURE ON THE STRATIFICATION TREE (ACCORDING TO ASHRAE 55-2013)

Height Above Floor	Notes
4.0 in (0.1 m)	Ankle level
11.8 in (0.3 m)	Knee level
24.0 in (0.6 m)	Waist level for seated occupants
43.0 in (1.1 m)	Head level for seated occupants
67.0 in (1.7 m)	Head level for standing occupants
86.6 in (2.2 m)	20 in below the drop ceiling
104.0 in (2.6 m)	4 in below the drop ceiling

FIGURE 7. A VERTICAL TEMPERATURE ARRAY

Each air temperature sensors was surrounded by a radiation shield, painted white to reflect solar radiation, as shown in Figure 8. The location of the vertical trees is shown in Figure 9. The position of the southern-most tree was selected to just avoid direct solar radiation on all but the lowest sensor.

FIGURE 8. THE RADIATION SHIELD FOR THE AIR TEMPERATURE SENSORS

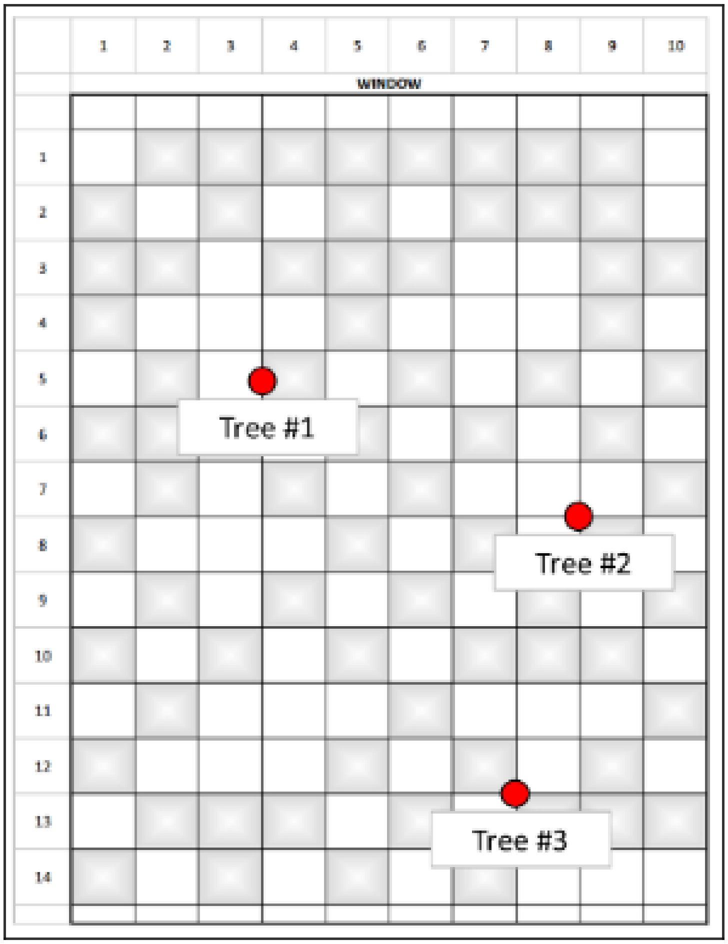


FIGURE 9. THE LOCATIONS OF THE TEMPERATURE STRATIFICATION TREES (SOUTH IS AT THE TOP)

Since globe temperature was an input to the calculation of operative temperature[1] (see Appendix 2), which was used for zone temperature control during the majority of the tests, a quick response globe temperature sensor was required. The response time of the 6" (150 mm) diameter globe thermometer specified in ISO 7726 [3] is ~ 20-30 minutes, which was considered to be too long for these tests. Three smaller globe thermometers were used, which were made by inserting Resistance Temperature Detector (RTD) sensors into table tennis balls coated with grey paint with an emissivity of ~0.90. According to Benton [4], this type of globe thermometer has a response time of ~6 minutes to reach 90% of its final value.

The chilled water supply temperature was measured at the outlet of the buffer tank; the water temperatures at the inlet and outlet of each water circuit were also measured.

Outdoor dry-bulb and dew-point temperature and direct and diffuse solar irradiance were measured on the roof of the adjacent double height test bed, the dry-bulb temperature

[1] Operative temperature is a weighted average of mean radiant temperature and dry-bulb air temperature and is indicator of the effect of these temperatures on thermal comfort.

sensor was housed in an aspirated enclosure and the dew point temperature was measured using a polished mirror sensor. Details of the sensors inside the test cell are presented in Table 4.

TABLE 4. INTERIOR SENSORS

Instruments	Sensor accuracy (manufacturer's spec)	Number	Measured variables
US Sensor PR103J2	±0.05 K	70+21	(a) Air temperature; (b) Vertical wall temperature; (c) Window temperature; (d) Drop ceiling temperature; (e) Slab surface temperature
Temperature RTD	±0.03 K	3	Globe temperature
MAG1100 PFA	±0.41% at max flow rate	3	Water flow rate of each radiant sub-circuit
BAPI RTD - sensor + transmitter	±0.03K	4	Chilled water supply and return temperatures
BAPI XXP thermistors	±0.05K	9	Slab temperatures

All the sensors were sampled by the data acquisition system at an interval of one second and one-minute averages calculated for analysis purposes.

4 TECHNICAL ANALYSIS

This section provides details of the experimental and simulation studies that were performed to address the project goals and objectives.

4.1 TASK1: STRATIFICATION AND EFFECT OF CEILING FANS

The goal of Task 1 was to quantify the stratification observed with the radiant panel cooling system and the radiant slab cooling system and describe the effect of ceiling fans on that stratification.

4.1.1 EXPERIMENT SETUP

Radiant panel cooling system setup

The chilled water flow rate in the radiant cooling panel system was maintained at 4.77 gpm (0.0631 l/s), equivalent to 0.0080 gpm/sf (0.00113 l/s.m^2), with the flow rate in each circuit proportional to the floor area served. The chilled water supply temperature was modulated between 55.0°F (12.8°C) and 68.0°F (20°C) to maintain the operative temperature at 73.4°F (23°C), 75.2°F (24°C) and 77°F (25°C) respectively, and then to maintain the dry-bulb temperature at 75.2°F (24°C). Each set point was maintained for at least 24 hours.

During the radiant panel cooling system experiment, the ceiling fan speed was switched between level 2 and level 3, when operating.

Radiant slab cooling system setup

The chilled water supply temperature was controlled at 55°F (12.8°C) and the water flow rate of each sub-circuit was maintained at 0.6 gpm (0.038 l/s).

4.1.2 RESULTS AND DISCUSSION

A preliminary step in comparing control on operative vs. dry-bulb temperature is to evaluate the magnitude of the temperature stratification of the air. Figure 10 shows the vertical temperature distribution, averaged over the three positions shown in Figure 9. The radiant panel system operates, and the internal gains are present, from 6:00 to 19:00, with approximately constant operative temperature. In the ASHRAE comfort standard, Standard 55.1, temperature stratification is defined as the vertical temperature difference between heights of 4 in and 67 in (0.1 m and 1.7 m), which correspond to the ankle and head heights for a standing occupant. The maximum value recorded was 2.9°F (1.6 K) at 11:00 on March 26th. Relatively modest stratification is expected in systems that cool from the ceiling, as opposed to the floor, because the cooling of the air by convective transfer at the ceiling drives convection cells, which serve to mix the room air and inhibit stratification.

Figure 11 shows the stratification effect observed with the radiant slab cooling system, operated with various combinations of chilled water supply temperature and flow rate, as noted in the figure. The stratification effect was significantly enhanced compared to that observed with the radiant ceiling panel cooling system; the largest stratification observed was 6.3°F (3.5 K), at 15:30 on April 7th, more than twice as much as with the ceiling panel system.

Figures 10 and 11 show a substantial diurnal variation in the temperature stratification. The absence of both solar and internal gains at night reduces very substantially the cooling load, resulting in very limited stratification.

The measurements are consistent with the expectation that radiant ceiling cooling systems, which generate convection cells, will exhibit less stratification than radiant floor cooling systems, which promote stable stratification. The opposite behavior is expected in heating mode, with the radiant ceiling panels generating substantial stratification, though no measurements were made during this project.

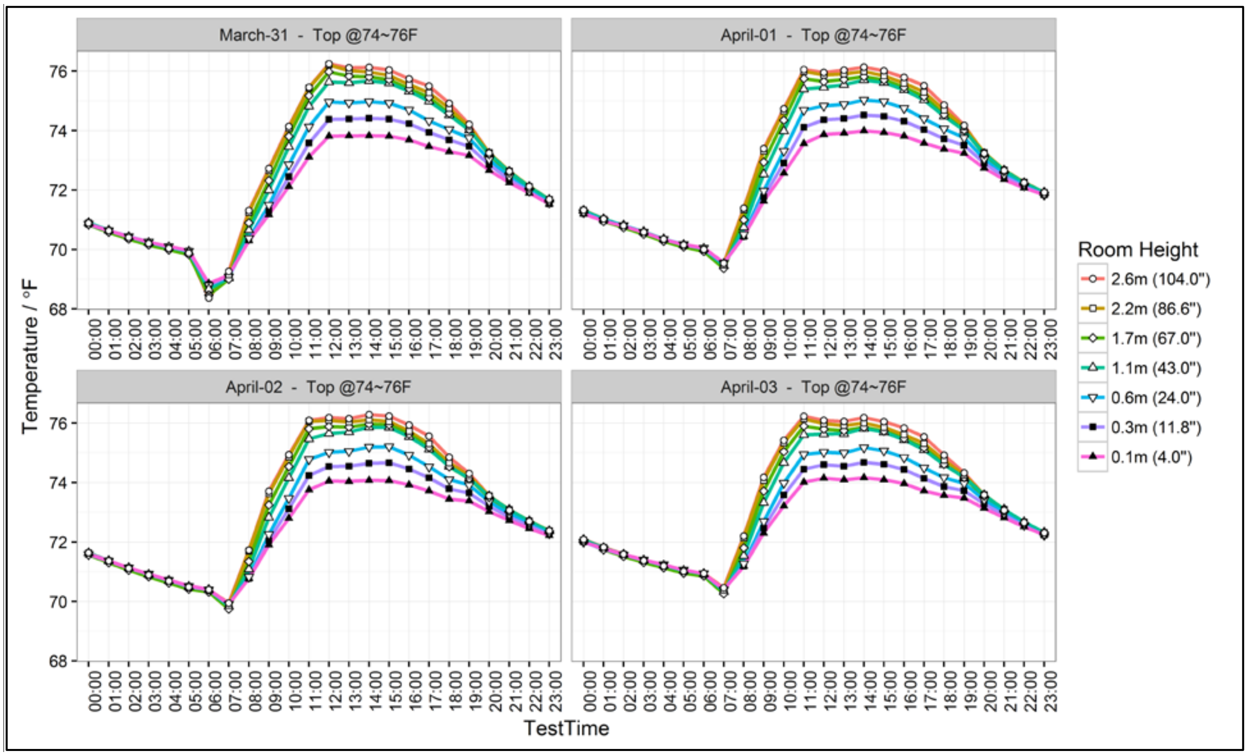


FIGURE 10. AIR TEMPERATURE STRATIFICATION IN THE RADIANT COOLING PANEL SYSTEM OPERATING 6:00 – 19:00

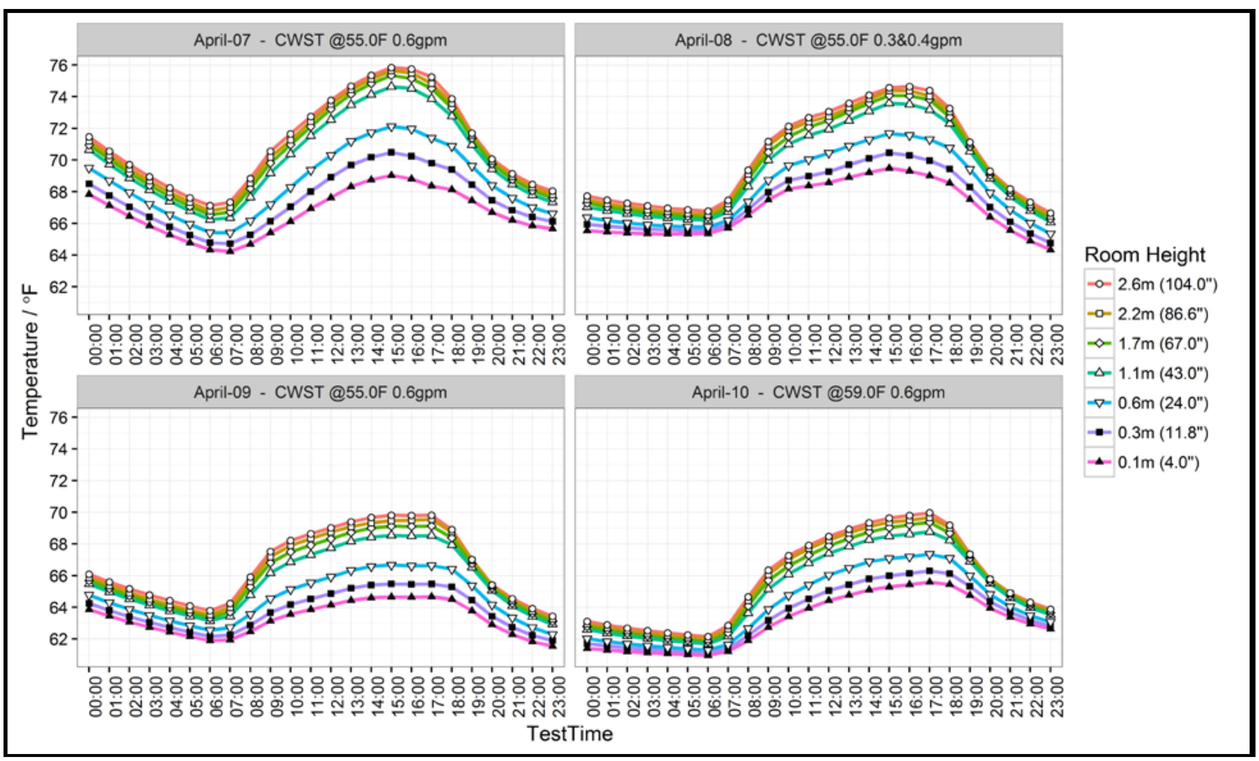


FIGURE 11. AIR TEMPERATURE STRATIFICATION IN THE RADIANT COOLING SLAB SYSTEM OPERATING 6:00 – 19:00

Figure 12 shows the effect of using ceiling fans on radiant ceiling panel cooling system performance. The three ceiling fans were switched on at Level (speed) 2 from 11:40 to 15:10, and at Level 3 since 15:10. Apart from the well-established effect of ceiling fans on thermal comfort [2], ceiling fans reduce temperature stratification, as illustrated in the figure, and can also promote increases in heat transfer at the surface of the radiant panels.

Figure 13 shows the effect of ceiling fans on temperature stratification in the radiant slab cooling system. There is a very substantial reduction in the significant temperature stratification, which improves thermal comfort in and of itself, in addition to the beneficial effect of the increased air motion and the increase in heat transfer at the surface of the radiant slab. The effect on the cooling capacity of the slab system is unclear from the available measurements, though other studies.

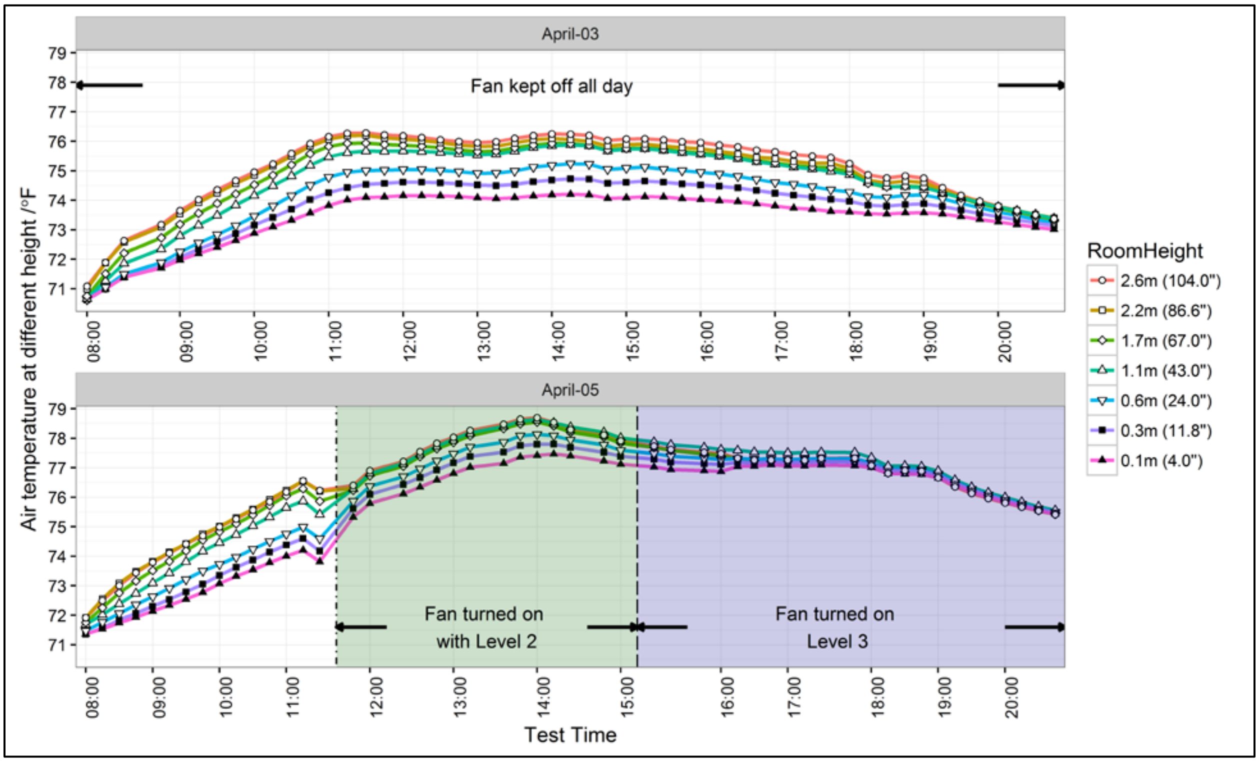


FIGURE 12. THE EFFECT OF CEILING FAN OPERATION ON AIR TEMPERATURE STRATIFICATION IN THE RADIANT CEILING PANEL SYSTEM

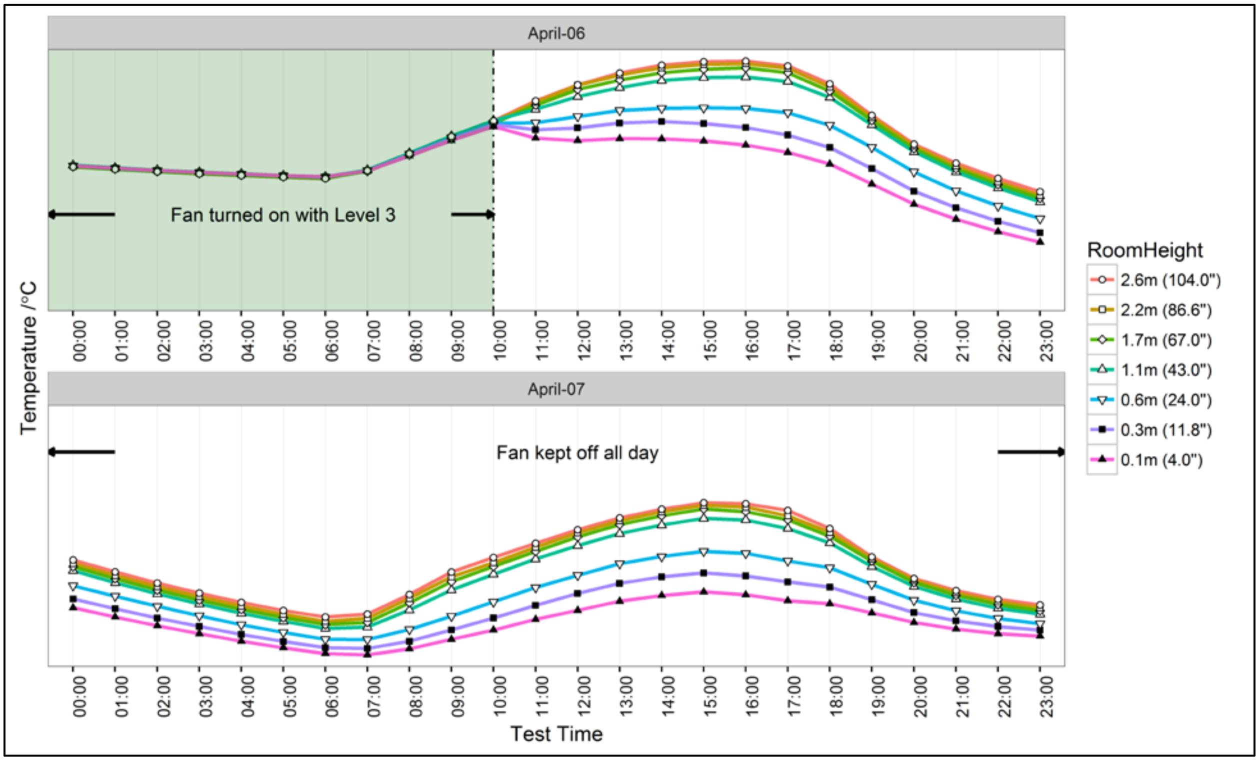


FIGURE 13. THE AIR TEMPERATURE STRATIFICATION COMPARISON BETWEEN DIFFERENT CEILING FAN RUNNING CONDITIONS IN EMBEDDED RADIANT COOLING SLAB SYSTEM

4.2 TASK 2: VARIATION IN COMFORT FROM PERIMETER TO CORE

The goal of Task 2 was to determine the variation in comfort conditions with distance from the south façade and compare the variation observed for the radiant panel cooling system and the radiant slab cooling system.

4.2.1 EXPERIMENT SETUP

The experiment setup was the same as Task 1 but with the ceiling fan off. Air temperature profiles were measured at the three locations shown in Figure 9 from March 25 to March 28 for the radiant panel cooling system and April 7 to April 10 for the radiant slab cooling system.

4.2.2 RESULTS AND DISCUSSION

Figure 14 and Figure 15 show the core-perimeter variation in dry-bulb temperature in radiant cooling panel system and the radiant cooling slab system, respectively. Each set of three lines represent the dry-bulb temperature vertical profiles at different locations (shown in Figure 7). The more perpendicular the lines, the less the variation in temperature with height. The larger the horizontal separations between the lines, the greater the dry-bulb temperature variations between the different locations. The larger horizontal separations in

Figure 14 indicates that there are greater positional variations in temperature in the radiant cooling panel system. The dependence of this effect on time of day, and the higher temperatures near the window, indicate the significant influence of solar radiation. The largest difference in temperature is between North and South (positions #3 and #1 respectively in Figure 9), and was 2.2°F (1.2 K) at 15:10 on March 25th at a height of 24 in (0.6 m) above the floor.

Figure 15 shows the corresponding variations in dry-bulb temperature observed in the radiant slab cooling system. However, compared to radiant cooling panel system, the core-perimeter variation in the radiant slab system was much smaller. Even though solar radiation still had a strong impact on the dry-bulb temperature in the space, the temperature differences between different locations had a relatively small maximum value of 1.6°F (0.9 K) at 15:00 on April 7th. The reason for this difference in behavior is unclear, especially since the radiant slab cooling system produces stable stratification, which might be expected to inhibit horizontal mixing. One possible explanation is that the solar radiation impinging on the floor is removed directly by the radiant slab system, leaving the more uniformly-distributed internal gains to drive the diurnal variation in room dry-bulb temperature.

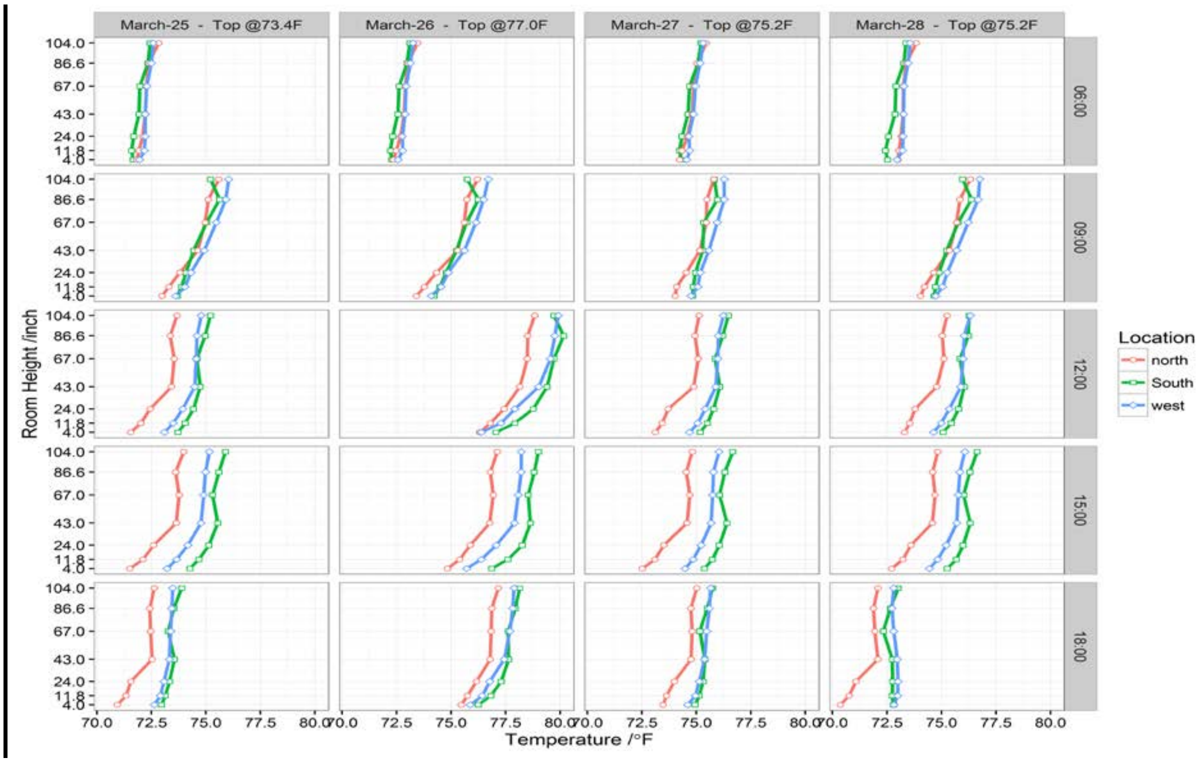


FIGURE 14. PERIMETER EFFECT OF DRY-BULB TEMPERATURE IN RADIANT COOLING PANEL SYSTEM

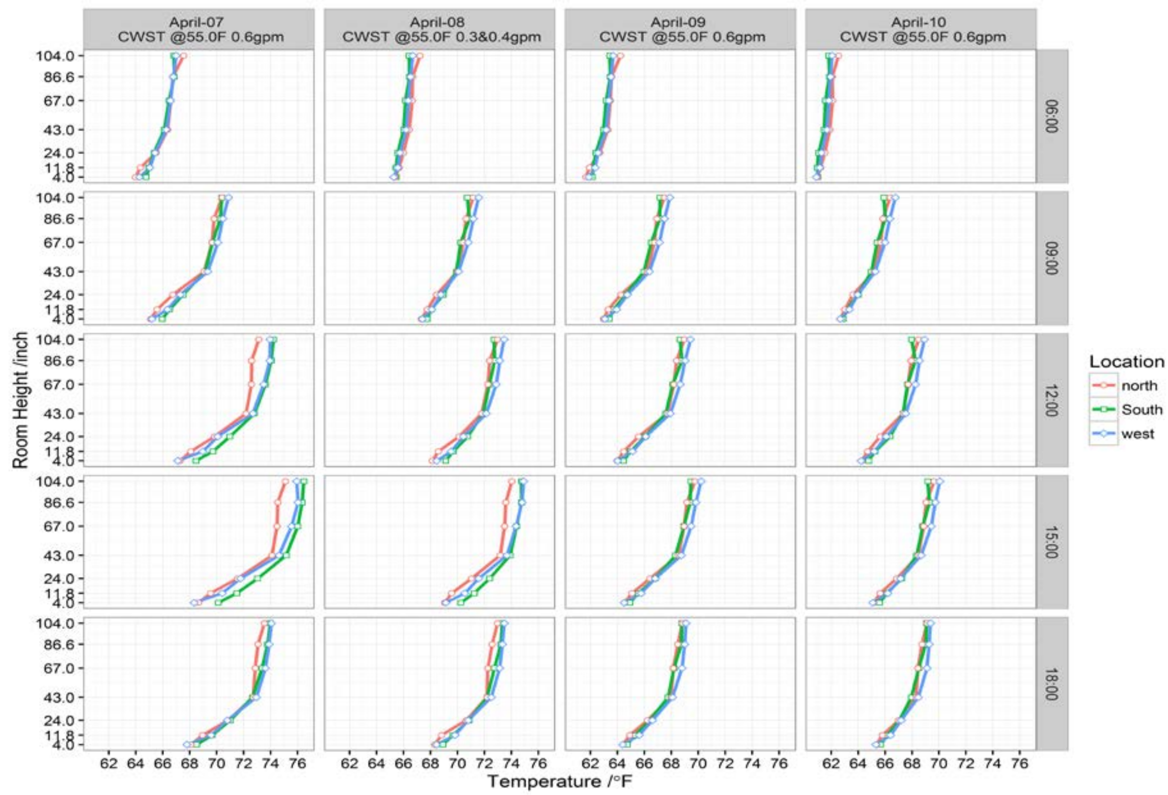


FIGURE 15. PERIMETER EFFECT OF DRY-BULB TEMPERATURE IN RADIANT COOLING PANEL SYSTEM

4.3 TASK 3: CONTROL ON OPERATIVE TEMPERATURE VS. AIR TEMPERATURE

The temperature of a radiantly-conditioned space is generally evaluated in terms of the operative temperature, T_{op}. Operative temperature can be calculated from the dry-bulb (air) temperature and either the mean radiant temperature or a globe temperature – see Appendix 2. It has recently been suggested that the difference between the air temperature and the operative temperature in radiantly-cooled commercial spaces may be small enough to allow the air temperature to be used as a proxy for the operative temperature. Since the temperature control of occupied spaces conventionally uses air temperature sensors and air temperature is easier and less expensive to measure than operative temperature, Task 3 is to investigate whether air temperature can be used in lieu of operative temperature as the feedback control variable for radiant panel systems.

4.3.1 EXPERIMENT SETUP

The experiment setup was the same as Task 1 but with the ceiling fan off.

4.3.2 RESULTS AND DISCUSSION

Figure 16 shows the core-perimeter variation in operative temperature in the radiant panel system and Figure 17 shows the corresponding variation for the radiant slab system. During the night, heat loss through the south window has a significant effect on the adjacent temperature in the radiant panel system, but not in the radiant slab system. From 07:00 to 10:00, the solar elevation is relatively low, distributing solar radiation throughout the cell. From 10:00 to 14:00, the solar elevation is higher, distributing solar radiation more towards the south end and substantially less towards the north end of the cell. In spite of cooling the slab substantially at night, the operative temperature in the slab system increases from ~66°F (19°C) at 7:00 to ~74°F (23°C) at 16:00, indicating that the slab system has limited cooling capacity, though the capacity in this case is limited partly by the carpet and partly by the masking effect of the furniture.

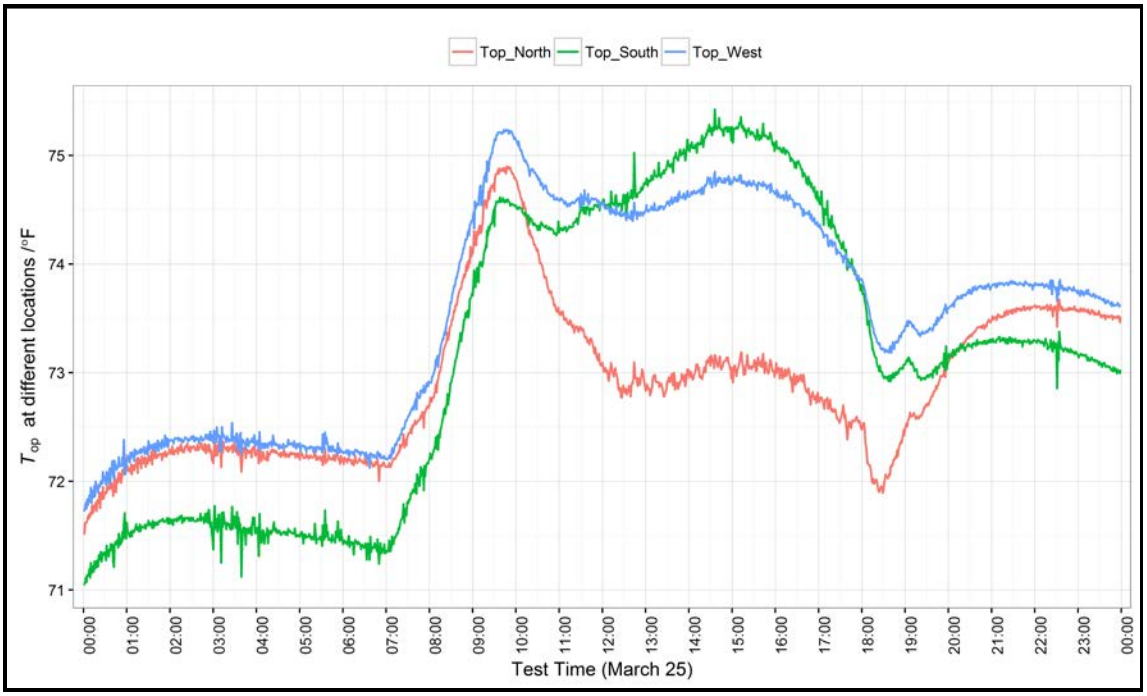


FIGURE 16. OPERATIVE TEMPERATURE AT DIFFERENT LOCATIONS IN THE RADIANT PANEL SYSTEM

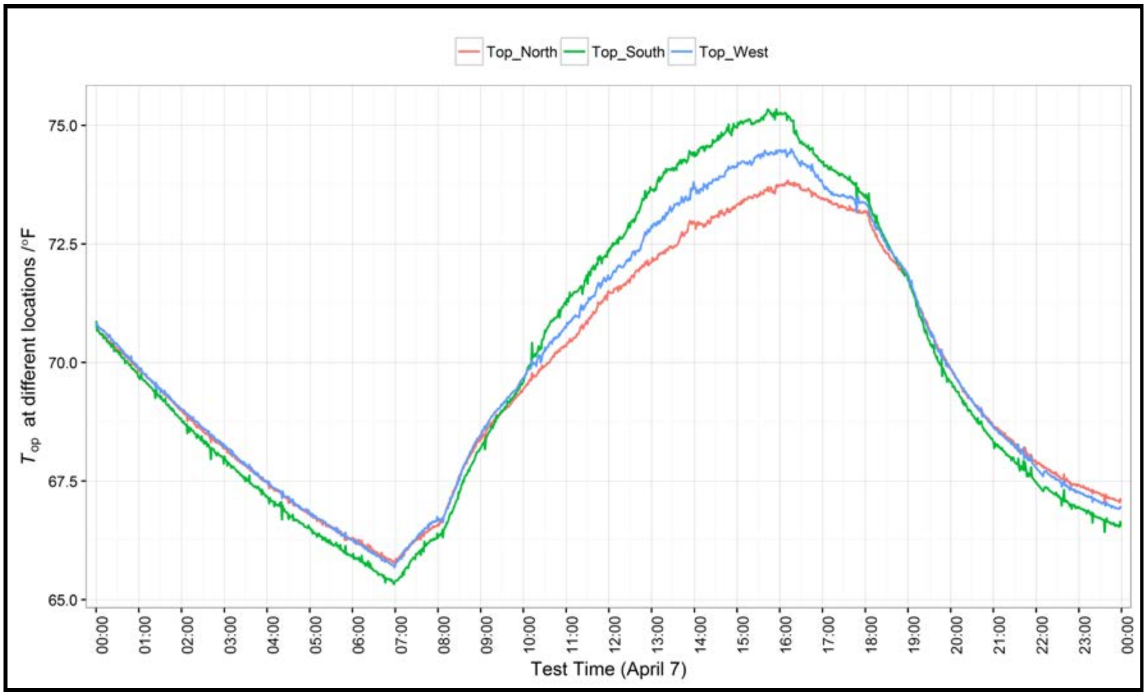


FIGURE 17. OPERATIVE TEMPERATURES AT DIFFERENT LOCATIONS IN THE RADIANT SLAB SYSTEM

Figure 18 shows the difference between the spatially-averaged operative and dry-bulb air temperatures (T_{op} - T_a) for the radiant panel system on four test days. The operative temperature set-points for each test day are shown after the date label in the figure. It can be seen that the difference between operative and air temperature is quite small, with a maximum value not exceeding 0.4^{o}F (0.2 K). Between 9:00 and about 19:00, the operative temperature was higher than air temperature and, for the rest of the day, it was lower than air temperature. The pattern of profiles seems to be independent of the operative temperature set-point. This diurnal variation may be partly accounted for by the surface temperature of the window being higher that the room air temperature during the day and lower than the room air temperature during the night. The small difference between air temperature and operative temperature indicates that it is possible to use air temperature instead of operative temperature for radiant ceiling panel cooling systems with the relatively low loads that typically characterize radiant cooling systems. More work is required to investigate feasibility at higher cooling loads and for heating.

Figure 19 shows the difference between the spatially-averaged operative and dry-bulb air temperatures (T_{op} - T_a) for the radiant slab system on four test days. The difference is larger than in the case of the radiant panel system, with a maximum value of 1.1^{o}F (0.6 K).

It should be noted that radiant slabs cannot be controlled with conventional feedback control strategies because of the long time constants involved. Model Predictive Control (MPC), in which a simple model is used to predict the response to different control actions, or heuristic strategies informed by MPC, offer potential ways forward.

The potential advantage of controlling radiant panel cooling systems on air temperature rather than operative temperature is that sensing air temperature is more conventional and simpler, and hence less expensive to implement and maintain.

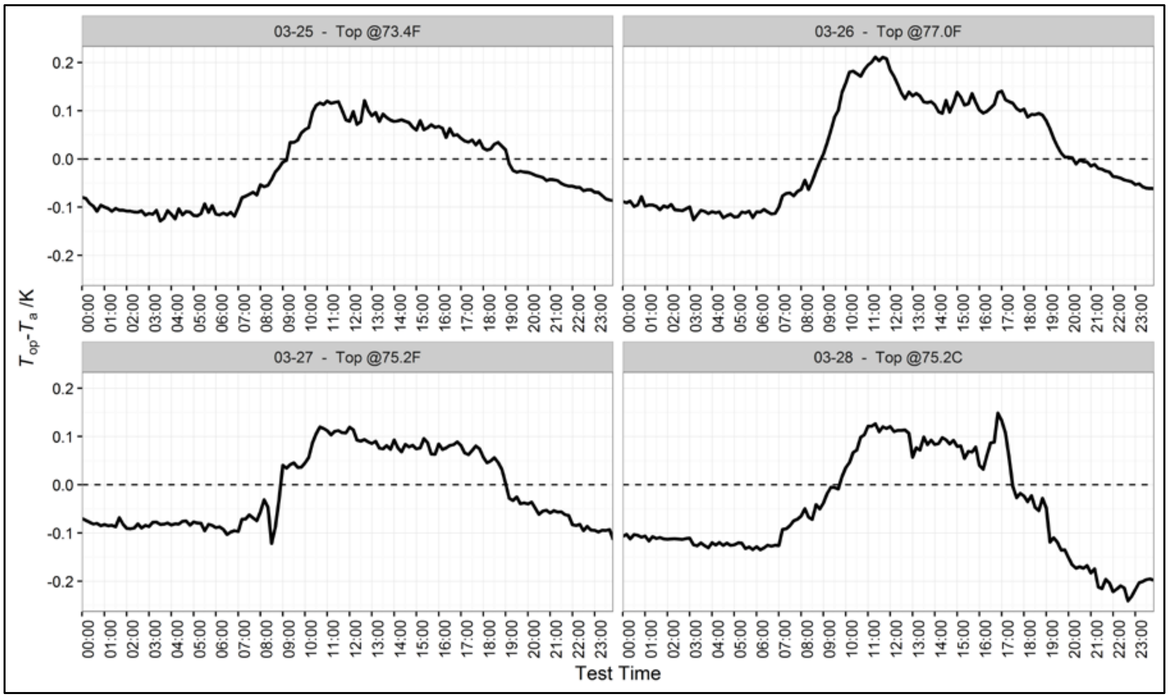


FIGURE 18. THE DIFFERENCE BETWEEN OPERATIVE AND AIR TEMPERATURE FOR THE RADIANT PANEL SYSTEM

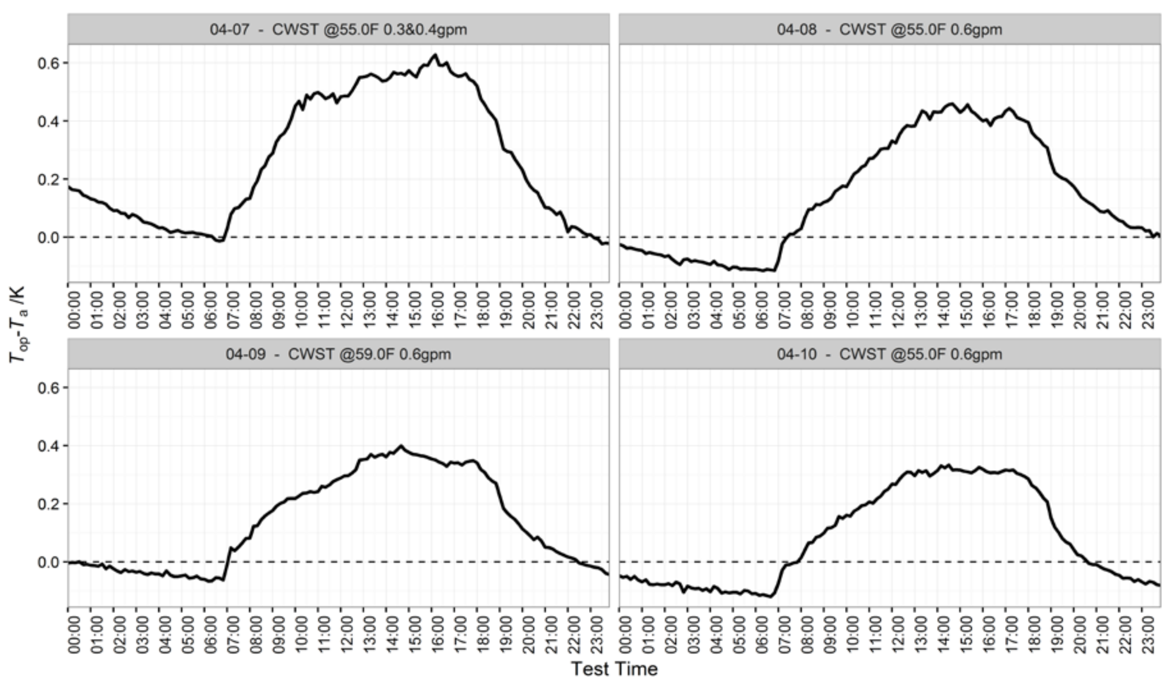


FIGURE 19. THE DIFFERENCE BETWEEN OPERATIVE AND AIR TEMPERATURE FOR THE RADIANT SLAB SYSTEM

4.4 TASK 4: EFFECT OF CARPET ON COOLING CAPACITY

The goal of Task 4 was to determine quantitatively the influence of the carpet on the cooling capacity of radiant slab systems. The carpet had been selected to have relatively low thermal resistance.

4.4.1 EXPERIMENT SETUP

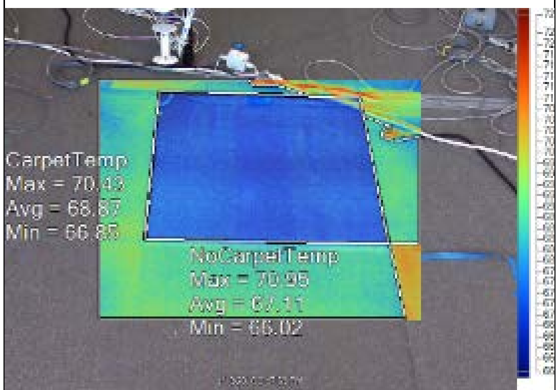
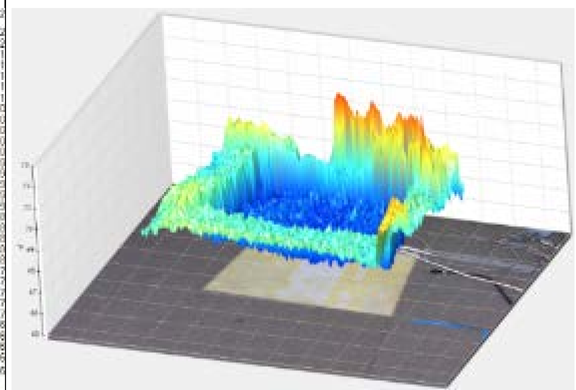
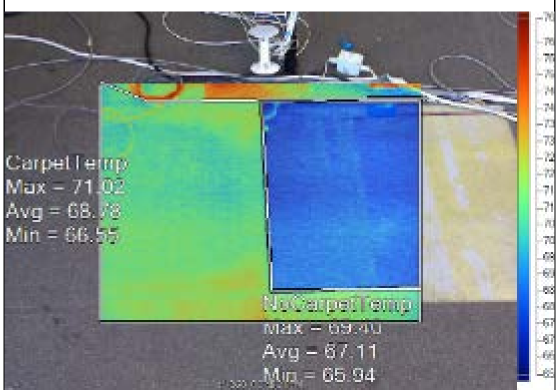
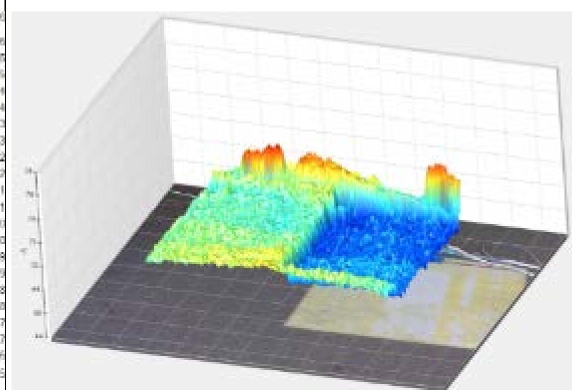
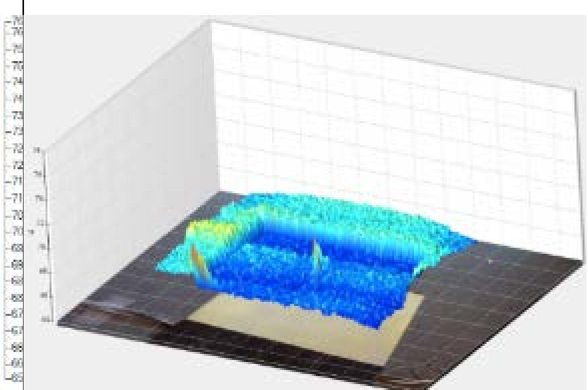
The chilled water supply temperature to the radiant slab was maintained at 55°F (12.8°C) with the flow rate of each sub-circuit at 0.60 gpm (0.038 l/s). The carpet used in the experiment has relatively low thermal resistant. Three carpet tiles were removed to expose the slad directly to the thermal camera. Therefore, infrared thermal images of slab and the carpet carpets can be obtained. A combination of slab surface temperature measurement and IR imaging was used to investigate the influence of the carpet on the floor temperature and cooling capacity of the radiant slab system.

4.4.2 RESULTS AND DISCUSSION

Temperature comparison using infra-red thermography

The IR images shown in Figure 20, which combine the infrared image and the photograph of the measured surface in a "Picture in Picture" mode, show the temperature distribution on the floor. There are two polygons, one enclosing an area of slab surface without carpet and one enclosing a neighboring area with carpet. The maximum, average and minimum temperatures measured in each polygon are included in the figure.

Three sites were measured. One was near the south wall where a sun patch may occur at noon; one was in the middle of the room; and one was near the north wall. Each site was measured twice. The site near the south wall showed a 1.7^{o}F (1.0 K) average temperature difference between the carpet and the exposed slab, while the other two sites showed very similar differences, with 1.0^{o}F (0.6 K) at both the middle site and the north site.

ID	IR image	3D-IR image
South site		
South site		
Middle site		

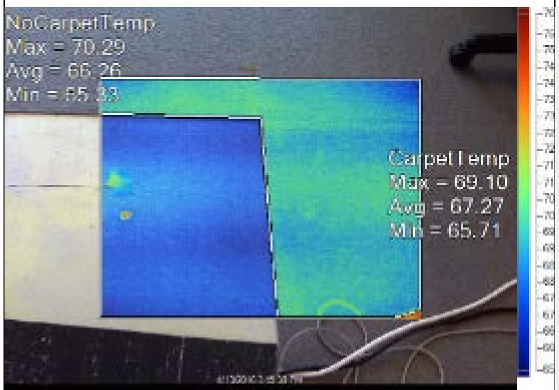
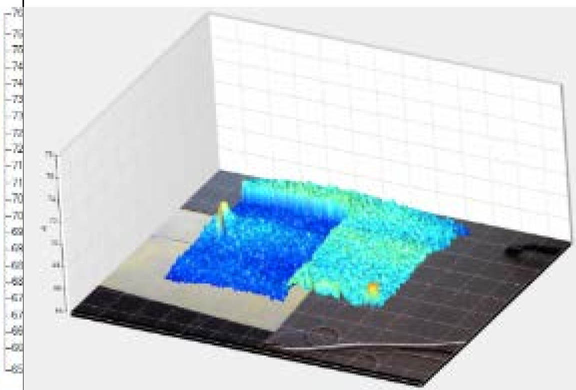
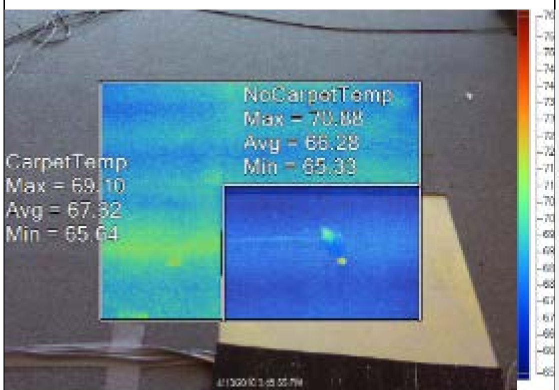
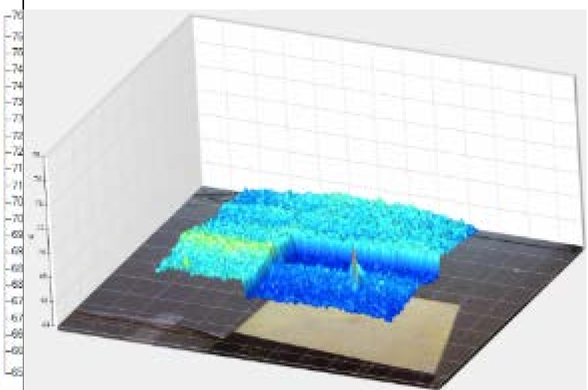
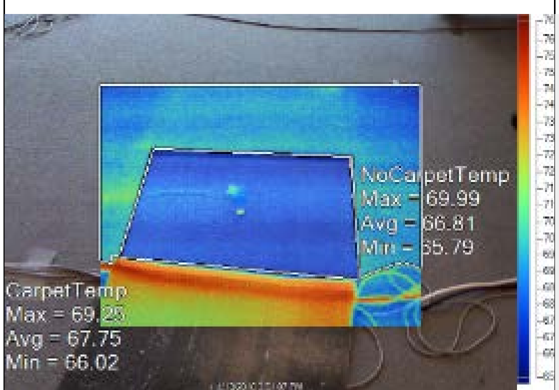
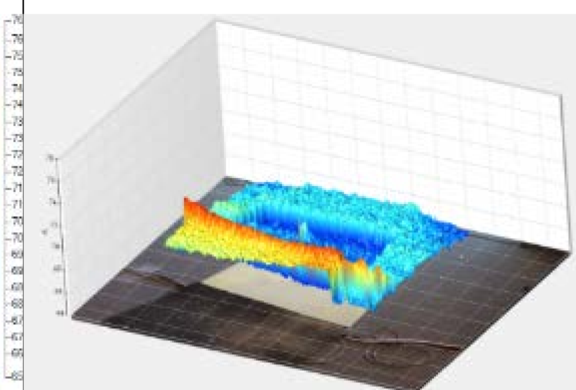
ID	IR image	3D-IR image
Middle site		
North site		
North site		

FIGURE 20. TEMPERATURE DISTRIBUTIONS BETWEEN AREAS WITH AND WITHOUT CARPET

Temperature comparison using temperature sensors

Figure 21 shows the location of the nine temperature sensors installed below the carpet. Figure 22 shows the profiles of the three sensors in the middle of the room. The red temperature line corresponds to the sensor without carpet (TMR_32), while the black and blue lines correspond to sensors underneath carpet tiles. When the carpet tile above TMR_32 was removed, at 13:00 on April 12, the thermal resistance between the surface of the radiant slab and the other uncontrolled surfaces was reduced, which increased the radiant heat transfer between the slab and those surfaces, producing an increase in the surface temperature of the slab. Based on these measurements, use of carpet similar in thermal properties to that studied in this test would require the supply chilled water temperature to be reduced by ~2 F (1K), with a corresponding reduction in the opportunities for free cooling. A possible increase in condensation risk in the carpet and the pipes connecting the chilled water plant to the slab should be considered, along with (modest) changes to the design and control of any dedicated outdoor air system providing mechanically conditioned ventilation air. Simulation tools, such as EnergyPlus, can be used to assess condensation risk in specific cases.

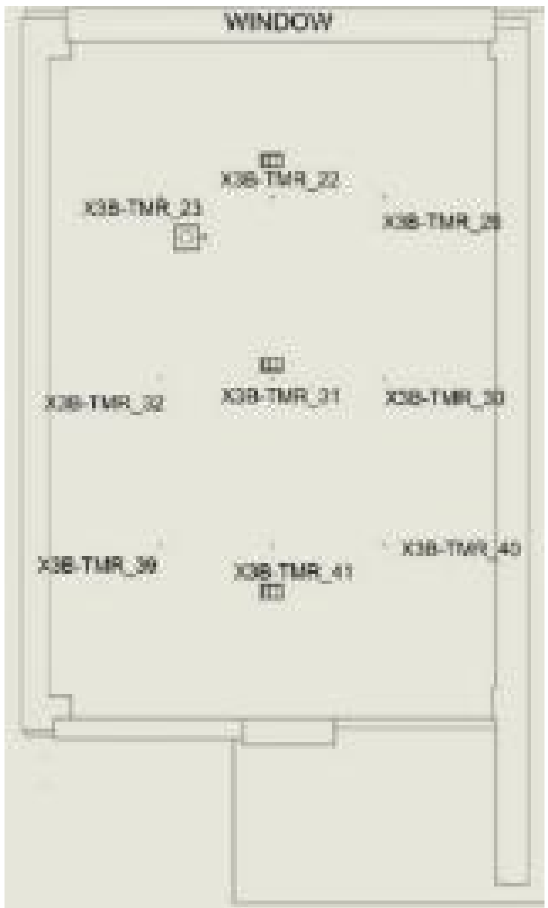


FIGURE 21. SENSOR LOCATIONS BELOW THE CARPETS

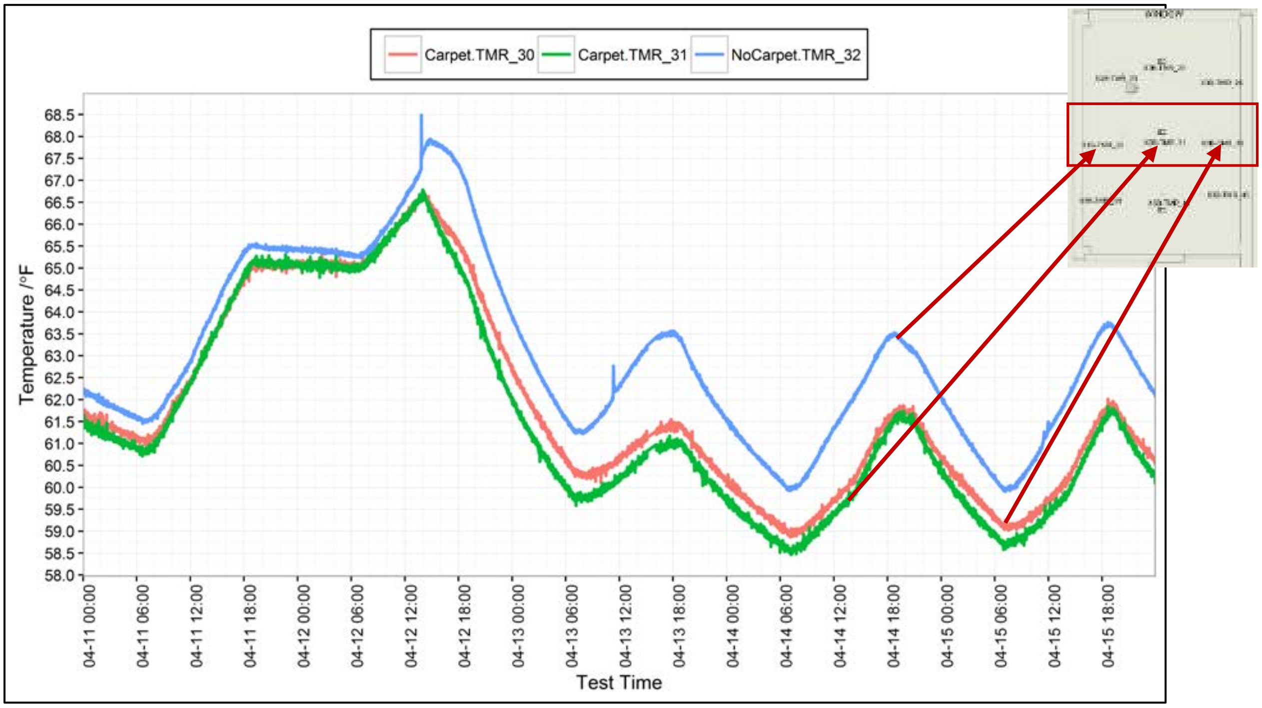


FIGURE 22. THE TEMPERATURE PROFILES OF SENSORS IN THE MIDDLE AREA ON APRIL 11-15

4.5 TASK 5: RADIANT SLAB CONTROL AND LOAD SHIFTING

The goal of Task 5 was to investigate the peak load smoothing and shifting effect of the radiant slab cooling system.

4.5.1 EXPERIMENT SETUP

The chilled water flow rate of each sub-circuit and the chilled water supply temperature of the embedded radiant cooling slab system were maintained constant at 0.60 gpm (0.038 l/s) and 55°F (12.8°C) respectively. The circulation pump was turned off at different times of the day for various lengths of time to investigate the load shifting patterns.

4.5.2 RESULTS AND DISCUSSION

In order to investigate the peak shaving potential of the radiant cooling slab system, the chilled water circulation pump was switched off from 14:00 for different periods of time. The off periods were 6 hours (with the pump turning on at 20:00), 5 hours (with the pump turning on at 19:00) and 10 hours (with the pump turning on at 24:00) on April 14, April 15 and April 16 respectively. The circulation pump ran continuously on April 13, which serves as a baseline

In Figure 13, the upper plot shows the global irradiation, which was measured by a weather station installed on the roof of the adjacent cell. The total irradiance was almost the same on April 13, 15 and 16 and somewhat lower on April 14. The lower plot includes the ambient dry bulb temperature, which had a similar profile April 13, 14 and 15 and was somewhat higher on April 16. The bottom, black, line shows the pump status. The slab surface temperature, indicated by the light blue line, cycles between similar maximum and minimum values, except for April 16, where the maximum is higher and persists longer because of a combination of higher operative temperature (possibly driven by higher ambient temperature) and longer off-time for the circulation pump. More data are required for a definitive assessment, but the limited data presented here suggest that turning off the chilled water plant for five or six hours may have only a relative modest effect on comfort. The benefit in terms of peak demand reduction and shifting of load from on-peak to off-peak periods could be quite substantial, though this effect has not been quantified here. This effect has been studied in the field by the Center for the Built Environment at UC Berkeley

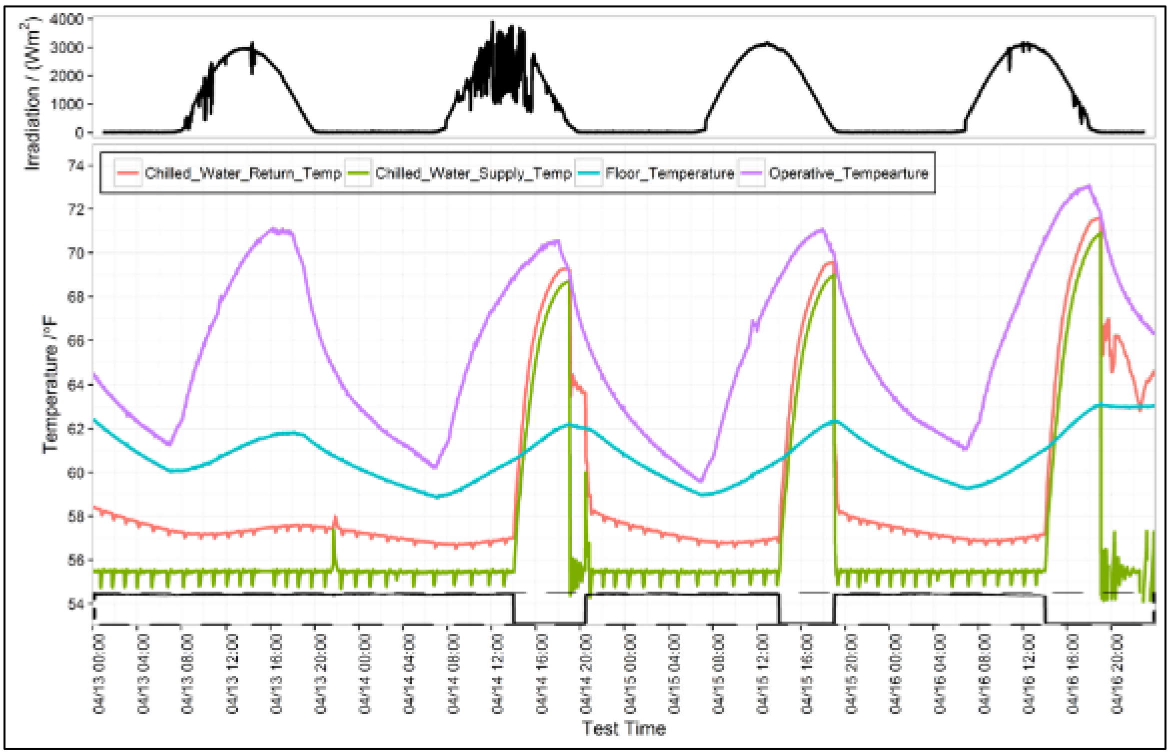


FIGURE 23. TEST RESULTS OF LOAD SHIFTING CONTROL OF RADIANT SLAB SYSTEM

4.6 TASK 6: ENERGY SAVINGS POTENTIAL OF RADIANT CEILING PANEL SYSTEMS VS. VARIABLE AIR VOLUME SYSTEMS

The goal of Task 6 was to investigate the energy saving potential of radiant ceiling panel systems compared to Title 24 2013-compliant VAV systems for California commercial buildings.

4.6.1 SIMULATION SETUP

This study uses the EnergyPlus new-construction medium-size office building model from the DOE Commercial Reference Building Models set as the starting point for the reference model.

Building Model Information

The U.S. Department of Energy has developed 15 reference-building models of three vintages (new, pre-1980, and post-1980 construction), which encompass most of the commercial building stock, for 16 locations (representing all the U.S. climate zones). The reference-building models represent approximately 70% of commercial building energy use based on the 2003 Commercial Building Energy Consumption Survey (CBECS).

This study used the new-construction medium-size office building reference model. Table 5 lists general information about the building, and Figure 33 shows the building geometry and the floor plan.

TABLE 5. BUILDING MODEL INFORMATION

Building Characteristic	Value
Total floor area	53,600 ft^2 (4,982 m^2)
Floor area per occupant	200 ft^2 (18.58 m^2)
Lighting load	1 W/ft^2 (10.76 W/m^2)
Plug load	1 W/ft^2 (10.76 W/m^2)
Aspect Ratio	1.5
Number of Floors	3
Window Fraction (window-to-wall Ratio)	0.33
Floor-to-Ceiling Height	9 ft (2.74 m)
Floor-to-Floor Height	13.12 ft (4 m)

Idealized operable shades, controlled to meet the illuminance set-point using daylight where possible, were added to both models.

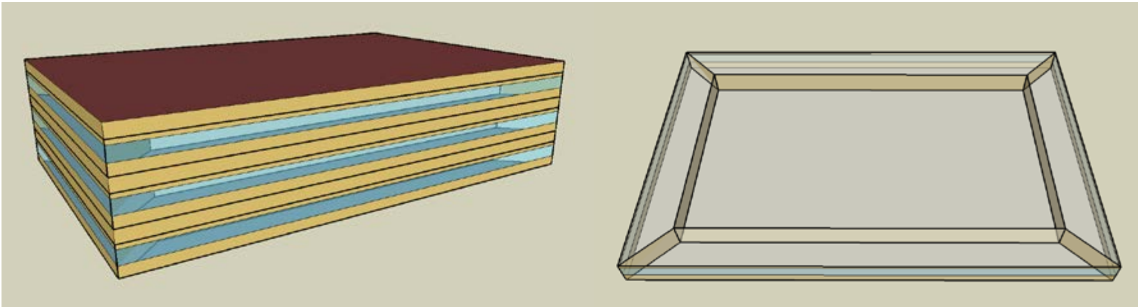


FIGURE 24. BUILDING GEOMETRY AND FLOOR PLAN LAYOUT

Mechanical system information

The HVAC system in the original model was modified to meet the needs of this study.

(a) Radiant panel system

Radiant ceiling panels covering 100% of the conditioned area were added to the model. The heating or cooling capacity is controlled by varying the water flow rate through the panels. A chiller, a cooling tower and a gas boiler were added to the model. The cooling tower has a variable speed fan and can also operate in free convection mode. A waterside economizer, based on a plate frame heat exchanger, was introduced in order to take advantage of free cooling from the cooling tower.

Ventilation air is supplied by three VAV packaged rooftop units, one for each floor, configured as dedicated outside air systems (DOAS) supplying 100% outside air with supply air temperature reset based on outside air dry-bulb temperature. Cooling is provided by direct expansion (DX) in order to avoid the need to produce low temperature chilled water just for dehumidification – radiant cooling systems operate with higher chilled water temperatures in order to avoid condensation and to maximize the use of waterside free cooling. Demand-controlled ventilation (DCV) is employed, with the airflow rate in each zone changing in response to the occupancy.

(b) Baseline VAV system

The packaged DX VAV units in the original DOE reference model were replaced by built-up VAV units with hot water and chilled water coils, enabling the use of water-cooled chillers and gas boilers. The model inputs of the HVAC zone level systems, secondary systems and the primary systems are in accordance with the Title 24 2013 Nonresidential Alternative Calculation Method Reference Manual requirements.

(c) System schedules

- Operation/occupancy schedule: Monday to Friday from 6:00 to 22:00, Saturday from 6:00 to 18:00, Holidays and Sunday off.
- Cooling mode:
    - Operative Temperature Set-point: 75.2°F (24°C) with a night set-up temperature of 86°F (30°C) for the VAV case, 73.4-77.0°F (23-25 °C) with a night set-back

range of 84.2-87.8°F (29-30ºC) for radiant case, i.e. a throttling range of 1.8ºF (1 K)
    - Radiant system supply water temperature range: 60.0-68.0°F (15.6-20.0ºC)
- Heating mode:
    - Operative Temperature Set-point: 69.8°F (21ºC) with a night set-back range of 50°F(10°C) for the VAV case, 68.0-71.6°F (20.0-22.0ºC) with a night set-back range of 48.2-51.8°F (9-11°C), i.e. a throttling range of 1.8°F (1 K)
    - Radiant system supply water temperature range: 86-122°F (30-50ºC)

4.6.2 Results and Discussion

San Jose (California climate zone 4), Los Angeles International Airport (California climate zone 6), Los Angeles Civic Center (California climate zone 9) and Stockton (California climate zone 12) were selected for the study.

Figure 25 compares the annual HVAC source energy consumption in the four climate zones. The percentage number above the red bar indicates either increased or reduced HVAC source energy use for the radiant system relative to the baseline VAV case. A positive value indicates energy savings while a negative number indicates increased energy use. As can be seen from the figure, the energy savings range from 13.9% to 25.5% among the four cities. Figure 26 correlates the energy savings with the Cooling Degree Days (CDD's) of the four cities. As shown in the figure, the energy savings increase nearly linearly with the CCD's.

One main advantage of the VAV system is the use of an air-side economizer, which is much more efficient than the water-side economizer in the radiant system. The fan energy is assumed to be unchanged by the increase in outside air fraction whereas the water-side economizer requires pumping energy, and also fan energy, if the cooling tower is not operating in natural draft mode. As the CCDs increases, the availability of air-side economizer operation decreases while the benefit of using higher chilled water supply temperature in the radiant system starts to emerge. Figure 27 shows the breakdown of the annual HVAC energy consumption. The radiant system uses more energy than the VAV system in the "cooling" category, which includes the energy consumption of the chiller, the cooling tower fan and the DOAS. This is because in the transition and winter seasons, the efficiency of the water-side economizer is less than that of the air-side economizer, as noted. Most of the energy savings for the radiant system come from the thermal energy transport (the fan and pump energy shown in Figure 27). This is because water is a significantly more effective heat transfer fluid than air, due to its higher specific heat capacity and lower viscosity.

As can be seen from Figure 27, the heating energy is a small fraction of the total energy consumption; however, the percentage savings range from 43.5% to 64.4%. This is due to the inherent limitation of single duct multi-zone VAV systems associated with the minimum airflow setting at the VAV terminal unit, which is required to ensure adequate mixing of the supply air and the air in the zone. When serving both core and perimeter zones, the VAV system has to supply cold air year-round to satisfy the cooling needs at the core zones. As the cooling load decreases at the perimeter zones, the supply airflow deceases until it reaches the minimum airflow setting. As the cooling load continues to decrease, the VAV terminal unit has to reheat the supply air to avoid overcooling the space. This form of simultaneous heating and cooling does not occur in radiant systems.

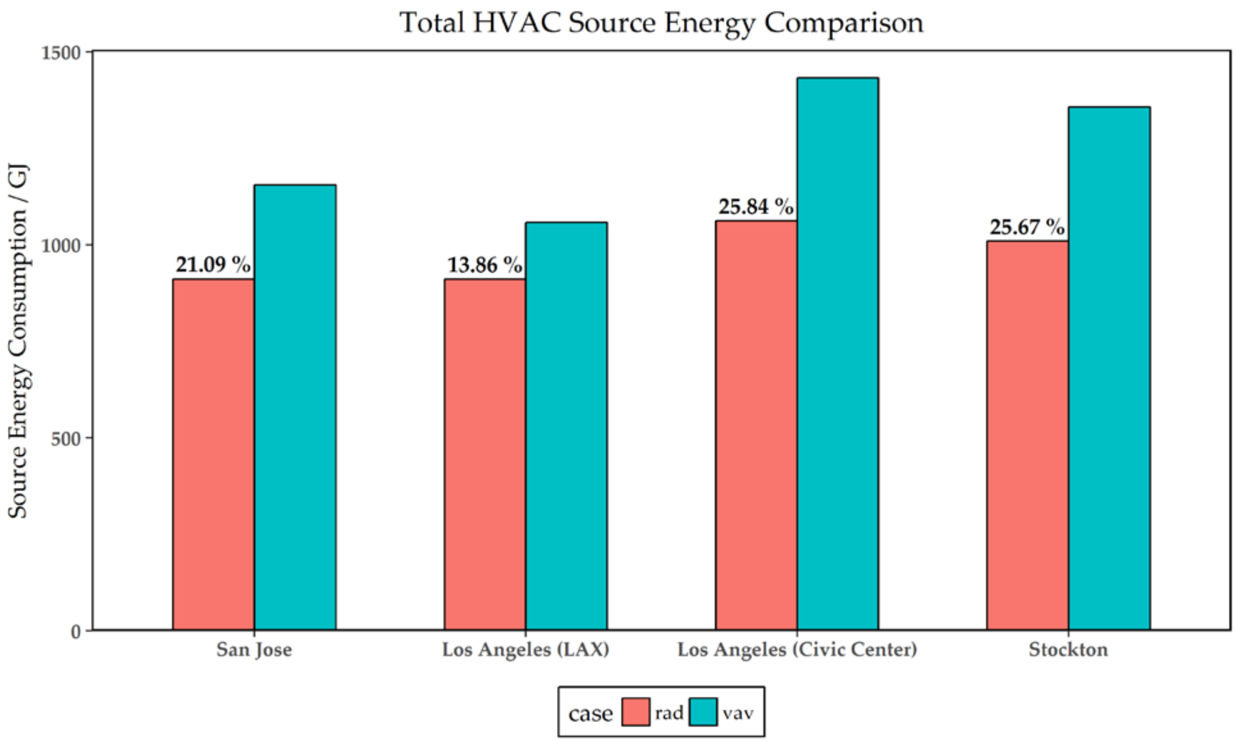
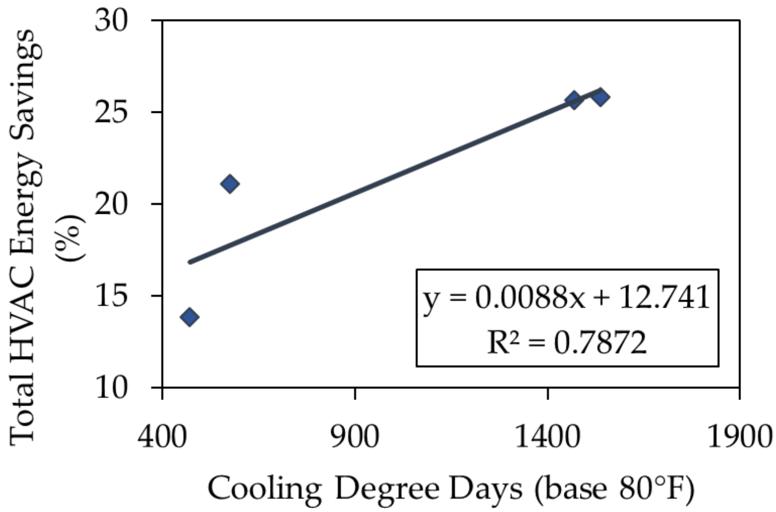


FIGURE 25. TOTAL ANNUAL HVAC SOURCE ENERGY COMPARISON



Cities	CDD	Savings (%)
San Jose	574	21.09
LA Airport	470	13.86
LA Civic Center	1537	25.84
Stockton	1470	25.67

FIGURE 26 TOTAL ANNUAL HVAC ENERGY SAVINGS VS. COOLING DEGREE DAYS

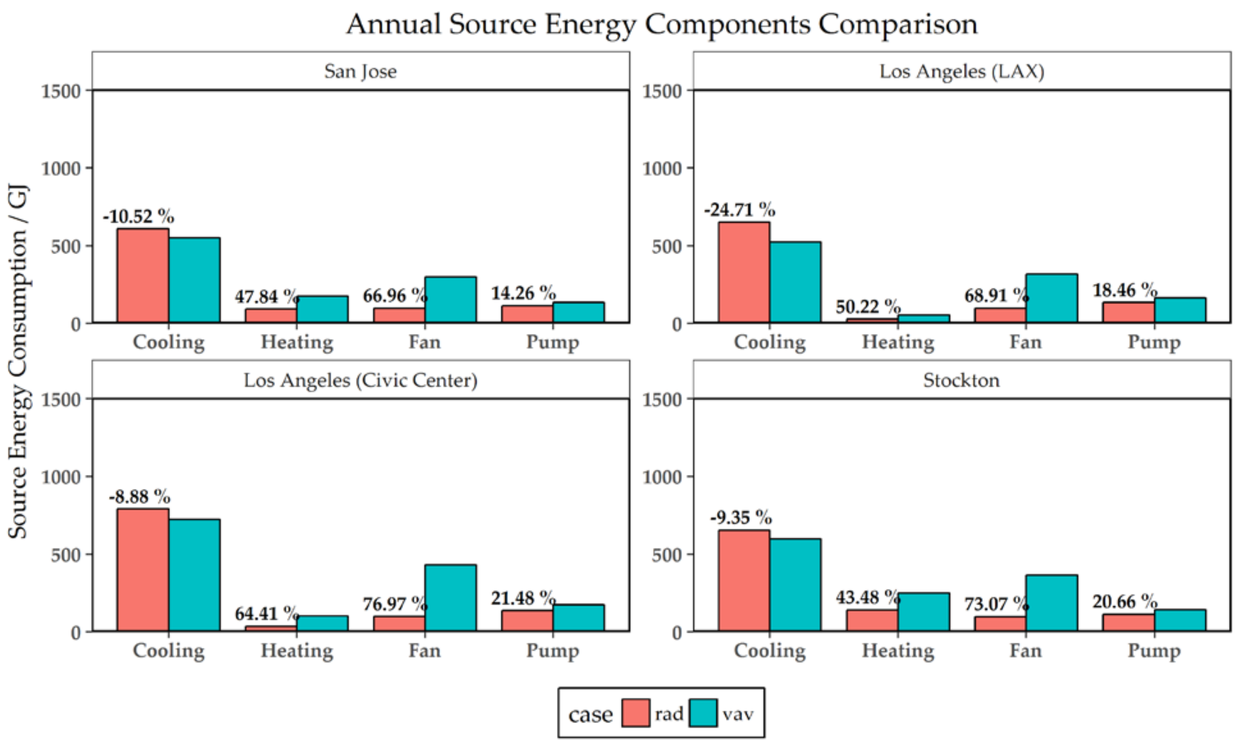


FIGURE 27 ANNUAL HVAC ENERGY BREAKDOWN

Figures 28-31 compare one week of indoor operative temperature in summer (for top floor south facing perimeter zone) and winter (for top floor north facing perimeter zone) for the four cities. The operative temperatures in cooling mode are very similar and the operative temperatures for the radiant system are slightly higher in heating mode, indicating that the superior energy performance of the radiant system was not obtained at the expense of thermal comfort.

A side-by-side comparison of a radiant slab system and a VAV system in an office building on the Infosys campus at Pocharam SEZ, Hyderabad, India, reported in [7], is shown in Figure 32. The east wing has radiant slab cooling and the west wing has a variable-air-volume (VAV) system. There is very little east- and west-facing glazing, so the two halves are effectively identical thermally. The VAV system uses approximately half the energy for cooling and ventilating used by the earlier VAV buildings on the campus but, for several years, the radiant slab system has used 1/3 less energy than the VAV system, as shown in Figure 33 [7]. Occupant satisfaction with the working environment was assessed by Infosys using an on-line survey developed by the UC Berkeley Center for the Built Environment, which showed significantly greater satisfaction with the radiant system than the VAV system – see Figure 34. These energy and occupant satisfaction differences have led Infosys to standardize on radiant cooling for its future buildings. Another advantage of hydronic systems over air systems is that the ducts and the shafts are smaller, since they only have to handle the outside air, leading to cost savings and/or more useable space.

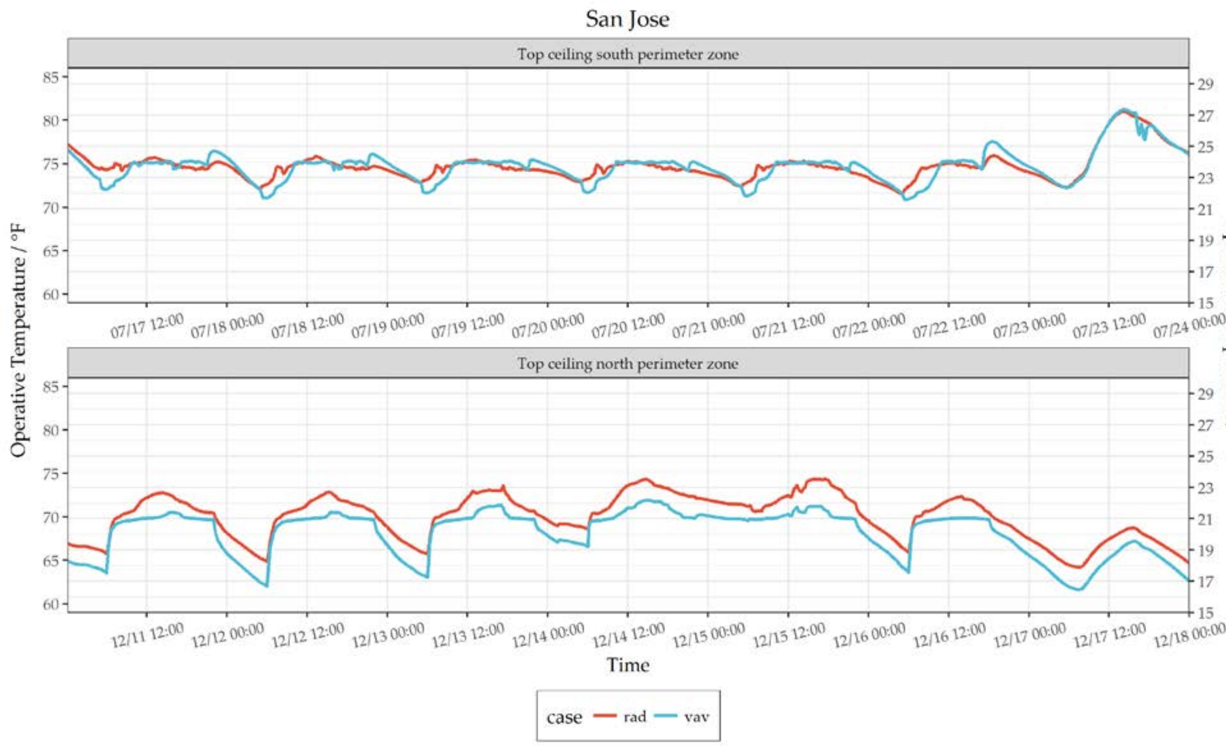


FIGURE 28 ONE WEEK OF INDOOR OPERATIVE TEMPERATURE COMPARISON FOR SUMMER AND WINTER IN SAN JOSE

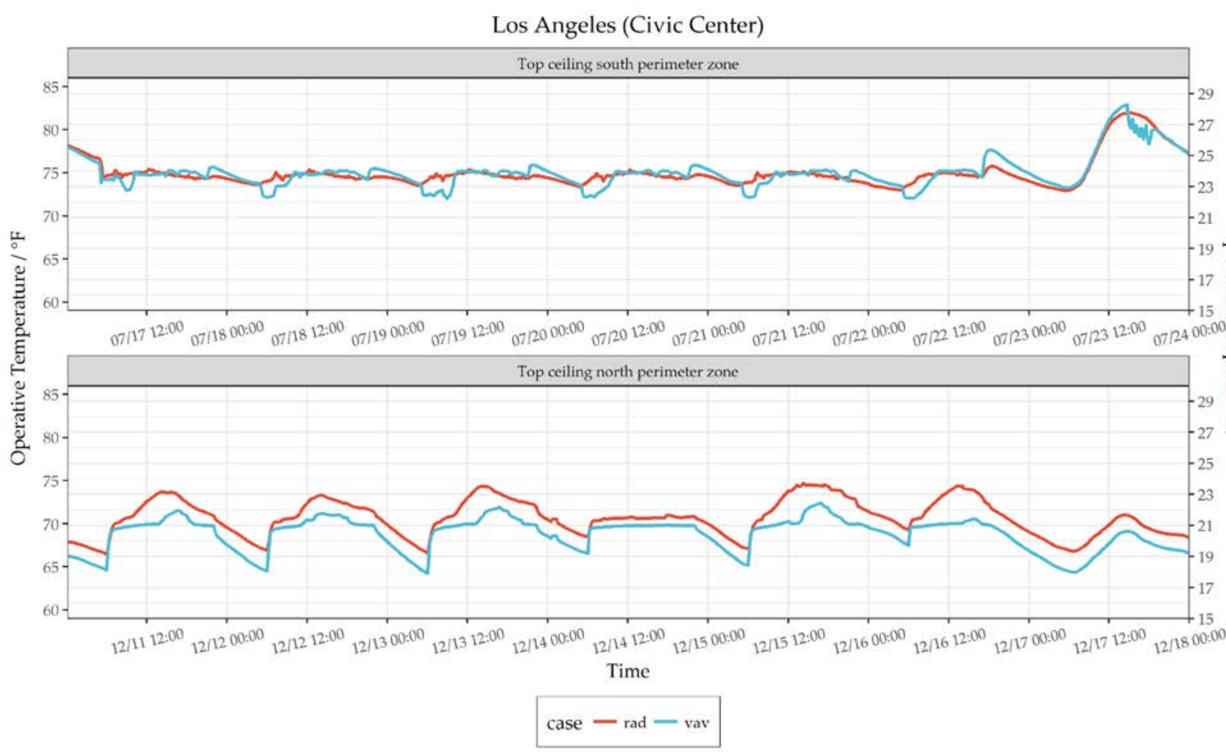


FIGURE 29 ONE WEEK OF INDOOR OPERATIVE TEMPERATURE COMPARISON FOR SUMMER AND WINTER IN LA CIVIC CENTER

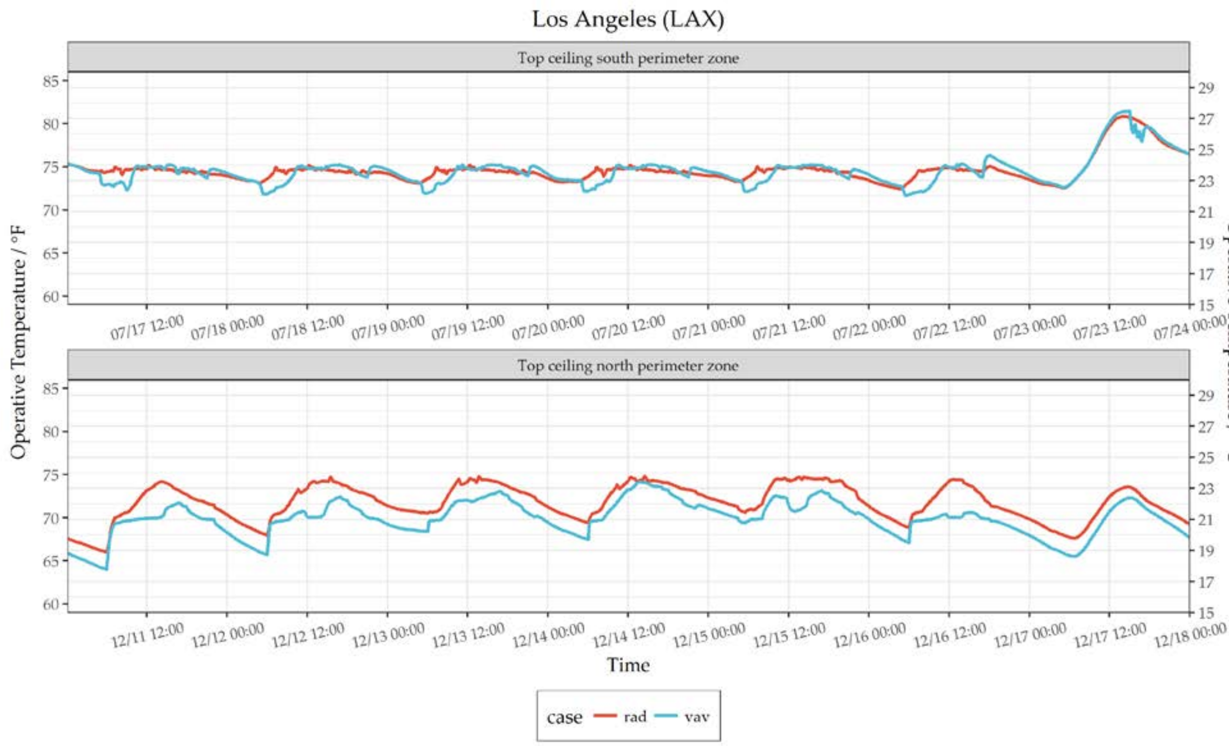


FIGURE 30 ONE WEEK OF INDOOR OPERATIVE TEMPERATURE COMPARISON FOR SUMMER AND WINTER IN LA AIRPORT

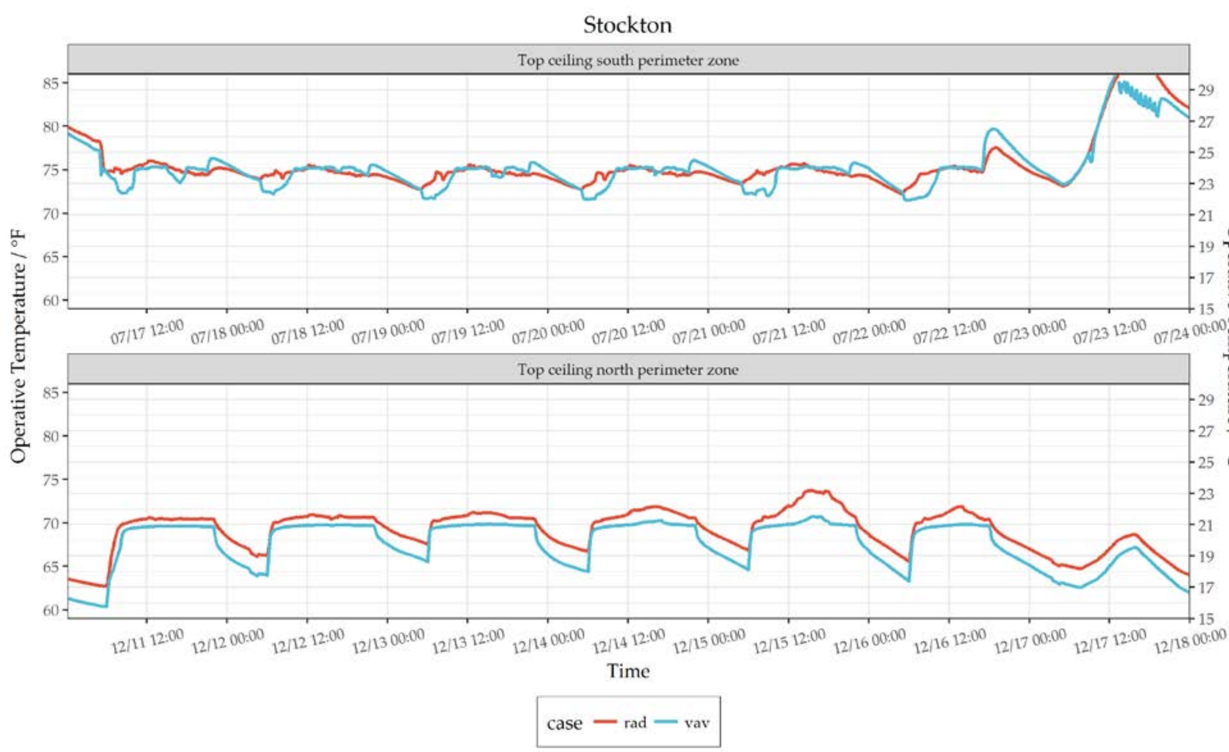


FIGURE 31 ONE WEEK OF INDOOR OPERATIVE TEMPERATURE COMPARISON FOR SUMMER AND WINTER IN STOCKTON

FIGURE 322 THE OFFICE BUILDING IN HYDERABAD, INDIA USED TO COMPARE THE ENERGY PERFORMANCE OF A RADIANT SLAB TO A VAV SYSTEM

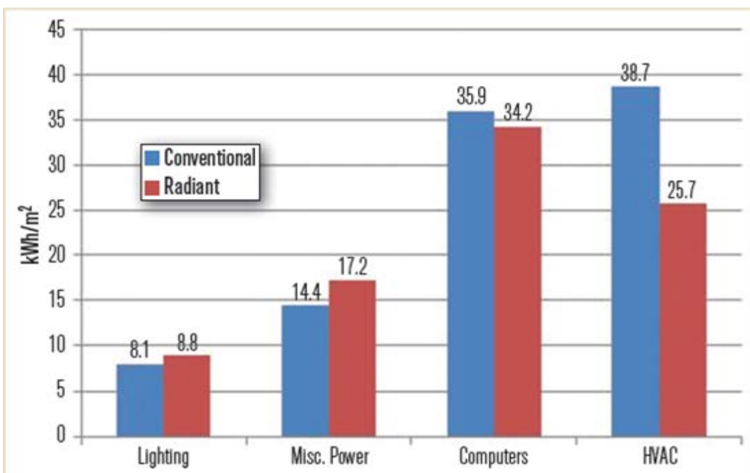


FIGURE 333 COMPARISON OF THE ENERGY USE IN THE TWO HALVES OF THE OFFICE BUILDING IN HYDERABAD. IN SPITE OF THE INTERNAL GAINS BEING SLIGHTLY HIGHER, THE COOLING ENERGY USED BY THE RADIANT COOLING SYSTEM IS 1/3 LESS THAN THAT USED BY THE CONVENTIONAL VAV SYSTEM [7].

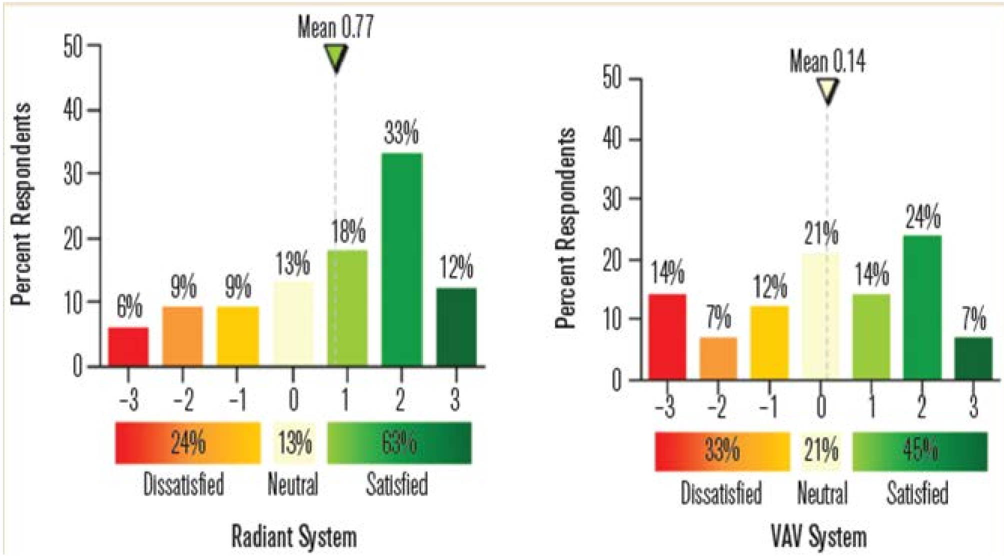


FIGURE 344 RESULTS OF AN OCCUPANT COMFORT SURVEY OF THE SDB1 OFFICE BUILDING SHOWING SIGNIFICANTLY GREATER SATISFACTION WITH THE RADIANT SYSTEM THAN THE VAV SYSTEM [7]

4.7 TASK 7: INTEGRATED CONTROL OF RADIANT CEILING PANELS, ACTIVE FAÇADES AND DAYLIGHT HARVESTING

The goal of Task 7 was to assess the benefits of integrating HVAC, facade and lighting control. The basis of the integrated control is the use of the HVAC mode (heating/floating/cooling) to determine whether to maximize or minimize solar gain, subject to glare and illuminance requirements. Radiant ceiling panel systems were selected as the HVAC system in this study in order to explore whether the energy benefits of such systems could be further enhanced by integrated control.

4.7.1 SIMULATION SETUP

In this study, the baseline model is the same as the radiant system model used in Task 6. The daylighting and façade controls are presented in more detail below in order to explain the operation of the integrated control strategy.

Radiant panel system

- Schedule: Monday to Friday from 6:00 to 22:00, Saturday from 6:00 to 18:00, Holidays and Sunday off.
- Heating mode
  - Set-point: 68-71.8°F (20-21ºC), i.e. a throttling range of 1.8 F (1 K)
  - Supply water temperature range: 86-122°F (30-50ºC)
- Cooling mode
  - Set-point: 73.6-80.8°F (22-26ºC), i.e. a throttling range of 7.2 F (4 K)
  - Supply water temperature range: 60-68°F (15.5-20ºC)

Shading and daylighting controls

- Window type: double glazing with air in the gap
- Shading: motorized blinds or shade. (In current version of EnergyPlus, modulating control of motorized blinds or shade to meet the illuminance set-point is not available; switchable glazing (electrochromic) was modeled as a proxy for the modulating control of the motorized blinds or shade)
- Daylighting control settings:
    - Reference point: 5 ft (1.52 m) from the exterior wall and 2.6 ft (0.8 m) above the floor.
- Illuminance set-point:
    - Baseline: 500 lux year-round
    - Integrated control: 500 lux when radiant system is in cooling mode or is floating in the dead band between heating and cooling; 10,000 lux when the radiant system is in heating mode
- Maximum allowable discomfort glare index: 22
    - Glare calculation azimuth angle: 90 degree clockwise from zone y-axis (i.e. direction of view is parallel to the window)

Control algorithms

Four control algorithms were studied; each algorithm is summarized in Table 6.

TABLE 6. SUMMARY OF THE CONTROL ALGORITHMS

Control algorithm	Description	Daylight Illuminance Set-point (lux)
Baseline (no integration)	Motorized blinds/shade and artificial lighting are sequenced to maintain a fixed illuminance set-point of 500 lux in the zone, subject to a maximum glare index of 22.	500 year-round
Seasonal Switch (no integration)	In winter-time, maximize the solar gain in the space, subject to a maximum glare index of 22 and a maximum illuminance of 10,000 lux. Use artificial lighting, as necessary, to maintain 500 lux. In the transition and summer seasons, use the same algorithm as the baseline.	10,000 (November-March); 500 (April-October)
Cooling-triggered integrated control	When the radiant ceiling panels run in cooling mode, use the same algorithm as the baseline; when the radiant ceiling panels run in floating or heating mode, maximize the solar gain in the space, subject to the glare and illuminance constraints.	500 (radiant system is in cooling mode); 10,000 (radiant system is in floating or heating mode)
Heating-triggered integrated control	When the radiant ceiling panels run in heating mode, maximize the solar gain in the space, subject to the glare and illuminance constraints; when the radiant ceiling panels run in floating or cooling mode, use the same algorithm as the baseline.	10,000 (radiant system is in heating mode); 500 (radiant system is in floating/cooling mode)

4.7.2 Results and Discussion

San Jose (California climate zone 4), Los Angeles International Airport (California climate zone 6), Los Angeles Civic Center (California climate zone 9) and Stockton (California climate zone 12) were selected for the study.

Annual Energy Performance Comparison

Figure 35 compares the annual source energy consumption, relative to the baseline, for the three control algorithms in four climate zones. As can be seen from the figure, the heating-triggered integrated control algorithm brings very limited amount of savings in a range of 0.01% to 0.02%. Other cases actually waste energy compared to the baseline case, except the seasonal switch algorithm for Stockton.

The cooling-triggered integrated control use more energy than the baseline for all the four climate zones, with the LAX the most with 0.53% energy waste and Stockton the least with 0.27% energy waste.

The seasonal switch algorithm has better performance in general compared to cooling-triggered integrated control but still wastes energy when compared to the baseline. Only Stockton shows savings (0.04%) when using the season switch algorithm.

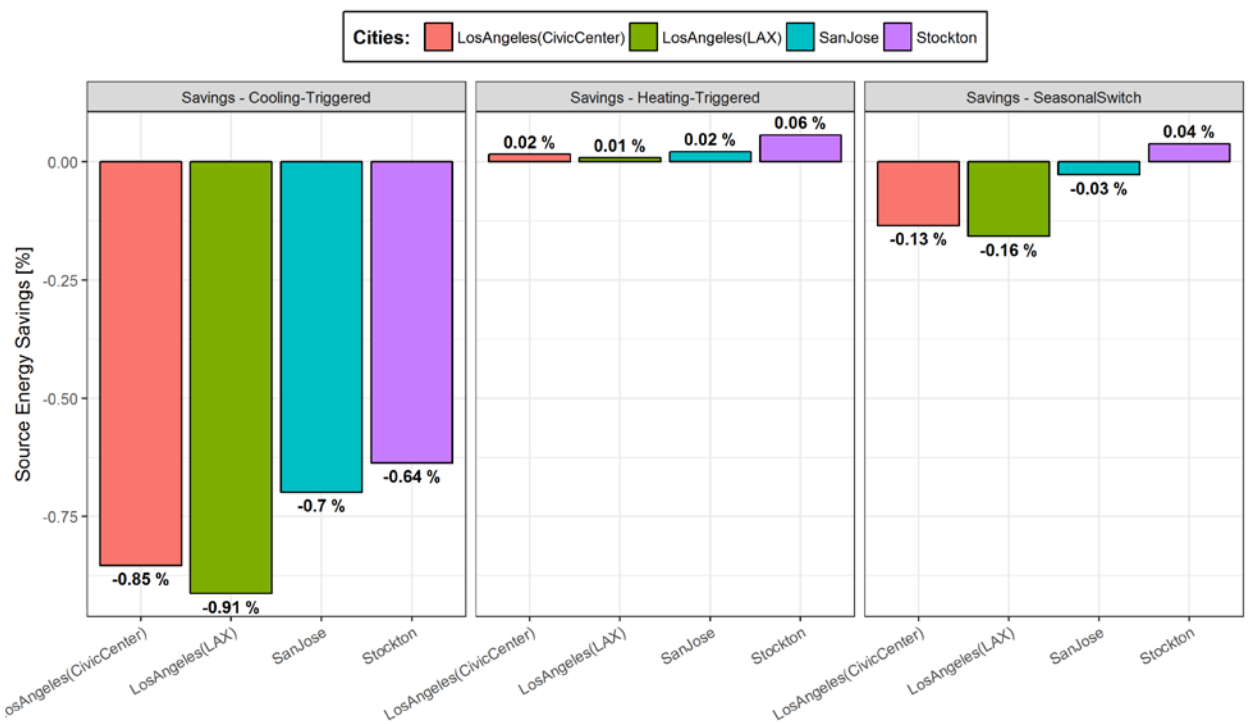


Figure 35. Annual source energy consumption comparison

Figure 36 shows the breakdown of the annual source energy consumption. As can be seen from the chart, though small heating energy savings was obtained, the seasonal-switch and the cooling-triggered integrated control algorithms consume more cooling and pump energy than the baseline, with the cooling-triggered integrated control algorithm being the worst case. This could partially explain the pattern seen in the comparison of the annual source energy consumption, which includes the HVAC components and the lighting. Not shown are miscellaneous equipment (plug loads) and exterior lighting and equipment (elevators), since they are not affected by the integrated control. Together with the HVAC and lighting in the core zones, which account for 59% of the total floor area, ~85% of the source energy consumption cannot be impacted by the integrated control, and so the savings from integrated control, such as they are, have an upper bound of 15%.

In-depth Analysis

In order to understand why the savings are small compared to the 15% upper bound, consider the behavior of a south facing zone in Los Angeles in December. Figure 37 shows the illuminance at the lighting control reference point for the zone. The red dashed lines indicate the two different illuminance set-points: 500 lux and 10,000 lux. In the cases of the Cooling-Triggered algorithm, the illuminance is not attenuated by the active facade because the HVAC system is either in floating or in heating mode. In fact, the system is rarely in heating mode, as can be seen by the very limited number of occasions when the Heating-Triggered algorithm causes the attenuation of the illuminance. The SeasonalSwitching algorithm causes no attenuation in winter.

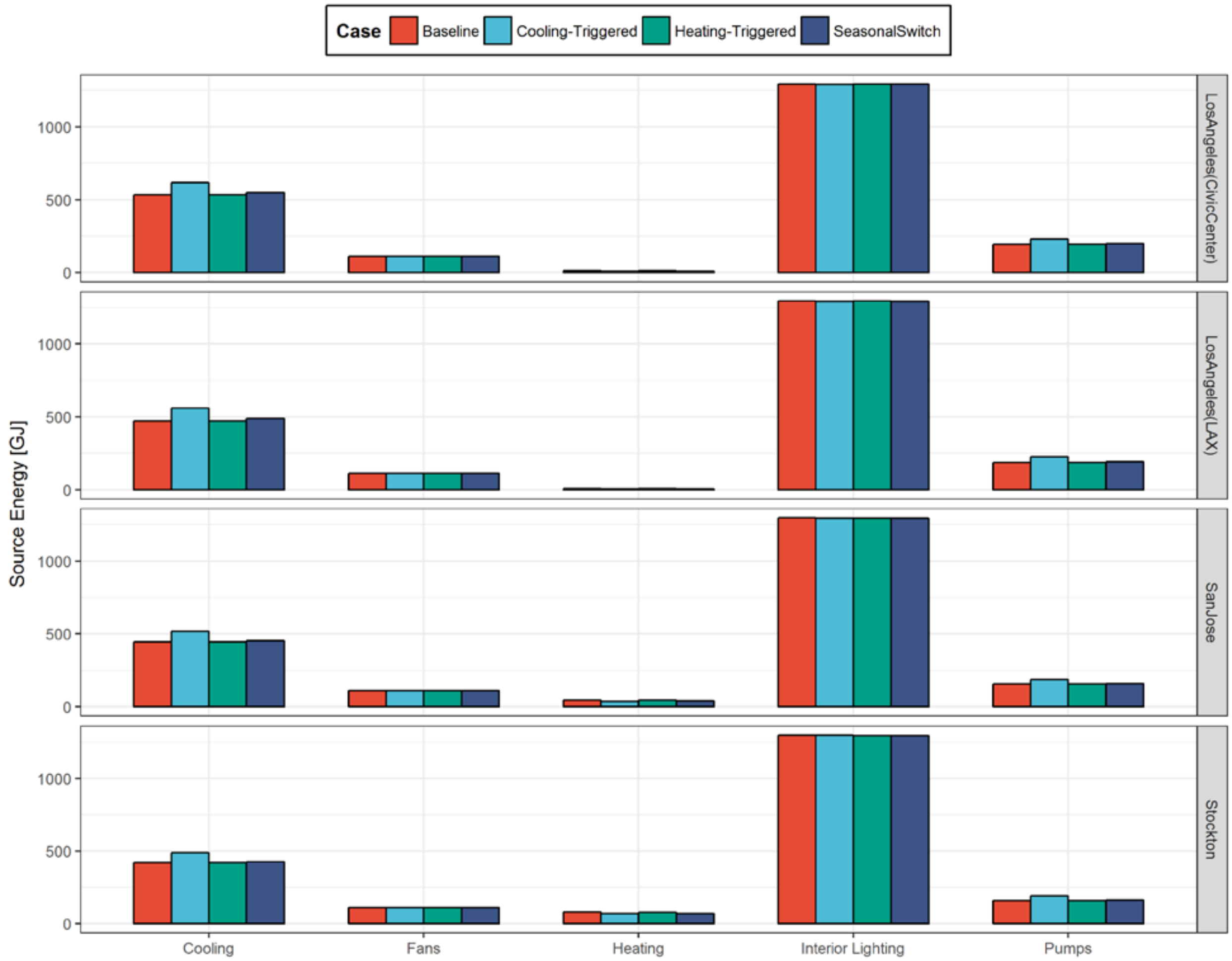


FIGURE 36. ANNUAL SOURCE ENERGY CONSUMPTION BREAKDOWN

Figure 38 shows the corresponding transmitted solar radiation in the south facing zone for the three control algorithms. The fact that the HVAC system is very rarely in heating mode results in solar radiation rarely contributing to reducing the HVAC heating load and hence reducing the energy consumption. This solar contribution is not quite so rare in Stockton, accounting for the slightly greater savings in the less temperature inland climate.

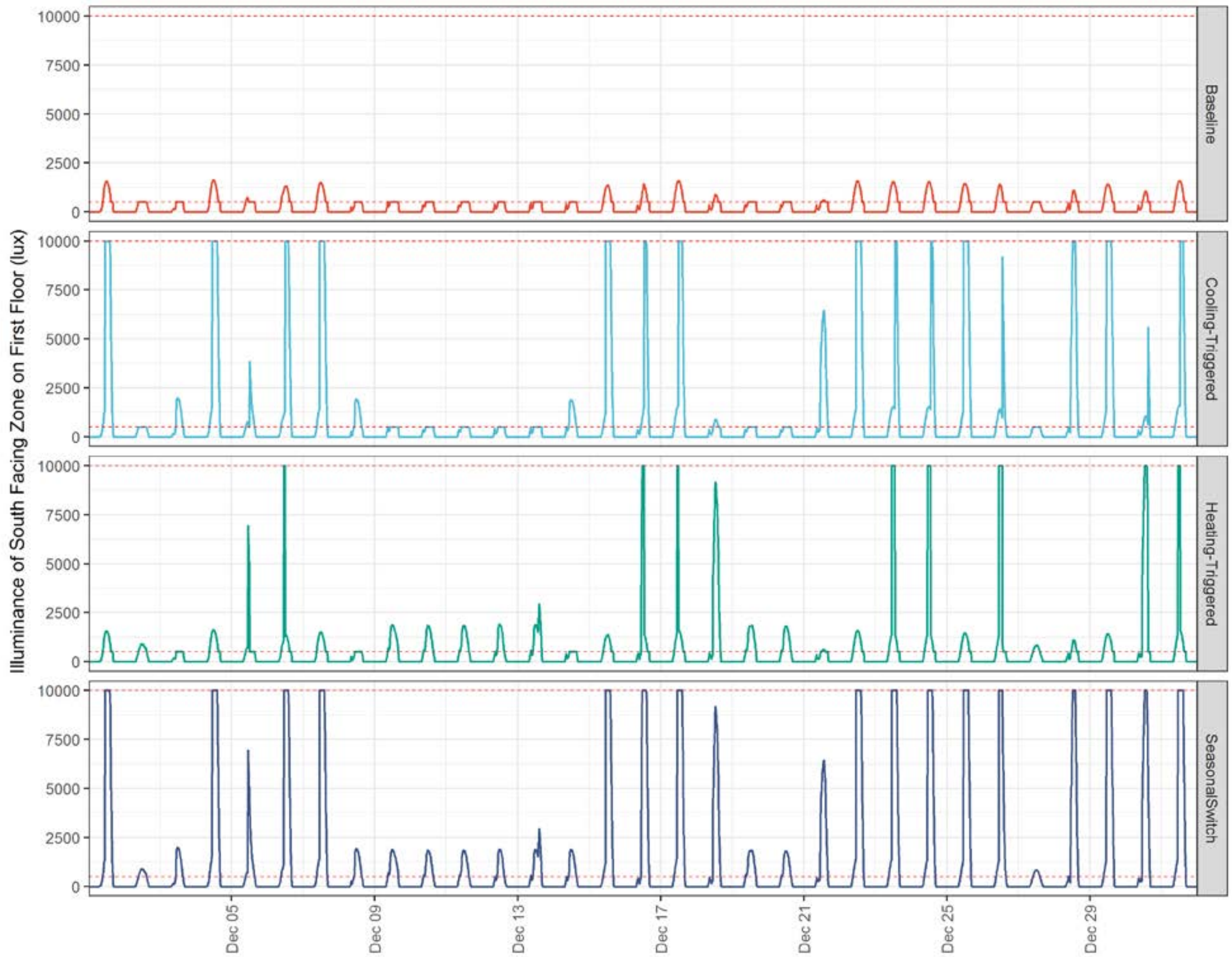


FIGURE 37. ILLUMINANCE AT THE REFERENCE POINT IN DECEMBER FOR THE SOUTH FACING ZONE

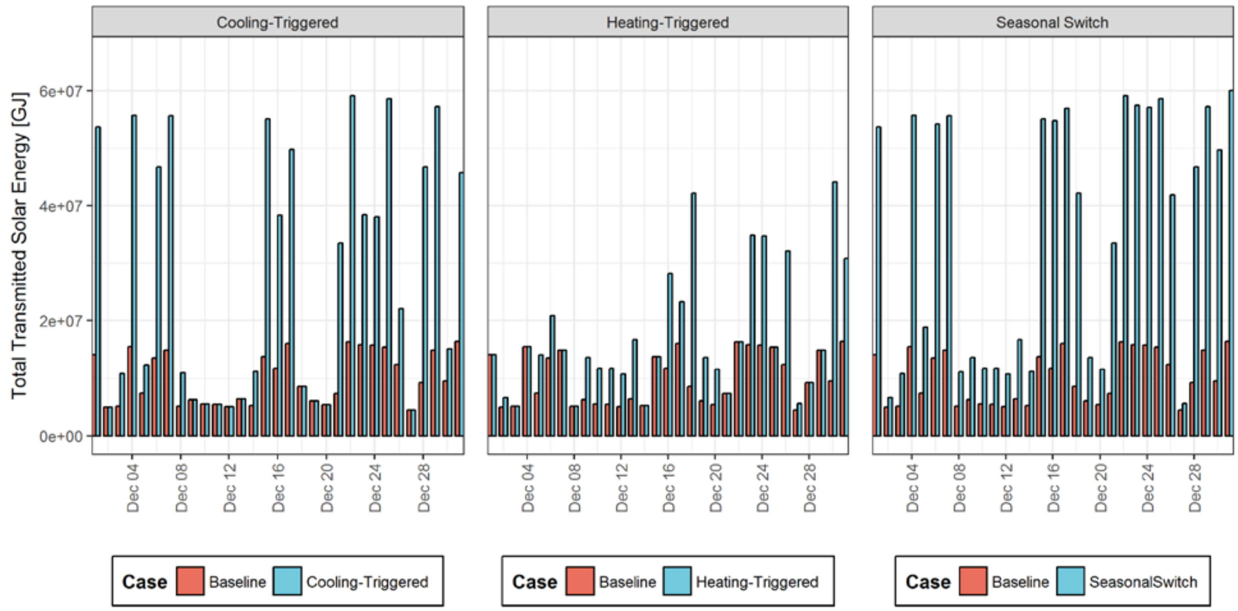


FIGURE 38. TRANSMITTED SOLAR RADIATION IN THE SOUTH FACING ZONE

5 SUMMARY OF RESULTS

Specific findings from the FLEXLAB tests and simulation analyses include:

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1. Radiant ceiling panel system:

a. Temperature stratification is modest in cooling mode, with a maximum value of 2.9°F (1.6 K) being observed during the tests – too small to have a significant impact on thermal comfort. Use of ceiling fans to provide comfort at higher zone temperatures (≤4°F (2 K)) would also serve to reduce any excess stratification.

b. The elevation of zone dry-bulb temperature near the south facade was primarily driven by solar radiation, with the highest value of 2.2°F (1.2K) usually occurring approximately three hours after solar noon, the time lag being due to the thermal capacity of the floor. Again, this effect is too small to have a significant impact on thermal comfort and so there is no need to incur the additional expense of a separate control zone at the perimeter of a 50% window-to-wall ratio unshaded south façade. Simulation analysis could be used to extrapolate this result to west-facing zones and south-facing zones with >50% window-to-wall ratio and shaded facades.

c. The difference between operative and air temperature is quite small if the loads are modest enough to allow them to be met by a radiant system (≤~10 Btu/hr.sf (30 W/m^2). The observed difference did not exceed 0.4°F (0.2 K) in cooling mode. This makes it possible to use air temperature as the controlled variable input to the zone temperature controller instead of operative temperature, with significant cost savings.

2. Radiant floor slab system:

a. The observed stratification effect was significantly greater in the radiant cooling slab system; the largest temperature difference observed was 6.3°F (3.5 K) - approximately twice that observed in the radiant ceiling panel system in cooling mode. This is slightly greater than is permitted by ASHRAE Standard 55.1 (ASHRAE, 2013) but is well within the 13°F (7 K) limit identified by UC Berkeley for conditions near thermal neutrality. Use of ceiling fans to achieve approximate thermal neutrality would also have the effect of significantly reducing stratification.

b. Similar to the radiant panel system, the perimeter effect was also driven by solar radiation; the dry-bulb temperature difference between the front of the cell, adjacent to the window, and the rear of the cell had a maximum value of 1.6°F (0.9 K).

c. The difference between the spatially-averaged operative and dry-bulb air temperatures for the radiant slab system is larger than in the case of the radiant panel system, with a maximum value of 1.1°F (0.6 K).

d. The use of thin carpet requires the supply chilled water temperature to be reduced by ~2°F (1K), with a corresponding reduction in the opportunities for free cooling. An increase in condensation risk may result.

e. Interruption of system operation for a period of ~5 hours in the afternoon and early evening appears to have only a slight effect on cooling performance, enabling the HVAC load (excluding ventilation) to be shifted away from times of peak electricity demand.

3. Energy savings potential of radiant ceiling panel systems vs. Variable Air Volume systems

EnergyPlus simulations of the medium-size office building model from the DOE Commercial Reference Building Models set were performed for a radiant ceiling panel system and a variable-air-volume (VAV) system. Simulations were performed for four climates: San Jose (California climate zone 4), Los Angeles International Airport (California climate zone 6), Los Angeles Civic Center (California climate zone 9) and Stockton (California climate zone 12) and the HVAC energy savings range from 14% to 26%. The radiant system savings are primarily due to the air handling unit fan energy reduction outweighing the additional cooling tower fan energy required to obtain free cooling.

A side-by-side comparison of a radiant slab system and a VAV system in an office building on the Infosys campus at Pocharam SEZ, Hyderabad, India, shows that the radiant slab system consistently used 1/3 less energy than the VAV system while providing greater occupant satisfaction with the indoor environment. These energy and occupant satisfaction differences have led Infosys to standardize on radiant cooling for its future buildings.

4. **Integrated control**

The potential benefit in terms of whole-building energy performance of integrating the control of an active façade with the control of the HVAC system in a typical medium-sized office was found to be very small. Factors include:

- The dominance of the core zones (~60% by floor area)
- HVAC source energy consumption in the perimeter zones only accounts for ~15% of the source energy consumption for the whole building
- The benefits of active facades are treated separately – this study only considers whether knowledge of the HVAC mode (heating, floating or cooling) can increase the benefit of an active façade
- For the four California climate zones studied in this project, the medium office building is cooling dominated, and thus maximizing solar gain in heating mode has limited benefits. The benefits are further limited by the limited solar radiation when the building is in heating mode.
- Simple integrated control based on HVAC mode can produce negative savings. This may be due to increased solar gain to offset morning heating load being stored by the thermal capacity of the fabric of the building, leading to increased afternoon cooling loads. Predictive control could address this issue, if the benefits were found to be worthwhile.

In hindsight, it might have been more useful to focus on small commercial buildings, which are more envelope-dominated, and hence likely to show greater savings. The motivation to assess the benefits of integrated control for buildings with radiant systems, thus providing some continuity with the FLEXLAB work, may have been counterproductive, since the chilled and hot water plant required for radiant systems are typically not used in small commercial buildings. A better application for integrated control may be small buildings with conventional direct-fired heating and direct expansion cooling, though producing active façade systems, with suitable controllers, that are robust enough and inexpensive enough for the small commercial sector is likely to be challenging.

6 RECOMMENDATIONS

1. Expand current knowledge and information regarding radiant systems:

   - Verify and extend the preliminary findings of this study through case studies of radiant systems in California commercial buildings that span design, construction and operation and address both technical and financial issues

2. Promote the appropriate deployment of radiant systems in commercial buildings, based on current knowledge, through:

   - Design guides and related training classes, based on results from this project, formal case studies and the expertise of the relatively limited numbers of designers and contractors working productively with radiant systems

   - Targeted incentives, including Savings by Design, based on demonstrated benefits; target radiant slab systems for new construction to gain load shifting benefits and target radiant panel systems for relative ease of retrofit

3. Control methods for radiant slabs need to be further developed and tools provided. Three current R&D projects are addressing this need in different ways:

   - A project at UC Berkeley Center for the Built Environment, funded by the CEC's EPIC research program, seeks to better characterize radiant systems and develop improved simple control algorithms

   - A joint LBNL / University of California Berkeley Mechanical Engineering Department project, funded by DOE and the Government of India, has developed a model predictive control (MPC) method and algorithms for the control of radiant slabs and a set of tools based on these algorithms is currently being ported to an open-source computing platform. In the short to medium term, MPC methods may be more robust when used to guide the development of heuristic strategies in the design phase than when implemented in control systems in buildings,

   - LBNL is leading a project, OpenBuildingControl, funded by DOE and CEC, to develop a controls design tool for buildings. The tool will enable designers to compare control strategies and configure sequences of operations for a variety of systems, including both conventional and MPC strategies for radiant system control.

   These projects can be expected to reduce the current difficulties in the design of the controls for radiant slabs. PG&E / State-wide programs can then help to disseminate these control strategies and tools and promote the adoption of radiant slabs, in particular, based on proven design methods and control strategies.

4. Further research on quantifying the impact of different factors on the benefits of integrating HVAC, façade and lighting:

   - Verify climate impact on implementation benefits of integrated control, especially for smaller buildings and colder climate zones

- Investigate the application of predictive control to diurnal load variations, especially for demand response
- Investigate the potential role of integrated predictive control for very high performance buildings and fluctuating availability of electricity due to increasing presence of power generated from renewable sources on the grid.

APPENDIX 1: INDUSTRY ADVISORY GROUP

The active members of the group were:

- Lisa Gelfand (Gelfand Partners Architects)
- Lisa Heshong (TRC Solutions)
- Kevin Hydes (Integral Group)
- Kyle Konis (University of Southern California)
- Claire Maxfield (Atelier Ten)
- Maurya McClintock (MCC Facades)
- Kevin Powell (GSA)
- John Pulley (Buro Happold)
- Alan Shepherd (PAE Consulting Engineers)

The group responded to a detailed web-based questionnaire on needs and priorities for testing low energy cooling systems and integrated control of HVAC, facades and lighting. An analysis of the responses was performed and the main outcomes can be summarized as follows:

- Testing of radiant slabs, radiant panels, natural ventilation / mixed mode and integrated control are all important
- Perimeter effects with radiant systems are of particular interest
- There is strong interest in both sizing and condensation control for radiant panels, slightly less with radiant slabs
- There is substantial interest in testing control strategies for mixed mode systems
- There is substantial interest in testing load shifting with radiant slabs
- There is significant interest in testing ceiling fans with radiant slabs, slightly less with radiant panels

APPENDIX 2: CALCULATION OF DERIVED QUANTITIES USED IN THE ANALYSIS

Operative temperature

As noted above, the operative temperature was used to control the output of the radiant systems. As prescribed in ISO 7726[3], in most practical cases where the relative velocity is small (<0.2 m/s) or where the difference between mean radiant and air temperature is small (<4 °C), the operative temperature, Top, can be approximated sufficiently accurately by the mean value of the mean radiant temperature, Tr, and the dry-bulb temperature, Ta.

EQUATION 1. OPERATIVE TEMPERATURE

$$T_{op} = \frac{1}{2}(T_r + T_a)$$

The absolute mean radiant temperature was calculated from the measured absolute temperature of the globe, T_g, and the absolute dry-bulb temperature using the general equation below, cited in ISO 9976 [3]:

EQUATION 2. MEAN RADIANT TEMPERATURE IN TERMS OF GLOBE TEMPERATURE - GENERIC

$$T_\mathrm{r} = \left[\frac{\mathrm{h_c}}{\sigma\varepsilon}\left(T_\mathrm{g} - T_\mathrm{a}\right) + T_\mathrm{g}^4\right]^{0.25}$$

specifically, the relationship used by Benton [4]:

EQUATION 3. MEAN RADIANT TEMPERATURE IN TERMS OF GLOBE TEMPERATURE - SPECIFIC

$$T_\mathrm{r} = \left[\frac{6.32D^{-0.4}V^{0.5}}{\sigma\varepsilon}\left(T_\mathrm{g} - T_\mathrm{a}\right) + T_\mathrm{g}^4\right]^{0.25}$$

where D is the diameter of the globe in meters, ε is the emissivity of globe, σ is the Stefan-Boltzmann constant (5.67 x 10^8 $Wm^{-2}K^{-4}$), and V is air speed (m s^{-1}).

APPENDIX 3: EXPERIMENT SCHEDULE AND SETTINGS

System type	Task No.	Test Date	Test condition				
			T_{op}	Ceiling fan	Flow rate (per sub-ciruit for slab)	Chilled water supply temp	Circulate pump
RCP	1	3/25/2016	73.4°F	Off	1.94 gpm (Cir. 1&2) 1.47 gpm (Cir. 3&4) 1.36 gpm (Cir. 5&6)	Not controlled	Always on
RCP		3/26/2016	77.0°F	Off	(as above)	Not controlled	Always on
RCP		3/27/2016	75.2°F	Off	(as above)	Not controlled	Always on
RCP		3/28/2016	75.2°F	Off	(as above)	Not controlled	Always on
RCP		3/31/2016	74.0~76.0 °F	Off	(as above)	Not controlled	06:00 – 19:00 (on)
RCP		4/1/2016	74.0~76.0 °F	Off	(as above)	Not controlled	06:00 – 19:00 (on)
RCP		4/2/2016	74.0~76.0 °F	Off	(as above)	Not controlled	06:00 – 19:00 (on)
RCP		4/3/2016	74.0~76.0 °F	Off	(as above)	Not controlled	06:00 – 19:00 (on)
RCP	2	4/5/2016	77.0°F After 12:00	11:40-15:10 (Level 2) 15:10-24:00 (Level 3)	(as above)	Not controlled	11:30-19:00 (on)
RCFS		4/6/2016	Not controlled	00:00-10:00 (Level 3)	0.60 gpm 11:00-24:00	55.0°F	06:00-24:00 (on)
RCFS	3	4/7/2016	Not controlled	Off	0.60 gpm 00:00-17:00	55.0°F	Always on
RCFS		4/8/2016	Not controlled	Off	0.30 gpm 10:00-15:00	55.0°F	Always on

					0.40 gpm 15:00-24:00		
RCFS		4/9/2016	Not controlled	Off	0.60 gpm 00:00-23:00	55.0°F	Always on
RCFS		4/10/2016	Not controlled	Off	0.60 gpm	59.0°F	Always on
RCFS		4/11/2016	Not controlled	Off	0.60 gpm	59.0°F 00:00-08:00 64.4°F 08:00-24:00	Always on
RCFS	4	4/12/2016	Not controlled	Off	0.60 gpm	64.4°F 00:00-14:00 55.0°F 14:00-24:00	Always on
RCFS		4/13/2016	Not controlled	Off	0.60 gpm	55.0°F	Always on
RCFS	5	4/14/2016	Not controlled	Off	0.60 gpm	55.0°F	14:00-20:00 (off)
RCFS		4/15/2016	Not controlled	Off	0.60 gpm	55.0°F	14:00-19:00 (off)
RCFS		4/16/2016	Not controlled	Off	0.60 gpm	55.0°F	14:00-24:00 (off)
RCFS		4/17/2016	Not controlled	Off	0.60 gpm	55.0°F	00:00-19:00 (off)

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