Title
Comparison of construction and energy costs for radiant vs. VAV systems in the California Bay Area

Authors
Feng, Jingjuan Dove
Cheng, Hwakong

Publication Date
2018-11-15

COMPARISON OF

CONSTRUCTION AND ENERGY COSTS FOR RADIANT VS. VAV SYSTEMS

IN THE CALIFORNIA BAY AREA

November 15, 2018

FINAL REPORT

Prepared by
Jingjuan Dove Feng, P.E., Senior Engineer
Hwakong Cheng, P.E., Principal

Prepared for
California Energy Commission
EPIC-14-009

Taylor Engineering
1080 Marina Village Parkway, Suite 501
Alameda, CA 94501
(510) 749 – 9135

Contents

1 Executive Summary

Building professionals have designed variable air volume (VAV) systems over the past several decades and have learned to design them to be cost effective and energy efficient (PG&E, 2007). In contrast, radiant systems are still unfamiliar to most professionals in the building industry and there are no well-established radiant system design guidelines. Even among the most experienced radiant designers, there is a diverse range of approaches for design and control of the systems (Paliaga, Farahmand, Raftery, & Woolley, 2017), and there is limited information on the cost effectiveness of these different design practices. The objectives of this study were to provide a realistic first cost comparison of the two systems types, to provide designers with feedback on cost-sensitive aspects of radiant system design, and to suggest control and design measures that could improve system energy efficiency by coupling the first cost comparison with predicted energy costs.

To provide a realistic comparison, alternative radiant and variable air volume (VAV) HVAC designs were developed for an office building in California that was designed with a radiant system in real life. The building is 4-stories with primarily open-plan offices totaling 112,000 ft^2 and is designed with very low internal loads with LED lighting and plug load management. The modeled building performance has exceptionally low site energy use intensity (EUI) of approximately 12 kBtu/ft^2-yr, far below the median 55 kBtu/ft^2-yr measured performance of office buildings in the same climate zone (U.S. Department of Energy, 2018). The building envelope includes high performance glazing with a window-to-wall ratio of 40 percent and exterior overhangs. The radiant design is a high thermal mass radiant system with tubing embedded in the structural slab for bi-directional heat transfer for both heating and cooling. Ventilation and supplemental cooling are provided by a dedicated outdoor air system (DOAS). The all-air design is a VAV system with hot water reheat coils at the terminals. Both designs are served by an air-source heat pump that can provide simultaneously-generated hot and chilled water. The designs were developed in detail and used as the basis for two sets of HVAC construction cost estimates. Overall results were consistent between the two estimates and averaged in the summary below. Only HVAC and control costs are included and common mechanical elements between the two designs, such as toilet exhaust system were not included. Major findings from the cost estimates include:

- The radiant HVAC design has a total cost of \$38.9/ft^2 compared to \$29.9/ft^2 for the VAV design, representing a \$9.0/ft^2 premium for the radiant design.
- The higher costs for the radiant system can largely be attributed to higher piping labor costs for piping and radiant equipment, which itself is \$9.8/ft^2 higher than that for the VAV design.
- There is a \$1.2/ft^2 premium in equipment cost for the radiant system, which is mainly associated with the radiant equipment.
- Though there are some sheet metal cost savings for the radiant design due to smaller ducts, the savings do not outweigh the increased piping costs. The total installed cost for sheet metal is \$4.5/ft^2 for the radiant design, compared to \$7.9/ft^2 for the VAV design.

- As much of the cost premium for the radiant design is associated with piping labor, the premium is more pronounced in the San Francisco Bay area with its high labor rates at about $120/hr. For the estimated national average labor rate of $85/hr, the premium for radiant is $6.8/ft^2, compared to the VAV system.

The high installed cost for the radiant equipment is partly a reflection of the current radiant manufacturers' pricing strategies and the contractors' bidding practices. The radiant market is relatively small and immature in the United States. Radiant system costs are likely to decrease due to economies of scale as the market grows and as uncertainties decrease as the design and construction industry gains more experience.

Though there is wide variability in how radiant systems are designed today, the radiant system in this study was intended to be representative, and one that carefully considers first cost. Nevertheless, the overall cost results are only directly applicable to the two designs that were studied, and care should be taken when applying these results broadly. Climate-specific factors and other design alternatives have a significant impact on first cost and energy use. Alternative design approaches are discussed that may reduce first cost and/or energy cost.

For designers, some aspects of the radiant system design have more significant impact on costs and warrant careful attention. These considerations include:

- Consider the use of radiant mats, instead of traditional radiant loops, to reduce cost through reduced labor. However, radiant mat designs may not be practical or as cost effective for buildings with smaller or oddly-shaped zones.
- Increase radiant tube spacing if possible to reduce material and labor costs, in particular for conventional loop designs. With extended operation, radiant slabs with wider spacing may achieve similar thermal performance as slabs with smaller spacing.
- Strategically design hydronic distribution systems to minimize total pipe length. We compared the installed cost differences for two different approaches: a single set of pipe risers vs. multiple pipe-risers. The former relies on a single set of larger risers and long horizontal distribution runs on each floor, whereas the latter employs multiple sets of smaller risers strategically located to minimize the overall amount of pipe length and overall piping costs by $2.5/ft^2 for the case study building. This strategy may reduce construction cost for any system with distributed piping but is particularly critical for high thermal mass radiant since there are both chilled and hot water pipe distribution systems.
- The study building utilizes a four-pipe system to each radiant zone in the baseline radiant system. Many designers employ a 2-pipe distribution approach or a combination of 2-pipe and 4-pipe approach. If the latter approaches were used, designers may need to consider the potential thermal comfort impacts. More research and design guidance is needed to help designers decide which approach works best for their buildings.
- Utilize large radiant zones to minimize the number of changeover assemblies to reduce the cost of the radiant design but may potentially sacrifice comfort depending on the layout. This is another area that needs more research and guidance.

- The middle floors of a thermally active multi-story building will generally have both the floor and ceiling as active radiant surfaces, whereas the ground floor may only have the ceiling activated if radiant tubing is not installed in the slab-on-grade (or similarly the top floor may only have the floor activated if radiant tubing is not installed in the roof). The N+1 slab, i.e. radiant in slab-on-grade floor or in-roof layer, adds significant cost. For the case study building, adding radiant in the slab-on-grade would increase the total cost by about $3.2/ft^2.
- For high thermal mass radiant system designs, there may be an opportunity to reduce the capacities of central plant equipment if load shifting control strategies are to be implemented. Though theoretically possible, this does not appear to be common design approach today, likely due to perceived risk of capacity shortfalls. If this is proven to be acceptable in the future, there would be some savings in central plant equipment costs.

Energy models of the two designs were developed in EnergyPlus to evaluate the corresponding energy and comfort performance. Radiant system energy use and energy cost vary significantly depending on the specific control strategy employed for the radiant and DOAS equipment. For this study, we implemented a set of radiant slab control sequences that modulate slab temperature settings based on zone conditions and allows for load shifting by locking out the radiant slab during certain periods of the day. Though not fully optimized, we evaluated a range of control settings and report those that provided the best energy performance and comparable thermal comfort to the modeled VAV design. The VAV design models best practice control strategies as described in ASHRAE Guideline 36.

- The annual simulation results show that the total site HVAC energy use is 16.2% higher for the radiant system (2.9 kBtu/ft^2) than the VAV design (2.5 kBtu/ft^2). The VAV design has significantly lower cooling energy use and benefits from the opportunity for free cooling from the airside economizer with mild San Francisco weather. The radiant design has lower heating energy use but slightly higher fan energy use, compared to the VAV design. DOAS fan are commonly expected to use less energy than VAV fans because of the much lower design airflows but, in fact, the opposite is often true due to the fact that VAV systems generally operate for the majority of time at lower part loads, where fan laws and differences in sizing result in significantly lower fan power than at design. The DOAS fan energy is significantly higher in this case, even with the DOAS duct mains and risers oversized compared to the VAV system.
- The annual HVAC electricity cost for the radiant design is 8.0% lower than for the VAV design, $1.12/ft^2 for radiant compared to $1.22/ft^2 for VAV design. The energy cost savings for the radiant design are due to reduced demand charges associated with peak demand shifting as the radiant slabs are only active from 6 am to 12 pm. Operating the radiant slab systems during different periods of the day may further reduce the total energy cost. For example, if running the slab from midnight to 10 am, the energy cost could be reduced to 1.06/ft^2, but with decreased comfort performance.
- Control sequences have significant impacts on the overall HVAC energy performance, and, in fact, some of the control approaches commonly used in the industry appear to be quite

energy inefficient. For example, the radiant design site energy use ranged from 2.7 kBtu/ft^2 to 4.4 kBtu/ft^2 for the study building simply by varying the DOAS supply air temperature control approach.

The radiant system design evaluated in this study is intended to be representative of current typical and good practice, though it also reflects novel control strategies that are not yet common in practice. However, there are many opportunities to improve the energy performance of radiant systems. Designers should consider the following:

- In mild climates, such as the Bay Area in California, HVAC designs should take advantage of the benefits of free cooling as much as possible either with airside or waterside economizers. The 2019 version of California Title 24 will newly require economizers for radiant systems above a certain size threshold. The lack of an economizer in the radiant design has a large effect on overall energy consumption. Designers should use a holistic approach that in cooperate design features would facilitate the use of a load shifting strategy such that the plant equipment size could be reduced to offset the cost of the waterside economizer.
- High thermal mass radiant systems allows great opportunity for the load shifting strategy. The benefits include significant savings of operation cost and installed cost by allowing equipment size to be reduced. With load shifting strategy, it is not as important for the plant to meet the instantaneous cooling or heating load. The plant can operate longer hours if needed to cool or heat the radiant slab to a prescribed temperature setpoint. From a building design perspective, one of the key elements to facilitate the adoption of load shifting strategy is to limit loads, in particular, solar gain in west and south perimeter zones to avoid space temperature spikes in late afternoon.
- Decoupling the cooling source for the radiant slab and the DOAS system, particularly when humidity is of concern, may provide improved efficiency for the cooling plant serving the radiant system by allowing the chilled water temperature to reset higher.
- The DOAS supply air temperature (SAT) control should include space humidity monitoring logic such that the supply air temperature can be reset higher when latent load is not a concern. This can significantly extend the free cooling period and effectively push the radiant slab to take on more cooling load. If the DOAS has a heating coil, a large deadband between heating coil setpoint and cooling coil setpoint can significantly reduce energy waste.

2 Introduction

High thermal mass radiant cooling and heating systems have become a popular alternative to all-air HVAC systems but there is limited information on the cost effectiveness of this technology compared to all-air systems. As part of the California Energy Commission (CEC) Electric Program Investment Charge (EPIC) project *Optimizing Radiant Systems for Energy Efficiency and Comfort*, and in conjunction with the Center for the Built Environment (CBE) at University of California, Berkeley, Taylor Engineering conducted research to compare the first costs and energy costs for high thermal mass radiant and variable air volume (VAV) with reheat. The objectives of this study were to provide a realistic fist cost comparison of the two systems types, and to provide useful information about aspects of radiant system design that have important impacts on first cost.

To provide a realistic comparison, an experienced mechanical engineering team developed comprehensive designs of each HVAC system type for a real building in California. The designs include construction document level HVAC design drawings, as well as control schematic drawings. Then, these drawings were sent to two experienced mechanical contractors to provide detailed HVAC construction cost estimations independently. The designs were also used for annual building simulations to evaluate their energy and comfort performance. In addition to comparing first costs of the base design, we evaluated various options for the radiant system that may have a large impact on first costs.

2.1 Background and Literature Review

Building professionals have designed variable air volume (VAV) systems over the past several decades, and have learned to design them to be cost effective and energy efficient (PG&E, 2007). In contrast, radiant systems are still unfamiliar to most professionals in the building industry and there are no well-established radiant system design guidelines. Even among the most experienced radiant designers, there is a diverse range of approaches for design and control of the systems (Paliaga, Farahmand, Raftery, & Woolley, 2017), and there is limited information on the cost effectiveness of these different design practices.

Anecdotally, we learned from some building professionals that installed costs of radiant systems are comparable to VAV systems, while others reported that the first cost premium for radiant systems is about \$10/ft^2, compared to VAV (Cho, 2017). We are not aware of any straightforward comparison of the two systems that includes both equipment and installation cost in the United States.

One study reported that the costs to install a VAV system vs. a radiant system are comparable, around \$5.1/ft^2, in an office building in India (Sastry & Rumsey, 2014). However, labor and equipment costs in India are much lower than other countries, so the comparison may not be applicable to buildings in the United States.

A report by the Pacific Northwest National Laboratory evaluated the simple payback of a package of energy saving measures that included embedded surface radiant slab system (i.e. with insulation to decouple the slab from the building structure) with a Dedicated Outdoor Air System (DOAS) (Thornton, Wang, Lane, Rosenberg, & Liu, 2009). Based on the experiences of several engineering consulting firms, the study assumed the total cost of radiant system equipment, including plant equipment, averages \$9.31/ft^2 for medium office buildings. The equipment cost for VAV system was estimated to be \$5.23/ft^2. The report also studied construction cost, but it lumped the cost of all energy saving measures evaluated such as upgrading building envelope, shading, and reducing plug load and improving lighting.

In a report by the National Renewable Energy Laboratory that evaluated strategies to achieve 50% energy savings in large office buildings, the equipment and construction cost of a radiant system with DOAS for the 90.1 prototype low-rise large office building was estimated to be \$22.1/ft^2, and the baseline VAV system cost was \$18.6/ft^2. (Leach, Lobato, Hirsch, Pless, & Torcellini, 2010). In the radiant design, the radiant slab and its associated hydronic system cost was \$10.89/ft^2. The source of the data was from RMH Group, a Denver-based engineering consulting firm. Even though this report provides installed cost breakdown for the major equipment categories, there are limitations in the design and cost estimation approaches used in the study. The study was based on a theoretical floor plan, simple occupancy patterns, and a conceptual system design that does not consider the complexities present in all real buildings. In addition, equipment sizing was limited to the major HVAC equipment, and there were no detailed design layouts of the air and hydronic distribution system were presented in the report.

2.2 Report Organization

Sections 3 and 4 of this report provide a summary of the base building and an overall description of the radiant and VAV systems. Section 5 provides a detailed summary of the system design criteria. Section 6 presents the results of the first cost comparison and discusses cost implications of radiant system design variations. Section 7 presents the results of energy simulations. Section 8 identifies design and control opportunities that require future research.

3 Base Building

The base building for this study is an actual building located in the San Francisco Bay Area in northern California (California Climate Zone 3, ASHRAE Climate Zone 3C) constructed with a high thermal mass radiant system. Starting with a building designed for radiant is critical because the limited cooling capacity of radiant systems effectively requires a building design that minimizes envelope loads for successful operation. Several other criteria were considered in the selection of the base building:

- Occupancy type: The desired building occupancy type is an office building with an open office floor plan. Though some designers indicate that office buildings are not ideally suited to radiant systems (Paliaga, Farahmand, Raftery, & Woolley, 2017), they represent the largest percentage of the 400 projects in the CBE radiant database (CBE, 2017). In those office buildings, we filtered for the radiant systems that are the primary HVAC system type, as opposed to just in lobbies and atria, where radiant is sometimes used as a dedicated, secondary HVAC system type.

- Climate: Among the California climate zones, the coastal climates are expected to represent the geographic area with the most commercial building construction and therefore provide wide applicability for future projects.
- Radiant slab type: A priority was placed on designs with high thermal mass radiant systems as opposed to embedded surface systems (ESS) as the primary source for both heating and cooling to match the overall focus of the EPIC research project.

3.1 Description

The original building design was modified to be more broadly applicable, representative of good design practice, and to streamline the analysis. The simplified building is four stories with a total floor area around 112,000 ft^2. The analysis is based on a typical floor layout that is representative of open plan office spaces.

The typical floorplan includes primarily open office spaces on the perimeters and some enclosed interior meeting rooms. The building was designed with various features to improve energy performance compared to California building energy efficiency standards, including:

- Low window-to-wall ratio of about 40 percent
- Exterior overhangs to reduce solar gain
- High performance glazing with an overall U-value at 0.40 and SHGC at 0.28
- Low energy LED lighting and automatic daylighting and occupancy sensing controls
- Low equipment power densities using advanced plug load management

These features can significantly reduce the HVAC heating and cooling loads, and reduced cooling loads are a critical factor for a radiant system to be successful.

4 HVAC System Descriptions

The HVAC system designs were developed based on the typical floor plan along with riser diagrams and roof top equipment plans. The designs were carefully evaluated to provide as much of a level comparison as possible. Nevertheless, there are many subjective design factors for each system type that impact the comparison. This section provides a general description of the two system types. Note that HVAC components common between both designs were not included (e.g. server room cooling, toilet/janitor exhaust, etc.) to simplify the cost estimating to focus on the major design differences. This reduces the overall HVAC costs, such that percentage differences between the two designs are not representative of the percentage difference in overall HVAC costs.

4.1 Radiant/DOAS System

The high thermal mass radiant system has tubing embedded in the structural slab such that heat can be exchanged with the occupied zone above and below each middle-story slab. Therefore, each space is generally conditioned by the radiant slabs both above and below, except for the ground floor which is only conditioned by the ceiling above – the slab-on-grade is not an active radiant surface. Figure 1 shows a schematic of the overall HVAC system design which utilizes the radiant surfaces as the primary heating/cooling system. Ventilation is provided by a variable speed dedicated outdoor air system (DOAS) with oversized maximum airflows to some spaces that require supplemental cooling, and pressure independent VAV boxes to meter an appropriate amount of air into each ventilation zone. Building pressure is maintained by two relief fans on the roof that exhaust air from each floor through the HVAC shaft. Chilled and hot water to the building are provided by a four-pipe air source heat pump located on the roof.

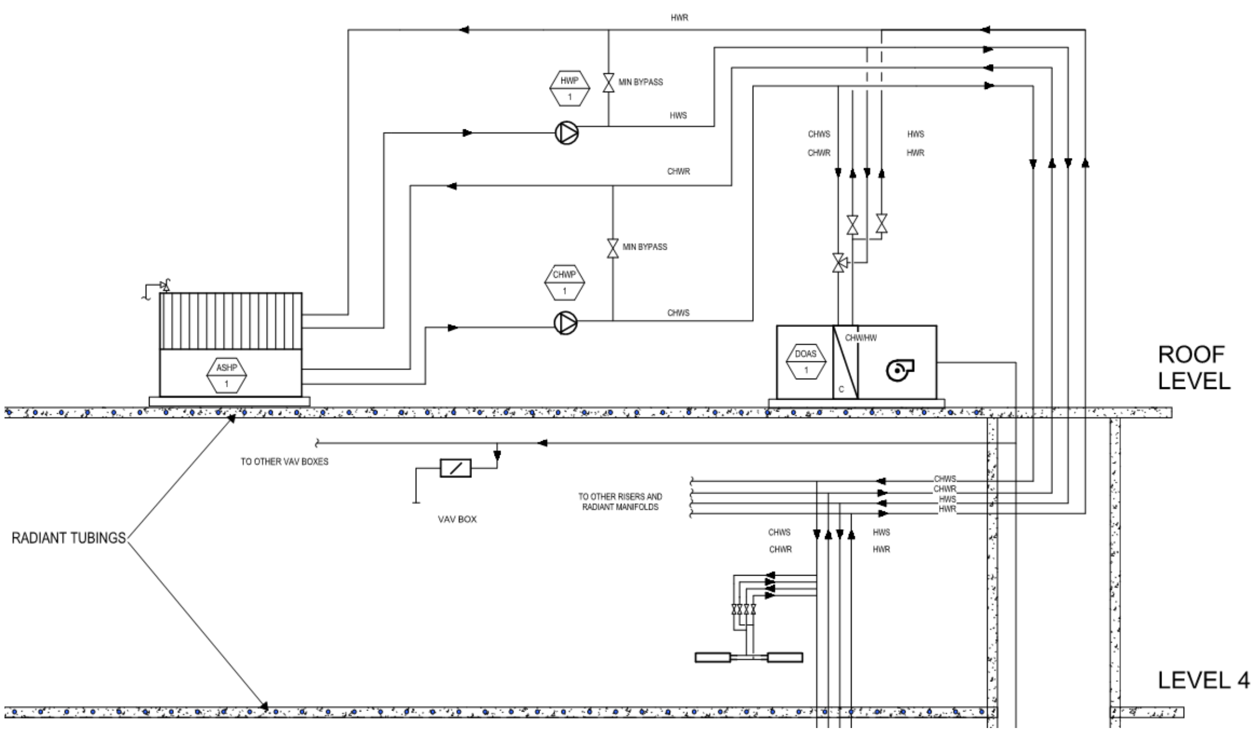


Figure 1: Radiant System Schematic

Figure 2 shows the radiant system zoning plan for each typical floor. The radiant thermal zones are separated based on orientation and exposure. Many of the radiant zones serve a mixture of open plan office and enclosed meeting rooms. Perimeter radiant zones provide both heating and cooling whereas the interior radiant zones only provide cooling.

Figure 3 shows the ventilation zoning plan for each typical floor. The open office areas are divided by quadrant and are distinct from the thermal zones. Enclosed interior spaces are served with VAV ventilation zones, with clusters of four to six small meeting rooms served by a common VAV terminal and each of the four large conference rooms served by dedicated VAV terminals with demand-controlled ventilation. The ground floor DOAS zoning is designed with two additional perimeter zones (for the east and west perimeters) to handle envelope loads for those exposures.

As the thermal and ventilation zones are not identical, the space conditioning and ventilation approaches are different for open offices, small conference rooms, and large conference rooms. The architectural features, such as floor to ceiling height and acoustic ceiling coverage, are also different for the three types of spaces. These architectural features have direct impacts on both the DOAS and radiant design and control. Table 1 summarizes the design features for each space type.

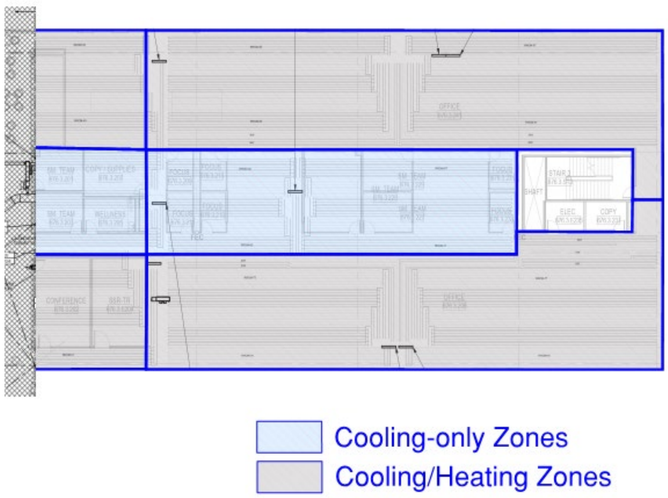


Figure 2: Radiant Zoning Plan for Portion of Typical Floor

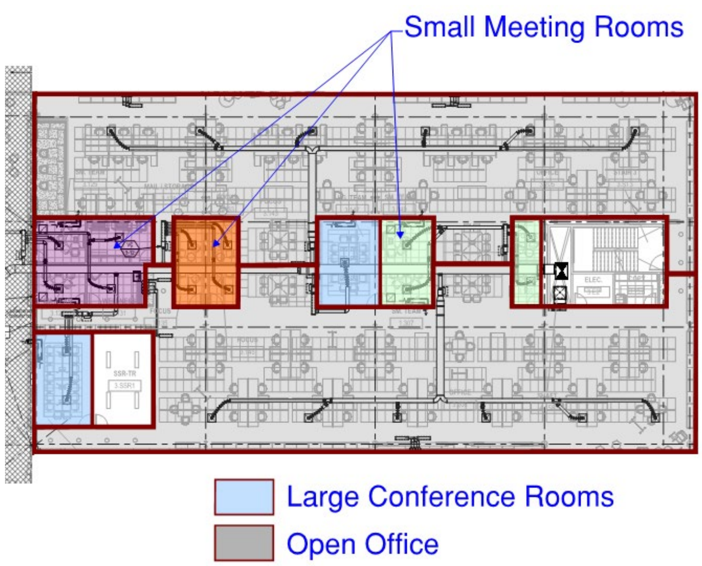


Figure 3: DOAS System Zoning Plan for Portion of Typical Floor (Ground Floor has Two Additional Perimeter Zones for West and East Exposures)

Table 1: Radiant System Design Features by Space Type

Design Features		Open Plan Office	Large Enclosed Conference Rooms	Small Enclosed Conference Rooms
Acoustic ceiling coverage		~65%	~50%	100%
Active radiant surfaces	Typical floor	Radiant ceiling + floor	Radiant ceiling + floor (One conf. room has cooling only)	Radiant floor (Cooling only)
	Ground floor	Radiant ceiling	Radiant ceiling (One conf. room has cooling only)	No effective radiant cooling due to 100% ceiling coverage
DOAS terminal unit	Typical floor	Cooling-only pressure independent terminal unit (without reheat coil)		
	Ground floor	Pressure independent terminal unit with reheat coil	Cooling-only pressure independent terminal unit (without reheat coil)	
DOAS Airflow	Typical floor	Constant flow for 15 cfm/person (~0.17 cfm/ft^2)	Flow resets from 0.15 cfm/ft^2 to 15 cfm/person by DCV or thermal load	Constant flow for 15 cfm/person (~0.45 cfm/ft^2)
	Ground floor	West/East: Flow reset from 15 cfm/person to design airflow for supplemental cooling Others: CAV sized for 0.15 cfm/ft^2	Flow resets from 0.15 cfm/ft^2 to 15 cfm/person by DCV or thermal load	Flow resets from ~0.45 to 0.62 cfm/ft^2 by thermal load

For the open offices on the typical floors, cooling and heating are primarily by radiant floors below and ceilings above. The acoustic ceiling coverage is about 65% in these areas, which will decrease the radiant ceiling system cooling capacity by about 15% as demonstrated by recent studies (Karmann, et al., 2017) (Dominguez, Kazanci, & Olesen, 2017). Nevertheless, the radiant systems can meet the full design cooling and heating loads in open office areas. The pressure-independent DOAS terminals provide constant ventilation airflow.

For the ground floor, we used slightly different approach as there is no radiant floor heating or cooling. In the open offices, the overall approach is to use a hybrid VAV-radiant approach where large DOAS VAV boxes have supply air diffusers directed to the perimeter open areas (no diffusers in the interior open areas). This allows the VAV boxes to provide supplemental heating/cooling where it is needed (i.e. perimeters) while minimizing the risk of overheating/overcooling interior areas, see Figure 3. Thermal loads in open interior areas are met by the radiant system and ventilation is distributed through the "sweep" effect of neutral transfer air. Unlike the typical floors, the ground floor is designed with two more perimeter zones for the air system (for the east and west perimeters) to handle envelope loads for those exposures. All perimeter zone terminals on the ground floor are designed to provide supplemental cooling and heating. With this approach, we sized the open office ventilation terminals for north and south exposures at the minimum ventilation rate of 0.15 cfm/ft^2. The east and west ventilation terminals are sized for cooling loads based on a supply air temperature of 65°F. Perimeter VAV boxes have a reheat coil sized to meet supplemental heating demand, assuming the radiant ceiling heating capacities are reduced by 35% due to acoustic ceiling panel coverage (Alvarez, Hviid, & Weitzmann, 2014).

Each large conference room is served by the radiant zone for the adjacent open plan office areas. In other words, the conference rooms do not have independent control over the radiant surface temperatures. Three of the four interior conference rooms are served by perimeter radiant zones which provide both heating and cooling. This configuration could result in conflicting control between the radiant and the air systems at times if an overall radiant zone is in heating mode but the conference room demands cooling. The acoustic ceiling coverage in the large conference rooms is about 50%, which causes about an 11% decrease in radiant ceiling cooling capacity. The dedicated cooling-only VAV boxes are sized to provide a maximum airflow of 15 cfm/person (~0.65 cfm/ft^2), which is sufficient to meet the supplemental cooling requirements at 65F supply air temperature – even when the radiant system is in heating mode.

Each small conference room has 100% acoustic ceiling coverage, which means that the floor is the primary active radiant surface on typical floors. For the ground floor, which does not have a radiant floor, the primary cooling source is the air system. Ventilation is provided by a cooling-only VAV box, that is shared by four to six adjacent small conference rooms. On the typical floors, these VAV boxes are sized for 15 cfm/person, which is about 0.45 cfm/ft^2, and is sufficient to meet the supplemental cooling requirements. Since there is no DCV control in these rooms, the boxes provide constant ventilation airflow when one or more of the rooms is occupied as indicated by an occupancy sensor. For the ground floor, the VAV boxes are oversized to provide up to 0.62 cfm/ft^2 to satisfy thermal load but can otherwise modulate down to the minimum ventilation requirement.

4.2 VAV with Terminal Reheat System

Design methods described in Advanced Variable Air Volume System Design Guide (PG&E, 2007) have been used for the VAV system design. The VAV system design consists of a central air handling unit with economizer sized for 100% flow and VAV reheat terminals at each thermal zone. Building pressure is maintained by the relief fans in the air handling unit that exhaust air from each floor through the HVAC shaft There are twenty-seven zones on each floor, and unlike in the radiant design, the thermal and ventilation zones are identical in the VAV design. Zoning is determined based on space usage, orientation and exposure (perimeter vs interior). Figure 4 shows the VAV zoning for the typical floor.

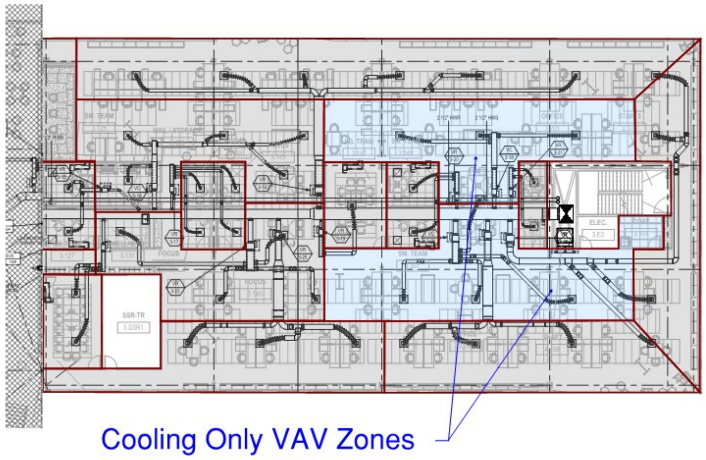


Figure 4: VAV Zoning Plan for Portion of Typical Floor

Compared to the radiant design, the VAV design provides more granular thermal and ventilation control with many more thermal zones. Large conference rooms and clusters of small meeting rooms are each served by an independent VAV terminal.

All zones are equipped with a reheat coil, except for four cooling-only boxes on each floor that serve interior open offices. Terminals serving enclosed, interior spaces are provided with reheat to avoid the risk of overcooling when providing minimum ventilation. The risk of overcooling is avoided in interior open office areas by allowing the zone airflows to drop to zero with ventilation maintained by transfer air from adjacent perimeter open office zones that have non-zero minimums.

On the plant side, the same type of air source heat pump is used to match the radiant design. The hot water distribution is primary-only, variable flow with variable speed pumps. The chilled water distribution is constant flow with the design flow set equal to the minimum requirement of the ASHP. The chilled water loop is close coupled on the roof, with a storage tank providing additional volume for stability and to minimize cycling. There is no control valve at the chilled water coil; capacity is modulated by resetting the supply temperature rather than varying the flow rate with a control valve.

5 HVAC System Design Criteria

This section summarizes the design criteria for the HVAC systems. Load calculations were performed in the Integrated Environmental Solutions Virtual Environment (IES VE) software.

5.1 Design Weather Conditions

Equipment sizing was based on the following design weather conditions:

- Design weather city: San Francisco AP, CA
- Summer design day 0.5% design drybulb/coincident wetbulb: 83/64°F

- Winter median of extremes: 31°F

5.2 Internal Loads and Environmental Conditions

A. Design Temperatures

Table 2 summarizes the design temperatures used for space cooling and heating load calculations. The design indoor air temperature setpoints for the VAV system are 70°F in heating and 75°F for exterior zones and 73°F for interior zones in cooling.

In the radiant design, the system is sized to maintain 78°F indoor air drybulb temperatures in cooling and 68°F in heating. This is a wider range compared to the VAV design, but typical of actual practice in radiant buildings to account for the thermal comfort benefit of the radiant surface temperatures. As space temperature in high thermal mass radiant buildings has long response time to changes in hydronic system temperature and flow rate, it is also impractical to maintain a narrow deadband. Both simulations and experimental data show that the difference between air temperature and operative temperature in spaces conditioned by air systems are usually greater than in radiant systems. In a simulated case (Feng, Schiavon, & Bauman, 2013), the differences for air systems were about 2.0-2.7°F, with the air temperature lower than the operative temperature during cooling. For the radiant system, the operative temperature and air temperatures were much closer, and with the air temperature higher than the operative temperature during cooling. Experimental data showed similar results (Karmann, et al., 2017). (Woolley, Schiavon, Bauman, Raftery, & Pantelic, 2018). Similarly, the heating setpoint of 68°F accounts for the operative temperature when the radiant surface temperatures are warm. Because of the radiant effect, similar thermal comfort can be achieved for the two designs based on the setpoints in Table 2.

For the VAV system, the design supply air temperatures at diffuser in cooling are 57°F for exterior zones and 62°F for interior zones (higher temperatures for interior zone sizing to allow for more supply air temperature reset).

For the radiant system, since the enclosed meeting rooms and two perimeter zones on the ground floor rely on the air system for cooling, the design airflows were sized for 65°F supply air temperature for the perimeter zones and 68°F for the interior zones. The high design supply temperature is used for these zones to allow the DOAS supply temperature to reset effectively and minimize the risk of overcooling.

Table 2: Design Temperatures

System	Design Condition (all temperatures are drybulb)	Heating (°F)	Cooling (°F)
Radiant	Offices, lobby and conference rooms	68	78
	Design supply air (at outlet) - exterior zones	68	65
	Design supply air (at outlet) - interior zones	68	68

VAV	Interior zones	70	73
	Exterior zones	70	75
	Design supply air (at outlet) - exterior zones	95	57
	Design supply air (at outlet) - interior zones	95	62

B. Internal Loads

Table 3 provides a summary of the internal load densities common to both designs. Occupant densities are based on furniture counts in the original design.

Table 3: Internal Load Summary

Space Type	Lighting (W/ft^2)	Equipment (W/ft^2)	Occupants (ft^2/person)
Open office	0.27 + 0.2 task lighting	0.8	~ 60-200
Conference/ Team Rooms	0.27	0.37	~ 20-45
Lobby	0.27	0.07	120

C. Ventilation Rates

The design ventilation flowrates for the three main space types in the radiant design are described in Section 4.1, and summarized in Table 1. For the VAV design, zone ventilation and VAV minimums are based on the greater of 0.15 cfm/ft^2 or 15 cfm/person for all spaces. Only large conference rooms have demand controlled ventilation, and the minimum flow rates are area-based ventilation requirements.

The DOAS is sized for a total flowrate of 19,400 cfm, and the sum of the non-coincident peak zone airflows is 22,751 cfm, representing 85% flow diversity. Note that the design DOAS flow rates in some enclosed spaces are only slightly less than the VAV design due to the relatively high supply air temperatures provided in the DOAS design and the use of the DOAS to provide supplemental cooling. The VAV design can deliver more cooling with less flow because of the lower supply air temperatures.

In many open office areas, occupant density is high enough that the occupant-based ventilation is greater than the area-based requirement. For the VAV design, the area-based ventilation requirement is 16,206 cfm and the occupant-based ventilation requirement is 19,144 cfm, reflecting a diversity factor of 85%. The outdoor air flowrate is equal to the lower value during normal operation and reset up to the higher value based on demand controlled ventilation logic.

5.3 Radiant Slab System

Tubing is embedded in the concrete structural slabs on top of the metal deck (see schematics in Figure 5). There are two types of slabs: the ceiling/floor slabs for the middle floors and the roof slab for the top floor. For the middle level slabs, heat transfers both up and down such that spaces are conditioned by radiant surfaces from both above and below. Since there is no active radiant system in the slab-on-grade, the radiant system only serves the first floor from above.

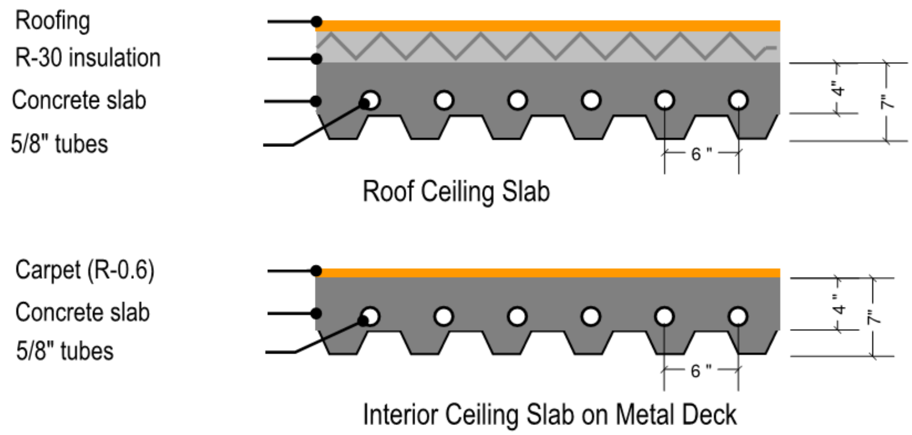


Figure 5: Radiant Slab Detail

Heating and cooling capacities of the radiant slabs are shown in Table 4. The capacities are based on finite element analysis for 6-inch tube spacing. When there are acoustic panels, the heating and cooling capacities are decreased by 35% and 15%, respectively, for the cases with 65% coverage (Karmann, et al., 2017) (Alvarez, Hviid, & Weitzmann, 2014). For each thermal zone, the combined radiant system capacity is compared to calculated cooling and heating loads to determine if supplemental heating and cooling are needed from the DOAS system. For the open offices and large conference rooms on the typical floors, the radiant system can meet all loads except for the east zones which have peak cooling load due to morning solar load. However, radiant system cooling capacity is known to be 50-100% higher when there is direct solar load (Odyjas & Gorka, 2013) (Feng, et al. 2016) (Pantelic, et al., 2018), which suggests that the radiant system can meet loads even in those east zones. In the small conference rooms that have 100% ceiling coverage, the radiant system alone cannot meet design cooling loads, but the additional cooling delivered by the ventilation of 15 cfm/person from the DOAS provides sufficient supplemental cooling even at a supply air temperature of 68°F.

On the hydronic side, under slab manifolds are installed above the ceiling cloud plane, and each radiant manifold is connected to the hot water riser and chilled-water riser by a two-pipe system via two sets of on/off control valves. See design drawings for piping and control details.

Table 4: Radiant Slab Capacity

Slab	Heating (Btu/hr/ft^2)		Cooling (Btu/hr/ft^2)	
	Up	Down	Up	Down

		No coverage	65% coverage		No coverage	65% coverage	100% coverage
Middle Slab	7.8	14	9.1	6.4	19.2	16.3	1.9
Roof Slab	1.8	14.7	9.6	1.0	20.0	17	2

5.4 Air Distribution System

A. Air Handler and Fan Sizing

For the radiant design, the dedicated outdoor air unit provides ventilation and supplemental cooling to the building and operates with 100% outdoor air, variable speed fans, and a single hydronic changeover coil that is served by either chilled or hot water. Two mixed flow fans provide building relief, assuming 0.10 cfm/ft^2 of exfiltration rate. The DOAS unit is sized for 19,400 cfm. See design drawings for DOAS details.

The VAV air handling unit (AHU) is sized for 63,000 cfm, which is sized based on the total coincident load multiplied by internal load diversity factors (0.8 for people, 0.9 for lighting and 0.5 for plug loads). Overall, the central fan design airflow shows a 0.7 diversity factor of the total VAV box design airflow. The VAV air handling unit consists of an economizer, filter, cooling coil, variable speed plug fan array for supply, and variable speed plug fan array for relief. See design drawings for AHU details.

The air handling unit design parameters and sizing criteria are summarized in Table 5.

For the supply fan external static pressure, the VAV system is sized for 2.75 inches, compared to 2 inches for the DOAS unit in the radiant design. The differential in static pressure between the two systems is to account for pressure drop for the following components:

- VAV reheat coil: reheat coils in the VAV design are sized for no more than 0.3 inches of pressure drop, and the reheat coils in the DOAS system are sized for no more than 0.2inches
- Pressure drop in return air path: 0.08 inches along return path + 0.15 inches at AHU return opening (for radiant, the relief fan takes care of this resistance)
- Added friction loss in VAV ducts

The DOAS air handling unit is sized for air velocities of 300 fpm compared to 500 fpm for the VAV system. The lower DOAS velocity is justified as it operates at or just below its design velocity most of the time. Therefore, the reduced pressure drop at lower velocities has a much larger impact on annual fan energy compared to VAV systems. Sizing VAV systems for peak flows at 500 fpm is typical of standard industry practice, though most VAV systems operate at far lower flows for most of the time.

Table 5: Air Handling Unit Sizing Summary

Design Parameters	VAV	Radiant/DOAS

Supply Fan	Design Flow (cfm)	63,000	19,400
	External Static Pressure (in)	2.75	2.0
	Total Static Pressure (in)	4.86	3.64
	Fan Type	Plug fan array	Plug fan array
	Design break horsepower, hp	68.61	15.92
Relief Fan	Design Flow (cfm)	47,500	7,800
	Total Static Pressure (in)	1.1	1.1
	Fan Type	Plug fan array	Mixed flow
Maximum Air Velocity (fpm)		500	300
AHU Supply Air Opening (fpm)		1300	1300
Outside air intake louver pressure drop (in)		0.3	0.3

B. Duct Sizing Criteria and Layout

The supply air riser is in a shaft at the far east end of the building. For both designs, the duct mains are run down the interior core of each floor where the 9 ft ceilings provide more clearance for larger ducts. Branch ducts and VAV boxes are located in the core areas with lower ceilings as much as practical. In the open perimeter areas, ducts are maintained above the 11 ft ceiling cloud plane. The DOAS ducts are limited to 8 inches tall in the perimeter areas when passing below beams with high ceiling clouds, and VAV ducts are limited to 10 inches tall (2 inches deeper than DOAS ducts to account for thinner slab thickness without radiant tubing in VAV design).

The radiant DOAS ducts and risers are sized using the equal friction method with friction rates of 0.05 inches of pressure drop per 100 ft throughout. The VAV ducts mains are sized at friction rates of 0.2 inches per 100 ft at the shaft transitioning down to low pressure ducts sized for 0.05 inches per 100 ft. These friction rates are uniformly reduced by 0.05 inches compared to conventional VAV design to match the DOAS sizing at the low-pressure end and to provide a more even comparison between the two designs.

The DOAS system has about 40% fewer supply diffusers compared to the VAV design, because of the lower overall airflow rates and because the DOAS design approach in open areas delivers the supply air to the perimeters only for supplemental temperature control (with no diffusers in open interior areas).

5.5 Hydronic System Design

A. Radiant System

Hot and chilled water are provided by an air source heat pump (ASHP) to serve the radiant slabs, DOAS changeover coil, and the hot water reheat coils on the first floor. The ASHP provides simultaneous hot and chilled water with heat recovery between the two hydronic loops. Though expensive, heat recovery chillers seem well suited to radiant applications because they cannot generate high temperature hot water (which is not needed for radiant) and the relatively high minimum flow rate requirements pair well with the low loop temperature differences associated with radiant systems.

The chilled water system serves the DOAS changeover coil and the radiant slabs. It is sized for the space sensible cooling load plus the load to condition the ventilation air from outdoor temperature at the cooling design condition to 62°F supply air temperature. The hot water system is sized for the design space heating load, plus the load to pre-heat the ventilation air from 31°F to 68°F. For the radiant slabs, the cooling design delta T is 5°F, and heating design delta T is 10°F. Chilled water is supplied at 57°F and hot water is supplied at 90°F.

For high thermal mass radiant system design, there is research showing the possibility to undersize the central plant cooling and heating equipment if load shifting control strategies are to be implemented. However, we know most of the designers are currently not comfortable with this approach due to the controls complexity it entails and a lack of guidance in design and equipment selection. Therefore, we have not explored this sizing approach for the study. However, in Section 7.3A, we used energy simulations to investigate the impacts of load shifting on site energy consumption, operating costs, thermal comfort and equipment sizing.

The hydronic distribution is primary-only, variable flow for both the hot water and chilled water systems, with bypasses to maintain minimum flows to the heat pump. The hot water pump operates at nearly constant flow due to the high minimum flow requirement. The design includes only a single heat pump, which represents a single point of failure (a more conventional approach would be to design around two heat pump units and a pair of pumps for each system).

The main hot and chilled water pipes run horizontally on the fourth level from east to the west, and feed into nine sets of pipe risers located close to the radiant manifolds serving identical zones on different floors. The ground floor VAV reheat coils are also fed from nearby hot water risers. For the radiant design, using multiple risers reduces overall pipe lengths by 30% compared to having a single riser, which can be a significant cost reduction. As the radiant design requires four pipes to most changeover assemblies, and the manifolds are located mostly in the perimeter areas, the benefits of multiple risers are critical for the radiant design.

B. VAV system

In the VAV design, the air handling unit cooling coil is the only load on the heat pump. Since the heat pump has a very high minimum flow requirement and the loop is very close coupled, the chilled water loop is designed to be a constant flow system without a control valve at the chilled water coil. Capacity modulation is provided by resetting chilled water supply temperature.

The hot water system is primary-only, variable flow with a bypass to maintain minimum flows to the heat pump. Chilled water is supplied at 45°F and hot water is supplied at 115°F. The latter is much lower than typical for VAV reheat systems but is limited by the ability of the heat pump unit to provide high temperature hot water.

A single set of hot water pipe risers at the east end of the building provides hydronic distribution to the floors. Since all VAV boxes are in the interior core of the building, we minimized the branch piping length.

Table 6 summarizes the hydronic system parameters for the radiant and VAV designs, and Table 7 shows the air source heat pump design information. Note that even though the design capacities are similar, the VAV design uses the model one size larger than the radiant design because of the higher design heating water temperature.

Table 6: Hydronic System Design Parameters

	Chilled Water					Hot Water				
System	Supply Temp (°F)	Delta T (°F)	Design Flow (gpm)	Minimum Flow* (gpm)	Pump Head (ft.)	Supply Temp (°F)	Delta T (°F)	Design Flow (gpm)	Minimum Flow* (gpm)	Pump Head (ft.)
Radiant	57	7	630	250	80	90	17	180	180	70
VAV	45	15	255	255	28	115	13	230	180	50

*Note: Minimum flow required by the heat pumps

Table 7: Air Source Heat Pump Design Specifications

	Model	Cooling		Heating	
System		Capacity (Ton)	COP	Capacity (kBtu/hr)	COP
Radiant	Aermec NRP 2250	196	3.93	1,455	2.82
VAV	Aermec NRP 2500	192	3.58	1,455	2.16

6 Cost Study

6.1 First costs

The mechanical construction cost estimates for each of the designs were developed independently by two mechanical contractors with experience in both VAV and radiant systems, based on design drawings, as well as control schematic drawings, developed to the construction documents level. Major equipment selections and quote by vendors, including the air handling unit and the air source heat pumps, were also provided to the two contractors.

Major assumptions in the construction cost estimates:

- Estimates assume typical industry practices to ensure consistency between the two design approaches and costs that are representative of standard practice; detailed specifications were not developed.
- Common mechanical elements between the two designs were deleted to simplify the cost estimating process (e.g. toilet/kitchen exhausts, server room conditioning, etc.). Overall costs and costs per unit area may be slightly lower than typical because of the deleted elements.

- Labor rates are based on the San Francisco Bay Area location ($123 per hour for sheetmetal and $118 per hour for piping in one estimate)
- Costs are for HVAC and controls only; cost impacts on architectural, structural, electrical, insulation requirements and other trades are not accounted for here. Potential differences in ceiling coverage between the two designs are not addressed.

Figure 6 presents the total HVAC construction costs for the VAV and radiant design from the two contractors. The two estimates are very similar. On average, the radiant HVAC design has a total cost of $4,349,000 or $38.9/ft^2 compared to $3,341,000 or $29.9/ft^2 for the VAV design, representing a $1,008,000 or $9.0/ft^2 premium.

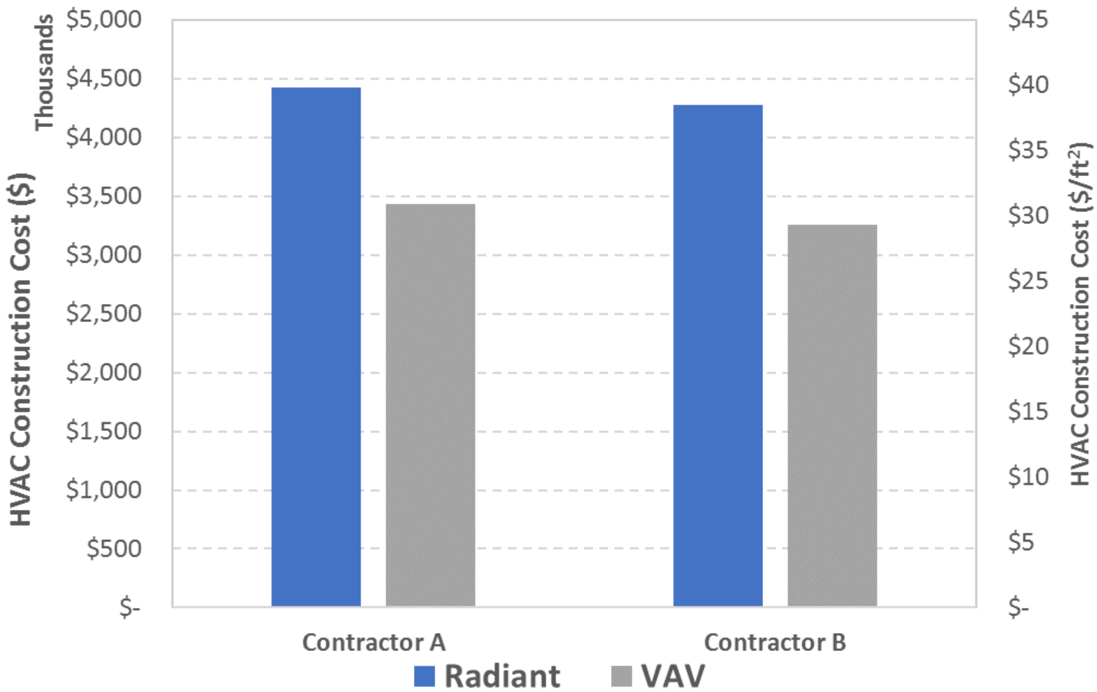


Figure 6: Total HVAC Construction Cost for Radiant and VAV Designs

Figure 7 compares the cost breakdown between the two designs. The bars are the average cost of the two estimates, and the whiskers show the estimate ranges. For sheet metal and piping, costs are broken out separately for labor versus material. The other categories shown in the chart combine costs for a range of several subcategories. They are combined for clarity and simplicity due to differences in the detail provided in the two estimates. In general, the two cost estimates aligned very closely with each other for the major categories. Since the two estimates aligned well and only one provided a high level of detail, the detailed estimate is used as the basis for the detailed cost breakdown shown in Table 8. Based on Table 8 and Figure 7:

- The cost premium associated with the radiant system is mainly due to labor for piping and radiant equipment, which itself is $9.8/ft^2 higher than that for the VAV design. Note also that the markup costs are based on multipliers of the other costs so most of that premium is also due to the high piping labor costs in the radiant design.

- There is a $1.2/ft^2 premium in equipment cost for the radiant system, which is mainly associated with the radiant equipment. See Table 9 for equipment cost breakdown.
- The sheet metal material and labor costs are 43% lower for the radiant/DOAS design. The total sheet metal cost is $4.5/ft^2, 12% of the total first cost, for the radiant design and is $7.9/ft^2, about 27% for the VAV design.
- Most of the subcontract costs are comparable between the two designs with the exception that controls costs are lower for radiant. This is mainly due to the much lower number of VAV boxes in the radiant/DOAS design, compared to the VAV design.

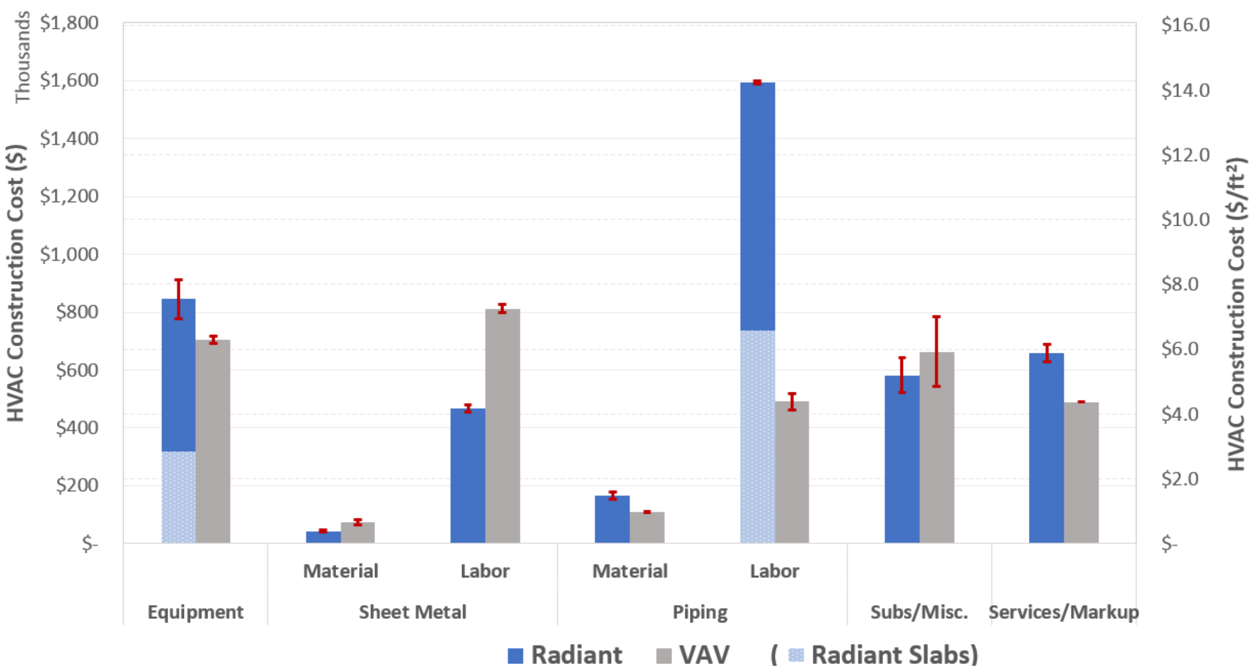


Figure 7: Cost Breakdown Comparison Between the VAV and Radiant Designs

The HVAC equipment costs account for roughly 20 to 25% of the overall HVAC costs for both systems, and are summarized in Table 9 with breakdowns illustrated in Figure 9 and Figure 10. The largest difference in equipment costs between the two designs is for the radiant equipment (loops, mats, manifolds) which is about $2.82/ft^2, accounting for nearly half of the equipment costs in the radiant design. However, the DOAS air handling unit is only $0.79/ft^2, about 32% of the cost of the larger VAV air handling unit ($2.52/ft^2), which makes up for much of the cost of the radiant equipment. There are other minor differences in equipment costs between the two designs but they are overshadowed by the radiant equipment and air handling unit costs.

Piping labor costs are $4.6/ft^2, accounting for about 19% of the overall costs in the VAV design, but $14.2/ft^2, about 44%, in the radiant design. In absolute terms, the piping labor for the radiant design is more than 3 times that of the VAV design. The radiant and VAV piping labor costs are summarized in Table 10, Figure 11 illustrates the radiant piping cost breakdown. When combining the labor cost for radiant floors and the manifolds/changeover assemblies, the radiant equipment labor is $6.6/ft^2, accounts for 46% of the total piping labor. About $1.9/ft^2, 13% of the total radiant piping labor, is associated with installing the hot and chilled water pipe distribution on the floors, with smaller portions attributed to the piping on the roof and the risers.

Table 8: Mechanical Construction Cost Summary

Category	Radiant ($)	VAV ($)	Radiant ($/ft^2)	VAV ($/ft^2)	Difference ($/ft^2)	Difference (%)	Radiant (% of total)	VAV (% of total)
Equipment	**$777,000**	**$717,000**	**$6.94**	**$6.40**	**$0.54**	**8%**	**21.7%**	**26.0%**
Sheet Metal	**$524,000**	**$882,000**	**$4.68**	**$7.88**	**-$3.20**	**-41%**	**14.6%**	**31.9%**
Material	$46,000	$83,000	$0.41	$0.74	-$0.33	-45%	1.3%	3.0%
Labor	$478,000	$799,000	$4.27	$7.13	-$2.87	-40%	13.3%	28.9%
Piping	**$1,766,000**	**$629,000**	**$15.77**	**$5.62**	**$10.15**	**181%**	**49.3%**	**22.8%**
Material	$178,000	$110,000	$1.59	$0.98	$0.61	62%	5.0%	4.0%
Labor	$1,588,000	$519,000	$14.18	$4.63	$9.54	206%	44.3%	18.8%
Subcontracts/misc.	**$514,000**	**$535,000**	**$4.59**	**$4.78**	**-$0.19**	**-4%**	**12.0%**	**16.5%**
Insulation	$174,000	$174,000	$1.55	$1.55	$0.00	0%	4.9%	6.3%
Controls	$250,000	$268,000	$2.23	$2.39	-$0.16	-7%	7.0%	9.7%
Miscellaneous	$21,000	$21,000	$0.19	$0.19	$0.00	0%	0.6%	0.8%
Rigging	$16,000	$16,000	$0.14	$0.14	$0.00	0%	0.4%	0.6%
SCBC	$41,000	$44,000	$0.37	$0.39	-$0.03	-7%	1.1%	1.6%
Rentals	$12,000	$12,000	$0.11	$0.11	$0.00	0%	0.3%	0.4%
Services/Markup	**$688,000**	**$489,000**	**$6.14**	**$4.37**	**$1.78**	**41%**	**19.2%**	**17.7%**
Services	$299,000	$193,000	$2.67	$1.72	$0.95	55%	8.3%	7.0%
Markup	$389,000	$296,000	$3.47	$2.64	$0.83	31%	10.9%	10.7%
Total	**$4,269,000**	**$3,252,000**	**$38.12**	**$29.04**	**$9.08**	**31%**	-	-

Notes:
Detailed summary based on costs from a single estimate
Services: Includes field supplies, vehicles, supervision, project management
SCBC: Startup, controls, balance & commissioning

Table 9: Mechanical Equipment Cost Summary

Equipment	Radiant ($/ft²)	VAV ($/ft²)
AHU	$0.79	$2.52
ASHP	$2.71	$2.87
Radiant	$2.82	$0.00
VAV/Dampers	$0.21	$0.36
Hydronic	$0.25	$0.44
Grilles	$0.14	$0.22
Total	**$6.93**	**$6.40**

Notes: Cost includes sales tax

Figure 9: Radiant Equipment Cost

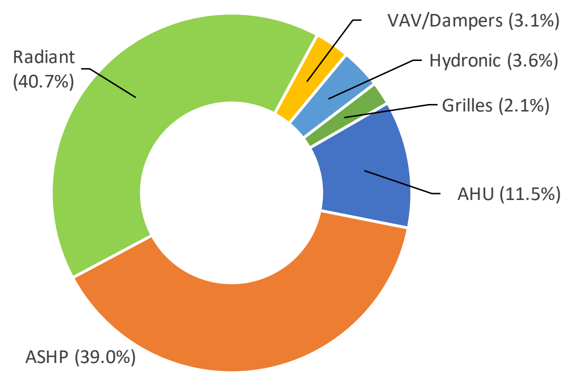


Figure 8: VAV Equipment Cost

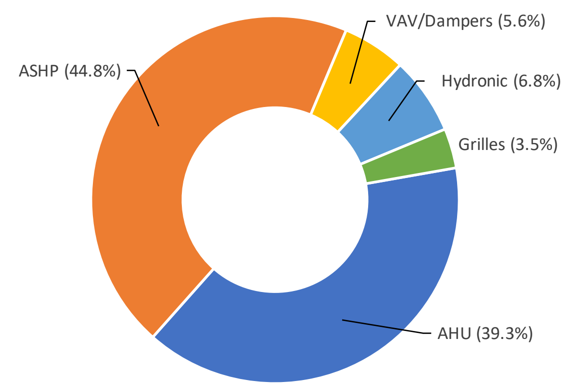
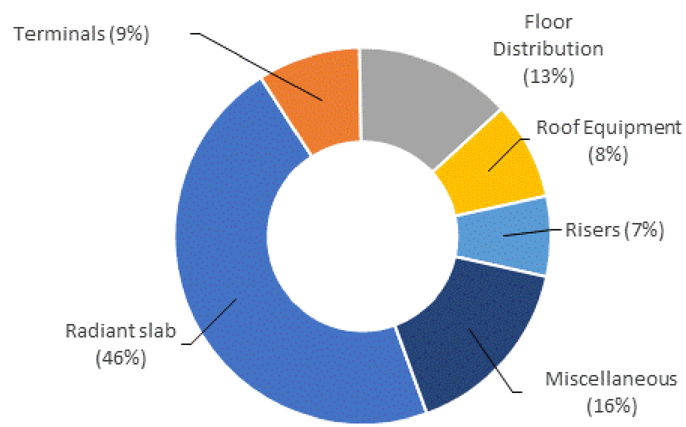


Table 10: Piping Labor Cost Summary

Category	Radiant		VAV	
	Cost ($/ft²)	Percent (%)	Cost ($/ft²)	Percent (%)
Radiant Tubes/Mats	$6.59	46%	$0.00	0%
Terminals	$1.25	9%	$0.94	20%
Floor Distribution	$1.91	13%	$1.10	24%
Roof Equipment	$1.19	8%	$1.19	26%
Risers	$0.98	7%	$0.12	3%
Miscellaneous	$2.27	16%	$1.27	28%
Total	**$14.19**	100%	**$4.63**	100%

Notes:

Figure 10: Radiant Piping Labor Cost



Detailed summary based on costs from a single estimate
Terminal piping labor costs include manifold/changeover valve or reheat valve connections.

6.2 Discussion

The designs compared in this evaluation represent a single data point that applies specifically to the designs described here. Though the radiant design was carefully screened and modified to be widely applicable and representative, care should be taken when generalizing the results beyond the scope of this study. Many design aspects are highly subject to engineering judgment and the preferences of individual designers. Though nuances in the VAV system design may also have an impact on the first cost comparison, the industry has agreed to a more common 'best practice' over many years and many projects for VAV systems, and thus this discussion focuses primarily on design choices for the radiant system with which the industry as a whole has had far less experience and opportunity for optimizing designs and reducing costs

The radiant design represents a \$9.0/ft^2 cost premium over the VAV design for this building application, which is consistent with the \$10-15/ft^2 premium reported anecdotally by some mechanical contractors. It is worth mentioning that the radiant system cost could be higher or lower if the project scope were different or different radiant system design approaches were used. More important than the bottom line comparison though, is the detailed breakdown of costs which can provide valuable insight on what aspects of the system design have the greatest cost impact and warrant careful attention.

A. Radiant System and Piping Costs

The cost for the radiant slab equipment is \$2.82 /ft^2, accounts for 40% of the total HVAC equipment cost, and installation labor cost is \$6.59/ft^2, about 46% of the total piping labor cost for that system. The percentages are higher if considering that the soft costs are a multiplier and should therefore be proportionally attributed to each major hard cost category.

To investigate the equipment cost range, we requested design layouts and prices from two radiant manufacturers. The two manufacturers each provided a design, which used different percentages of radiant mats and conventional loops, and different numbers of manifolds. Radiant mats are pre-assembled networks of headered tubing which can be rolled out to cover a large area with less labor, compared to laying out individual loops for conventional tubing. Both designs used 6" tube spacing to maximize radiant slab output.

Though the piping labor costs in the two cost estimates are very similar, the consensus from the contractors and manufacturers that we interviewed was that actual labor hours can vary widely based on a number of factors, such as the mechanical and general contractor's experience with radiant systems, coordination during the construction process, and the specific installation detail for the radiant system. As detailed cost breakouts were only provided for one of the cost estimates, union labor estimates for radiant installation from one of the manufacturers were used to provide a second point of reference for installation costs.

Table 11 provides the ranges of material and installation labor costs for the two radiant slab designs for the case study building. The unit material costs are based on estimated contractor pricing for the overall system (with typical discount from the list price), but do not include contractor markup, costs for start-up, overhead, etc. In general, the contractor pricing depends on wholesaler's discount from the manufacturers and/or the profit goal of the wholesalers, and both of which could vary drastically. Note that the total contractor cost will not be the sum of the cost for mats, loops and manifolds, and it depends on the percentages of mats and loops used in each design, and the number of manifolds also depends on the layout and whether mats or loops are used, and their percentages.

Table 11: Radiant Slab Material and Labor Cost of the Two Radiant Designs

Radiant Component	Material ($\$/ft^2$)[1,4]	Labor ($\$/ft^2$)[4]	Combined Material and Labor Cost[3,4] ($\$/ft^2$)
Mats	1.7 - 2.3	2.1 - 3.9	3.8 - 6.2
Loops	0.9 - 2.0	5.2 – 6.2	6.1 - 8.3
Manifolds [2]	0.67 - 0.78	0.27 - 0.46	0.94 - 1.25

1. The material costs are estimated contractor pricing, The material costs include wholesalers' margin, and may vary depending on a number of factors.
2. Manifold cost includes fittings, loop flow meters, loop valves, etc.
3. Total prices do not include contractor mark-up, startup, overhead, etc.
4. Costs are normalized based on the area of radiant surface, not building square footage.

Based on Table 11, for a 50/50 mixture of mats and loops design, the labor cost for slab installation ranges from $4.15 -$4.55/ft2. At $118/hr of labor rate, it equals roughly 38 man-hours per 1000 ft^2. This number may seem high, but it appears to reflect the current contractors' bidding strategy. From interviews of contractors and manufacturers, we found that, besides the experience of the mechanical contractors, radiant installation cost also depends highly on the general contractor's experience with radiant slab project and their project management skill. This is because radiant installation requires a lot more coordination between trades, and careful scheduling is extremely important during the construction process. Unfortunately, poor project management is not uncommon in the industry. This kind of uncertainties are probably also reflected in the high labor cost.

In the design evaluated for this study, the manifolds are installed directly under radiant slab. The more common method is to install them in a recessed wall cabinet. The first costs, including material and labor, for the under-slab manifold are less than for the wall cabinet manifolds, but the maintenance and testing labor could be higher. For each under-slab manifold, one estimate assumed 4 hours of labor for installation, not including connecting to the main hydronic system. Though actual manifold costs are better represented on a per-unit rather than a per-unit-area basis, the latter is used to allow comparison with other costs with acknowledgement that the per-unit-area cost is primarily dependent on the area served by each manifold.

<u>Radiant mats vs. loops</u>

Radiant mats can be used to reduce field labor cost. Table 11 shows the labor hours to install radiant loops is 35 to 200% higher than to install mats. For mats, labor is estimated at 0.018 to 0.033 hr/ft^2 to install, while for loops, it is 0.044 to 0.053 hr/ft^2 (at 6-inch tube spacing).

The material cost shown in Table 11 conflated many factors such that it cannot provide a clean price comparison between mat vs. loop design. In the base design, Manufacturer A provided a design that uses roughly 70% mats and 30% conventional loops. To compare the total cost between mat and loop installation, we requested Manufacturer A to also provide a design using 100% conventional loops. The total radiant material cost for the all-loop design was $2.72/ft^2, which is 8 percent higher than the mixed mat and loop design. The lower material cost for radiant mats may possibly be attributed to Manufacturer A's marketing strategy as the mats are a relatively new product. The overall installed cost is about $0.7 ~ 2.4 /ft^2 higher than the design using a mix of mats and loops.

In summary, in areas with high labor rates, designers may consider facilitating the use of radiant mats to reduce labor costs. However, radiant mat designs may not be practical or as cost effective for buildings with smaller or oddly-shaped zones. In addition, since the mats are generally assembled on a made-to-order basis, the unit prices may be much higher for smaller orders.

Radiant tube spacing

Since radiant tubing represents the largest cost category in the radiant design, we investigated the option to increase tube spacing from 6 to 9 inches, to potentially reduce both material and labor cost. The cost savings depend on the type of radiant system to be installed, i.e. pre-manufactured roll-out mat or conventional loop system:

- Conventional loops: contractors estimated about 0.022 to 0.026 hour per linear foot of installation, translating to $1.7/ft^2 of labor cost savings plus $0.6/ft^2 of material savings for the wider 9-inch spacing
- Radiant mats: there were 5 to 15 percent savings in material costs for the wider 9-inch spacing. The small cost difference is because the mats require the manufacturer to customize the design for each project which has unique zoning and layout, and most of the costs are for assembly and production rather than the material. Manufacturer's marketing strategies may also play a role here.

A potential issue with increasing tube spacing is the reduction of radiant slab capacity. Depending on the slab configuration, the theoretical steady state cooling capacity may be reduced by 5 to 25% with tube spacing changes from 6 to 9 inches. Due to design uncertainties and possibility of future need, designers and manufacturers often maximize the steady-state radiant slab output to meet peak design load. However, radiant slabs are so massive that they never work in steady state condition. Dynamic simulations show that it is possible to achieve very similar peak cooling capacity with 9-inch and even 12-inch spacing by opening the valves longer. Furthermore, there are many examples of buildings designed to use 9" spacing. More research is needed to investigate the impact of tube spacing and dynamic control in high thermal mass radiant systems, and the cost implications of these decisions.

In summary, increasing tube spacing saves 30% costs for conventional loop designs. For radiant mat installation, we do not have accurate information on potential labor savings to install the wider 9-inch spacing, but one of the contractors estimated about 5% less labor cost for different tube spacing. With the current radiant market pricing strategy, there may not be appreciable material cost benefit to increase tube spacing. For thermal zones with less heat gains, designer should evaluate the option of wider spacing to reduce cost.

Hydronic distribution layout

The installation of hot and chilled water distribution piping is a significant cost component, particularly in the radiant system design. We compared the installed cost differences for a hydronic distribution approach of using a single set of piping risers against an alternative of using multiple-risers. The former relies on a single set of larger risers and long horizontal distribution runs on each floor, whereas the latter employs multiple sets of smaller risers strategically located to minimize the overall amount of pipe length (a 30% reduction in pipe length in this case). With multiple risers, smaller copper risers can be used instead of larger steel risers. The former is much less expensive to install due to lighter weight materials and because it is faster to solder than to weld.

The baseline radiant design employed the multiple riser approach though the VAV reheat design utilized the single riser approach to be more representative of common practice. The cost penalty for increased pipe length associated with single risers is much more significant for the radiant design since it requires both hot and chilled water pipe distribution on every floor.

In summary, strategically designing hydronic distribution systems to minimize total pipe length can have a significant impact on overall system cost. The multiple riser approach evaluated here resulted in a $2.5/ft^2 first cost reduction in the radiant design. Similarly, locating changeover valve assemblies as far upstream as practical can minimize piping costs by reducing the length where 4 pipes are required.

Hydronic system type

In the baseline radiant system, we designed a four-pipe system to each radiant zone, i.e. there are four pipes connecting to each radiant zone switchover assembly. Based on a design survey (Paliaga, Farahmand, Raftery, & Woolley, 2017), many designers also employ the 2-pipe distribution approach or use a combination of 2-pipe and 4-pipe approach for radiant slab distribution system design. With the building level changeover systems where the two-pipe distribution is used, all radiant zones in the building can be only in either cooling or heating mode. This approach could reduce piping material and labor costs but may provide inferior temperature control. Another common approach is to use whole floor changeover where only the risers are 4-pipe and the floor distribution is 2-pipe only. The cost impact of this strategy would need to be carefully considered: though it reduces the length of 4-pipe distribution and the number of changeover assemblies, the 2-pipe distribution may be relatively long depending on the building geometry, and the impacts on thermal comfort also need to be evaluated as multiple exposures and orientations are all grouped into a single radiant zone.

Designers who have used the 2-pipe approach claimed to have little concern on thermal comfort control in the buildings. Using giant radiant slabs that continually exchanging heat with each other, radiant buildings may experience a very different thermal dynamic environment from the buildings conditioned by all-air systems, especially for those buildings equipped with load reduction measures. More comprehensive field studies of real radiant buildings would help designers better understand of how the systems operate and regulate space temperatures and provide guidance on which distribution design approach to select.

Radiant zone size

Utilizing large radiant zones to minimize the number of changeover assemblies can also contribute to reducing the cost of the radiant design, but may or may not be at the cost of comfort depending on the layout. It is not uncommon to see large zoning that combines open and enclosed spaces, which may potentially result in comfort issues and energy waste if the hybrid ventilation and radiant systems are not properly designed and control is not optimized.

In open plan layouts, large radiant zones may not necessarily result in comfort problems. The radiant design evaluated in this study utilized 10 radiant zones per floor for about 2500 ft^2 per zone because of the large open office plan. This large zoning approach extends the depth of a perimeter zone to 30 feet, which is double the depth of a traditional perimeter zone in VAV design. A potential concern may be maintaining comfort in a large zone where heat gain distribution is non-uniform. However, radiant systems may be more resilient to non-uniform distribution of heat gains because heat exchange potential is spread out across large areas (by radiant exchange between the exposed slab surfaces above and below, and by the fluid flowing in the individual radiant piping circuits which typically span the longest dimension of these large zones) and because of the "self-regulating" effect, which is the capability of radiant slabs to passively regulate the heating/cooling output at the slab surfaces based on the varying temperature differential between the radiant surface and the temperature of the air and other surfaces in the space. These phenomena are thought to be a reason why comfort can be maintained throughout large radiant zones even though slab surface temperatures change slowly. There is, however, little research to provide guidance on how to take advantage of it at design, and where the limitations are.

While smaller zoning can theoretically provide more granularity in radiant zone control, it comes at a cost premium, is often not necessary, and does not take advantage of the self-regulating aspect of radiant slabs.

Number of radiant slabs

The middle floors of a thermally active multi-story building will generally have both the floor and ceiling as active radiant surfaces, whereas the ground floor may only have the ceiling activated if radiant tubing is not installed in the slab-on-grade (or similarly the top floor may only have the floor activated if radiant tubing is not installed in the roof). Where the number of radiant slabs is equal to the number of floors, we refer to the design as having N layers of radiant slabs. Designs that have one additional radiant slab than the number of floors are referred to as having N+1 layers of radiant slabs. The N+1 slab, i.e. radiant in slab-on-grade floor or in roof layer, adds significant cost considering that it can only provide heating or cooling in one direction. For the case study building, adding radiant in the slab-on-grade would increase the total cost by about \$350,000, which is about \$3.2/ft^2.

Among five thermally active building projects that we reviewed, two have N+1 radiant layers and three have N layers. The two N+1 designs are school and commercial office buildings, and the three N layer designs are a lab and two office buildings. Among the three N layer designs, only one building addresses the temperature control for the ground floor differently than the rest of the floors, with four-pipe fan coils to provide additional capacity since there is only one active radiant surface.

In the case study building where thermal loads have been minimized, the radiant ceiling or floor alone can meet 70 to 90% of the load. In cases such as this, adding the N+1 radiant layer will generally be less cost effective than oversizing the DOAS system and adding reheat coils to the perimeter VAV boxes.

For buildings with higher thermal loads, the cost for the N+1 radiant layer may be justified to avoid the energy penalty associated with oversizing the DOAS or the cost to install alternative supplemental temperature control. The cost benefits should be carefully weighed during the design.

B. Air System Duct Sizing and Distribution

One common claim that a radiant design could be cheaper than a VAV design is that the DOAS systems use significant less amount of sheetmetal than VAV systems. In this case, the sheetmetal construction costs in the radiant design is \$4.7/ft^2, and for the VAV design it is \$7.9/ft^2. The cost difference is much less significant than generally claimed, and sheetmetal cost is not the determining factor when comparing the two in this case. Though DOAS distribution systems generally deliver far less airflow compared to equivalent VAV systems (19,400 vs 63,000 cfm here), the heavy turndown and typical operation at part load for VAV systems allows for VAV ducts to be more aggressively sized. Diversity can be applied in the duct mains and risers because the higher friction rates and fan power required at full design only occur for relatively few hours of the year. By contrast, DOAS distribution systems generally operate at constant flow (or near constant flow) conditions and therefore are typically sized at lower friction rates to avoid excessive energy use associated with the near-design friction losses. Some previous studies (Stein & Taylor, 2013) and simulation results from this study comparing DOAS and VAV systems have documented higher fan energy use associated with DOAS. Therefore, sizing DOAS distribution system for lower friction rate is critical for energy efficiency purposes. The same comparison applies to air handler sizing. The result of the different sizing criteria is that duct mains and risers of the DOAS design may not be significantly smaller than the VAV design despite the large difference in peak flow.

For this reason, building owners need to be cautious about another commonly claimed non-HVAC cost saving by DOAS designs that the mechanical shaft size could be significantly reduced in exchange for more rentable floor space. In this case, the supply duct risers are only about 25% larger for the VAV design than the DOAS. Even though the VAV mechanical shaft may be also used for air return, it is unrealistic to expect the shaft size for DOAS to be significantly smaller.

Similarly, care should be taken regarding the claim of potentially lowered floor-to-floor height for a radiant building due to smaller HVAC duct size.

C. DOAS Zoning and Supplemental Cooling

Thermal and ventilation zones are generally identical in all-air VAV designs where the primary air also provides ventilation. In contrast, the ventilation and thermal zones are generally distinct in HVAC designs that rely on DOAS units to provide ventilation. In addition, radiant systems often may not have sufficient cooling capacity to meet design loads, requiring the need for supplemental cooling. Oversizing the DOAS unit and zones is one common approach to provide supplemental cooling. However, the energy penalty to oversize a DOAS system is much larger than oversizing air handling unit with an economizer. To minimize the need for oversizing as much as possible in the radiant design for this study, we use an approach that distributes the conditioned air directly to the perimeter open spaces where supplemental heating and cooling are primarily needed. Open interior areas do not have direct air distribution, to avoid over-heating and over-cooling, and the ventilation requirements in these areas are met by transfer air, see Figure 3.

It is relatively common practice for DOAS ventilation zones to encompass relatively large areas, including a mix of occupancies and exposures. This approach provides simplicity and reduces the number of terminals required. One drawback with large ventilation zones is that it is more difficult to reset ventilation based on demand controlled ventilation and occupancy sensing, since the airflow rate cannot be individually modulated to track the occupancy patterns of each separate space.

When airflow is oversized for supplemental cooling for individual spaces within a larger ventilation zone, there is also a significant risk of overcooling at off-design conditions because the airflow rates cannot be individually modulated to each different space according to thermal demand as all spaces share one thermostat. Neutral supply air temperatures may reduce this risk but would require more airflow to deliver the same amount of supplemental cooling, and modeling studies show that providing consistent neutral temperatures consumes far more energy than DOAS units with variable supply temperatures in climates with significant economizer potential. Maintaining elevated constant airflow rates in certain spaces for supplemental cooling may also significantly impact energy use at the DOAS air handler due to operating the fan at near-design conditions and additional outdoor air that must be conditioned. Though it increases first cost, providing separate DOAS VAV terminals to enclosed spaces that require supplemental cooling may improve thermal comfort and operational energy efficiency. In the case study building, the total installed cost for the DOAS VAV boxes is less than 5% of the total cost.

D. Central Plant Design

The selected building design was somewhat unique in featuring an all-electric design. First costs for both designs could be reduced by substituting the ASHP with an air-cooled chiller and condensing boiler. This alternative was not considered because of the extensive downstream design changes that would be required.

For high thermal mass radiant system design, there is research showing the possibility to undersize the central plant cooling and heating equipment if load shifting control strategies are to be implemented. However, we know the majority of the designers are currently not comfortable with this approach due to the controls complexity it entails and a lack of guidance in design and equipment selection. If this approach is proven to be acceptable by the industry in the future, there would be some savings in central plant equipment costs. In some cases, these strategies allow for quite different cooling plant designs, with much lower first and operating costs, such as those only using a cooling tower.

In Section 7.3A, we used energy simulations to investigate the impacts of load shifting on site energy consumption, operating costs, and thermal comfort.

E. Use of Ceiling Fans

Ceiling fans are sometimes used in the open spaces with an intent to provide increased air movement to expand the upper limit of space temperatures that provide acceptable occupant thermal comfort and to give occupants instantaneous control over thermal comfort conditions. This concept is especially appealing in radiant buildings as the high thermal mass radiant systems have limited cooling capacity and are slow in response to sudden control changes. The cost benefit is that ceiling fans may eliminate the need of supplemental cooling in zones that have cooling load just above the radiant slab capacity. The energy benefit is that the primary HVAC system does not need to run as hard and therefore saving energy during normal cooling operation. However, it is likely difficult in open offices to guarantee that every occupant will experience similar increased air speeds such that the thermostat setpoint could be raised safely and not have complaints. Adding to the complexity is the open office dynamics in terms of occupants' preference for control over their thermal environment. Therefore, the practicality of the concept to raise the setpoint with ceiling fans likely depends on the application, design and layout of the fans, and occupant's willingness to engage in the control of the fans.

Another benefit of ceiling fans in radiant building is to increase the radiant cooling capacity by increasing the air velocity at the active ceiling surfaces (Karmann, Bauman, Raftery, & Schiavon, 2018). However, ceiling fans are not a necessity if the combined radiant and DOAS system has sufficient capacity to meet the thermal loads. We estimated the total cost, including equipment, control, installation and profit for contractors, is about $3000/fan.

F. Impact of Labor Rates

A substantial portion of the radiant system installation costs are associated with labor, but labor rates can vary widely by geographic region. The study site is based in the San Francisco Bay Area, where the labor rates are among the highest in the nation. To evaluate the impact of varying labor rates, we repeated the cost estimate over a range of values. Figure 11 shows the overall cost results of the radiant and VAV designs at different labor rates. The red line shows the cost premium for the radiant design over the VAV system on a unit area basis. The national average labor rate is roughly represented by the $85/hr value, which equates to a $6.8/ft^2 premium for radiant over the VAV system. The labor rates used in this study shown second from the right, resulting in $9.1/ft^2

premium. The bars at the far right represent costs at San Francisco labor rates, which increase the premium to $11.2/ft^2$.

Overall, the cost premium for radiant varies by nearly a factor of two over the range of labor rates evaluated. Note that this is a simplified evaluation to illustrate the direct impact of vary labor rates only – costs may also be impacted by other geographic differences which are not considered here, such as utility rate tariffs and design weather conditions.

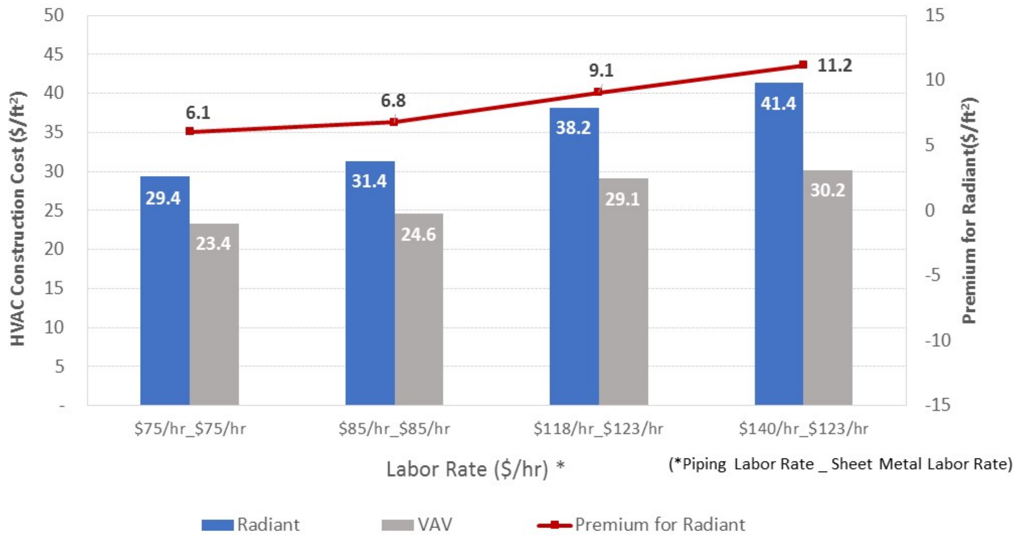


Figure 11: Impact of Labor Rates

7 Energy Performance Evaluation

The energy performance of the two designs are evaluated in EnergyPlus Version 8.7, which is one of the few building simulation packages that can effectively model high thermal mass radiant systems. It performs a fundamental heat balance on all surfaces in spaces, and can integrate the heat transfer calculations in the radiant slab hydronic system with changing space conditions (U.S. Department of Energy, 2016).

For any HVAC system type, energy and comfort performance depend highly on the control sequences. In the VAV system model, the controls are generally based on the recently published ASHRAE Guideline 36 (ASHRAE, 2018), which provides high performance sequences of operation for VAV systems that have been widely acknowledged in the building industry. However, for the hybrid radiant slab and DOAS system, there are no well-established control sequences readily available. Some of the control approaches commonly used in the industry appear to be quite energy inefficient. The results presented in the report represent the performance of a specific combination of control sequences tested during the simulation. To demonstrate the significant impact of the control approach, we have simulated a range of control options for radiant systems and documented the results in Section 7.3.

7.1 Modeling Methods

This section describes the modeling methods and assumptions that are critical for understanding the annual energy and comfort performance. **Error! Reference source not found.** shows the image of the energy model for the building. Both energy models are developed to reflect the design as much as possible, except when we are limited by the software's modeling capability.

A. Radiant Slabs

EnergyPlus module "ZoneHVAC: LowTemperatureRadiant:VariableFlow" is used for modeling the slabs. This module works with a special construction type that allows hydronic piping to be embedded in between construction layers.

The built-in EnergyPlus radiant slab control was overwritten using the Energy Management System module to implement a control strategy developed by Raftery et al. (Raftery, Duarte, Schiavon, & Bauman, 2017). Figure 12 shows a schematic diagram of the controller in cooling mode. The controller responds to both zone and slab temperature conditions and allows a user to specify periods during the day in which the radiant system cannot operate. The primary control loop is an on/off controller that controls the radiant zone valve in response to the error between the temperature sensor in the slab, placed close to the surface, and the slab setpoint.

The slab setpoint control loop then uses a proportional controller that operates using the error between the maximum/minimum zone air temperature during occupied hours on the previous day relative to the comfort setpoint for cooling/heating. This secondary controller activates once at the end of the occupied period each day, and makes the change to the slab setpoint. The comfort setpoint is 1°F (adjustable) above or below the heating and cooling limits (respectively) of the comfort bounds defined for the zone. In this way, the controller gradually responds to changes in the zone loads over the course of several days.

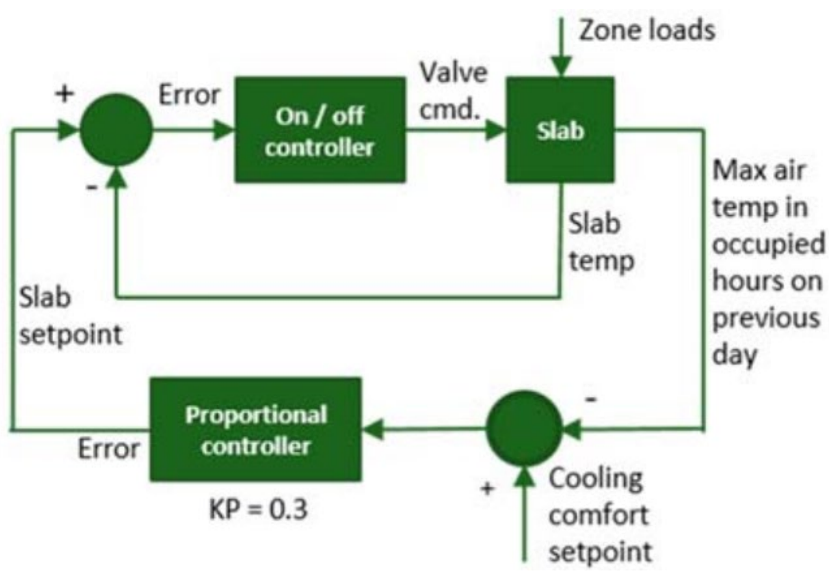


Figure 12: Schematic diagram of the controller in cooling mode. The same approach applies in heating mode, but using the minimum instead of maximum air temperature on the previous day, and heating instead of cooling comfort setpoint (Raftery, Duarte, Schiavon, & Bauman, 2017)

In addition, the controller can only operate in one mode each day – either intermittent cooling, off for the entire day, or intermittent heating. This ensures a 24-hour period between mode changes to avoid wasted energy use from heating and cooling during the same day.

Lastly, the designer selects a period in which the radiant system does not operate- e.g. shutoff from 12 pm to 2 am. This feature allows building owners to minimize utility charges at peak periods.

B. Controls and Schedules

Table 12 summarizes the control strategies implemented in the baseline energy models.

Table 12: Summary of Controls

Control	Radiant	VAV
Cooling/ Heating setpoint	• Space temperature high/low limit: 78.8/68°F • Radiant slab setpoint low/high limit: 68/71.6°F • Radiant slab setpoint resets based on previous day error from space temperature setpoints • Radiant zone locked out from mode (heating vs cooling) changes over 24-hour period • Radiant zone valve shuts off if chilled water supply temperature is less than 1°F above then zone dew point • DOAS VAV setpoint: 78.8/68°F	• Weekday occupied: 75/70°F • Weekday unoccupied: 60/85°F • Weekend: 60/85°F
Ventilation availability [1]	• Weekday: 7 am-7 pm • Weekend: no	• Weekday: 7 am -7 pm • Weekend: no
System availability schedule	• Radiant slab operation: 6 am to 12 pm (Locked out otherwise) • DOAS: same as ventilation schedule	Available 24/7 to maintain zone setpoint
Supply air temperature (SAT) control	• Changeover coil cooling setpoint: reset from 68 to 58°F based on outside air temperatures from 58 to 68°F • Changeover coil heating setpoint: 55°F.	• Range: 55 to 65°F • Setpoint resets based on warmest zone cooling demand (see note 2)
VAV terminal control	See Table 1 for DOAS VAV control	Dual-Maximum Logic (Taylor, Stein, Paliaga, & Cheng, 2012)
Zoning	See note 4	As designed
AHU supply fan	Variable flow based on demand	Variable flow based on demand
Economizer	DOAS: 100% outdoor air unit	Integrated economizer

Chilled water flow	Variable flow based on demand	Constant flow
Chilled water supply temperature (CHWST)	Constant at 57°F.	Reset between 45 and 55 °F to maintain 60°F return water temperature (see note 3)
Hot water flow	Constant flow	Constant flow
Hot water temperature	Constant at 90°F	Constant at 115°F

Notes:

1. Ventilation requirements are only enforced during this period.
2. The SAT warmest zone reset logic is not from Guideline 36. EnergyPlus does not have the capability to model Guideline 36 SAT control logic
3. This is different from design intent which resets CHWST to maintain AHU supply air temperature at setpoint. This sequence cannot be modeled directly in EnergyPlus.
4. EnergyPlus cannot model different thermal and ventilation zoning which is the case in the radiant design. Large and small conference rooms, which share radiant zone with the adjacent open offices in the design, are modeled to have a dedicated radiant zone. This prevents the model from capturing potential fighting between the DOAS and the radiant slab.

C. Air Source Heat Pump

EnergyPlus cannot directly model the centralized four-pipe air source heat pump. Instead, the chilled water and hot water plant are modeled separately as air-cooled chiller with scroll compressor and district heating plant. Hourly cooling electricity consumption is directly reported from EnergyPlus, while hourly heating electricity consumption is calculated using the plant heating loads from EnergyPlus and COPs calculated from heat pump heating mode regression models. Full load and part load power and capacity performance data from the heat pump manufacturer are used to calibrate the heat pump regression models in heating and cooling mode, but this approach does not account for the potential heat recovery opportunities. Appendix 11 presents the modeling details and the regression models.

7.2 Results

A. Thermal Comfort

To evaluate thermal comfort performance, we describe the percentage of occupied hours when any zone operative temperature exceeds the specified range from the annual simulations. Even though air temperature sensors were used in the design and control for both systems, operative temperature can better represent thermal comfort condition, and is the temperature used in the Graphic Comfort Zone Method in ASHRAE Standard 55 for acceptable thermal condition evaluation (ASHRAE, 2013). During summer operation, for clothing level at 0.55, air speed at 20 fpm, metabolic rate at 1.1 and humidity level at 50%, we use percentage of occupied hours exceeds 78.8 °F, which corresponds to a PPD level of 6% and PMV value of 0.2. For winter operation, for clothing level at 1.0, we use percentage of occupied hours lower than 68 °F, which corresponds to a PPD level of 12% and PMV value of -0.58. Figure 13 shows the two designs have achieved similar thermal comfort level in the building.

Figure 14 and Figure 15 plot the distributions of space operative temperature and air temperature (respectively) for all thermal zones on a typical floor, and the bars are color coded for different space types. Comparing the two figures we can see that even though the air temperatures in the all-air system building are controlled well within 70-75°F (bottom chart of Figure 15), the operative temperatures range from 69-78°F. In the radiant building, the operative and air temperature distributions are similar, and are controlled within 68-78°F most of the time.

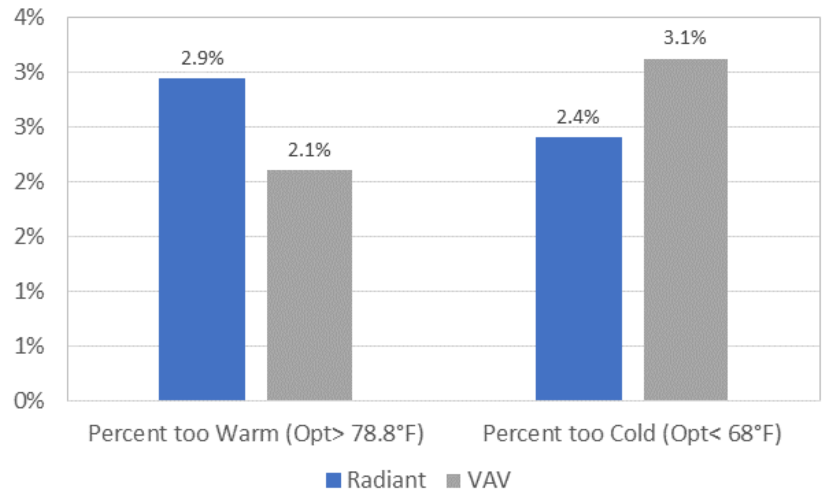


Figure 13: Thermal Comfort Performance of the Radiant vs VAV design

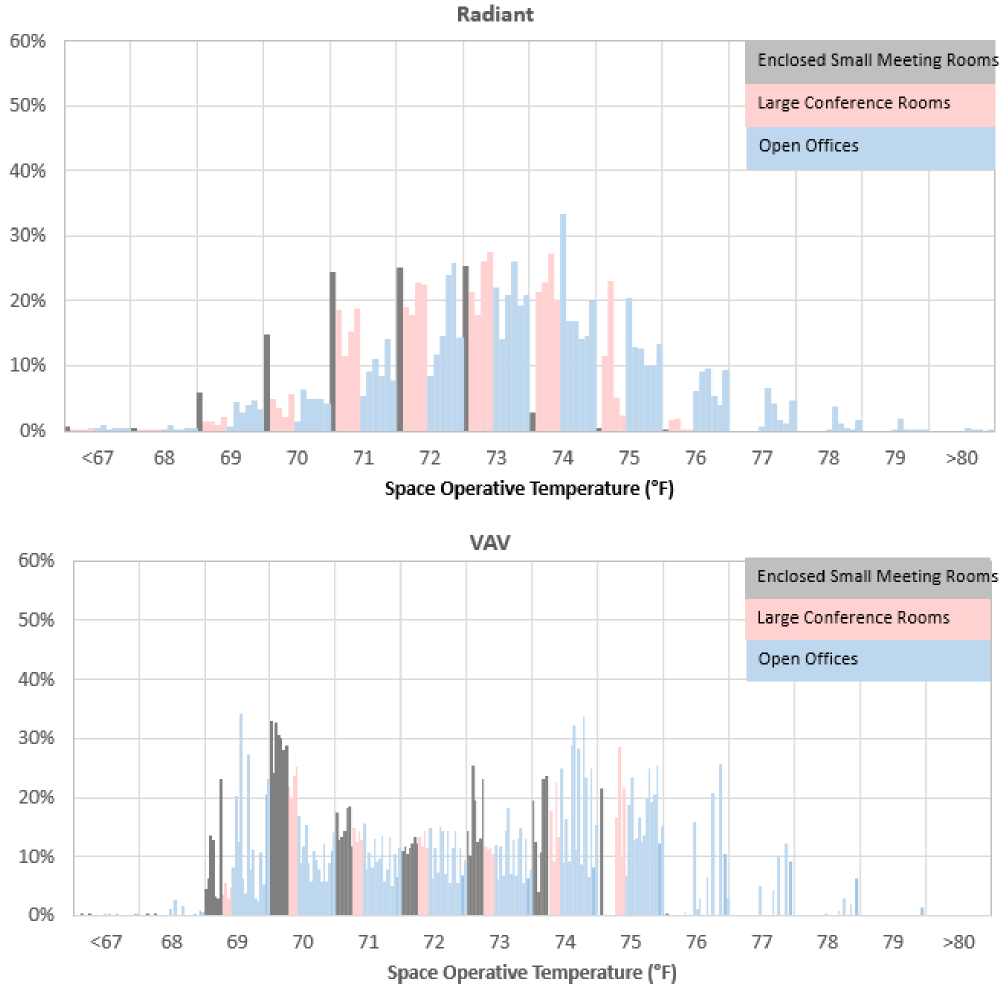


Figure 14: Space Operative Temperature Distribution in Different Space Types: (Top) Radiant Building; (Bottom) All-Air Building

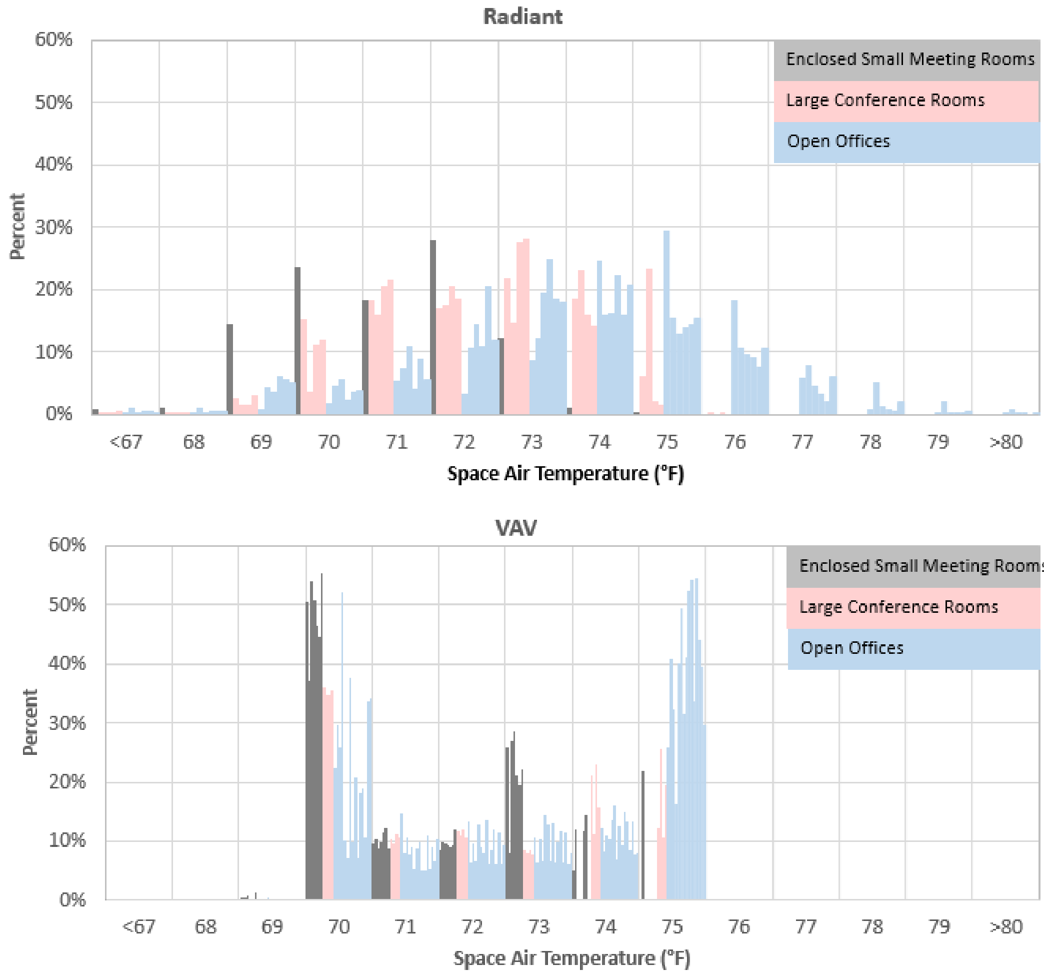


Figure 15: Space Air Temperature Distribution in Different Space Types: (Top) Radiant Building; (Bottom) All-Air Building

B. Energy and Cost

<u>Energy</u>

Figure 16 shows the HVAC annual site electricity consumption for the two designs. The total energy consumption is 2.9 kBtu/ft^2 for the radiant design and 2.5 kBtu/ft^2 for the VAV design. The radiant system HVAC energy use is 16.2% higher than the VAV design. The modeled whole building energy use intensity (EUI) is 12.8 kBtu/ft^2 for the radiant and 12.4 kBtu/ft^2 for the VAV building, both of which are exceptionally low EUIs, which is in large part a reflection of the extremely low internal loads and good envelope.

However, if evaluated using the time-dependent valuation (TDV) energy, the radiant system consumes 26.5 kBtu/ft^2 compared to 34.5 kBtu/ft^2 for the VAV design, and outperforming the VAV system by 23.1%, see Figure 17.

The TDV energy was calculated using 2016 TDV values for climate zone 3 based on 15 year forecasts. In California, the TDV is used in the cost effectiveness calculation for Title 24 code updates. "The concept behind TDV is that energy efficiency measure savings should be valued differently depending on which hours of the year the savings occur, to better reflect the actual costs of energy to consumers, to the utility system, and to society. The TDV method encourages building designers to design buildings that perform better during periods of high energy cost." (California Energy Commission, 2016)

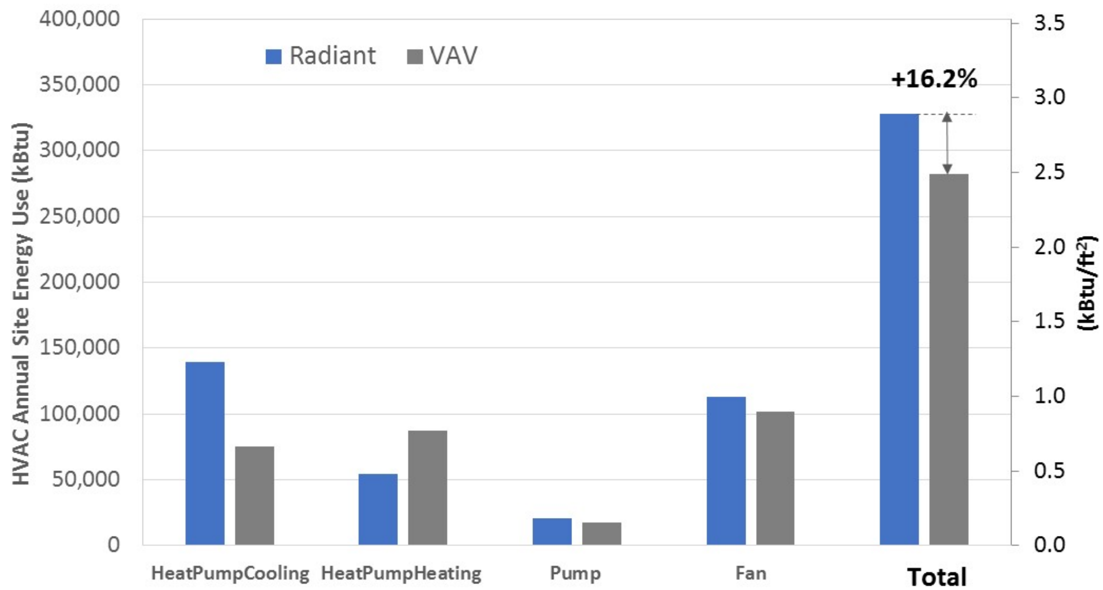


Figure 16: HVAC Annual Site Energy Use for the VAV and Radiant Designs

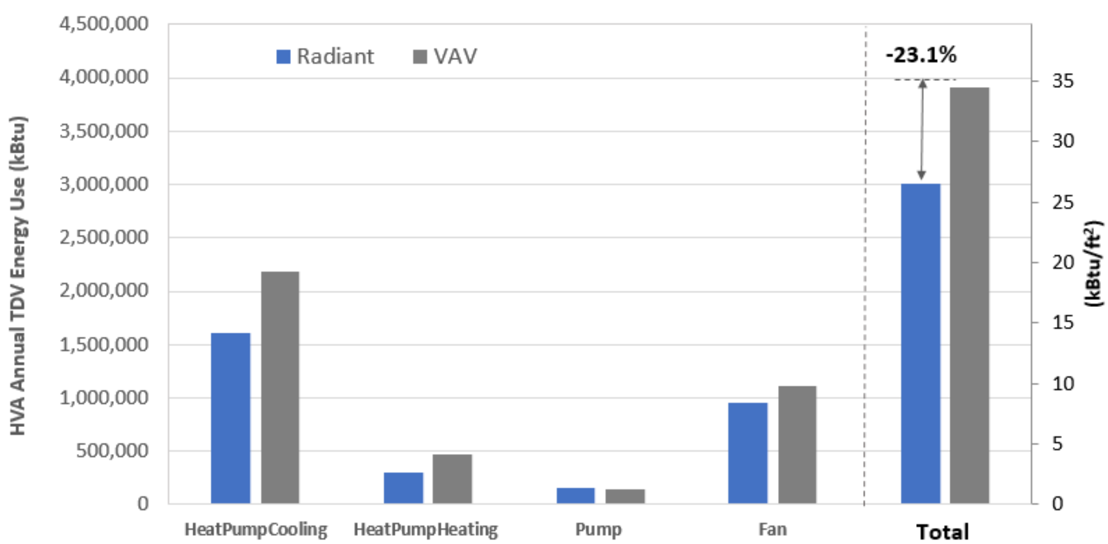


Figure 17: HVAC Annual TDV Energy Use for the VAV and Radiant Designs

Energy Cost

Building total electricity consumption was used for energy cost calculation. For both designs, the non-HVAC electricity uses, including lighting and internal equipment, were 3.15 kWh/ft^2. For the VAV building, the HVAC total electricity consumption is 20.1% of total building electricity use, and for radiant building, the HVAC total electricity is 22.6% of the total building electricity use.

Total building electricity cost was calculated using the PG&E E19 rate schedule (PG&E, 2018). It consists of three parts: customer mandatory charge (a flat rate per day), energy charge and demand charge. The PG&E E19 rate tariff and Time-of-Use schedule are summarized in Table 13, and the monthly building electricity costs for the VAV and radiant building are summarized in Table 14.

Table 13: PG&E E19 Utility Rate Tariff and Time-of-Use (TOU) Schedule

Season	Rate Period	Energy Charge ($/kWh)	Demand Charge ($/kW)		Time-of-Use Period
			TOU Demand	Maximum Demand	
Summer	Peak	0.16055	19.65	17.74	12 noon to 6:00 pm M-F (except holidays)
	Part-Peak	0.11613	5.4		8:30 am to 12 noon M-F (except holidays) 6:00 pm to 9:30 pm M-F (except holidays)
	Off-Peak	0.08671	-		9:30 pm to 8:30 am M-F (except holidays) All day Saturday, Sunday, and Holidays
Winter	Part-Peak	0.11004	0.12	17.74	8:30 am to 9:30 pm M-F (except Holidays)
	Off-Peak	0.09401	-		9:30 pm to 8:30 am M-F (except Holidays) All day Saturday, Sunday, and Holidays

Table 14 Monthly Whole Building Electricity Cost

Month	Mandatory Charge ($)	Radiant			VAV		
		Energy Charge ($)	Demand Charge ($)	Total ($)	Energy Charge ($)	Demand Charge ($)	Total ($)
1	$611	$3,914	$4,028	**$8,552**	$4,016	$4,305	**$8,932**
2	$552	$3,503	$3,312	**$7,367**	$3,486	$3,450	**$7,488**
3	$611	$4,070	$3,741	**$8,423**	$3,938	$3,182	**$7,731**
4	$591	$3,974	$2,806	**$7,371**	$3,866	$2,942	**$7,399**
5	$611	$4,583	$7,242	**$12,436**	$4,436	$9,268	**$14,315**
6	$591	$5,121	$7,762	**$13,474**	$5,114	$11,521	**$17,227**
7	$611	$5,521	$7,681	**$13,813**	$5,370	$10,656	**$16,637**
8	$611	$4,997	$7,074	**$12,682**	$4,746	$7,159	**$12,516**
9	$591	$5,251	$9,126	**$14,969**	$5,367	$11,658	**$17,617**

10	$611	$4,916	$6,969	**$12,496**	$4,652	$8,231	**$13,493**
11	$591	$3,491	$3,303	**$7,385**	$3,387	$3,402	**$7,380**
12	$611	$4,031	$3,605	**$8,247**	$4,017	$2,963	**$7,591**
Total ($)	**$7,195**	**$53,371**	**$66,648**	**$127,215**	**$52,396**	**$78,735**	**$138,326**
Total ($/ft^2)	$0.06	$0.47	$0.59	**$1.12**	$0.46	$0.69	**$1.22**

For this case study building, the annual building electricity cost for the radiant design is $1.12/ft^2, compared to $1.22/ft^2 for the VAV design, representing an 8.0% savings:

- The total energy charges for the radiant building is $0.47/ft^2, which is 1.9% higher than the VAV design.
- The total demand charge is for the radiant building is $0.49/ft^2, which is 15.3% lower than the VAV design. Operating the radiant system from 6 am to 12 pm does not significantly reduced the maximum peak demand of the month, but it shifts the peak electricity demand to "Part-Peak" period. We investigate the impact of the radiant operation schedule in detail in the discussion section.

C. Fan Energy

It may be counter-intuitive that the DOAS supply fan consumes more energy than the VAV supply fan given that the total design flow is only one third of the VAV design flow. However, we can explain this when plotting the power vs. flowrate correlations of the two fans, see Figure 18. The Fan Laws suggest that fan power is proportional to the cube of flow rate. Even though the cubic relationship cannot be achieved during real operation, the VAV supply fan power reduces significantly at part load conditions. Based on Figure 18, the VAV fan power consumption is less than the DOAS fan power when the flow rate drops below 36,000 cfm. Figure 19 plots the DOAS and VAV supply fan flowrate distributions from the annual runs. It shows the total number of hours of each airflow range. While the DOAS mostly runs at constant volume, the VAV unit runs at low part load conditions most of the time, with more than 90 percent of the time lower than 36,000 cfm. For more than 50% of the time when the fan is running, the VAV system only needs to provide minimum ventilation at low fan power.

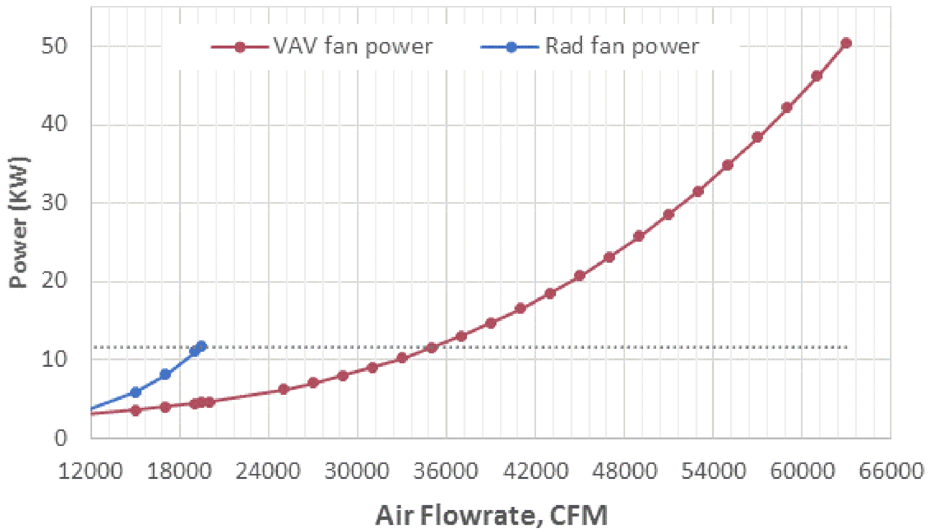


Figure 18: Power vs. Air Flowrate Correlation for the VAV and DOAS Supply Fans

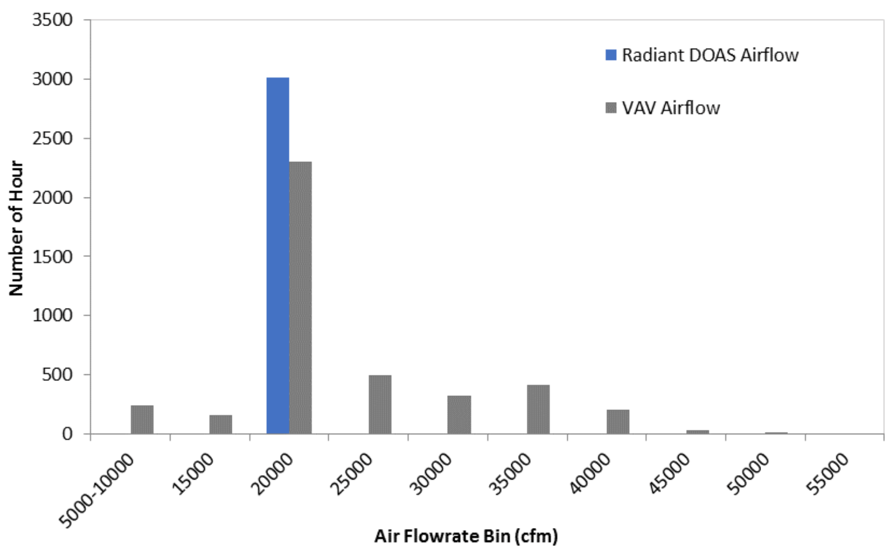


Figure 19: Annual Air Flowrate Distribution for the VAV and DOAS Supply Fan

D. Cooling

Figure 20 shows the monthly heat pump cooling load (not electricity consumption) for both designs, together with the monthly average outdoor air drybulb temperature. For the radiant design, we breakdown the total cooling load into radiant slab cooling load and the DOAS cooling coil load. It shows that the radiant system design consumes more cooling energy in winter and summer months due to lack of air-side economizer. The San Francisco weather is very mild all year long. Even during the summer months, the average dry bulb temperatures are in the mid-60s, which is in a range where an air-side economizer works effectively to provide free cooling.

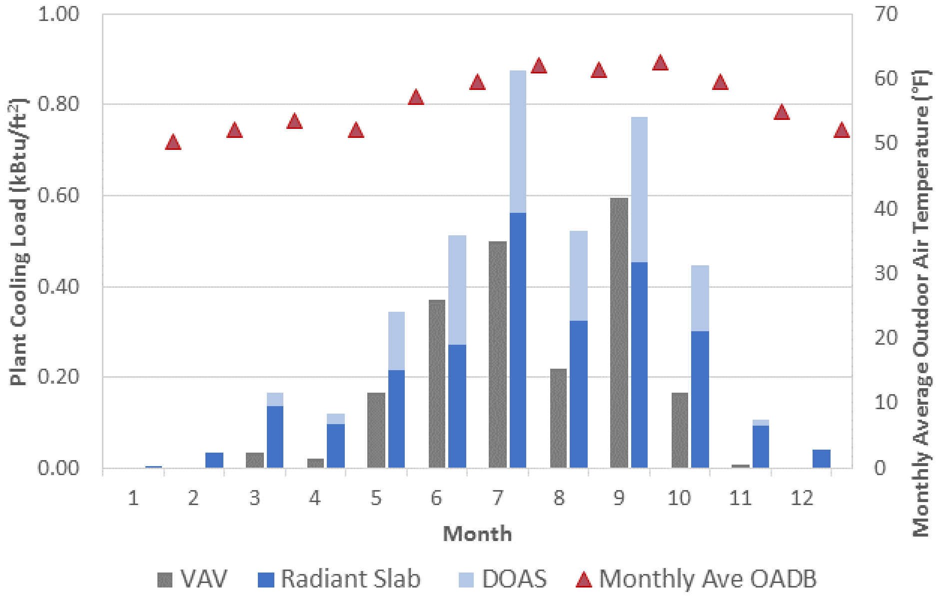


Figure 20: Monthly Plant Cooling Load for the VAV and Radiant Designs

Figure 21 plots the heat pump cooling load profiles for both designs for four representative days in August, together with the outdoor air temperature and the air system supply air temperatures. In the mornings when the outdoor air temperature was lower or just above 60°F, the VAV design operated without mechanical cooling, while the radiant slabs were activated to remove heat from the spaces. When the outdoor temperature raised in the afternoons, the VAV system started to use mechanical cooling. But with the integrated economizer strategy, the total cooling demand was not significantly higher than the radiant design.

The mild San Francisco weather also benefits the VAV design by allowing the supply air temperature to reset aggressively without dehumidification concern. In humid climates, a DOAS could potentially save energy by reducing simultaneous cooling and reheat that would otherwise be needed to provide adequate dehumidification in the case of a traditional VAV design. From Figure 21, we see that the DOAS supply air temperature, which reset from 58 to 68 °F based on outside air temperature from 68 to 58°F, was lower than the VAV supply air temperature, which reset based on zone cooling demands. Because of the relatively low DOAS SAT setpoint, the DOAS cooling energy consists of a significant portion of the overall cooling consumption, about 35%, see Figure 20. If a higher DOAS reset range were used, the radiant slab cooling would be a bigger portion. See more discussion in paragraph 7.3B on this topic. With the radiant slabs locked out in the afternoon, the DOAS provided some supplemental cooling to the building.

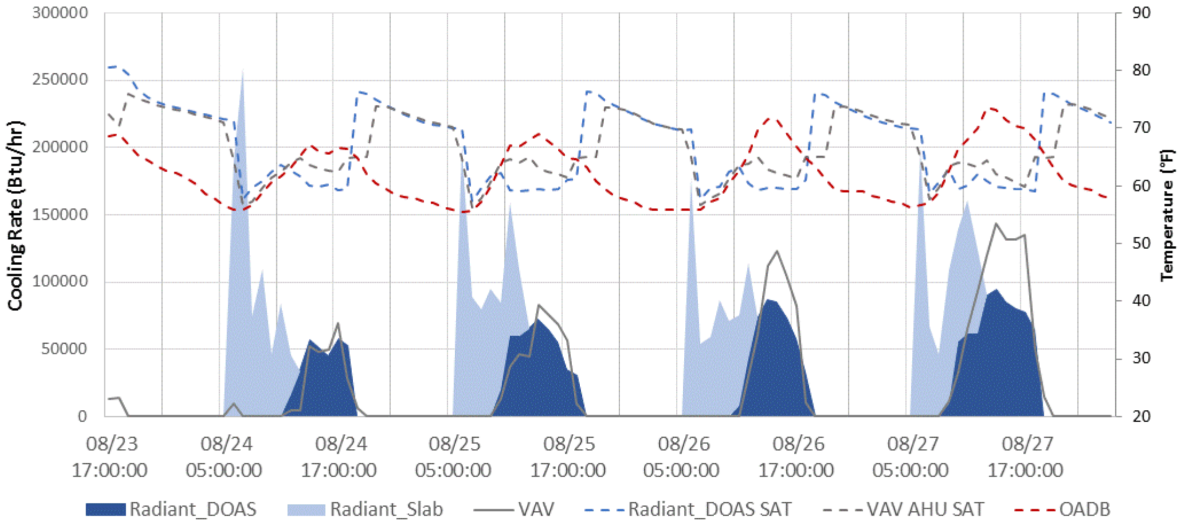


Figure 21: Radiant and VAV System Cooling Rate during Summer

On the plant side, the radiant system heat pump has an annual average cooling COP of 3.21, and the VAV heat pump annual cooling COP is 3.12. Figure 22 shows the heat pump cooling COPs from the annual simulation runs. The radiant heat pump operated during many more periods of low ambient temperature, contributing to the slightly improved plant efficiency. The VAV system heat pump also operated at relatively high efficiency due to aggressive reset of chilled water supply water temperature to meet the cooling load most of the time.

To ensure the DOAS has adequate dehumidification capability, we have limited the chilled water supply temperature to a constant 57°F. This is a slightly conservative approach and unique to this design as the DOAS and radiant slab use the same cooling source, which is not recommended in applications in more humid climates. We could also decouple the cooling source for the DOAS from the radiant system by using a conventional DX cooling coil and gas heating but with added cost. For more humid climates, the added cost may be justifiable with improved chiller plant efficiency by supplying higher chilled water temperature.

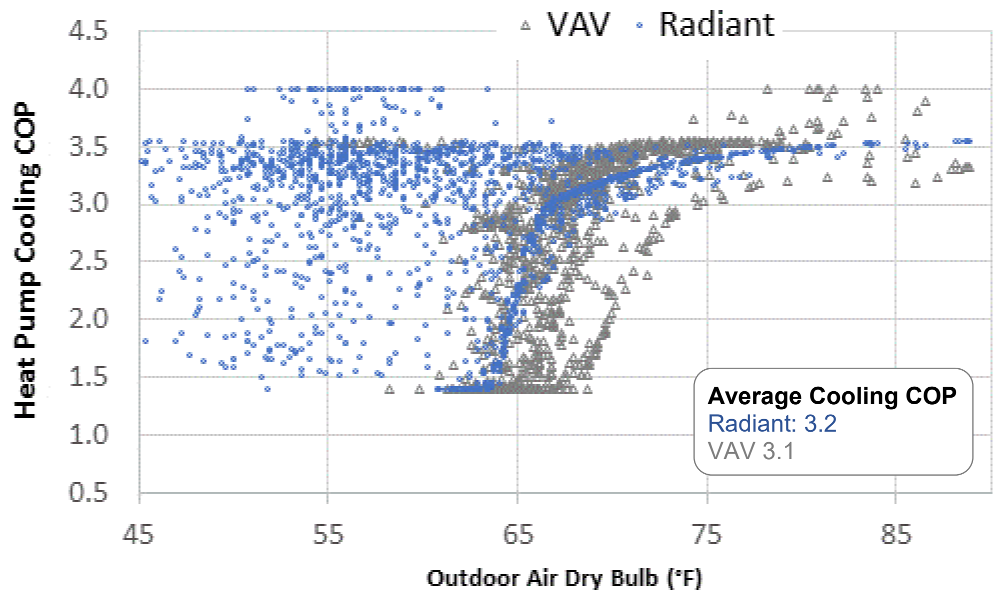


Figure 22: Heat Pump Cooling COP Comparison

E. Heating

Figure 23 shows the monthly heat pump heating load (not electricity consumption) for both designs. The monthly heating loads are consistently higher for the VAV system than for radiant. Figure 24 plots the heat pump heating COPs from the annual simulations of for the radiant and VAV design. The heat pump for the radiant system operates at much higher efficiency than that the VAV system due to lower hot water supply temperatures.

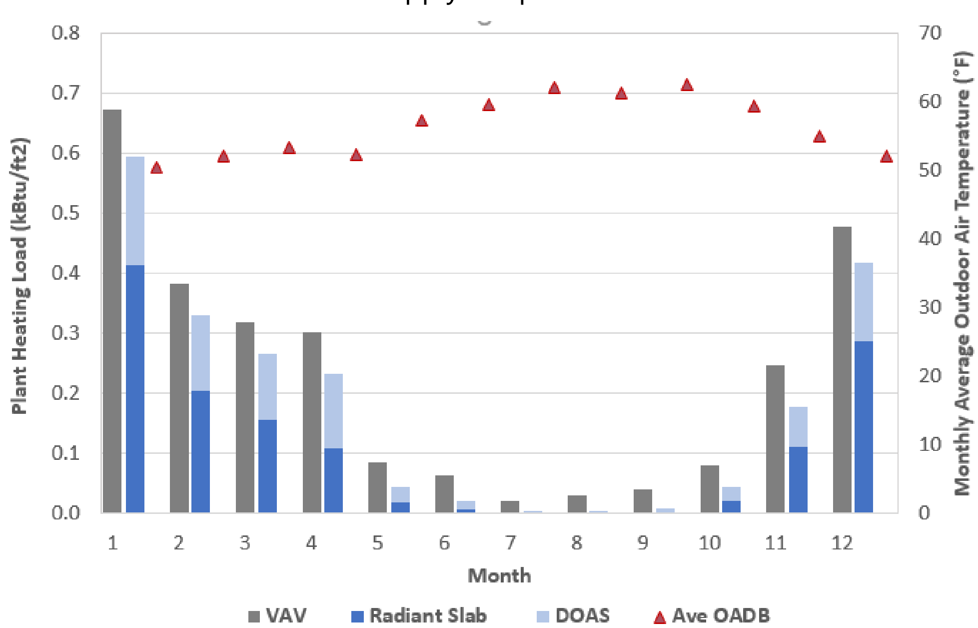


Figure 23: Monthly Heating Plant Heating Load for the VAV and Radiant Designs

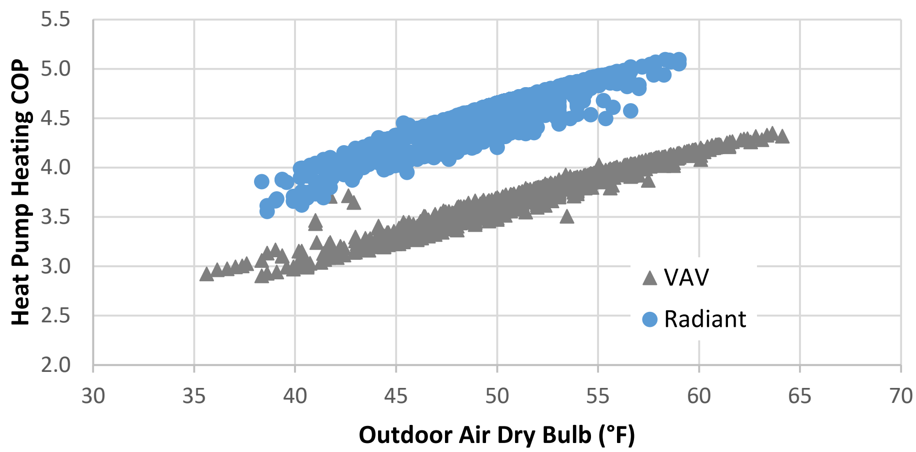


Figure 24: Radiant vs. VAV Heat Pump Heating COPs from the Annual Simulations

7.3 Discussion

A. Impact of Radiant System Operation Schedule

For high thermal mass radiant system design, there is research showing the possibility to undersize the central plant cooling and heating equipment if load shifting control strategies are to be implemented. The load shifting strategies also have the potential to reduce operating costs as we have shown from the simulations. In this section, we investigate the impacts of radiant operating schedule on site energy consumption, operating costs, and thermal comfort.

The radiant system controller developed by Raftery et. al allows the users to lock out the operation of the radiant system during predefined hours of the day and thus explore the possibility of load shifting to reduce demand charge (Raftery, Duarte, Schiavon, & Bauman, 2017). Figure 25 compares the HVAC site energy consumption and building electricity costs when the radiant slab system uses different operating schedules: Figure 26 presents the thermal comfort performance of each operating schedule based on operative temperature. The results apply to the case study building, and the optimal operating schedule depends on the load profiles of an individual building, the amount of mass activated, and controls of the DOAS system. To isolate the impacts of operating schedule, we have fixed all other control parameters. That means the overall energy performance may not be the optimal if other control parameters are fine tuned to pair with the slab schedule. For example, there is a possibility that when running the slab system 24/7, the DOAS supply air temperature could be reset higher to yield better overall energy performance while achieving similar comfort level in the building.

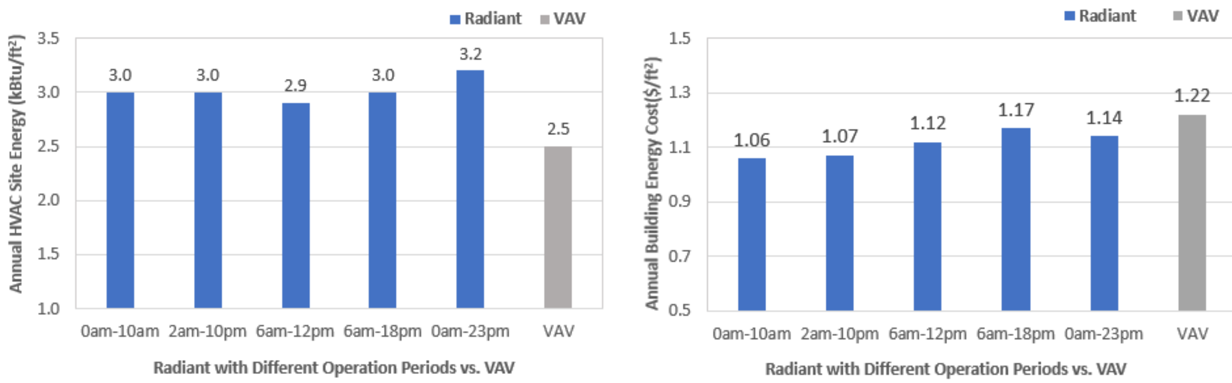


Figure 25: Comparison between VAV and Radiant System using Different Lockout Schedules: (Left) Annual HVAC Site Energy Use; (Right) Annual Building Electricity Cost

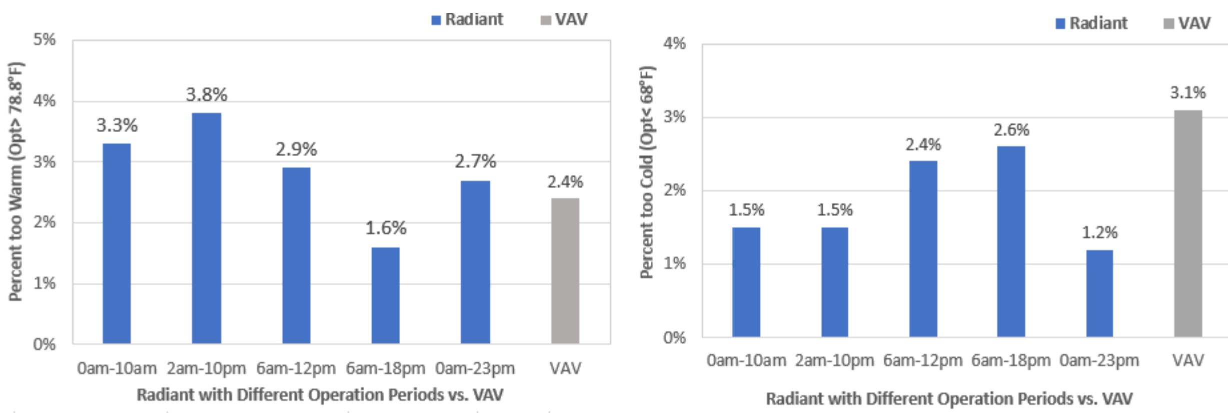


Figure 26: Comparison between VAV and Radiant System using Different Lockout Schedules: (Left) Annual Total Percent Occupied Hours too Warm (>78.8 °F); (Right) Annual Total Percent Occupied Hours too Cold (<68.0 °F)

Among the cases we simulated, running radiant slab system from 6 am to 12 pm yields the lowest site HVAC energy use, but running the slab system from midnight to 10 am yields the lowest electricity cost. With the latter schedule, the spaces in the west perimeter zones were often too warm in the late afternoon when solar load hit. On the other hand, there were fewer cold hours in the early morning.

Operating the slabs from 6 am -18 pm represents a more traditional operation strategy. Figure 25 shows that the resulting site HVAC energy consumption is slightly higher than using the baseline schedule (from 6 am – 12 pm), but the total building energy cost is 5% higher due to the large increase in demand charges.

Figure 27 and Figure 28 show the cooling plant load profiles for the VAV system and the radiant system using the 0 am - 12 pm schedule and 6 am-18 pm schedule respectively during a typical cooling week. With the 0 am - 12 pm schedule, radiant slab cooling occurs early in the mornings and the peak load on the plant side is flattened. In theory, this control approach allows for reduced equipment size. When controlling from 6 am-18 pm, more of the radiant cooling load occurs during the late afternoon during, where it is coincident with the DOAS cooling load and on-peak utility period, and results in a higher peak cooling load; contributing to higher demand charges.

To optimize the overall energy cost and improve thermal comfort, designer could adjust the operation schedule based on seasons or weather forecast. For example, allowing the radiant slab system to run 24/7 annually may result in energy waste but using this schedule during the peak cooling and heating demand periods could potentially improve thermal comfort while allowing the possibility to reduce plant equipment size. During partial load periods, the radiant system could be locked out for longer hours. On sizing of the equipment, because the radiant slabs act as a thermal storage mechanism, it is not important for the plant to meet the instantaneous cooling or heating load as long as the plant can cool or heat the radiant slab to a prescribed temperature setpoint within a specified period.

From building design perspective, one of the key elements to facilitate the adoption of load shifting strategy is to manage solar heat gain, in particularly in west and south perimeter zones to avoid space temperature spikes in late afternoon.

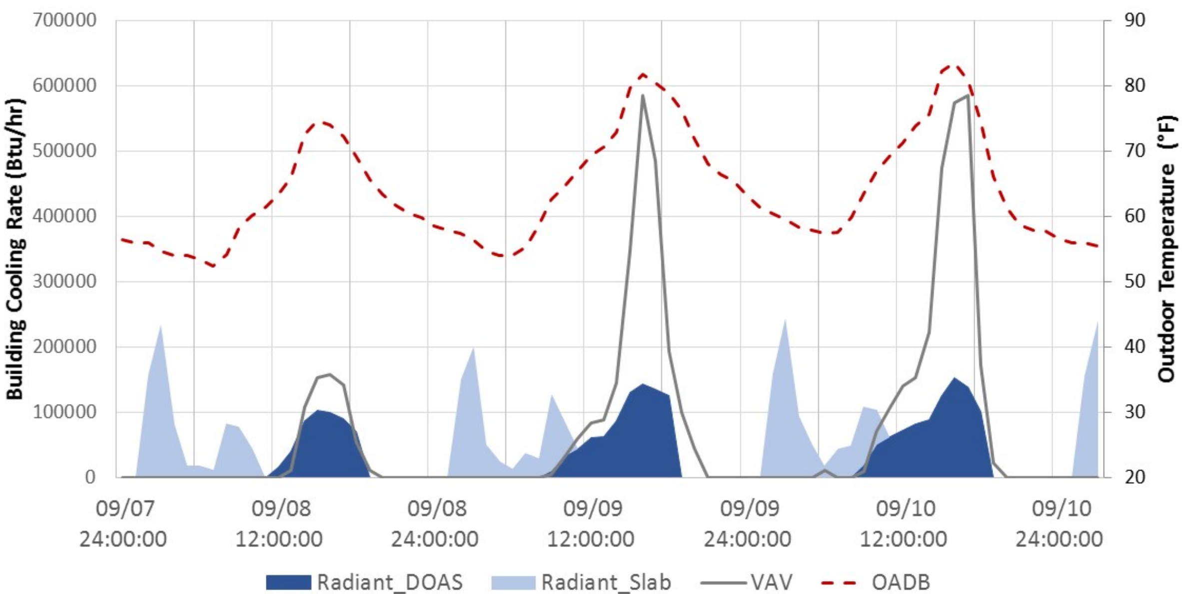


Figure 27: Cooling Plant Load Profiles for VAV vs Radiant (Radiant Slab 0 am – 12 pm during weekdays)

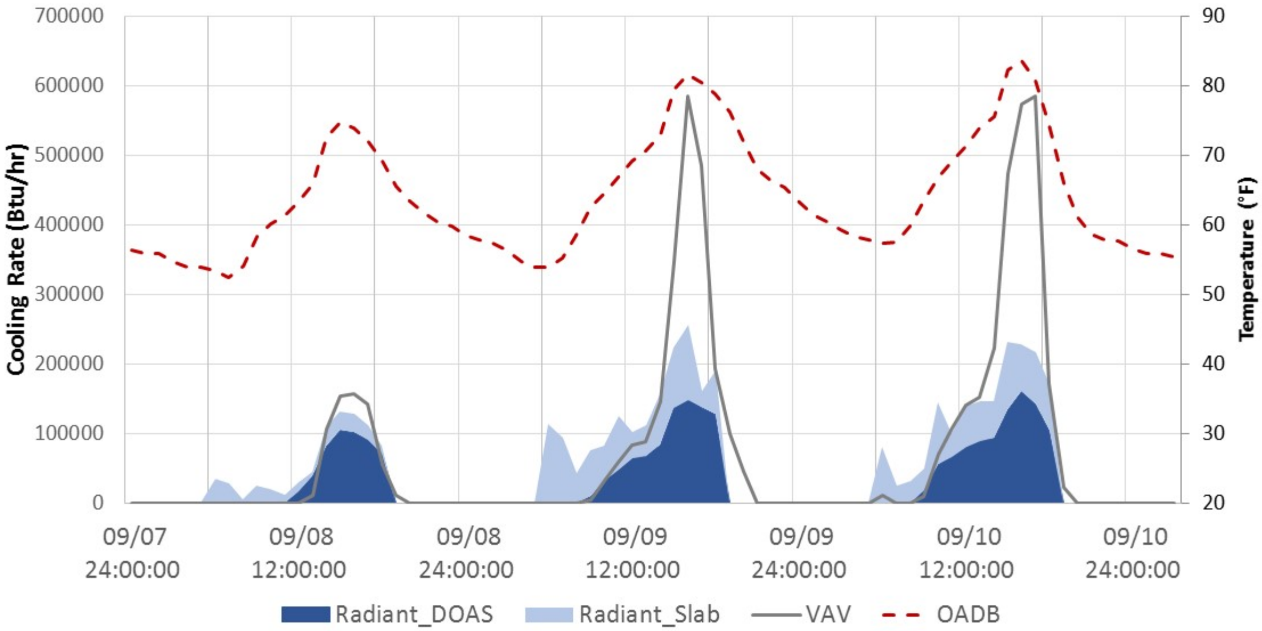


Figure 28: Cooling Plant Load Profiles for VAV vs Radiant (Radiant Slab 6 am – 18 pm during weekdays)

B. Impact of Dedicated Outdoor Air System (DOAS) Supply Air Temperature Control

The DOAS supply air temperature control is another challenging area that is critical to the overall energy performance of a radiant design. The TRC survey report shows that there is a wide range of DOAS SAT methods used among experience designers, including constant supply air temperature, supply air temperature adjusted for supplemental cooling or adjusted according to dehumidification needs (Paliaga, et al. 2017).

One of the strategies that is commonly employed is to supply air at neutral temperature, which involves cooling the air to remove moisture and then heated it back to neutral temperature. One possible benefit of this strategy is that it simplifies the radiant slab control by not providing cooling or heating from the air side. With heavy weight building mass, the space temperature can swing slowly such that no frequent mode switch is necessary and a whole floor or even a whole building can operate well with a 2-pipe radiant system. However, supplying neutral air during those economizer hours could result in significant heating and cooling energy waste, both at the DOAS and at the slabs, if radiant slabs are predominantly in cooling mode. The penalty is worse if the DOAS is oversized either for over ventilation (such as to gain the LEED point) or supplemental cooling.

Another study (Shank & Mumma, 2001) suggests the supply air temperature leaving the DOAS should be no higher than 55°F, and the supply air dew-point temperature should be kept at 44°F. However, this conclusion was based on simulations using Atlanta weather data, which is very humid and hot. In addition, it assumes the internal loads are at 3-5 W/ft^2, which are much higher than the current code minimum requirements.

In the mild and drier climates, as is the case in this study, lowered DOAS supply air temperature setpoint could maximize the use of free cooling and reduce mechanical cooling. On the other hand, with the high mass radiant system, it is inefficient to use the slabs as a mean of reheat, which some other fast responsive terminal units that commonly paired with DOAS would usually do if the DOAS overcools the space. Therefore, it is not desirable to reset the DOAS supply air temperature too low when there are simultaneous heating and cooling demands in different zones. In addition, it is likely to have interior zones that have cooling-only radiant system, as is the case in this study, supplying low temperature air from the DOAS even at minimum ventilation flowrate may over cool those spaces at part load conditions.

We have tested several control approaches for their energy impacts for the case study building:

- Deadband control: with this method, the changeover coil uses a large deadband between the heating and cooling setpoints. Heating and cooling setpoint can be individually reset based on predefined logic. For example, the cooling setpoint can be reset based on outside air temperature or cooling demands from all DOAS zones, while heating setpoint can be a constant value. Figure 29 shows an example of this approach with the cooling setpoint reset based on outside air temperature. The cooling setpoint reset range is adjustable. Another approach to reset the cooling setpoint based on zone cooling demands. In EnergyPlus, this strategy is called "Warmest Zone Supply Air Reset". This controller attempts to establish a supply air temperature setpoint that meets the cooling load of the zone needing the coldest air at the maximum zone supply air flow. This is the strategy used to control the supply air temperature of the VAV system.
- Single setpoint control: it is common that the DOAS heating and cooling coils (or changeover coil here) are controlled to maintain a single setpoint with a small deadband to avoid fighting. This single setpoint could be reset based on outside air temperature, based on season (using a constant value for each predefined period of the year), or based on zone demands. Figure 30 shows an example of this approach that reset the SAT based on outdoor air temperature. The SAT setpoint reset range is adjustable.
- Constant neutral supply air temperature: this approach requires separate cooling and heating devices in the DOAS such that the air can be cooled for humidity control and then heated back to a neutral temperature, as shown in Figure 31. In the radiant design for this study, the DOAS uses a changeover heating/cooling coil, and therefore, this approach is not feasible without causing condensation in the building. We modified the DOAS configuration in EnergyPlus to simulate this approach just for comparison purpose.

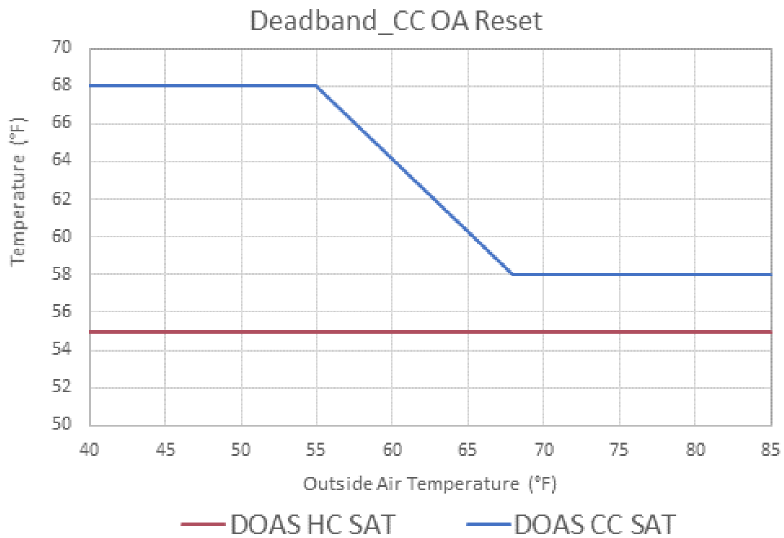


Figure 29: Example DOAS Supply Air Temperature Control Approach using Large Deadband with OA Reset for the Cooling Setpoint

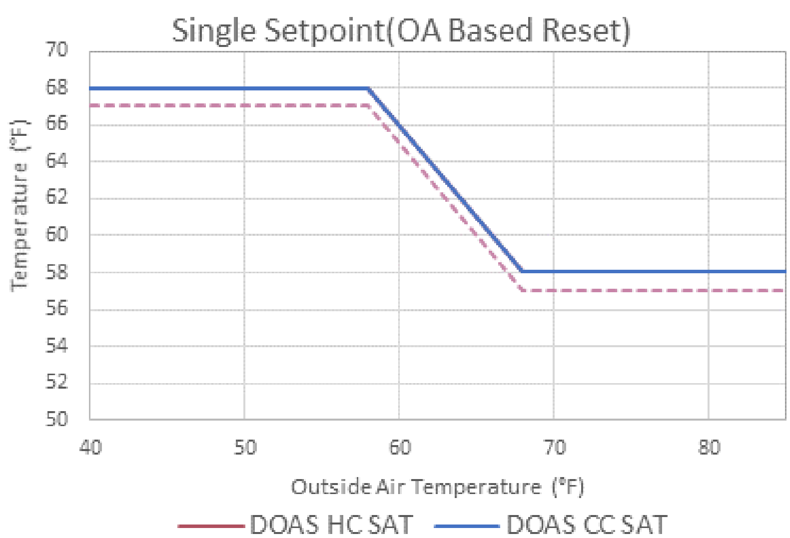


Figure 30: Example DOAS Supply Air Temperature Control Approach using Single Setpoint which Reset Based on Outdoor Air Temperature

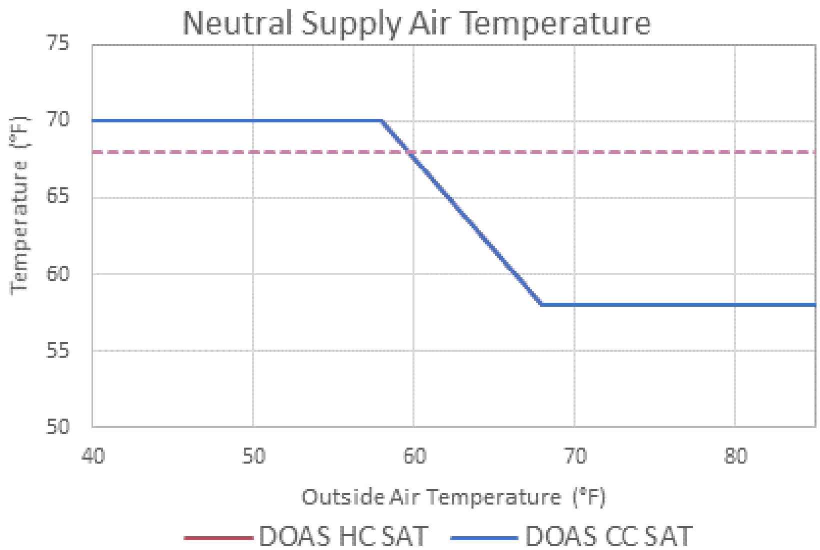


Figure 31: DOAS Supply Air Temperature Control Approach using Neutral Temperature at 68°F

For each supply air temperature reset approach, there are numerous combinations for heating and cooling setpoint logics and reset ranges. This study is not intended to find the optimal combination, rather to show the energy impact of the control sequences. Table 15 summarizes the settings for the control approaches we tested, and Figure 36 compares the site HVAC energy uses. For all runs, the radiant slab control was kept the same as the baseline control.

Among the approaches tested, the deadband approach with cooling setpoint reset based on outside air temperature appears to perform the best. The approach that uses neutral supply air temperature has the highest heating and cooling energy consumption.

Table 15: DOAS Supply Air Temperature (SAT) Control Sequences Settings

Approach	Heating Setpoint Logic	Cooling Setpoint Logic	OAT Low (°F)	OAT High (°F)	SAT High (°F)	SAT Low (°F)
Deadband	constant at 55F	OA reset 1	55	68	68	68
	constant at 55F	OA reset 2	60	75	71.6	60
	constant at 55F	OA reset 3 (Baseline Radiant)	55	68	68	58
	constant at 55F	Warmest 1	n/a	n/a	68	58
	constant at 55F	Warmest 2	n/a	n/a	68	55
Single setpoint	OA reset 1		55	68	68	58
	OA reset 2		58	72	68	58
	Warmest 1		n/a	n/a	68	58
	Warmest 2		n/a	n/a	68	55
Neutral	Constant 68	OA reset 1	55	68	68	68
VAV	n/a	Warmest (Baseline VAV)	n/a	n/a	65	55

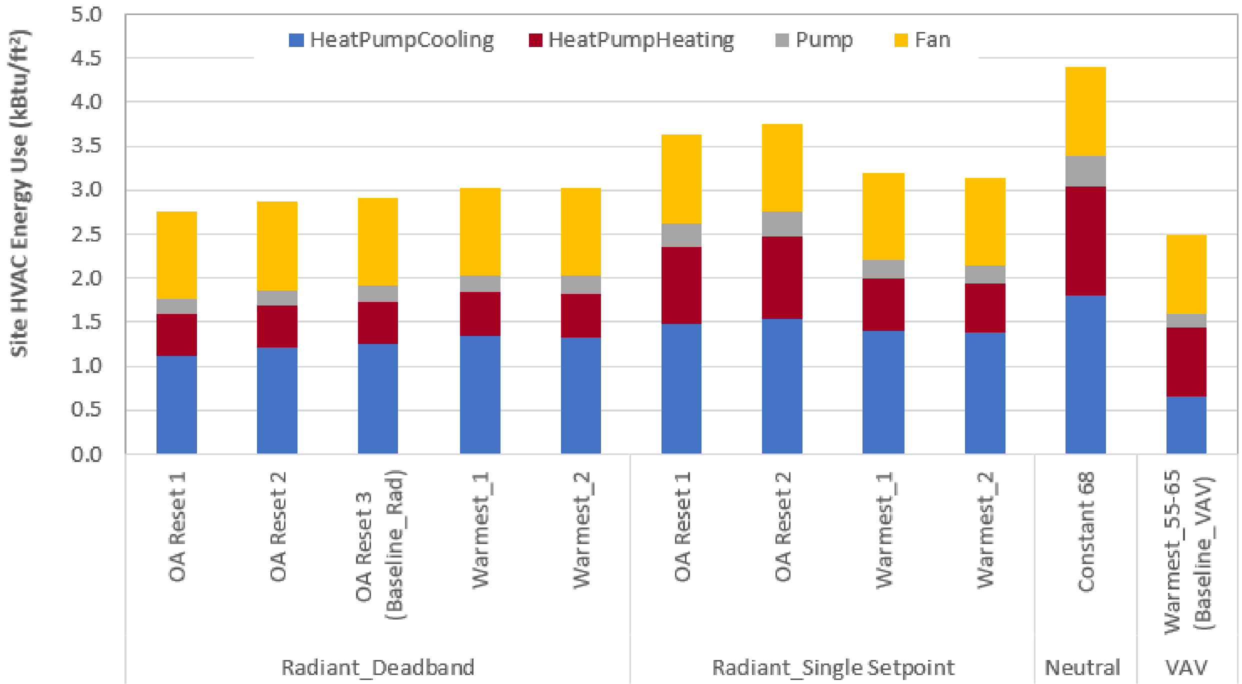


Figure 32: Site HVAC Energy Use with Different DOAS SAT Control Approaches

Among the different SAT reset ranges tested in the deadband approach, the strategy that maximizes free air-side cooling has the lowest energy consumption (OA Reset 1 in Figure 32). It supplies unconditioned outside air into the space when outside air temperature is between 55-68 °F. This approach also pushes the radiant slab system to take on more cooling responsibility, see Figure 33, which translates to lower energy cost as load has been shifted to partial peak demand period. However, this approach is not practical all the time as the DOAS also needs to dehumidify the ventilation air when necessary. As there is no active humidity monitoring in the model, we conservatively used a lower SAT range in the radiant baseline model, which results in about 5% more energy consumption and an even larger cost penalty as the DOAS must take on more cooling load.

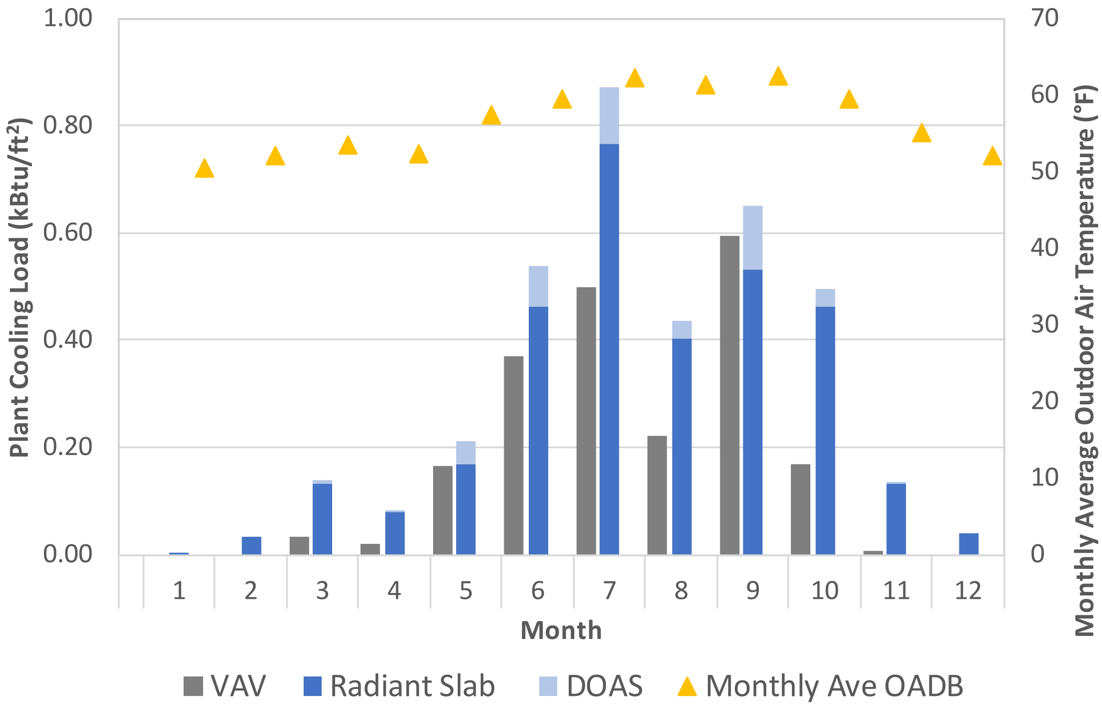


Figure 33: Monthly Plant Cooling Load with DOAS Supplying Unconditioned Ventilation Air When Outside Air Temperature Between 55-68°F

In the Bay Area climate, dehumidification is only needed for limited hours in a year. To maximize free cooling from the DOAS and push more load to the radiant slab system, designers should allow a higher DOAS supply air temperature cooling setpoint and use active humidity monitoring to reset supply air temperature lower only when it is needed to prevent condensation. Space dew point sensors are a common control feature in radiant buildings and could be used for this purpose.

C. Impact of Economizer

In mild climates, such as the Bay Area in California, HVAC designs should take advantage of the benefits of free cooling as much as possible either with airside or waterside economizers. In paragraph 7.2D, we have shown that the VAV design uses much less cooling energy than the radiant design because of the airside economizer. In other climates, the relative energy comparison may shift because of reduced airside economizer opportunities. Though it was outside of the scope of this study to examine other climates, we evaluated the impact of the airside economizer by simulating the VAV system with the economizer disabled. The radiant system uses similar but slightly less site energy compared to the VAV system without the airside economizer, as shown in Figure 34, and the annual total building energy cost for the VAV system would increase to $1.24/ft^2$, which is 13.2% higher than the baseline radiant design.

In the 2019 version of the California Title 24 energy code, air or waterside economizers will be prescriptively required for chilled water cooling systems without a fan with cooling capacities greater than the climate-specific thresholds. For California climate zone 3, the threshold is 940,000 Btu/hr (78.3 tons). ASHRAE Standard 90.1-2016 has a similar requirement.

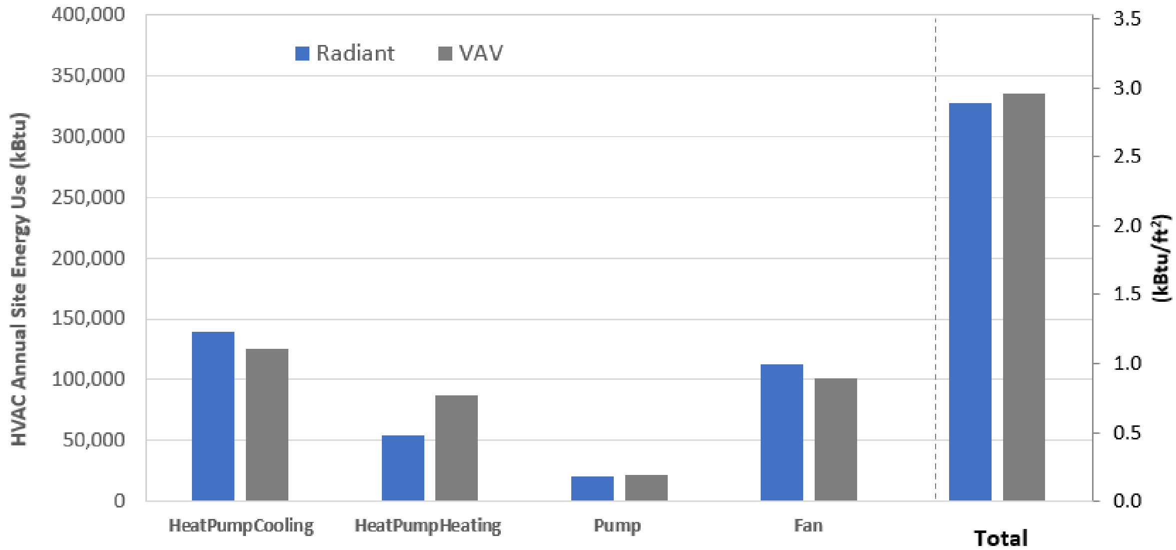


Figure 34: Comparison of HVAC Annual Site Energy Use for the Radiant vs. VAV Design without Airside Economizer

To lower the overall construction cost when including waterside economizers, designers should consider a holistic approach that would facilitate the use of a load shifting strategy such that the plant equipment size could be reduced to offset the cost of the waterside economizer.

Research from modeling studies has demonstrated that it is possible to use relatively high-water temperatures to cool down radiant slabs and still maintain thermal comfort (Duarte, Raftery, Schiavon, & Bauman, 2018). Use of high cooling water supply temperatures allows the use of evaporative cooling, such as cooling towers, as the sole source of cooling for the slab.

D. Impact of Control and Operation on Performance

The simulated energy performance represents high performance design and control strategies and idealized operation for both system types. Typical building energy use for similar buildings and systems in reality will likely be higher than reflected here due to factors such as sub-optimal control strategies and operational issues.

The discussions in paragraphs 7.3A and 7.3B describe how different approaches to controlling the radiant slab and the DOAS supply air temperature can have a large impact on the energy performance of the radiant system. For example, controlling the DOAS to a constant neutral supply temperature increases HVAC energy consumption by 50%. Actual energy use may increase due to other control factors, such as by not preventing rapid switching between heating and cooling modes. Though the impact of varying control strategies for VAV systems was not studied here, previous simulation studies have shown that poor control strategies may increase HVAC energy use by over 50% compared to average practice (Pang, Piette, & Zhou, 2017) in VAV systems.

Operational deficiencies and operator overrides may also significantly increase actual building energy use in real life. Issues commonly observed in VAV systems include, but are not limited to:

- Inoperable economizer dampers due to linkage failure, rusted dampers, etc.
- Control valves that bypass flow to coils when commanded shut
- Supply air temperature and duct static pressure setpoints that do not reset due to “rogue zones” or operator overrides

Though similar issues may impact radiant systems, the potential negative impact on energy performance may be less severe for radiant systems since there are fewer controlled devices. Further research is needed to better understand the common operational issues impacting on radiant system performance.

8 Opportunities for Further Research

The first cost analysis, coupled with energy simulation analysis, suggest opportunities to reduce installed cost and improve energy efficiency. However, there are many questions raised that were not able to answer as part of this project. There is a diverse range of commonly used approaches for design and control of the systems that we have not thoroughly evaluated, and there are also approaches that are pushing the current limit and appear to offer great opportunities based on preliminary results. More research is needed to comprehensively study those topics from energy, thermal comfort, and overall first cost perspective.

- Radiant system energy efficiency could be significantly improved if the design can take advantage of the benefit of free cooling either by airside or waterside economizer. In the 2019 version of the California Title 24 energy code, air or waterside economizers (WSE) will be prescriptively required for chilled water cooling systems without a fan with cooling capacities greater than the climate-specific thresholds. Research has demonstrated that it is possible to eliminate compressor cooling for most of the time for many California climates by using waterside economizer as the only cooling source (Duarte, Raftery, Schiavon, & Bauman, 2018). However, we have not found study that comprehensively investigates the cost, energy and comfort impacts of this design approach. In addition, with requirement for building humidity control and extreme weather caused by climate change, eliminating compressors may not be feasible in reality. Future study should focus on whole building HVAC design that includes consideration of cooling approaches for humidity control and periods when a WSE cannot satisfy comfort requirement as a real design must provide thermal comfort at all time, including at design conditions.
- For high thermal mass radiant system design, research has demonstrated the possibility to reduce the size of the central plant cooling and heating equipment if load shifting control strategies are to be implemented. Even though most designers understand the concept of load shifting, there are not enough precedents that give designers confidence

to adopt it. There is also no concrete guidance on how to size the equipment. More field measurement and testing of the comfort and energy performance of load shifting strategies would provide valuable feedback on the validity the measure.

- Increasing radiant tube spacing could reduce installed cost. Due to design uncertainties and possibility of future need, designers and manufacturers often maximize the steady-state radiant slab output to meet peak design load. However, radiant slabs are so massive that they never operate in steady state condition. Dynamic simulations show that it is possible to achieve very similar peak cooling capacity with 9-inch and even 12-inch spacing by opening the valves longer. Furthermore, there are many examples of buildings designed to use 9″ spacing. More research is needed to investigate the impact of tube spacing and dynamic control in high thermal mass radiant systems, and the cost implications of these decisions.
- The self-regulating nature of radiant appears to be one design consideration that allows designers to use large radiant zoning approaches to reduce cost. There is, however, little research to provide guidance on how to take advantage of it at design, and where the limitations are.
- Some designers employ the 2-pipe approach or use a combination of 2-pipe and 4-pipe approach for radiant slab distribution system design. Though this approach could potentially reduce first cost, the impacts on thermal comfort need to be evaluated as it provides less granularity for temperature control. Multiple radiant zones on the same branch or the whole building, if building level changeover system was used, can be in either cooling or heating mode. Though not shown here, the energy simulations exhibited periods where different radiant thermal zones were in different modes at the same time. We also tested the whole building changeover control approach in some runs, and found increased discomfort in some zones. Future research should investigate the hydronic design approach for buildings using realistic internal load schedules and with different envelope profiles. More comprehensive field studies of real radiant buildings could also help designers better understand of how the systems operate and regulate space temperatures and provide guidance on which distribution design approach to select.
- Related to the topic of hydronic design approach is whether heat recovery chiller or four-pipe heat pump is a cost effective central plant option. If whole building changeover system is a viable solution, this implies little heat recovery potential at the central plant, especially if the DOAS system is on a separate cooling/heating source.
- In the baseline radiant design, we oversized some DOAS terminal boxes for supplemental cooling and added reheat coils for some critical zones for supplemental heating. Other commonly used approaches for supplemental cooling/heating include four-pipe chilled beam, radiant panels, fan coils, VRV systems, etc. First cost comparison of those design approaches could provide valuable information to the design industry.
- In the case study design, each radiant zone manifold is connected to the hot water main and chilled-water main via two sets of on/off control valves. There are other radiant slab zone control approaches, including recirculation pumps, modulating valves, etc.

Comparison of cost and control performance of the different approaches could be valuable.

- DOAS system supply air temperature control is another challenging area. The control of these systems must be optimized with the local radiant system for control of ventilation and thermal load, if they are designed to do so, or significant amounts of energy can be wasted. The best control depends on the DOAS configurations, climates, building types. While there are design guides that offer general considerations and principles to control DOAS supply air dry bulb and dew point temperature, it is difficult for designers to translate principles into concrete control sequences that will function in practice. More research is needed to provide designers guidance on which sequence works best for their specific application, and to provide them with detailed control sequences, together with instrumentation requirement, that are ready for implementation.

9 Conclusions

In the cost comparison study, we presented a radiant design that is intended to be representative of the industry good practice and compared the first cost with a VAV system designed for the same building. There are other design approaches commonly used that could result in significantly different installed costs. Coupled with energy simulations, we also explored advanced design and control approaches that have the potential to reduce first cost and improve energy efficiency. Here is a summary of the findings.

9.1 Construction Cost

Major findings from the cost estimates include:

- The radiant HVAC design has a total cost of $38.9/ft^2 compared to $29.9/ft^2 for the VAV design, representing a $9.0/ft^2 premium for the radiant design.
- The higher costs for the radiant system can largely be attributed to higher piping labor costs for piping and radiant equipment, which itself is $9.8/ft^2 higher than that for the VAV design. Based on interviews of contractors and manufacturers, labor hours can vary widely based on a number of factors, such as the mechanical and general contractor's experience with radiant systems, coordination between trades during the construction process, and the specific installation detail for the radiant system.
- There is a $1.2/ft^2 premium in equipment cost for the radiant system, which is mainly associated with the radiant equipment.
- Though there are some sheet metal cost savings for the radiant design due to smaller ducts, the savings due not outweigh the increased piping costs. The total installed cost for sheet metal is $4.5/ft^2 for the radiant design, compared to $7.9/ft^2 for the VAV design.
- As much of the cost premium for the radiant design is associated with piping labor, the premium is more pronounced in the San Francisco Bay area with its high labor rates at about $120/hr. For the estimated national average labor rate of $85/hr, the premium for radiant is $6.8/ft^2, compared to the VAV system.

Though there is wide variability in how radiant systems are designed today, the radiant system in this study was intended to be representative, and one that carefully considers first cost. Nevertheless, the overall cost results are only directly applicable to the two designs that were studied, and care should be taken when applying these results broadly. Climate-specific factors and other design alternatives may have a significant impact on first cost and energy use. Alternative design approaches are discussed that may reduce first cost and/or energy cost.

The high installed cost for the radiant equipment is partly a reflection of the current radiant manufacturers' pricing strategies and the contractors' bidding practices. The radiant market is relatively small and immature in the United States. Radiant system costs are likely to decrease due to economies of scale as the market grows and as uncertainties decrease as the design and construction industry gains more experience.

9.2 Design Strategies to Reduce Cost

For designers, some aspects of the radiant system design have more significant impact on costs and warrant careful attention. These considerations include:

- Consider the use of radiant mats, instead of traditional radiant loops, to reduce cost, through reduced labor. However, radiant mat designs may not be practical or as cost effective for buildings with smaller or oddly-shaped zones.
- Increase radiant tube spacing if possible to reduce material and labor costs, in particular for conventional loop designs. With extended operation, radiant slabs with wider spacing may achieve similar thermal performance as slabs with smaller spacing.
- Strategically design hydronic distribution systems to minimize total pipe length. We compared the installed cost differences for two different approaches: a single set of pipe risers vs. multiple pipe-risers. The former relies on a single set of larger risers and long horizontal distribution runs on each floor, whereas the latter employs multiple sets of smaller risers strategically located to minimize the overall amount of pipe length and overall piping costs by $2.5/ft^2 for the case study building. This strategy may reduce construction cost for any system with distributed piping but is particularly critical for high thermal mass radiant since there are both chilled and hot water pipe distribution systems.
- The study building utilizes a four-pipe system to each radiant zone in the baseline radiant system. Many designers employ a 2-pipe distribution approach or a combination of 2-pipe and 4-pipe approach. If the latter approaches were used, designers may need to consider the potential thermal comfort impacts. More research and design guidance are needed to help designers decide which approach works best for their buildings.
- Utilize large radiant zones to minimize the number of changeover assemblies to reduce the cost of the radiant design but may potentially sacrifice comfort depending on the layout. This is another area that needs more research and guidance.
- The middle floors of a thermally active multi-story building will generally have both the floor and ceiling as active radiant surfaces, whereas the ground floor may only have the ceiling activated if radiant tubing is not installed in the slab-on-grade (or similarly the top

floor may only have the floor activated if radiant tubing is not installed in the roof). The N+1 slab, i.e. radiant in slab-on-grade floor or in-roof layer, adds significant cost. For the case study building, adding radiant in the slab-on-grade would increase the total cost by about $3.2/ft^2.

- For high thermal mass radiant system designs, there may be an opportunity to reduce the capacities of central plant equipment if load shifting control strategies are to be implemented. Though theoretically possible, this does not appear to be common design approach today, likely due to perceived risk of capacity shortfalls. If this is proven to be acceptable in the future, there would be some savings in central plant equipment costs.

9.3 Energy and Operation Cost

Energy models of the two designs were developed in EnergyPlus to evaluate the corresponding energy and comfort performance. Radiant system energy use and energy cost may vary significantly depending on the specific control strategy employed for the radiant and DOAS equipment. For this study, we implemented a set of radiant slab control sequences that modulate slab temperature settings based on zone conditions and allows for load shifting by locking out the radiant slab during a certain periods of the day. Though not necessarily optimized, we evaluated a range of control settings and report those that provided the best energy performance and comparable thermal comfort to the modeled VAV design. The VAV design models typical best practice control strategies.

- The annual simulation results show that the total site HVAC energy use is 16.2% higher for the radiant system (2.9 kBtu/ft^2) than the VAV design (2.5 kBtu/ft^2). The VAV design has significantly lower cooling energy use and benefits from the opportunity for free cooling from the airside economizer with mild San Francisco weather. The radiant design has lower heating energy use but slightly higher fan energy use, compared to the VAV design. DOAS fan are commonly expected to use less energy than VAV fans because of the much lower design airflows but, in fact, the opposite is generally true due to the fact that VAV systems generally operate for the majority of time at lower part loads, where fan laws and differences in sizing result in significantly lower fan power than at design.
- The annual building electricity cost for the radiant design is 8.0% lower than for the VAV design, $1.12/ft^2 for radiant compared to $1.22/ft^2 for VAV design. The energy cost savings for the radiant design are due to reduced demand charges associated with peak demand shifting as the radiant slabs are only active from 6 am to 12 pm. Operating the radiant slab systems during different periods of the day may further reduce the total energy cost. For example, if running the slab from midnight to 10 am, the energy cost could be reduced to 1.06/ft^2, but with decreased comfort performance.
- Control sequences can have significant impacts on the overall HVAC energy performance, and, in fact, some of the control approaches commonly used in the industry appear to be quite energy inefficient. For example, the radiant design site energy use ranged from 2.7 kBtu/ft^2 to 4.4 kBtu/ft^2 for the study building simply by varying the DOAS supply air temperature control approach.

9.4 Design Strategies to Improve Energy Efficiency

The radiant system design evaluated in this study is intended to be representative of typical and good practice today. However, there are opportunities to improve the energy performance of radiant systems. Designers should consider the following:

- In mild climates, such as the Bay Area in California, HVAC designs should take advantage of the benefits of free cooling as much as possible either with airside or waterside economizers. The 2019 version of California Title 24 will newly require economizers for radiant systems above a certain size threshold. Designers should use a holistic approach that in cooperate design features would facilitate the use of a load shifting strategy such that the plant equipment size could be reduced to offset the cost of the waterside economizer.
- High thermal mass radiant system allows great opportunity for the load shifting strategy. The benefits include significant savings of operation cost and installed cost by allowing equipment size to be reduced. With load shifting strategy, it is not as important for the plant to meet the instantaneous cooling or heating load. The plant can operate longer hours if needed to cool or heat the radiant slab to a prescribed temperature setpoint. From building design perspective, one of the key elements to facilitate the adoption of load shifting strategy is to manage solar heat gain, in particularly in west and south perimeter zones to avoid space temperature spike in late afternoon.
- Decoupling the cooling source for the radiant slab and the DOAS system, particularly when humidity is of concern, may provide improved efficiency for the cooling plant serving the radiant system by allowing the chilled water temperature to reset higher.
- The DOAS supply air temperature (SAT) control should include space humidity monitoring logic such that the supply air temperature can be reset higher when latent load is not a concern. This can significantly extend the free cooling period and effectively push the radiant slab to take on more cooling load If the DOAS has a heating coil, large deadband between heating coil setpoint and cooling coil setpoint may significantly reduce energy waste.

10 References

AERMEC. (n.d.). *AERMEC NRP 0800-1800 Technical Manual.* Retrieved 06 22, 2018, from http://aermec.us/products/airtowater/NRP800-1800/Technical%20Manual.pdf

Alvarez, M. A., Hviid, C. A., & Weitzmann, P. (2014). Experimental investigations of heat transfer in thermo active building systems in combination with suspended ceilings. *Proceedings of the 10th Nordic Symposium on Building Physics* (pp. 735-741). Lund University. Retrieved from http://orbit.dtu.dk/files/102497601/Alvarez_Experimental_investigations_of_heat_transfer_in_thermo_active_building_systems_in_combination_with_suspended_ceilings_2014.pdf

ASHRAE. (2013). *ANSI/ASHRAE Standard Thermal Environmental Conditions for Human Occupancy*. Atlanta.

ASHRAE. (2018). *ASHRAE Guideline36: High performance sequences of operation for HVAC systems*. ASHRAE.

California Energy Commission. (2016). *Time Dependent Valuation of Energy for Developing Building Efficiency Standards.* Retrieved from http://www.energy.ca.gov/title24/2016standards/prerulemaking/documents/2014-07-09_workshop/2017_TDV_Documents/

CBE. (2017). *Center for the Built Environment Radiant System Map*. Retrieved September 2017, from https://www.google.com/maps/d/u/0/viewer?mid=1LiVk7CwSdIlwZp5YX_hWasEkv0s&ll=18.243426357150128%2C2.611045900000022&z=2

Cho, P. (2017, October). *Cost-Effective Comfort with Radiant Heating and Cooling*. Retrieved November 2017, from Glumac : https://www.glumac.com/cost-effective-comfort-radiant-heating-cooling/?utm_source=sharpspring&utm_medium=email&sslid=MzO0NDQxNDYxNTYAAA&sseid=MzQ1NzA2sjQzMAIA&jobid=1014efd6-3177-4c1d-a748-e9b436634549

Dominguez, M., Kazanci, O., & Olesen, B. (2017). Experimental and numerical study of the effects of acoustic sound absorbers on the cooling performance of Thermally Active Building Systems. *Building and Environment*, 108-120.

Duarte, C., Raftery, P., Schiavon, S., & Bauman, F. (2018). How high can you go? Determining the highest supply water temperature for high thermal mass radiant cooling sytems in California. *Proceedings of the 4th International Conference on Building Energy & Environment.* Retrieved from https://escholarship.org/uc/item/0s06q03g

Feng, J., Schiavon, S., & Bauman, F. (2013). Cooling load differences between radiant and air systems. *Energy and Buildings*, 310-321.

Feng, J., Schiavon, S., & Bauman, F. (2016). New method for the design of radiant floor cooling systems with solar radiation. *Energy and Buildings*, 9-18.

Hyderman, M., Webb, N., Sreedharan, P., & Blanc, S. (2002). Development and Testing of a Reformulated Regression-Based Electric Chiller Model. *ASHRAE*.

Karmann, C., Bauman, F., Raftery, P., & Schiavon, S. (2018). Effect of acoustical clouds coverage and air movement on radiant chilled ceiling cooling capacity. *Energy and Buildings*. doi: https://doi:10.1016/j.enbuild.2017.10.046

Karmann, C., Bauman, F., Raftery, P., Schiavon, S., Frantz, W., & Roy, K. (2017, March). Cooling capacity and acoustic performance of radiant slab systems with free-hanging acoustical clouds. *Energy and Buildings*, 676-686. doi:https://doi.org/10.1016/j.enbuild.2017.01.002.

Leach, M., Lobato, C., Hirsch, A., Pless, S., & Torcellini, P. (2010). *Technical Support Document: Strategies for 50% Energy Savigns in Large Office Buildings.* Golden, Colorado: National Renewable Energy Laboratory.

Odyjas, A., & Gorka, A. (2013). Simulations of floor cooling system capacity. *Applied Thermal Engineering*, 84-90.

Paliaga, G., Farahmand, F., Raftery, P., & Woolley, J. (2017). *TABS Radiant Cooling Design & Control in North America: Results from Expert Interviews.* TRC. Retrieved from http://escholarship.org/content/qt0w62k5pq/qt0w62k5pq.pdf

Pang, X., Piette, M. A., & Zhou, N. (2017). Characterizing variations in variable air volume system controls. *Energy and Buildings*, 166-175.

Pantelic, J., Schiavon, S., Ning, B., Burdakis, E., Raftery, P., & Bauman, F. (2018). Full scale laboratory experiment on the cooling capacity of a radiant floor system. *Energy and Buildings*(170), 134-144.

PG&E. (2007). *Advanced Variable Air Volume System Design Guide.* Retrieved from http://www.taylor-engineering.com/Websites/taylorengineering/images/guides/EDR_VAV_Guide.pdf

PG&E. (2018). *PG&E Tariffs.* Retrieved from https://www.pge.com/tariffs/assets/pdf/tariffbook/ELEC_SCHEDS_E-19.pdf

Raftery, P., Duarte, C., Schiavon, S., & Bauman, F. (2017). A new control strategy for high thermal mass radiant systems. *Proceedings of Building Simulation Conference 2017.* San Francisco. Retrieved from www.escholarship.org/uc/item/5tz4n92b

Sastry, G., & Rumsey, P. (2014). VAV vs. Radiant: Side-by-Side Comparison. *ASHRAE Journal*, 16-24.

Shank, K., & Mumma, S. (2001). Selecting the supply air conditions for a dedicated outdoor air system working in parallel with distributed sensible cooling terminal equipment. *ASHRAE Transactions, 107.*

Stein, J., & Taylor, S. (2013). VAV reheat versus active chilled beams & DOAS. *ASHRAE Journal*, 18-32.

Taylor, S., Stein, J., Paliaga, G., & Cheng, H. (2012, December). Dual maximum VAV control logic. *ASHRAE Journal*, 16-24.

Thornton, B., Wang, W., Lane, M., Rosenberg, M., & Liu, B. (2009). *Technical Support Document: 50% Energy Savings Design Technolology Packages for Medium Office Buildings.* Richland, Washington: Pacific Northwest National Laboratory.

U.S. Department of Energy. (2016). *EnergyPlus Version 8.7 Documentation - Engineering Reference.*

U.S. Department of Energy. (2018). *Building Performance Database*. Retrieved from https://www.energy.gov/eere/buildings/building-performance-database

Woolley, J., Schiavon, S., Bauman, F., Raftery, P., & Pantelic, J. (2018). Side-by-side laboratory comparison of space heat extraction rates and thermal energy use for radiant and all-air systems. *Submitted to Energy and Buildings.*

11 Appendixes: AERMEC Heat Pump Modeling

EnergyPlus cannot directly model the centralized four-pipe air source heat pump. Instead, chilled water and hot water plant are modeled separately as air-cooled chiller with scroll compressor and district heating plant. Full load and part load power and capacity performance data from the heat pump manufacturer are used to calibrate the chiller and heat pump heating regression models.

11.1 Cooling

The air-cooled chiller, representing the heat pump for cooling, is modeled using the regression-based electric chiller model based on condenser entering temperature in EnergyPlus (U.S. Department of Energy, 2016). Figure 35 compares the calculated KW/tons using the regression models and the data provided by the heat pump manufacturer. Even though the calculated chiller efficiencies are lower than the efficiencies provided by the manufacturer, especially when in the low outside air temperature range, the modeled chiller can accurately capture the performance of the heat pump.

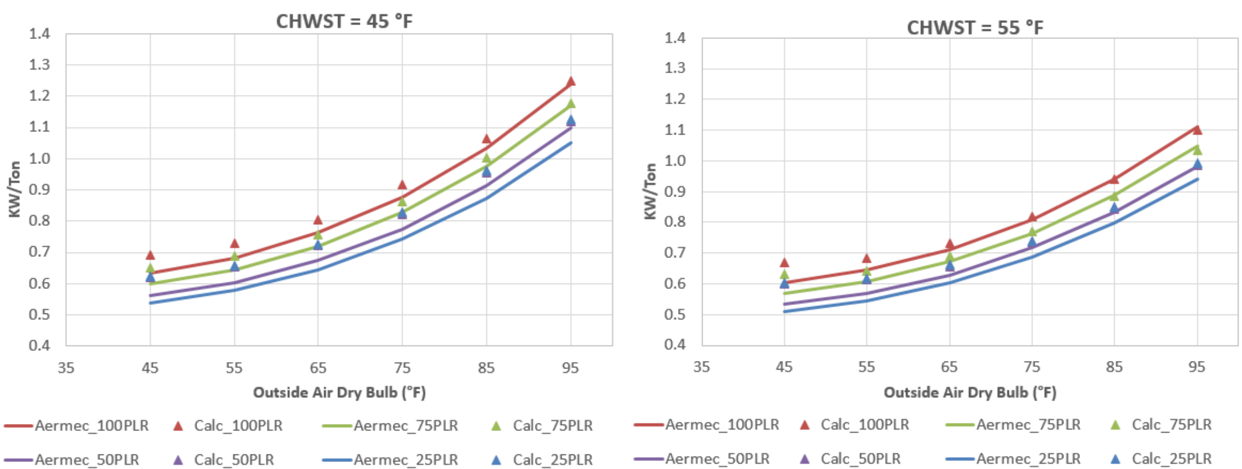


Figure 35:Comparison of KW/Ton from the Manufacturer vs. Model Calculations for Chilled Water Supply Temperature at 45°F and 55°F at 9°F

11.2 Heating

Hourly heating electricity consumptions are calculated using plant heating loads from EnergyPlus and COPs that are calculated from regression models. Five performance curves are developed to determine the heat pump heating performance at off-reference conditions, including:

- Heating capacity curve as a function of outdoor air dry-bulb temperature and hot water supply air temperature
- Energy input to heating output ratio curve as a function of outdoor air dry-bulb temperature and hot water supply air temperature
- Energy input to heating output ratio curve as a function of part load ratio

- Heating capacity correction curve as a function of the difference between return and supply water temperature
- Energy input correction curve as a function of the difference between return and supply water temperature

Figure 36 compares the calculated COPs using the regression curves models and the performance data points provided by the heat pump manufacturer.

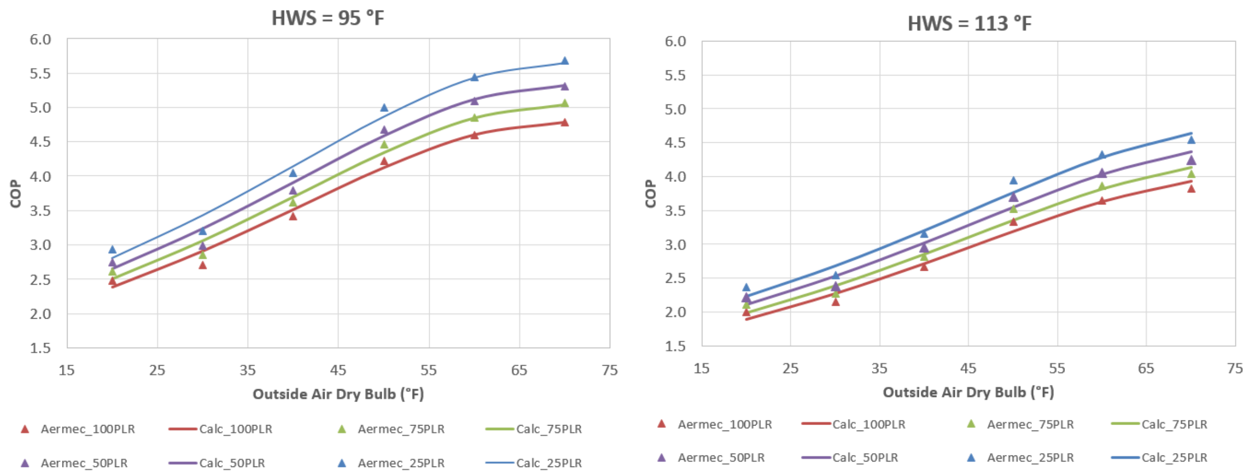


Figure 36: Comparison of COPs from the Manufacturer vs. Model Calculations for Hot Water Supply Temperature at 95°F and 113°F at 9°F Delta T