INFLUENCE OF DESIGN AND OPERATING CONDITIONS ON UNDERFLOOR AIR DISTRIBUTION (UFAD) SYSTEM PERFORMANCE

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ABSTRACT

Various methods are used to design and operate underfloor air distribution (UFAD) systems. There are a number of factors that affect UFAD performance: air distribution strategies in the supply plenum, system configuration and diffuser types, slab insulation, air handler supply temperature setpoints, operation of blinds at peak conditions, impact of occupant control, and the effect of climate differences. Generally, these factors influence performance indicators, such as plenum "thermal decay" (supply air temperature gains) and room air temperature stratification, which in turn affect system energy use and comfort conditions. Previously, the impact of design and operating strategies has been difficult to evaluate analytically due to the lack of simulation tools that accurately model the complex heat transfer processes involved with thermal decay and stratification. The development of EnergyPlus along with the recent addition of the UFAD module has progressed to the point that a systematic comparison of these strategies is now possible.

In this paper, we take a detailed look at the impact of a number of design and operating variations for a medium office building prototype in Sacramento CA. A comparison to a baseline conventional VAV overhead (OH) system is included to understand better the potential energy and comfort differences between the two technologies.

INTRODUCTION

A UFAD system primarily delivers conditioned air from a pressurized plenum through floor-mounted diffusers into the room (zone). Compared to conventional overhead (OH) mixing systems, where the air in the zone is well-mixed, UFAD has several potential advantages such as improved thermal comfort and indoor air quality (IAQ), layout flexibility, reduced life cycle costs and improved energy efficiency in suitable climates (Bauman 2003). However, previously two important features of UFAD systems, room air stratification and thermal decay (Lee 2012) in the underfloor supply plenum, could not be properly represented by most of the energy simulation programs widely used by the industry. Now the situation has improved with the development of a dedicated UFAD module in EnergyPlus. (Bauman et. al. 2007, Webster et al., 2008, DOE, 2010). The authors have used EnergyPlus/UFAD extensively and participated in the design and implementation of refinements to the UFAD module. Lee et al. (2011) describes lessons learned from this experience and guidance for how to model these systems properly.

With these tools, it is now possible to study ways to optimize the performance of the system using design and operating principles that can minimize energy use while maintaining comfort.

In this paper we analyze three design and operating strategies that affect UFAD system performance: plenum configuration and number of diffusers, which affect thermal decay and room air stratification, and real (or perceived) impacts of personal cooling control provided by the adjustable floor diffusers, which can lead to reductions in cooling and airflow energy by raising zone thermostat cooling setpoints.

SIMULATION SOFTWARE

The authors implemented the office-building prototype described below for development, testing, and performance studies using the publicly available EnergyPlus/UFAD simulation program. (DOE 2010) This paper reports results using a development version of EnergyPlus v6.0 that includes UFAD modules. A detailed description of these UFAD capabilities and why EnergyPlus is an ideal program for simulating UFAD systems can be found in a previous paper by Lee et al. (2011). Webster et al. (2008) discusses validation of the UFAD simulation capabilities based on laboratory testing, and details of laboratory testing appears in Bauman et al. (2007).

SIMULATION MODEL

Building model

A three-story prototype office building, located in Sacramento CA, is a rectangular shape (75 m x 51 m (246 ft x 167.3 ft)) with an aspect ratio of 1.5. The floor plate size is 3,716 m^2 (20,000 ft^2) (total floor area is 11,152 m^2 (60,000 ft^2)) and each floor is composed of four perimeter zones 4.5 m (15 ft) wide, an interior, which respectively represent approximately 39% and 61% of the floor area. Table

1 summarizes the building characteristics. Constructions and thermal properties of windows, walls (insulated stucco with steel framing), and roof can be changed based on climate zone to comply with ASHRAE 90.1 (2010). Design day specifications conform to ASHRAE 0.4% summer and 99.6% winter design conditions. The development version contains a preliminary sizing procedure for zone terminal units that attempts to accurately represents the effects of thermal decay on terminal unit entering temperatures. For details, see (Lee et al. 2011).

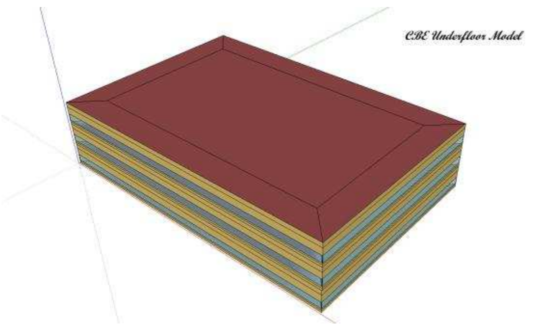

Figure 1. Illustration of building model and zoning

Table 1. Building model characteristics

Feature	Overhead	UFAD
Floor plate size	1858 m^2 (20k ft^2)	Same
Number of floors	3	Same
Floor to floor height	4.9 m (13 ft)	Same
Return plenum height	1 m (3.3 ft)	0.58m (1.9 ft)
Supply plenum height	NA	0.4 m (1.3 ft)
Skin/glazing	90.1 2010	Same
Window/wall ratio	33%	Same
Room setpoints, Occ [Unocc]	23.9/21.1 [29.5/15.5]°C (75/70 [85/60])°F	Same
Internal loads:		
Lights	10.8 W/m^2 (1.0 W/ft^2)	Same
Equipment	8.1 W/m^2 (0.75 W/ft^2)	Same
People	1.86 m^2/Person (201 ft^2/Person)	Same

HVAC systems

From 7:00 until 22:00 the system controls the internal air temperature to a cooling and heating temperature setpoint of 23.9°C (75°F) and 21.1°C (70°F), respectively. Internal load schedule maximums are 90-95% between hours of 9:00 to 18:00. The system does not operate during the night. Infiltration was assumed equal to 0.33E-03 $m^3/(s\ m^2)$ (0.11 cfm/ft^2) (flow per exterior surface area), when fans are off and 25% of that when fans operate (i.e., assumes a pressurized building when operating).

The minimum outdoor airflow rate was set to be 0.76 E-03 $m^3/(s\ m^2)$ (0.15 cfm/ft^2) flow per gross floor area.

Distribution of supply air to the zones occurs through swirl diffusers in interior zones and linear bar grille diffusers in the perimeter zones. Variable speed fan coil units (VSFCU) provide air to perimeter zones during cooling mode when the fan is on (and heating coil is off); during heating mode, the fan and the heating coil are on. Due to pressure in the plenum, airflow through the VSFCU (based on field measurements by the authors) occurs when the fan is off and the zone temperature is in the deadband.

The building, for both systems, is served by a single variable speed central station air-handling unit (AHU) including an airside economizer, a chilled water cooling coil, and a relief fan. A simulated static pressure reset strategy controls the AHU fan. In both UFAD and OH systems, supply air temperature (SAT) is reset as shown in Table 2 based on an outdoor air temperature (OAT) range of 18.3 to 21.1°C (65-70°F).[1] The central plant consists of a central scroll chiller with variable speed pumps and a two-speed cooling tower. A gas fired forced draft hot water boiler provides hot water to all heating coils. Table 2 shows further details of system and plant inputs.

Table 2. Summary of HVAC system configurations

HVAC	OH	UFAD
AHU supply air temperature (for OAT range)	15.6 to 12.8°C (60 to 55°F)	18.3 to 15.6°C (65 to 60°F)
AHU fan design static pressure	See Table 3	See Table 3
AHU fan efficiency	75%	75%
AHU part load shutoff[2]	125 Pa (0.5 iwc)	Same
Minimum outside air rate	7.62 E-04 $m^3/s/m^2$ (0.15 cfm/ft^2)	Same
Airside economizer; differential dry bulb	Yes	Yes
System cycles at night	No	No
Zone minimum airflow	7.62 E-04 $m^3/s/m^2$ (0.15 cfm/ft^2)	Same
Interior zone reheat	Yes	No

[1] This range was in error, should have been wider; it will be corrected in future studies.

[2] Represents fan static pressure operating curve extrapolated shutoff pressure. (iwc = inches water column, Pa = Pascals)

VSFCU design static pressure	NA	125 Pa (0.5 iwc)
VSFCU design efficiency	NA	15%
Plant		
Chiller design COP	5.0	Same
Cooling tower	2-speed	Same
Boiler design efficiency	80%	Same

Table 3. AHU design fan static pressures (FSP)

System/UFAD Plenum configuration	AHU Design FSP
Overhead system	1075 Pa (4.3 iwc)
Common plenum	700 Pa (2.8 iwc)
Series plenum	700 Pa (2.8 iwc)
Ducted perimeter	1075 Pa (4.3 iwc)

MODELING OVERVIEW

In the following, we describe the modeling of each of the three factors that are the subject of this study.

Plenum configuration

Plenum configuration simulation options reflect variations in methods used to distribute air in the supply plenum, which, because of thermal decay, has an impact on the supply air temperature to the zones and thus its airflow requirements. The intent of these idealized models is to capture the impact of different ways to configure plenum distribution; real systems will seldom conform perfectly to any of these models.

One of the goals of improved plenum design is to try to deliver the coolest air possible to perimeter zones, since the loads are greater there. In the most common plenum design used in today's practice, a large open plenum serves both interior and perimeter zones of the conditioned space. Due to the location of HVAC shafts in the core, air usually enters the plenum in the interior, although various forms of ductwork can distribute air across the floorplate to or toward the perimeter. Generally, elimination of thermal decay is not possible, but its impact is manageable. Likewise, distribution methods cannot guarantee exactly how the temperatures are distributed.

Figure 2 shows illustrations of three idealized cases for plenum distribution. These are plan views that represent slices of the supply plenum. For example, in Figure 2a, the injection point for AHU air is on the right and flows to the perimeter zone on the left as indicated by the arrows. EnergyPlus/UFAD models these plenums as fully mixed zones, which are idealized representations of actual distributions.

Figure 2a depicts an open "series" plenum distribution method, in which cool air from the AHU is delivered first to the interior portion of the open plenum. In this idealized model, the plenum airflow first gains heat (raising the temperature) from the interior zone before entering the perimeter portion of the plenum, where it gains additional heat. This plenum configuration results in the perimeter zone having higher thermal decay (i.e., difference between plenum and AHU SAT) than the interior zone.

For the "common" open plenum depicted in Figure 2b the entire plenum is mixed so both interior and perimeter zones receive the same temperature air derived from the combined heat gain from the two zones.

Figure 2c shows a third idealized approach that has been approximated in practice, where air is ducted directly from the AHU to the perimeter zone diffusers in parallel to that entering the interior zone. This of course eliminates heat gain to the air entering the perimeter, thereby reducing airflow rates, but at an extra cost for ductwork and increased reheat.

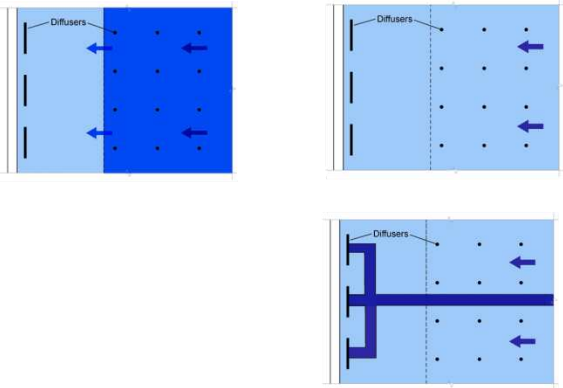

Figure 2. Supply plenum configurations; Clockwise, (a) Series, (b) common, and (c) ducted perimeter

Increased stratification

Room air stratification is a key factor in reducing energy use of UFAD systems because it determines how much of the room energy is distributed to the occupied zone (per ASHRAE Standard 55 (ASHRAE 2010), the region between 0.1 m (4 inches) and 1.7 m (67 inches) from the floor; the foot-head region). It also allows the thermostat setpoint to be increased to account for the lower temperatures in the occupied zone.

Stratification is produced by a complex interaction between thermal plumes from heat loads in the space and the turbulent mixing caused by the floor diffusers. If mixing is too high, there will be little or no stratification. EnergyPlus/UFAD contains semi-empirical algorithms based on laboratory testing of commonly used diffusers provided by various vendors (Bauman et. al. 2007). Internal studies (field and simulation) by the authors have shown that many of the diffusers, and especially linear bar grilles, produce little stratification. The lack of standardized design methods exacerbates this situation (Bauman et al. 2010). (An online version of a new tool that will help mitigate this situation is available at http://www.cbe.berkeley.edu/research/ufad_designtool-download.htm.)

Generally, stratification performance of the types of diffusers[3] simulated in this study improves (larger stratification) by increasing their number so that airflow at peak conditions is relatively low thereby reducing the throw height. To test the sensitivity of this we doubled the normal design (based on manufacturers rated airflows) number of diffusers.

Personal control

Previous studies (Bauman et al. 1998) have shown that occupants tolerate wider variations in indoor environmental conditions if they perceive they have control over them, or actually have control such as with workstation personal control systems. This potential benefit is realized in an UFAD system, by allowing the occupant to control the nearby floor diffuser to provide more or less cooling. We modeled this option by assuming an increase of cooling setpoints from 24 to 25°C (75 to 77°F) which represents the approximate cooling effect of increasing air velocity from 0.10 m/s to ~0.25 m/s (20 – 50 fpm). (See ASHARE Standard 55-2010 (ASHRAE 2010))

RESULTS

Typically, interior zones of UFAD systems have no terminal heating equipment. It is common practice in California not to use a central heating coil in the AHU. The purpose of the heating coil is to maintain thermal comfort in interior zones and is required for cold climates for both UFAD and OH systems. Lee et al. (2011) discusses some of the ramifications of this choice; also shown is the comfort impact due to various AHU supply air temperatures.

Energy performance

Table 4 summarizes the various cases simulated. All UFAD cases used minimum ventilation rates to allow for apples-to-apples comparisons to the "best practices" OH system. Furthermore, we assume UFAD systems operate better at low minimums and avoid problems of dumping and poor heating performance that sometimes occur with OH systems. The best practices OH system departs from standard 90.1-2010 by using zone minimum ventilation rates rather than the 20% specified in Appendix G. Using 20% results in minimum zone rates of ~0.00127 $m^3/s/m^2$ (0.25 cfm/ft^2).

[3] Certain types of VAV diffuser designs do not exhibit this behavior because they maintain constant throw height throughout their operating range.

Table 4. Simulated strategies summary

Case	Label	Description
1	OH - MinOSA (Base)	VAV box minimums set to "best practices" consistent with OSA requirements shown in Table 2
2	OH – 20% min	VAV box minimums set to 20%, as per ASHRAE 90.1(2010)
3	OH – MinOSA, no core htg	Case 1 but with no reheat for interior boxes; similar to UFAD
4	UF - common plenum	UFAD with common plenum
5	UF – series plenum	UFAD with series plenum
6	UF- ducted perimeter	UFAD with ducting directly to perimeter diffusers (no thermal decay)
7	UF – Increased stratification + common plenum	UFAD with increased stratification by doubling number of perimeter diffusers
8	UF – occupant control + common plenum	UFAD with cooling setpoints increased to 25°C (77°F)
9	UF – combo	Cases 6, 7, 8 combined

Figure 3 shows results from a comparison of energy performance between the strategies described above as well as an additional "combo" case, which shows the combined effects of increased diffusers, ducted perimeter plenum, and personal control.

These results are preliminary to a larger study that will incorporate additional strategies as well as five US climate zones. Included in this figure is the percentage difference (shown as percentage change) between each of the cases and the baseline OH simulation. Negative numbers indicate energy reductions (i.e., savings).

It is clear from Figure 3 that most of the savings results from savings in heating and only for the cases on the far right of the figure (cases 8 and 9 in Table 4) are there savings in both electric loads and heating. The decrease in heating energy for UFAD is about 45% overall. The overall HVAC savings shown in Figure 3 reflects the net effect of these trends. The heating trends tend to mask the impact on the electric loads that support cooling; for example, electric energy use increases from ~7-12% for the three plenum configurations. In the ducted perimeter case, the electric energy penalty is least (~7%) but heating energy is increased by 6 percentage points (due to less reheat) so the impact on overall HVAC energy is about the same for all plenum cases. Although these plenum cases are idealized versions of real systems, the results indicate that designers should strive to avoid designs that tend to produce the series case. Cases 8 and 9 show positive savings for both gas and electric, which yield decreases of 17% and 22%, respectively. It is clear that combining strategies delivers the best energy performance.

The large heating differences are somewhat explained by the results shown in Figure 4, showing a breakdown of heating components. We know that UFAD systems reduce reheat due to thermal decay in the supply plenum. In addition, the OH reheat shown in Figure 4 includes about 17% of interior zone reheat, which helps to account somewhat for the large disparity. On the other hand, it is not completely clear why the actual zone heating loads are so different. We know that there is some effect of the cool supply plenum causing extra zone heating load, but we do not believe it addresses the entire magnitude of the difference. This is the subject of ongoing research.

Thermal comfort

In this paper, we provide thermal comfort results in two ways: (1) a comparison between OH and UFAD of zone temperatures setpoints not met (TNM), and (2) some examples of predicted percentage of dissatisfied (PPD), based on operative temperature, for selected zones.

Table 5 shows results of temperature setpoints not met comparing OH and UFAD. For perimeter and interior cooling, UFAD has a higher percentage of hours not met but (except for ducted perimeter) still well below standards of ~300 hours per year (~10%) specified in ASHRAE 90.1 2010.

Differences between all the cases for cooling are largely due to sizing issues. For example, terminal unit sizes for the common plenum case were relatively smaller than for the series case, resulting in more unmet temperatures. South zones are a particular problem and require cooling design days in the fall. Complicating sizing procedures for UFAD is the lack of knowledge of thermal decay during sizing runs; we are currently developing alternative methods to resolve this problem.

Table 6 shows example PPD results for the interior (core) zone and West zone. For the interior, the results are not markedly different between OH and UFAD, only slightly higher for UFAD. This may not be true in colder climates, but in that case a central heating coil would help mitigate comfort problems in the interior. Results for the West zone indicate that OH systems have greater discomfort in winter. This is a counterintuitive result, but upon further study, we found that the mean radiant temperature is lower for the OH system in the West perimeter zone. Detailed data (e.g., surface temperatures) was not available to investigate further; this will be the subject of additional research.

Although these results are interesting, simulation only captures the effects of surface and air temperatures, not other real world effects such as drafts. However, they are somewhat consistent with our experience from field studies where interior zones are often too cool. However, in real systems, cool drafts can occur under conditions when cool air enters a supply plenum that behaves like a series plenum (e.g., when economizer is at minimum and outside air is lower than AHU setpoint), or if the SAT setpoint is lowered to ensure perimeter zones have adequate capacity.[4]

CONCLUSION

This study indicates that optimized design and operating strategies can deliver significant benefits relative to conventional OH systems. For example, increased stratification indicates 11% savings, an occupant control strategy yields 17% savings, and the combination case shows savings of 22%. The results also show that, at least for the Sacramento climate, plenum configuration options have little impact relative to one another. Their overall impact on HVAC energy use is about 8% relative to a “best practices” case for overhead systems. However, the common plenum assumption yields slightly better performance than the other configurations. Comparing to a normal practice overhead system conforming to ASHRAE 90.1-2010, savings for all UFAD cases are 8% (percentage points) greater than the best practices comparison.

Overall, the results suggest that simulated thermal comfort does not appear to be significantly different between the two technologies for any of the various options studied. Unexpectedly, in winter (for the Sacramento climate) indications are that overhead systems are slightly less comfortable in some areas (e.g., West perimeter zone) due to lower mean radiant temperatures. However, these results may not accurately reflect real world conditions because it is difficult to model effects such as drafts.

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ACKNOWLEDGMENT

The present work was supported by the Center for the Built Environment, University of California, Berkeley and the California Energy Commission - PIER project 500-08-044.

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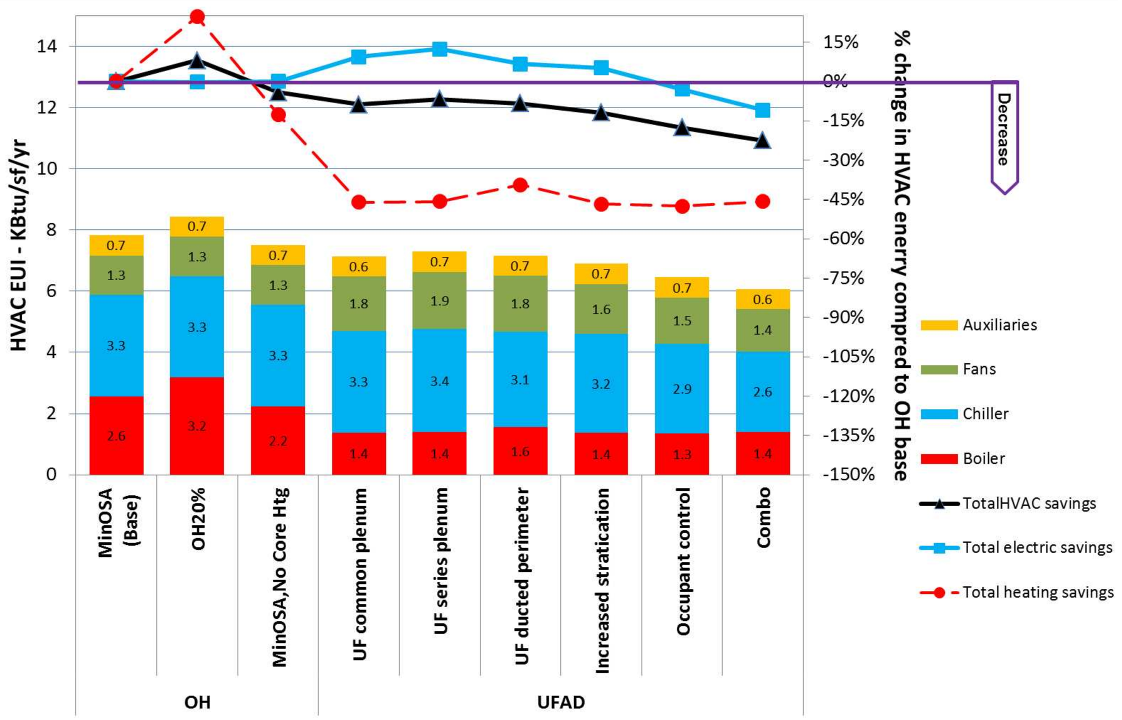

Figure 3: Energy performance comparison, Site HVAC EUI, Sacramento

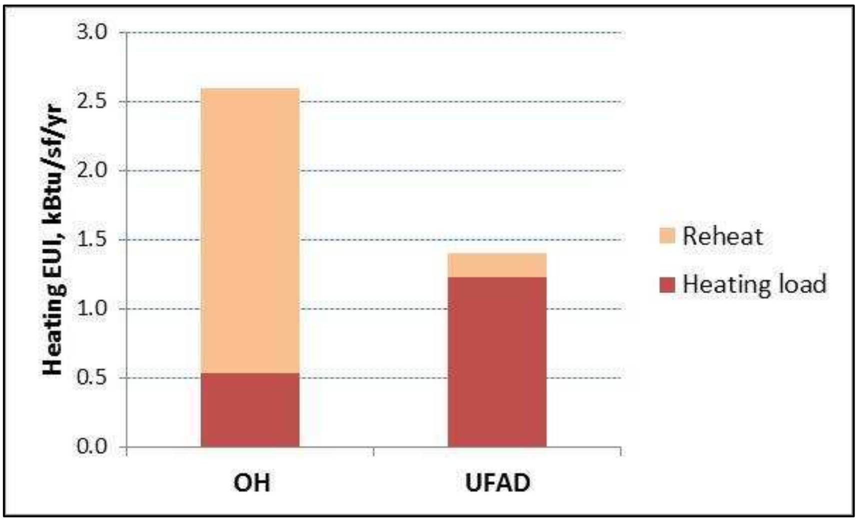

Figure 4: Example heating breakdown

Table 5. Thermal comfort - zone temperature setpoints not met, Sacramento

	Percentage of occupied hours with cooling or heating setpoint not met								
	Overhead			UFAD					
	OH MinOSA (base)	*OH 20% minimum*	*OH No Core Heating*	*Series plenum*	*Common plenum*	*Ducted perimeter*	*Increased stratification*	*Occupant Control*	*Combo*
Perimeter cooling	0.0	0.0	0.0	3.7	7.2	10.8	3.3	7.2	6.7
Interior cooling	0.0	0.0	0.0	0.0	0.1	0.2	0.1	0.0	0.1
Perimeter heating	1.9	1.9	2.0	1.5	1.4	1.9	1.4	1.4	1.8
Interior heating	1.6	0.5	8.7	9.6	7.1	5.7	7.4	6.3	5.1

Table 6. Thermal comfort – PPD for selected zones, Sacramento

UFAD Common	Monthly average Fanger PPD – Zone :										MF	Core
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Too cold	19.4	15.3	10.8	10.0	10.2	10.1	9.4	9.5	9.9	9.5	11.9	19.1
Too hot	0.2	0.3	0.4	0.5	0.4	0.4	0.5	0.5	0.4	0.5	0.4	0.2

OH MinOSA	Monthly average Fanger PPD – Zone :										MF	Core
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Too cold	16.6	12.9	9.8	8.5	8.2	7.8	7.0	7.0	7.3	7.7	10.8	15.6
Too hot	0.3	0.4	0.5	0.6	0.6	0.7	0.8	0.8	0.7	0.7	0.5	0.3

UFAD Common	Monthly average Fanger PPD – Zone :										MF	West
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Too cold	17.5	15.6	10.9	9.1	7.9	7.0	6.2	6.6	7.7	9.0	13.0	17.4
Too hot	0.3	0.5	0.8	1.2	1.4	1.6	1.9	1.9	1.5	1.1	0.6	0.3

OH MinOSA	Monthly average Fanger PPD – Zone :										MF	West
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Too cold	27.7	22.6	14.4	10.0	7.5	5.9	5.2	5.5	6.8	9.7	17.5	26.5
Too hot	0.2	0.3	0.6	1.0	1.4	1.7	1.9	1.9	1.5	1.0	0.4	0.2