Title

Development of guidelines for Modeling Underfloor Air Distribution (UFAD) Systems in EnergyPlus, eQUEST, and EnergyPro for use in California non-residential Building Energy Efficiency Standards

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Development of guidelines for Modeling Underfloor Air Distribution (UFAD) Systems in EnergyPlus, eQUEST, and EnergyPro for use in California non-residential Building Energy Efficiency Standards

Final Deliverable

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Purpose

The overall goal of this project is to develop ways that UFAD systems can be successfully simulated in California climates with DOE-2 programs that have no explicit UFAD models. These new methods should improve upon the current “work around” solutions offered in Title-24 alternative calculation methods (ACM) manuals in 2005 and 2008 standards [ACM 2005, 2008] [CEC 2008] and other ad hoc methods used by designers.

Background:

On February 22-23, 2006, at a public CEC Standards Workshop in Sacramento, several measure proposals were presented for new mechanical systems to be included in the 2008 Nonresidential Standards as Compliance Options. In particular, the Measure Information Template on Underfloor Air Distribution outlined proposed new specific language for inclusion in the ACM manual to facilitate modeling of UFAD systems with the current tools in use in California [EnergySoft 2006]. At this meeting, CBE presented comments and recommendations regarding these UFAD proposals, based on recent research findings from CBE’s past and ongoing PIER-sponsored projects: Energy Performance of UFAD Systems [Bauman, Webster et al. 2007]; Advanced Design and Commissioning Tools for Energy Efficient Building Technologies [Bauman, Webster et. al. 2011 [in press]; and currently, Advanced Integrated Systems Technology Development (500-08-044) to develop energy modeling and design tools for UFAD systems.

This project was formulated as a response to a request from CEC to help develop language and modeling guidelines for the ACM manual for the 2008 Standards. As originally proposed, this project was to include (1) development of refined methods for simulating UFAD in DOE-2 programs using EnergyPlus UFAD modeling tools that CBE developed; (2) refine the EnergyPlus perimeter zone models based on a new series of tests for perimeter zones in a full scale testing laboratory; and (3) compare EnergyPlus results to an anticipated UFAD version of eQuest. Since the 2008 standards are now published and the UFAD version of eQuest has been delayed, the objectives for this project have changed with more focus on development of methods for ways to simulate UFAD with existing DOE-2 tools. It is now anticipated that results of this work will be incorporated in the 2013 standards, and will be appropriate for use until EnergyPlus is approved for compliance purposes.

Deliverables

The following deliverables were embodied in the original scope of work but the nature of the work conducted under them has changed due to developments outside the control of this project. The results from this project are discussed under three tasks corresponding to these deliverables in the sections below.

1. Perimeter zone empirical correlations suitable for use in UFAD versions of EnergyPlus and possibly eQUEST.
2. Reports summarizing results of verification studies
3. Modeling guidelines and associated language changes to the ACM manual to support appropriate modeling of UFAD systems using DOE-2 programs.

These deliverables are discussed under three corresponding tasks below.

Task 1 - Perimeter zone empirical correlations

Objective

The key outcome of this task was to develop improved models, namely Gamma-Phi correlations that are critical to modeling stratification in perimeter zones of UFAD systems.

Task 1 results

Work in this task covered full-scale testing at Walnut Manufacturing’s (Walnut) UFAD laboratory in Kansas City, KN to update the perimeter zone models in EnergyPlus and the CBE UFAD cooling load design tool. This laboratory was designed and constructed according to CBE specifications to support this and other UFAD studies. Figure 1.1 shows a schematic of the laboratory configuration; a description of the test facility is included in Appendix A.

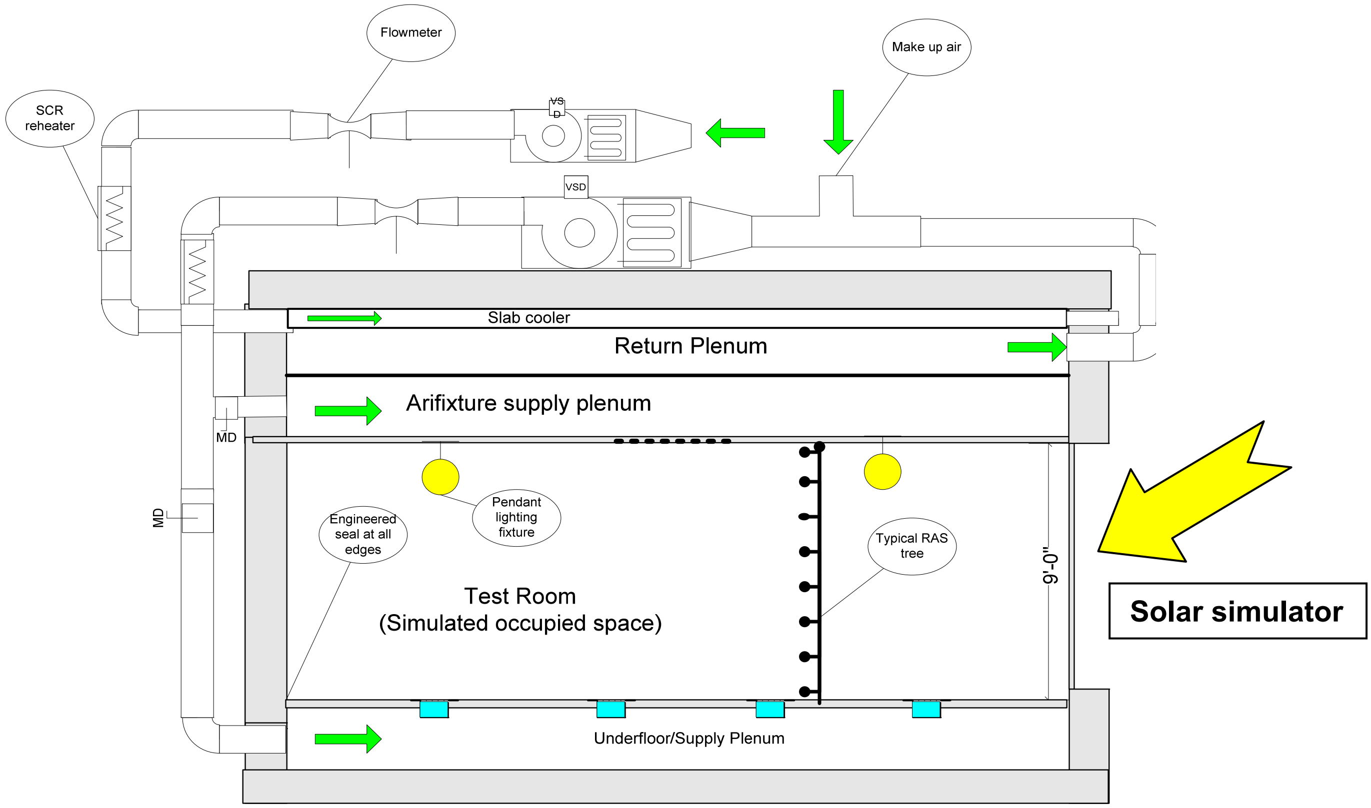


Figure 1.1: Walnut laboratory configuration

UFAD perimeter zone model

Stratification in UFAD systems is simulated as two mixing layers as shown in Figure 1.2. The temperatures in each layer result from a heat balance on each layer. The effect of stratification is simulated by dividing the total room heat gain between the two layers as specified by the UFAD semi-empirical models described below.

These models are semi-empirical correlations based on theoretical formulations derived from the theory of thermal plumes described in Chapter III of Bauman, Webster et al. These models, called Gamma-Phi correlations, are shown in Figure 1.3 for both sets of tests. Phi is a dimensionless parameter that is used to describe the vertical distribution of temperature (stratification) in the room. In the case of the two-zone model of stratification used in EnergyPlus, Phi represents the ratio of heat gain in the lower layer (occupied zone) to the total heat gain in the room. Gamma, on the other hand, is a dimensionless parameter that embodies the effects of diffuser characteristics, load, and airflow caused by the combined interactions of buoyant thermal plumes and turbulent “fountains” (diffusers). Previous stratification models for perimeter zones were developed from a limited number of tests conducted at York International’s laboratory in York PA.

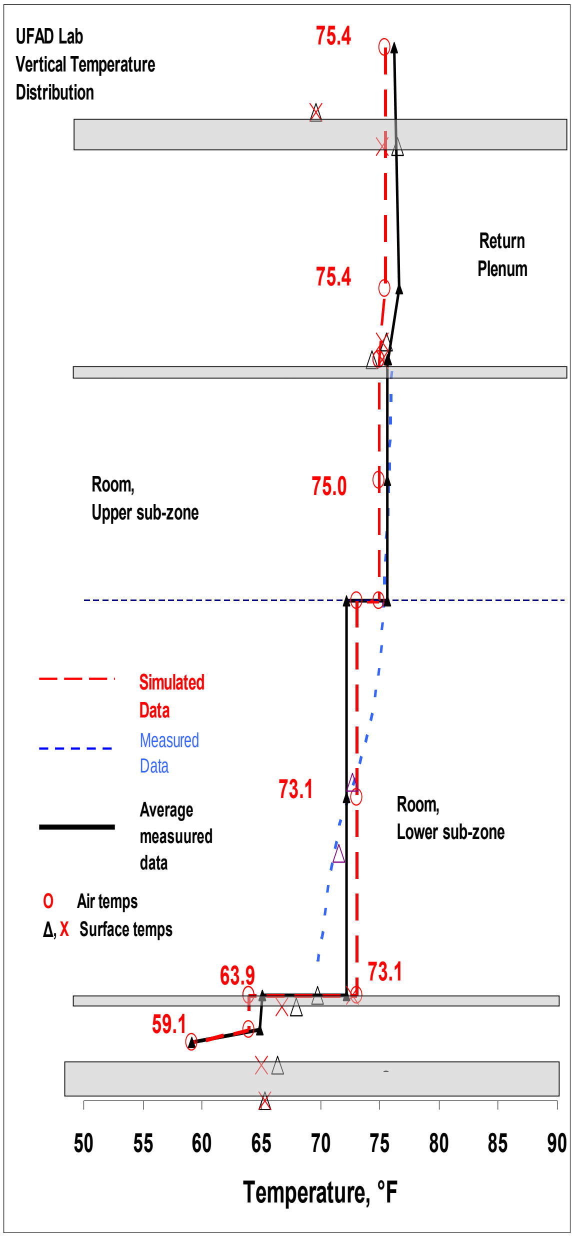


Figure 1.2: Estimated stratification using EnergyPlus showing mixed sub-zones

Walnut lab testing

Stratification measurements were made for two types of linear bar grill diffusers (Airfixture/Walnut and Titus) and two “VAV directional” diffusers set to their fully open positions. These diffusers were located at the perimeter of the test room next to the window with the solar simulator on. Tests were made at several different internal loads, air flow rates, supply air temperature and “solar” intensity selected in such a way as to keep the occupied zone temperatures in the comfort range and produce a large range of Gamma. The intensity of the light incident on the window from the solar simulator was adjusted by placing screens in front of the lights. The air temperature of the “outside” of the window was kept at about 90 °F.

Testing results

Figure 1.3 shows the Gamma-Phi (OZ) relationship for tests with the blinds at the window full open. Results from comparable tests from the York lab are also shown. The Gamma-Phi data for the two linear bar grill diffusers tested at the Walnut lab showed no significant difference and are plotted together. At high Gamma these have a Phi greater than 1.0 indicating that the air flowing out of the diffuser hits the ceiling before mixing with the room air lowering the return temperature so the occupied zone temperature is greater than the return temperature. This was not seen in the “VAV directional” tests, although Gamma was never as large as in some of the linear bar grill tests. The linear bar grill tested at the York lab appears to have a lower Phi, but the scatter is large. Uncertainty in the air flow discharge angle and differences in diffuser configurations may be factors in this difference; standard ways to measure these parameters by manufacturers is needed. Two trends seem clear from these results; for Gamma values less than ~10, Phi levels out at ~0.7; for Gamma in the range of 15-25, Phi tends to 1.0. The fact that the linear bar grills tested at York exhibit lower Phi in the 15-20 range of Gamma suggests that the design and configuration differences for these diffusers may have a significant impact on performance at high Gamma conditions; i.e., high load

For the “VAV directional” diffuser (a square diffuser with vanes that cause air to discharge in four different directions), although the points indicate a trend similar to the bar grilles in lower ranges of Gamma, in practice this dynamic range does not occur. These diffusers are modulating so in operation they operate at nearly constant Gamma, very close to the original results of Phi ~0.88 as shown.

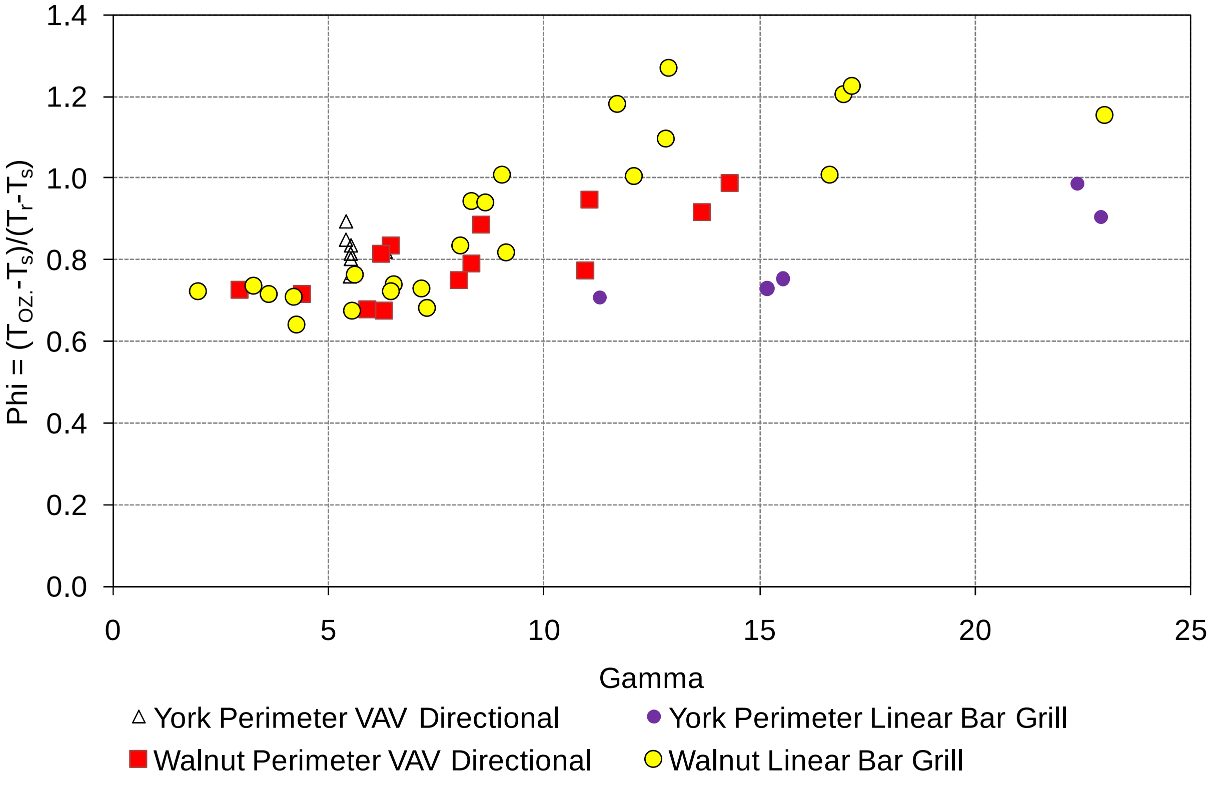


Figure 1.3: Gamma-Phi (OZ) (average of the occupied zone) plot of perimeter diffusers from the Walnut and York laboratory testing.

Other improvements in EnergyPlus

All modeling previous to this study was done with EnergyPlus V3.1. Since then many refinements have been made culminating in a V6.0 development version that contains the following UFAD improvements:

1. In the release version of EnergyPlus, the UFAD terminal unit is turned off during the deadband and thus there is no airflow entering the space. In practice this is not true because plenum pressure will cause a leakage airflow through the FCU. We fixed this bug to allow a minimum airflow through the unit when the FCU is off in the deadband. This is not entirely accurate, however, since the plenum pressure varies which will cause variable flow through the FCU.

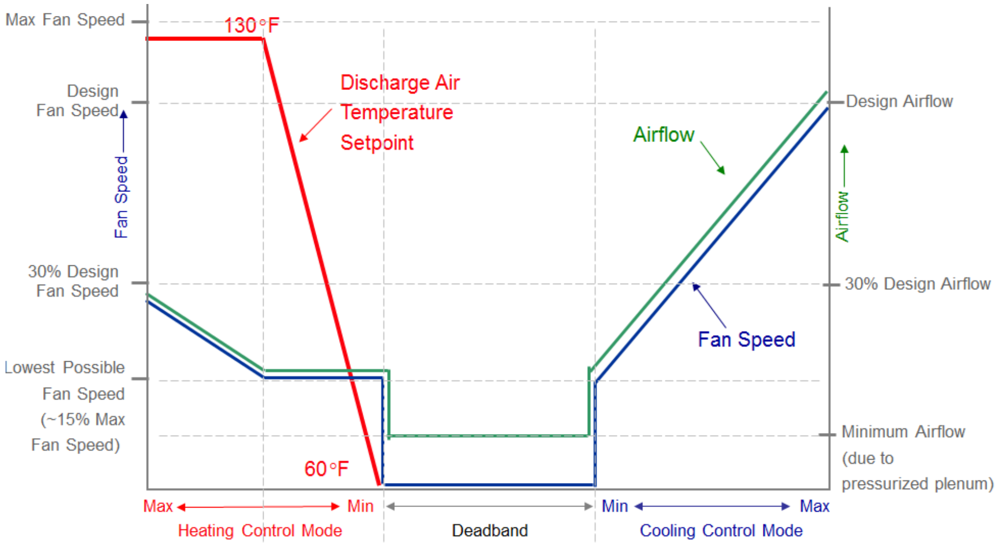


Figure 1.4: UFAD Terminal Unit Control

2. We implemented a “dual minimum” logic for UFAD terminal unit as shown in Figure 1.4. During the deadband when the small variable speed fans are switched off, the minimum airflow, which is 6% of design airflow, is supplied into the space. When the FCU is turned on in either heating or cooling mode, a minimum airflow is supplied according to minimum airflow settings, currently hardwired in the code at 20% consistent with T-24-2008 requirements.
3. In the release version of EnergyPlus, the terminal unit discharge air temperature can reach unrealistically high values. We implemented a change to provide a maximum discharge temperature limit, currently set to 130°F as shown in Figure 1.4.
4. Since the default convective coefficients in the release version of EnergyPlus are not representative of conditions in the underfloor supply plenum, we implemented a plenum convective coefficient model that correlates the coefficient with the airflow coming through the plenum based on Bauman and Webster’s work [Bauman, Webster et.al. 2006]. The correlation was developed from empirical data.
5. We identified a number of significant issues with the way that EnergyPlus “autosizes” system components. In many cases the EnergyPlus algorithms, especially those related to UFAD, do not work correctly so we developed work-arounds to size certain components (e.g., number of diffusers) that replace the default routines used in EnergyPlus auto-sizing. To overcome this limitation the following procedure was implemented in the EnergyPlus interface. First a design day simulation is run where the

peak system demand is determined. The peak demand is selected and the sizing factor of 1.2 is applied prior to the annual run.

6. In the release version of EnergyPlus there was no equation type available that can represent realistic part load models for boilers. Therefore, we implemented a piecewise solution in the code to better match the real boiler curve illustrated in Figure 1.5.

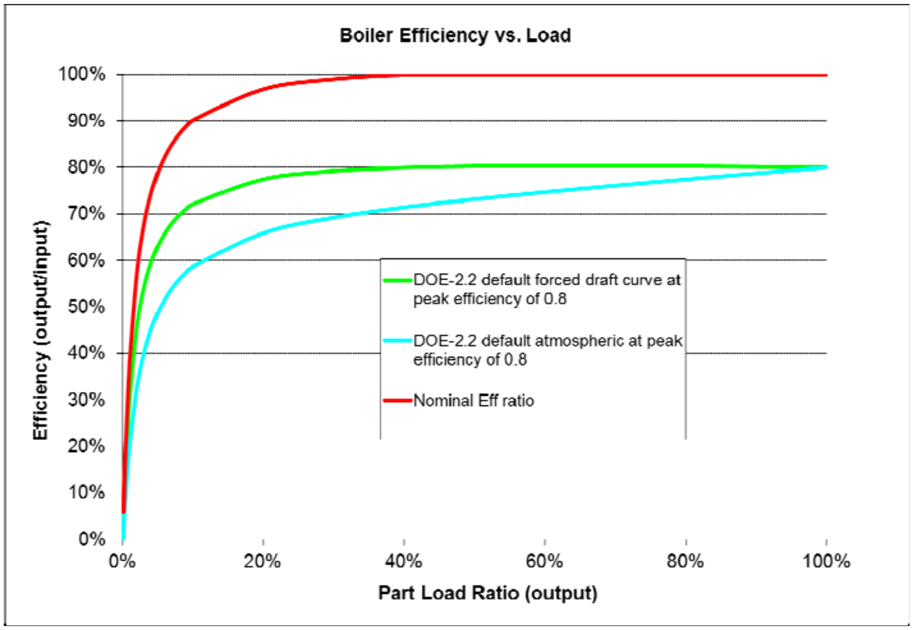


Figure 1.5: Boiler curves implemented by a piece wise soltuion

Task 2 – Verification studies

Objective

To determine the effects of differences between DOE-2 and EnergyPlus modeling for OH systems and the impact of other design, modeling, and operational parameters that might affect the accuracy of prediction.

Task 2 results

eQuest vs EnergyPlus comparison

To understand the fundamental difference between EnergyPlus and eQUEST in preparation for developing UFAD work-arounds we compared overhead versions of the two programs using identical models. To exclude confounding factors, we matched the construction, window, heating and cooling setpoints, schedules for HVAC operation, occupancy, lighting and equipment, radiative/convective splits, and climate (Sacramento), as closely as possible. The model is a three story building with 20,000 ft^2 floor plates and a VAV reheat system with single AHU with chilled water cooling and hot water heating. Version 3.63/3.64 of eQuest and V3.1 of EnergyPlus were used.

As the first step, a detailed comparison study was conducted at the thermal zone level, aiming to minimize the impact on the simulation of the air handling unit and plant operation.

Table 2.1 summarizes the eQuest and EnergyPlus parameters used for generating charts in this section.

Table 2.1: Parameters used for charts

Items		Thermal zone parameter
Individual thermal Zone	eQquest Parameter	1. *Current hour heat extraction rate, <QNOW> in FORTRAN* 2. *Calculated heat extraction rate*
	EPlus Parameter	*Sensible Cooling/heating Rate*
Total thermal zone	eQquest Parameter	Summation of all 15 zone's *Current hour heat extraction rate* or *calculated heat extraction rate*
	EPlus Parameter	Summation of all 15 zone's *sensible cooling/heating rate*

Current hour heat extraction rate: This is defined to be the rate at which sensible heat is removed from the conditioned space in order to maintain setpoint in the conditioned zone. This value is positive if the zone is in cooling mode, and negative if in heating mode. This represents the zone cooling/heating load demand. However, this parameter gives an unreasonable value when the system is switching between heating and cooling setpoint or "not meaningful when room temperature is outside the throttling range" of setpoint (DOE2 manual). We still report it to track differences and because it appears this value is passed to upstream HVAC system components for load calculation.

Calculated heat extraction rate: Due to the unreliability of the detailed heat extraction rate parameter, we calculate the actual hourly sensible heat extraction rate using the equation below:

$$(1) \qquad Calculated\ cooling(heating)rate = 1.08CFM \times (Room\ temp - discharge\ temp) \\ = \begin{cases} if\ room\ temp > dischage\ temp, & cooling\ rate \\ if\ room\ temp < dischage\ temp, & heating\ rate \end{cases}$$

Where,

$$Discharge\ air\ temp = \begin{cases} 55 & if\ reheat\ coil\ output = 0 \\ \dfrac{reheatcoil\ output}{1.08CFM} + 55 & if\ reheat\ coil\ output \neq 0 \end{cases}$$

Individual zone results

Middle floor interior (Core) and West perimeter zones are used to exemplify behavior differences at this level. The results are shown in Figure 2.1. Room setpoints are 21°C for heating and 24°C cooling.

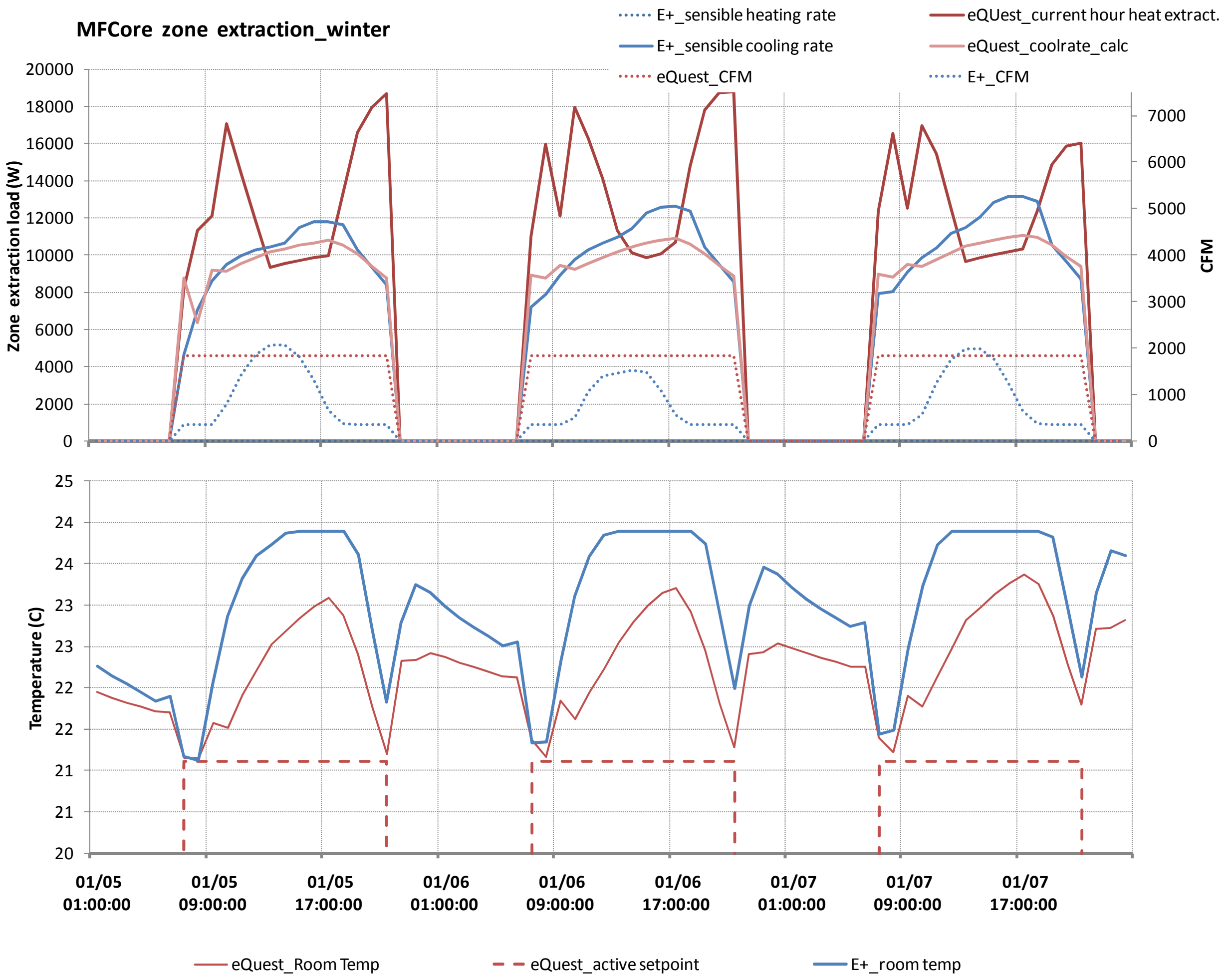


Figure 2.1(a): Comparison results of middle floor *Core* in *winter* (Sacramento)

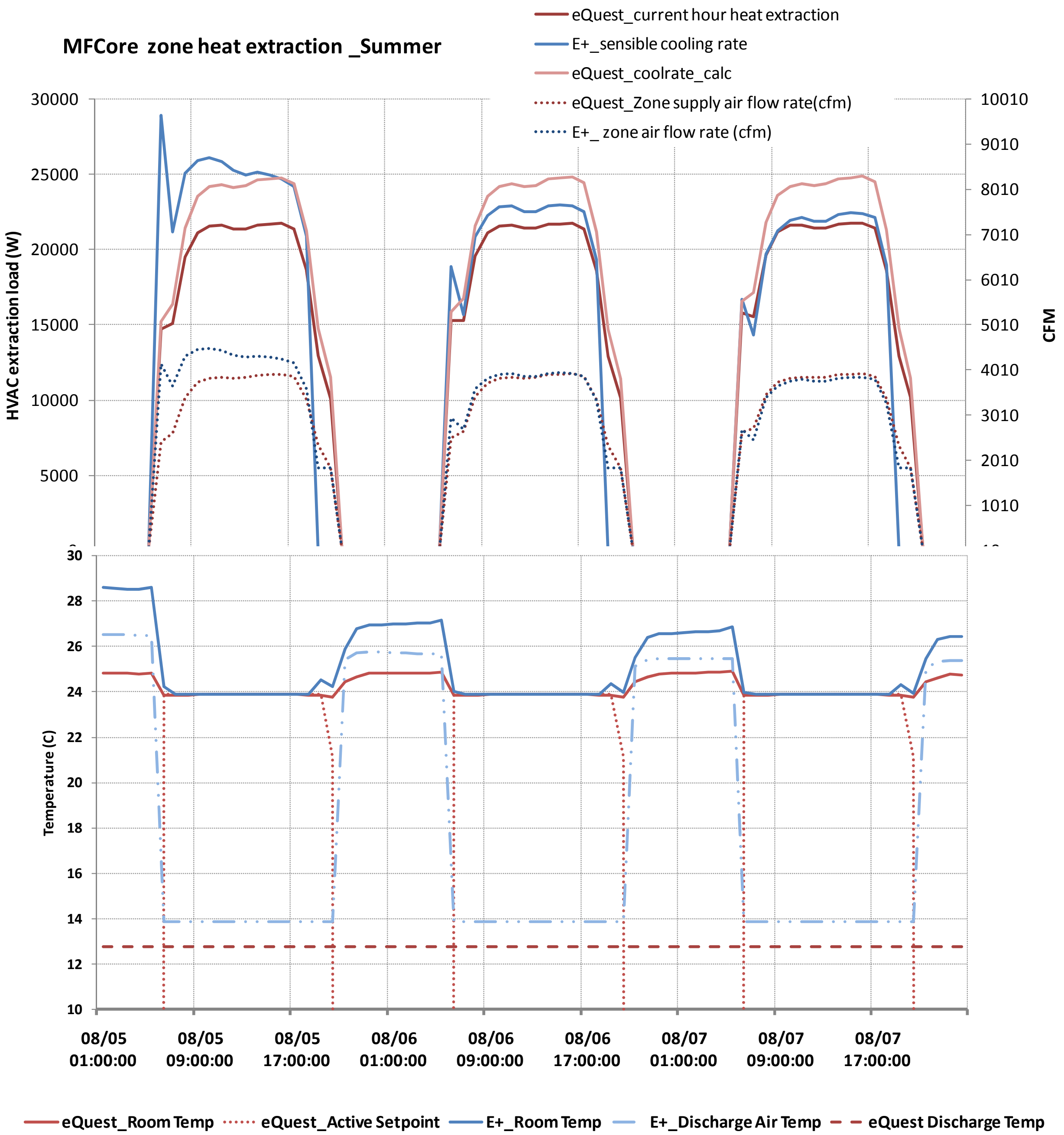


Figure 2.1(b): Comparison results of middle floor *Core* in *summer* (Sacramento)

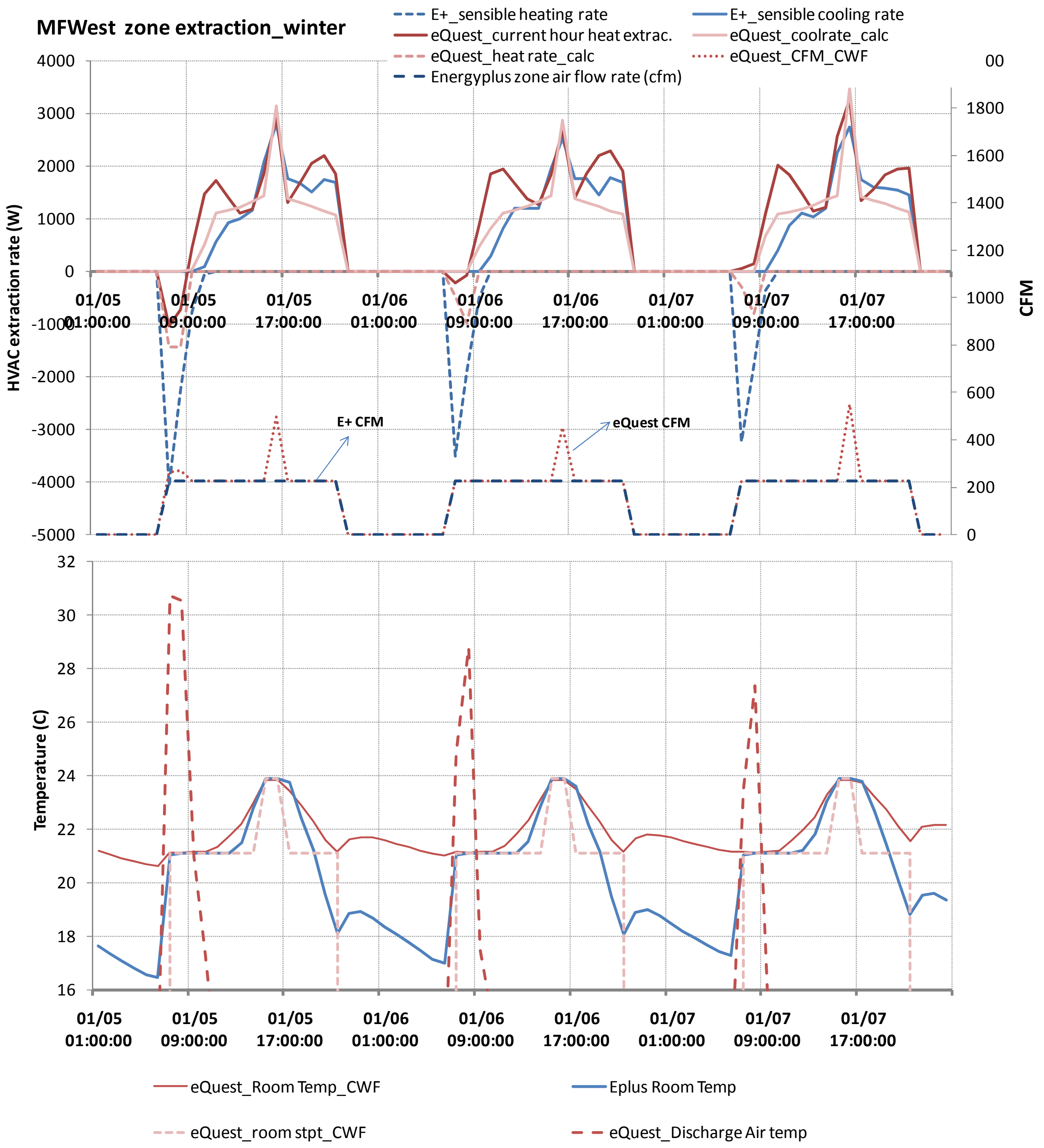


Figure 2.2(a): Comparison results of middle floor *west* in *winter* (Sacramento)

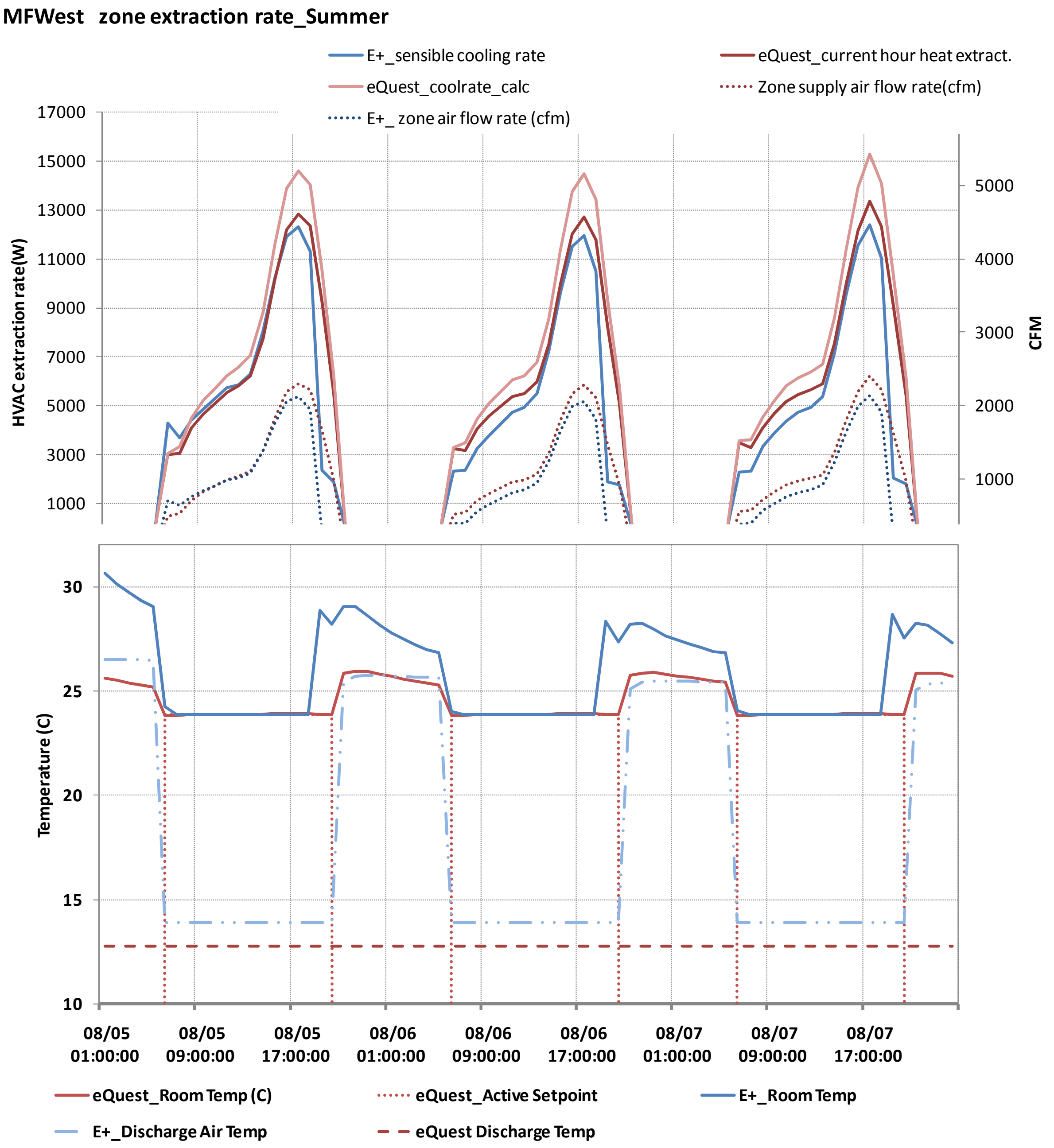


Figure 2.2(b): Comparison results of middle floor *west* in *summer* (Sacramento)

Key findings from these comparisons are:

a) Figure 2.1 (middle floor core zone) has the simplest heating/cooling load. Although the load levels (eQuest calculated vs. EPlus) match closely the resulting zone temperatures do not. Figure 2.1(a) shows how the zone load for EPlus is just high enough to control the room to the cooling setpoint, but for eQuest the load is never high enough to offset the cooling effect of the minimum airflow so the zone temperature drifts in the deadband. This figure also illustrates the large differences between using the eQuest extraction rate parameter vs. calculated results.

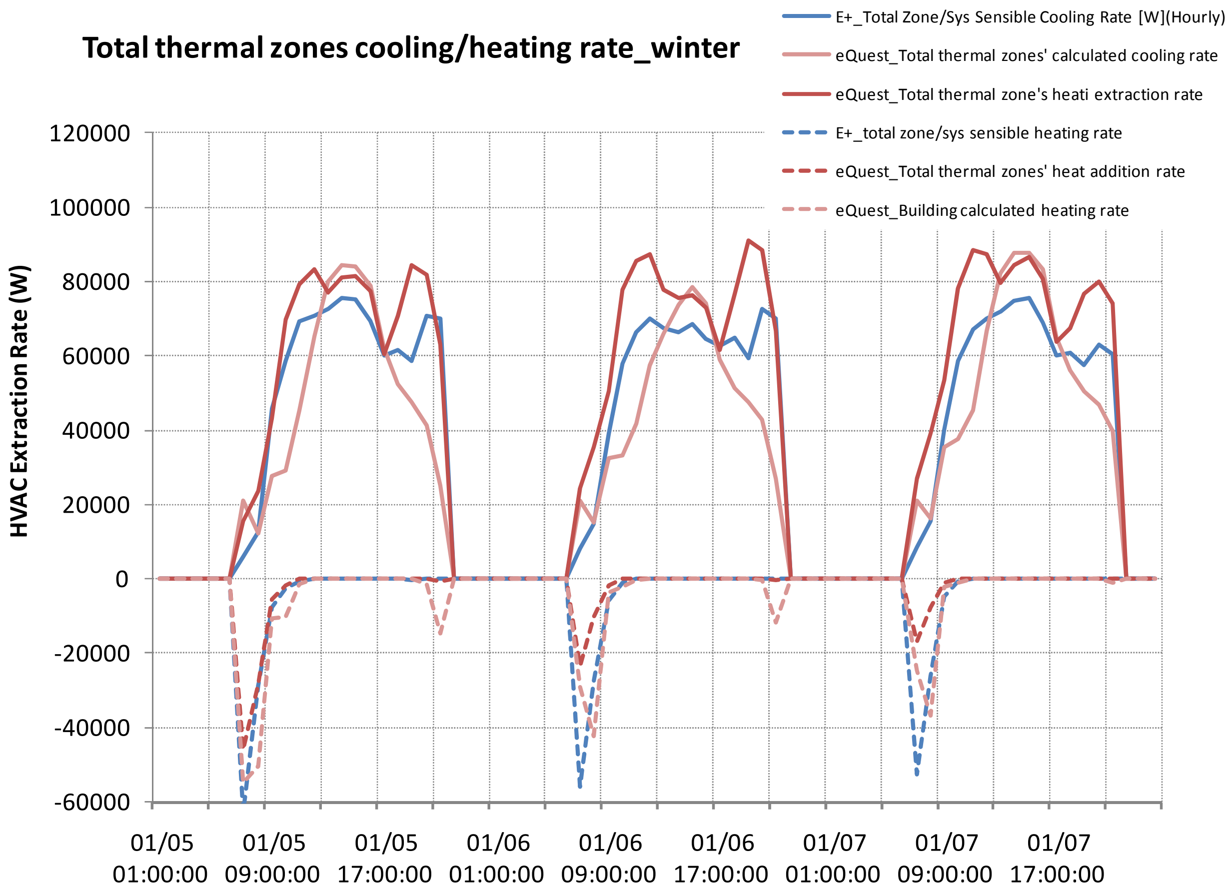
Figure 2.1(b), core zone summer, shows eQuest loads greater than EPlus but the airflow rates virtually the same; this is clearly due to the fact that the supply air temperature is lower for eQuest than for EPlus.

b) In Figure 2.2(a) for the West zone, the calculated cooling rate profile in eQuest matches reasonably well with EPlus results in winter. This figure again demonstrates that "current hour heat extraction rate" from eQuest only matches well when the room setpoint is well maintained, but exhibits strange behavior when the room temperature is in between the heating cooling setpoint and when the room temperature setpoint is outside of the throttling range. During the daytime the zone airflows are at minimum, so the zone temperatures are floating in the deadband and reflect the load profiles. Note also that the room temperature profiles exhibit large differences with EPlus temperatures dropping to low levels at night resulting in a large pickup load at startup in the morning.

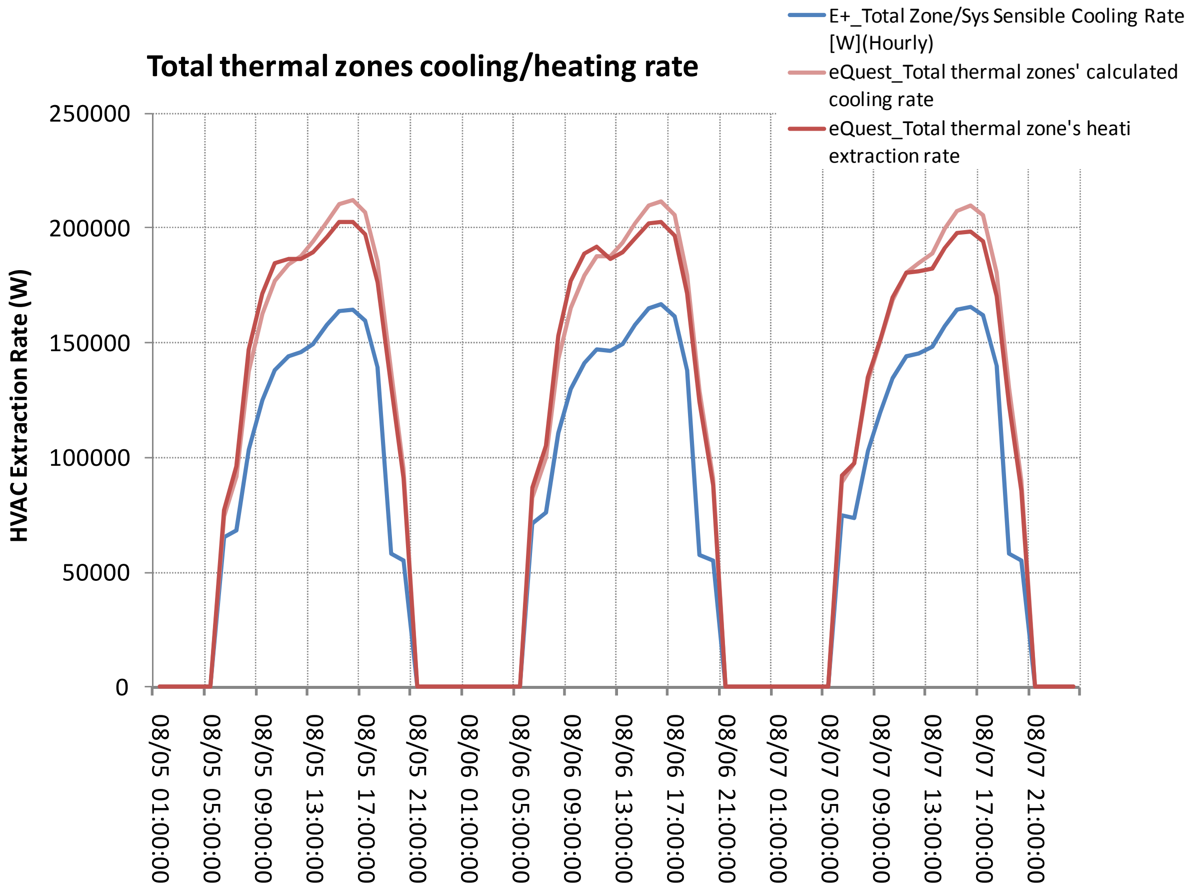
c) In Figure 2.2(b) for West zone in summer, however, when the room temperature can be well maintained and when a constant active setpoint is maintained, the reported value "current hour heat exchange rate" seems to matchup better with EPlus results. The differences in zone airflow reflect the differences in load, but again night time zone temperatures vary widely between the two programs.

Thermal zones total heating and cooling

Figures 2.3(a) and (b) show daily load profiles for heating and cooling load HVAC system sensible heating/cooling demand for the entire building in winter and summer respectively. The parameters used for comparison are the summation of zone level loads as described in Table 1.1.



Figures 2.3(a): Total thermal zone heat extraction/addition rate in winter (Sacramento)



Figures 2.3(b) : Total thermal zone heat extraction rate in summer (Sacramento)

Key findings for total demand as shown in Figures 2.3(a), (b) are:

a) In winter, the cooling rate profile reported by eQuest shows the strange “shoulders” at the beginning and the end of the day, which reflects the same behavior exhibited at the individual zone level; calculated cooling rate does not exhibit this behavior and appears more reasonable but it is somewhat different than the EPlus profile which lies in between the eQuest calculated and reported. On the heating side, we can clearly see that calculated heating rate in eQuest matches EPlus results closely.

b) In summer, Figure 2.3(b), however, the two eQuest parameters are very close but peak values are about 20% higher for eQuest than for EPlus.

Annual System annual load breakdown

Figure 2.4 shows results of heating and cooling demand by summing zone extraction rate for the entire year for all zones. We can see that for the annual results, the cooling load in eQuest is about 20% higher than the EPlus results.

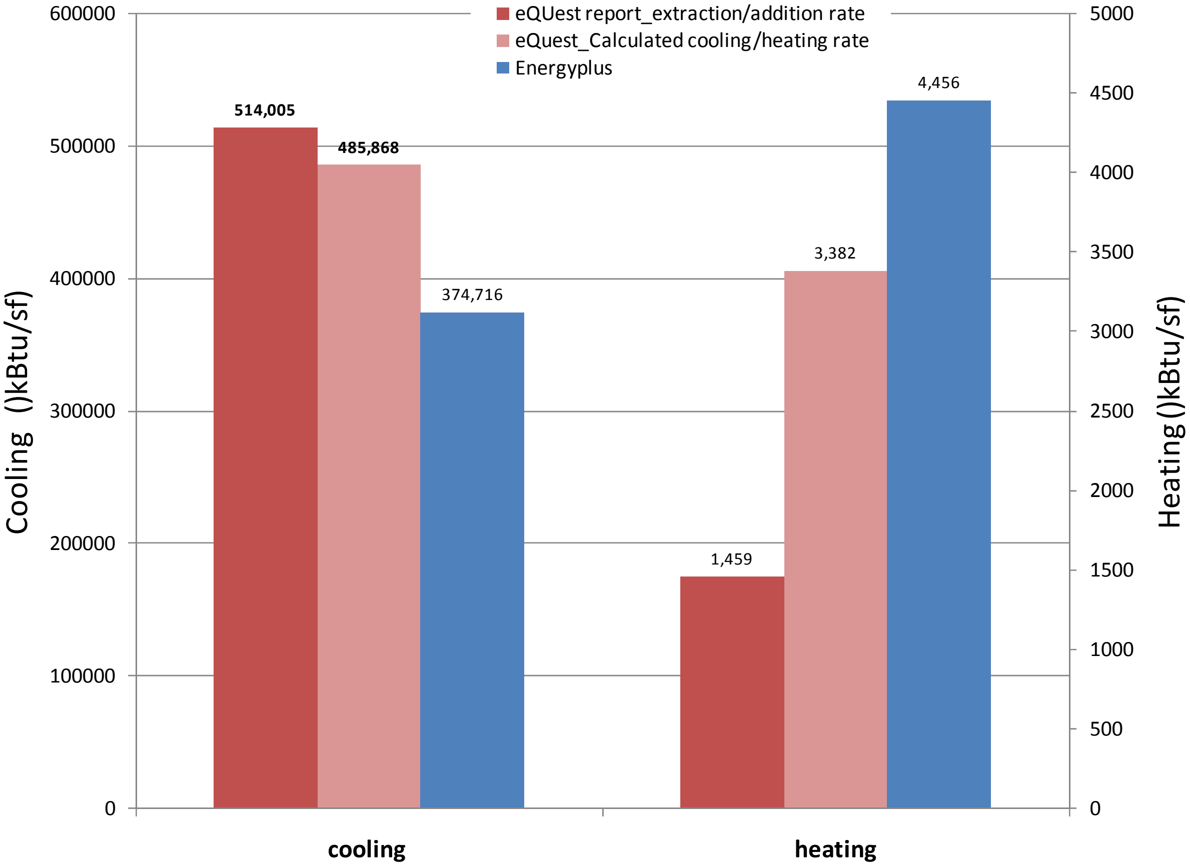


Figure 2.4: Annual total thermal zones' sensible cooling and heating energy consumption (Note scale change between heating and cooling (Sacramento)

HVAC System Level

For this study system component part load curves were NOT normalized between the two programs. These results, shown in Figure 2.5 only show order of magnitude comparison and a rough indication of the relative proportions of component energy use. While the cooling and airflow reflect previous results, auxiliaries are significantly different as is heating energy.

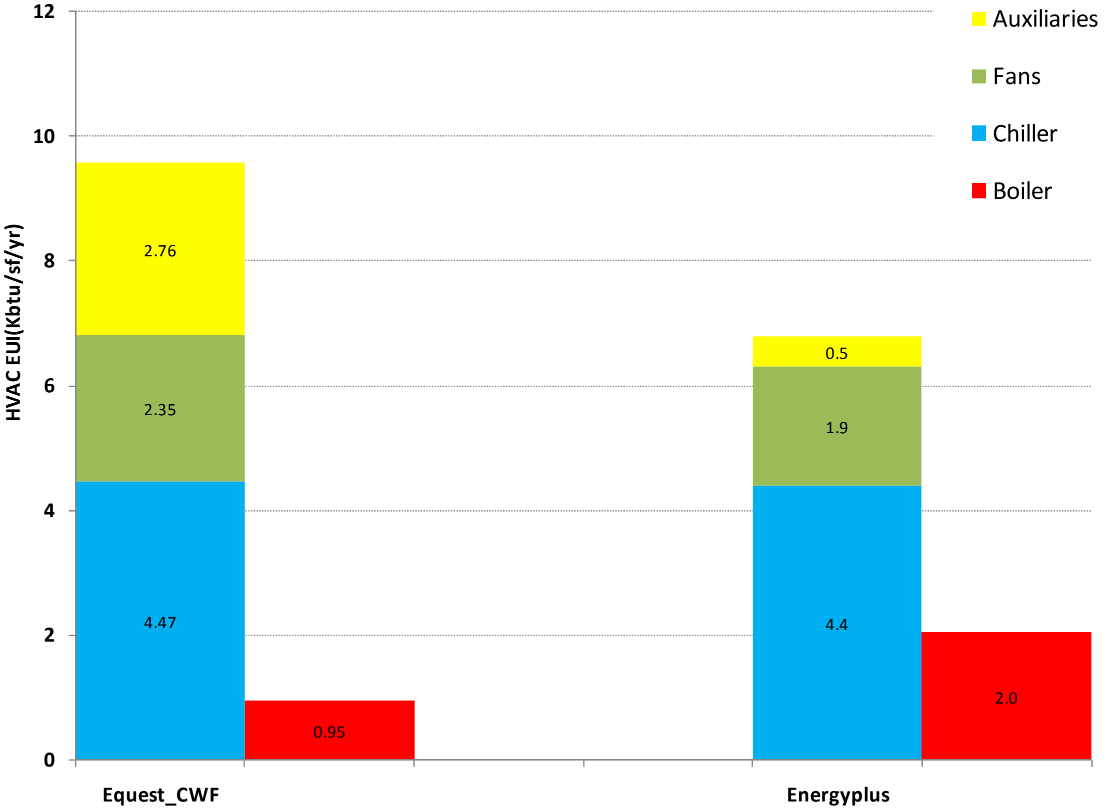


Figure 2.5: Total HVAC end use comparison between EQuest and EPlus.

Overall these eQuest studies provide insight into fundamental differences between eQuest and EPlus, however it does not answer all the questions. The greater cooling loads for eQuest shown in the hourly profiles are reflected in the zone level summaries (Figure 2.4) but not as much in the end use summary shown in Figure 2.5. Similarly, fan power and auxiliary energy are significantly higher for eQuest at the end use level. These discrepancies could be related to not having the system components “tuned up” properly between the two programs. The heating energy on the other hand, is significantly lower for eQuest compared to EPlus. This could be a result of differences in morning startup conditions as shown in Figure 2.2(a) that results from wide differences in night time zone temperature decay in perimeter zones. Further research seems warranted to better understand these differences. However, as discussed below, the methods for determining UFAD savings are assumed to be somewhat decoupled from these considerations.

Effects of whole building model differences

These tests were conducted to test and verify that performance comparisons that represent “UFAD” savings and trends are consistent with previous results, and if not to determine why they are different. It also ensures that problems are not introduced during the parametric run process; the check runs will give a basic benchmark to compare against. This helps to ensure no significant bugs were introduced when the whole building model was updated to T-24-2008 specifications. Two sets of simulations were run, one manually, and one using the newly created parametric run generator.

The results below show the comparison between the latest results from our current model and the previous results summarizing the source HVAC end use energy consumption breakdown. Fig. 2.6 shows the latest results from the current model, while Fig. 2.7 summarizes the results from the old model [Webster 2010]. Chiller/cooling energy is consistent between two sets of runs but boiler energy decreased and fan energy increased compared to previous results. Reasons for the discrepancies are summarized in the following:

1. The old model was based on ASHRAE 90.1-2004 construction specifications although the window thermal properties used were the same as the new model. Wall construction specifications are about 3 times more stringent in T-24-2008.
2. Thermostat settings for the new model are 73°F cooling and 70°F heating while in the old model they were 75/70°F. It has been shown that this difference can cause about a 10-15% change in cooling (and fan) energy.
3. Infiltration is scheduled off in the current model based on California Title-24-2008 specifications, while infiltration was modeled during the occupied hours in the old model (Figure 2.7). Detailed analysis revealed that this is the primary cause of the reduced boiler energy for the new model despite the increase in minimum airflow which increases reheat energy.
4. Minimum airflow for each UFAD terminal unit is set at 20% of the design airflow consistent with California Title-24-2008 in the current model, while it was set at 12% in the old model. This causes fan energy and heating (reheat) to increase.
5. Internal loads for people were increased in the current model, heat gain from zone equipment is increased, and lighting energy was decreased compared to the old model.
6. Internal mass is modeled in the current model, while it was not modeled in the old model.
7. Schedules are significantly changed from the old model.
8. Service cores are deleted in the current model but this should have little impact since adjustments were made in the internal load specifications to ensure total internal loads were the same for models with and without service core.
9. For the regressions and in the last run in Figure 2.6 the UFAD fans were modeled with the same design static pressure as OH. However, a direct comparison is cannot be made since the Fan energy for UFAD is modeled with same static pressure as OH.
10. The new model has refined convection coefficients for the supply plenum.

Figure 2.6 also indicates that for comparable design parameters UFAD consumes more energy than OH at equal SATs (e.g., 57°F). Also, the tradeoffs between fan and cooling energy as SAT is increased is not consistent with previous results (see Figure 2.7) which showed net positive savings due to this tradeoff. Further research will be required to fully determine the causes for this but the following two factors may be key contributors:

- Previous UFAD design sizing studies [Schiavon 2010] has shown that zone loads are greater throughout the day for UFAD, not just at peak conditions. This effect has been attributed to the use of relatively light mass raised floor panels that results in more immediate release of heat gain (especially in solar-loaded perimeter zones), as opposed to the greater storage effects of the floor slabs that are exposed to solar gain with OH systems. This may affect energy performance by increasing the cooling load (and airflow) during occupied hours for UFAD as opposed to storing energy in the floor slab for the OH system. Further research is warranted to more clearly quantify this effect; it will be the subject of ongoing research at CBE.

- Contrary to previous studies where a minimum ventilation strategy was used for UFAD (i.e., minimum airflow was fixed at the required ventilation rate, not a fixed percentage)[1] the current study uses a 20% minimum. This combined with increased terminal unit sizes causes terminal unit fan energy to increase. However, this strategy may unfairly bias UFAD since with large terminal unit sizes UFAD would have higher zone ventilation rates; the two absolute ventilation rates should be made equivalent. This will also be the focus of ongoing UFAD research. In addition, no credit is given to UFAD for the beneficial effects on ventilation effectiveness associated with stratification as is done with ASHRAE standards.

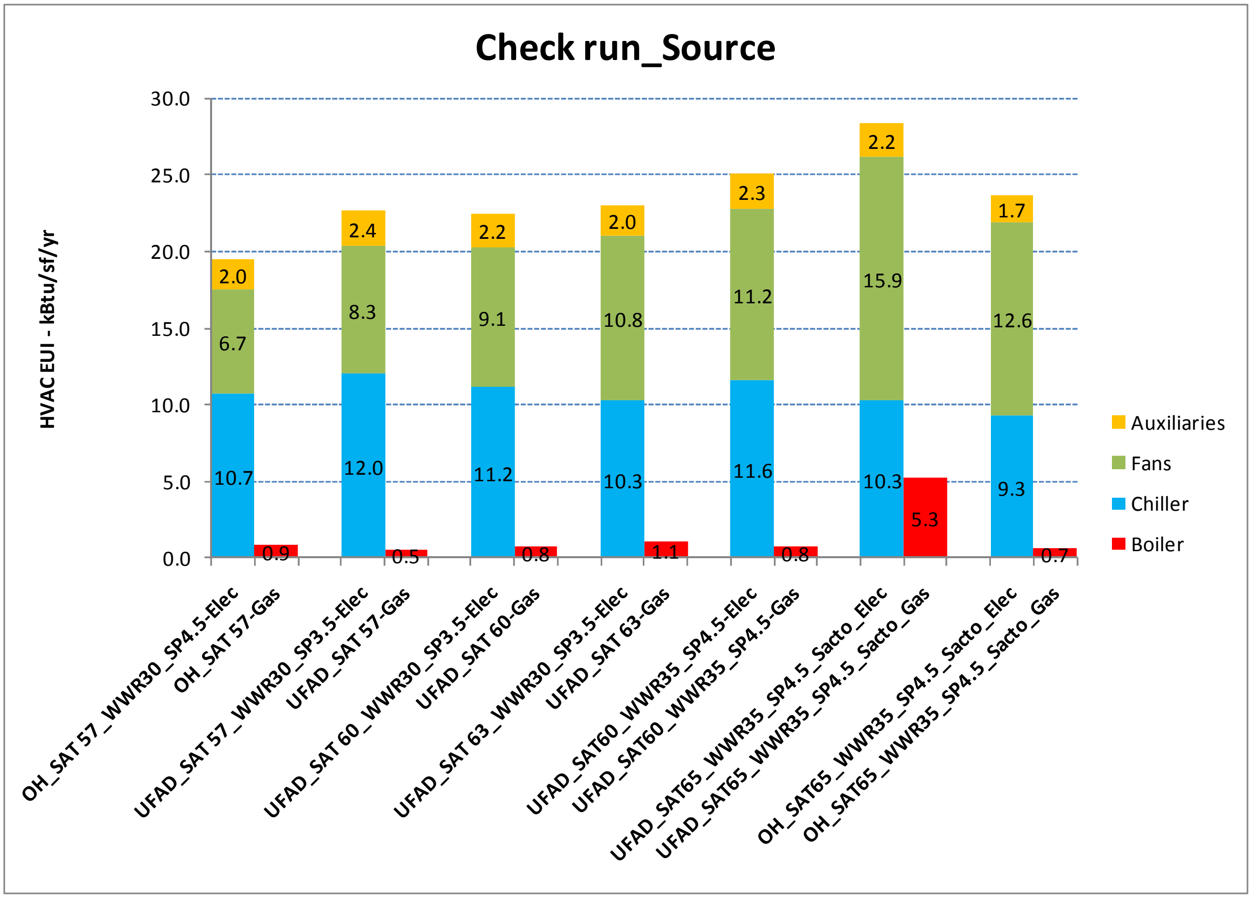


Figure 2.6: Latest energy breakdown results from the current model. (Source energy, Sacramento; 20k Gsf floorplate, 20% min ventilation)

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[1] More recent studies were conducted with a 12% minimum terminal unit airflow which is more consistent with how fan coils with variable speed drives operate.

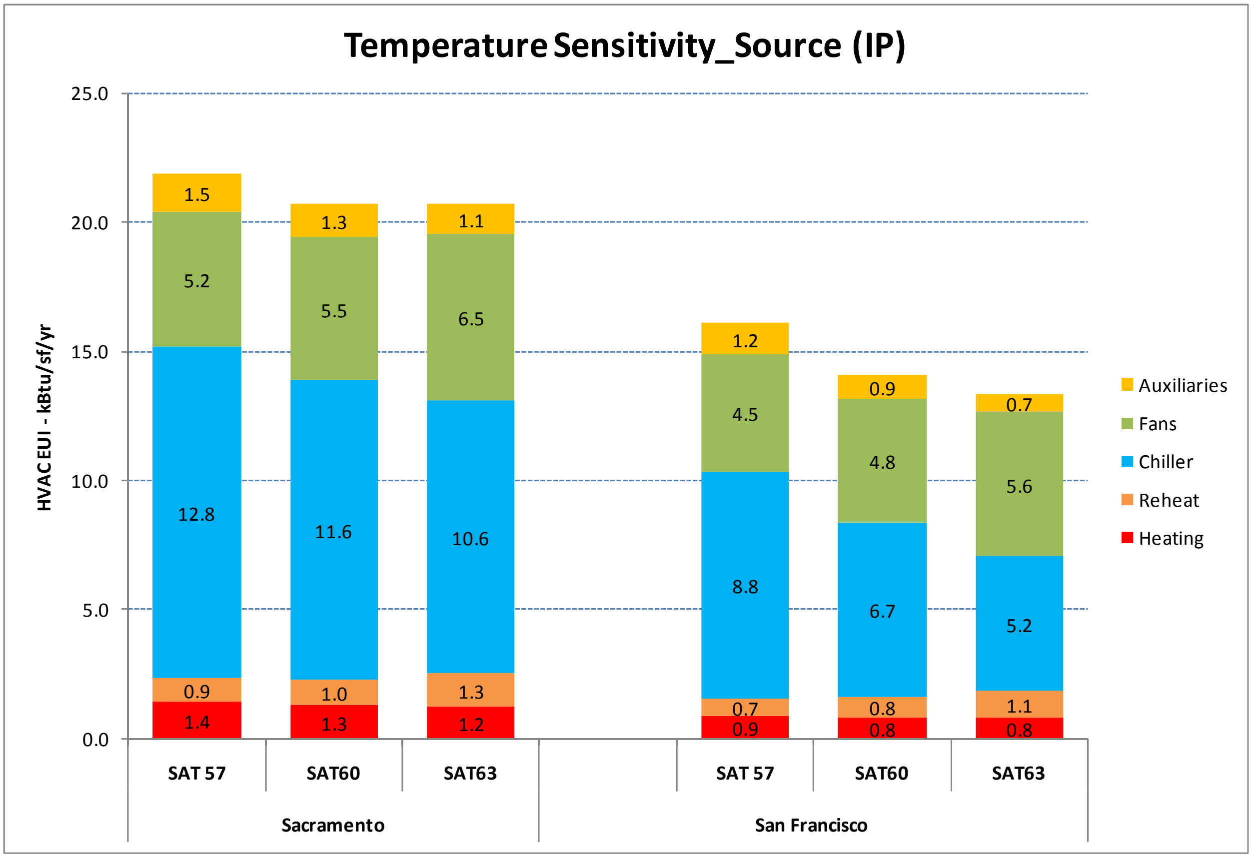


Figure 2.7: UFAD HVAC source energy breakdown results from the old model. [SimBuild 2010 paper; 38% WWR, 20k Gsf floorplate, min ventilation, 3.1 iwc design static pressure]

Based on the analysis above it appears that the results are comparable to previous work in this area.

We also performed two sets of simulations to test the reliability of the new parametric run generator: one run manually and one run using the parametric run generator. The purpose of this test is to ensure that any problems are not introduced during the automated parametric run process. This process led to several refinements in the parametric run generator.

EnergyPlus Furniture/ thermal mass preliminary study

There are two potential factors that could influence the peak load and energy performance of simulations for both eQuest and EnergyPlus: (1) solar distribution algorithms, and (2) interior thermal mass that represent the combined effects of interior structures and furnishings. In eQuest, solar distribution is assumed to be split between the floor and other surfaces as specified by a fractional distribution keyword. The effect of furniture and other internal structures is covered by an explicit thermal mass object. Neither program applies transmitted solar gains to these internal mass objects.

In EnergyPlus there are five different methods that users can select from to specify solar distribution. However, there are only two that are relevant for this study; FullExterior and FullInteriorandExterior. (The other methods include reflections and shadowing.) The former assigns all directly transmitted solar gain to the floor (much like eQuest) while the latter uses a ray tracing algorithm to determine which surfaces receive solar gain.

Test cases were run to identify the impact of these various methods on heat gain distributions and energy performance in EnergyPlus. There appears to be no significant differences between the two types of solar distribution. However, there are some noteworthy impacts due to thermal mass effects.

Due to the proximity of the underfloor plenum and its lower operating temperature in UFAD systems, a significant amount of heat can be transferred to the supply plenum. However, in reality, in perimeter zones this heat transfer can be impacted by the amount of solar gain that occurs on the floor which in turn can be affected by how much of it is intercepted by furniture. (This is not likely to be a major factor in OH systems since there is no underfloor plenum). A test was conducted where the floor was made up of two elements, standard raised floor, and an “artificial furniture” floor element. The special floor element consisted of thermal mass that represents the furniture (equivalent to the amount used in eQuest) and an insulating layer between the mass and the underfloor plenum. The other case, Standard Mass, had the same amount of mass but without solar exposure. Both cases used the FullExterior solar distribution algorithm which puts all the transmitted solar gains on the floor. The furniture element was assumed to have the same coverage as the eQuest furniture object, namely 85%.(See Figure 2.8)

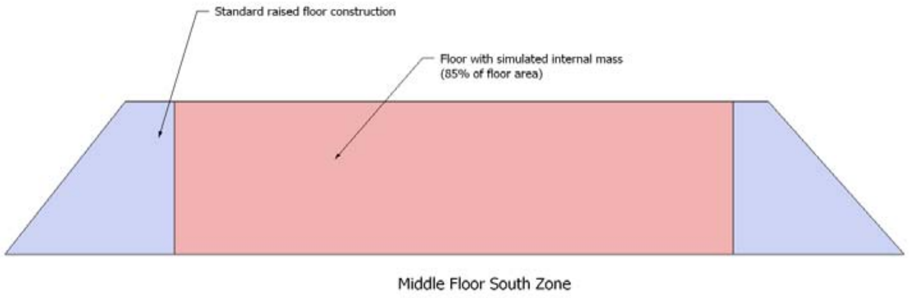


Figure 2.8: South zone Furniture Mass case illustration

Figures 2.9-2.12 show EnergyPlus annual simulation results for a south perimeter zone comparing a case using the (Standard Mass) internal mass object with a specially created artificial furniture case (Floor Mass). Figures 2.9(a), and 2.9(b) show the comparison for the zone cooling rate, and Figures 2.10(a) and 2.10(b) show the comparison for the supply plenum cooling rate. The room cooling rate is changed very little, with the median increasing by a small percentage for the Floor Mass case. The corresponding effect on supply plenum heat gain shown in Figures 2.10(a), (b) shows a decrease in median heat gain. These results indicate the effect of solar gains being intercepted by the furniture causing an increase in room load and thus reducing the amount of energy transferred to the supply plenum. These results are preliminary in that more analysis needs to be done to compare room and supply plenum for coincident cooling hours and the effects during heating hours need to be evaluated.

Figures 2.11(a), and 2.11(b) show the impacts on thermal decay associated with the differences in plenum heat gain; larger thermal decay for the Standard Mass case. Thermal decay is greater in the morning despite the fact that heat gain is less because airflows are lower in the morning.

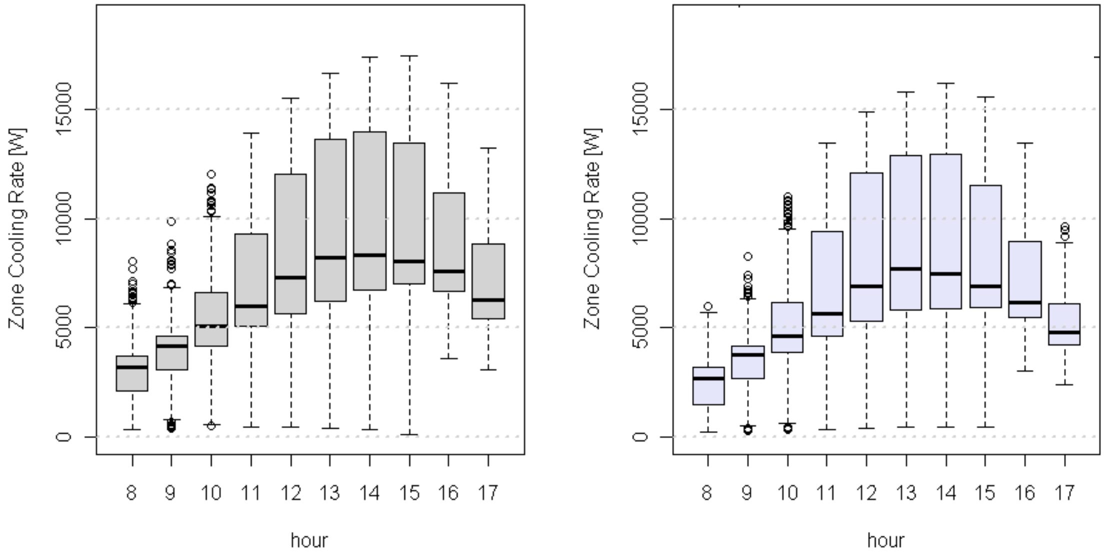


Figure 2.9(a): Zone cooling load with Floor Mass; (b): Zone cooling load with Standard Mass

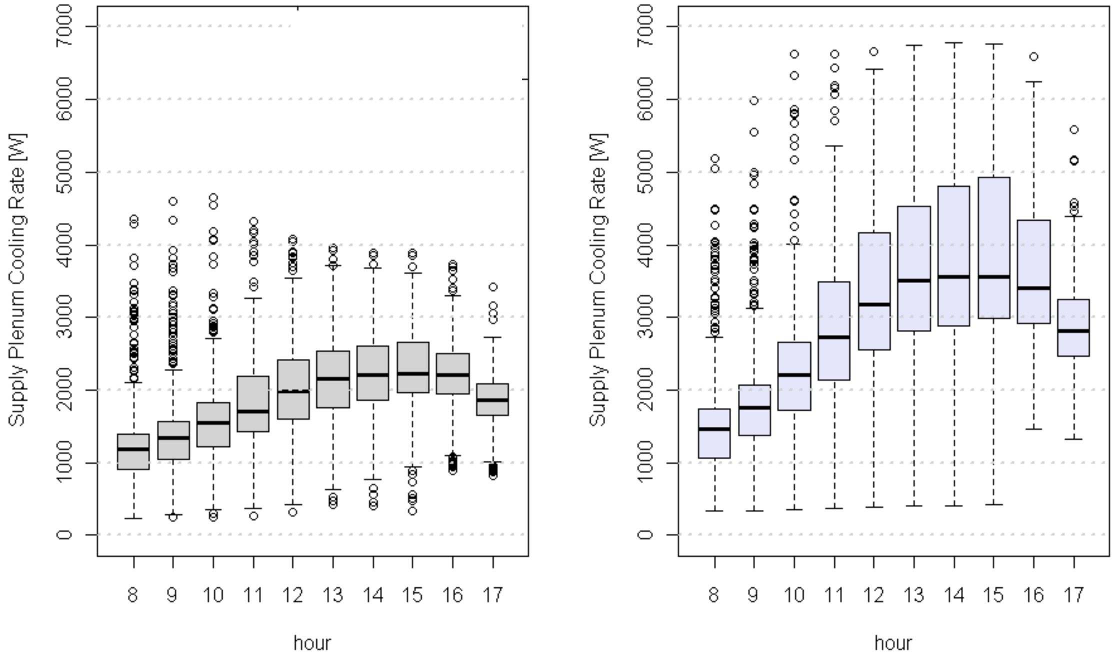


Figure 2.10(a): Supply plenum cooling load, Floor Mass (b): Supply plenum cooling load for Standard Mass

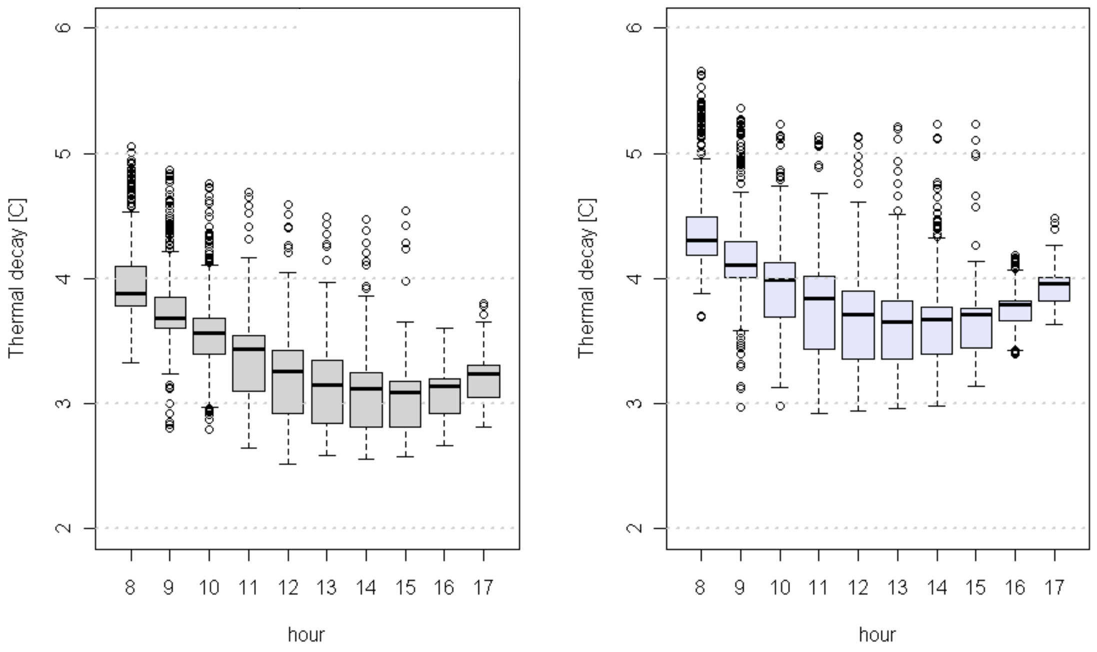


Figure 2.11: Supply plenum thermal decay (defined as difference between average perimeter zone plenum temperature and AHU supply temperature) for room cooling hours; (a) Floor Mass, (b) Standard Mass.

Task 3 – Modeling guidelines development

The original proposed approach to developing adjustment factors to allow DOE-2 programs to estimate UFAD performance were centered on improvements in concepts currently specified in the ACM. These consist of (1) reducing the zone loads by a room cooling load factor (RCLR) that accounts for heat transferred to the supply plenum; and (2) splitting the resulting zone heat gains between the room and the return plenum to simulate the effects of stratification. The effect of higher AHU SAT was covered by allowing higher SAT setpoints (or reset strategies); fan energy reductions due to reduced static pressure requirements for UFAD was covered by allowing for reduced design static pressure. Since there are several known flaws in this approach, the research team proposed to modify these procedures to produce a more accurate result based on using the UFAD version of EnergyPlus to help develop the improved methods.

Objective

Develop new methods and guidelines for modeling energy performance of UFAD systems with DOE-2 energy simulation programs.

Task 3 results

Evaluation of approaches

Upon review of the approach described above and new knowledge gained from the research team's use of EnergyPlus/UFAD, it became clear that the known flaws in the overall approach were not to be easily overcome for the following reasons:

1. The factors used to apportion energy to different sections of the system were static and not dynamic, thus the assumed fixed factors did not represent temporal variations adequately.
2. The assumed energy transfer to the supply plenum was left unaccounted for in the overall system heat balance
3. Perimeter zone solar gain was not accounted for properly
4. Merely increasing the SAT, while capturing the effect on economizer performance, did not adequately represent the effect of thermal decay on room airflow requirements.

A new approach was conceived that uses the UFAD version of EnergyPlus to compute differences between overhead and underfloor end use component energy consumption, which can then be applied to DOE-2 OH run results. This method requires generation of regression equations based on a large number of EnergyPlus runs for OH and UFAD made over a wide variation in load, design, and operating conditions.

EnergyPlus modeling

To conduct the regression analysis, CBE's three story, 20,000 ft^2 (nominal floorplate size) whole building model was upgraded to T-24-2008 prescriptive standards for building design parameters and schedules and performance compliance specifications for HVAC and plant systems from the ACM. Details of the CBE whole building model can be found in a Simbuild 2010 paper {Webster, Lee, et. al. 2010]. Modifications made to this model to conform to T-24-2008 are included in Appendix B. Simulations were run for three climates that represent the locations where the predominant number of UFAD systems occur; Los Angeles, San Francisco/Oakland, and Sacramento (weather files CZ12, CZ03, and CZ09, respectively). Variables used for the regression are shown in Table B-1. Besides the three climates and two system types (OH and UFAD), floorplate size, window to wall ratio, and lighting load are included to provide a wide load variation and to determine the sensitivity on UFAD savings of these factors; supply air temperature is the independent variable for the regressions. A full combinational set of runs was made for three levels of each of the variables resulting in a total of 486 runs. Due to time limitations these runs were made with the internal mass object only, not the artificial floor element.

Sensitivity study

A sensitivity study was conducted to investigate the relative importance of independent variables used in creating the regressions equations. It was determined that all the following variables have a statistically significant effect on the UFAD/OH ratios for end use components.

Regression correlations

The following equations were derived from a multi-variate regression analysis of the results of the EnergyPlus runs.

Heating model:

$$\mathit{Heating\ ratio} = 10^{-0.3120+0.0078\cdot SAT-0.6114\cdot INT+0.0810\cdot SAT\cdot INT+Climate}$$

With Climate coefficients:

- Los Angeles=0
- San Francisco=-0.2127
- Sacramento=-0.3578

Units: SAT in °C and INT in W/ft^2

Adjusted R-squared=0.93. Linear hypotheses verified

<u>Cooling model:</u>

$$Cooling\,ratio = \frac{1}{0.789334 + 0.007163 * SAT - 0.033136 * INT - 0.000463 * WWR + 0.000323 * Floor\,Area}$$

Units: SAT in °C and INT in W/ft^2 and Floor Area in kft^2

Adjusted R-squared=0.67. Linear hypotheses verified

<u>Auxiliary pumps model:</u>

$$Pump\,ratio = 1.6721 - 0.0324 * SAT + 0.1534 * INT + Climate + COEFF1 * INT$$

With climate coefficients:

- Los Angeles=0
- San Francisco=-0.1039
- Sacramento=-0.1717

And COEFF1:

- Los Angeles=0
- San Francisco=-0.0104
- Sacramento=0.1333

Units: SAT in °C and INT in W/ft^2

Adjusted R-squared=0.84. Linear hypotheses verified

<u>Fan model:</u>

$$Fan\,ratio = \sqrt{\frac{1}{-0.055595 + 0.017207 * SAT + 0.033494 * INT + 0.002553 * Floor\,Area + Climate}}$$

With climate coefficients:

- Los Angeles=0
- San Francisco=0.0145
- Sacramento=-0.042064

Units: SAT in °C and INT in W/ft^2 and Floor Area in kft^2

Adjusted R-squared=0.74. Linear hypotheses verified

Not included in the regression analysis were the following factors:

- Differences due to using York/JCI vs. fan coil/linear bar grille systems. Previous studies [Bauman, Webster et. al. 2006] indicate that there are no substantial annual energy performance differences between these two systems except for a small effect on fan energy.
- SA fan static pressure. UFAD systems generally have lower static pressure requirements than OH depending on supply plenum air distribution configuration. This can affect fan energy consumption by about 5-6% per 1 iwc change.
- Effect of number of diffusers. It has been shown [Bauman, Webster et. al. 2011] that the number of diffusers, especially with perimeter linear bars grilles can have a significant impact on stratification, a key factor in UFAD energy use.
- Return air system. Only relief fans were included in the regression modeling, not return fans. It is assumed that this return solution is used widely.
- Gamma-Phi. The new Gamma-Phi correlations (discussed above) were not used in the regression analysis.
- Cooling tower. Cooling tower energy use was inadvertently left out of the analysis.
- EnergyPlus UFAD refinements: There are known problems that need to be repaired; e.g., heating occurs in the deadband near the cooling setpoint; there is no winter design cooling day in EnergyPlus which is important for sizing south zones; leakage flow through the FCU is actually variable for some designs, so a better method of calculating this needs to be developed, and terminal unit sizing to properly account for thermal decay. As pointed out above, the methods to model minimum volumes for UFAD need to be improved so that ventilation rates between OH and UFAD are comparable.

Incorporating these factors into the methodology is important to improve the accuracy of the results and so should be the focus of future research efforts. In addition, further studies need to be made to understand the sources of misalignment between heating and cooling results between the two programs.

Guidelines and procedures

To apply the regression results the following procedure is used:

1. Run DOE-2 for proposed design building with an OH system with SAT set equal to the UFAD desired SAT.
2. Using regression equations, compute UFAD/OH ratios for each end use component (EUC), R_{euc}. Since there is some sensitivity to design variables internal load (INT), window to wall ratio (WWR), floorplate size (floor area), and climate (Climate) inputs of these variables into the regression equations must be the same as those of the proposed building design.
3. Multiply the R_{euc} for each end use by the corresponding OH end use energy consumption derived from a design run for the proposed building assuming an OH system operating at the UFAD design SAT (Step 1). This yields the UFAD end use components. Sum the components to yield the annual UFAD system HVAC energy consumption.
4. Re-run the design model for a selected OH baseline SAT. This yields the OH baseline end use components. Sum these components to yield the baseline OH HVAC system usage.
5. Determine the difference between UFAD and OH using the two annual sums of UFAD and OH baseline energy use.

Demonstration example

To demonstrate the efficacy of the methods and verify the accuracy of the results, the methods were used to create a comparison between simulated and calculated results (i.e., “True” vs “Calculated” UFAD end use energy consumption). This study was conducted only with EnergyPlus results; eQuest runs were not made due to aforementioned problems with getting the results between the two programs to correspond well. Figure 3.1 shows an example comparison for Sacramento for various SATs based on a building with 20,000 ft^2 floorplate, 35% WWR, and 0.85 W/ ft^2 internal load. Overall, the calculated vs. simulated results agree to within 10% or less.

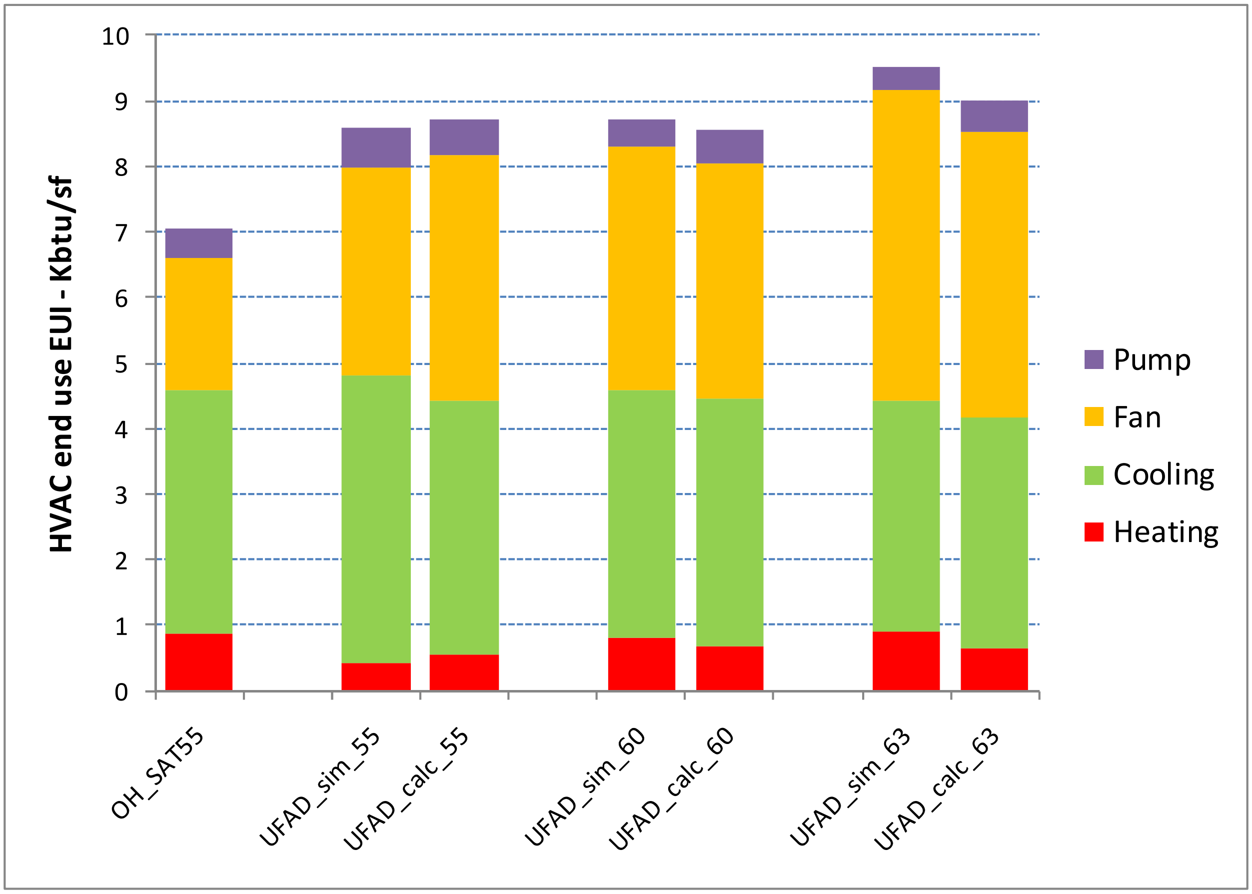


Figure 3.1: Simulated vs calculate UFAD energy use comparison (Sacramento)

Discussion, conclusions and recommendations

Considerable progress was made in the development of alternative methods for estimating the performance of UFAD systems derived from simulations for conventional overhead systems. The results look promising that appropriate regression equations can be developed by using the UFAD version of EnergyPlus.

A number of issues still remain to be resolved so the results of this work should not be construed to be a complete answer to the problem or to accurately represent UFAD performance savings. In particular, the results reported here are known to be negatively biased against UFAD due to the many unresolved issues noted above. These include the following:

- Minimum ventilation rates are not consistent between OH and UFAD

- EnergyPlus sizing issues for UFAD result in oversized terminal units
- Fan energy consumption does not account for lower static pressure requirements for UFAD
- UFAD systems may require a different approach to model interior furnishings thermal mass
- Unresolved differences between DOE-2 and EnergyPlus results point to fundamental differences in simulation techniques that may impact results
- Correcting known errors in EnergyPlus and in simulation procedures

In conclusion, the results indicate that these methods, when fully developed, would provide a relatively simple and accurate way to determine UFAD savings relative to conventional overhead systems free of the complications inherent in other “work arounds” being employed today. These methods also may be applicable to other systems types that are difficult to model in DOE-2 programs.

Recommendations

Recommendations can be summarized by emphasizing the need to resolve the issues exposed during this research as outlined throughout this report. Development work for EnergyPlus would be required as well as methods to refine inputs to both EnergyPlus and eQuest to better align the two programs. EnergyPlus modifications would include refinements in input procedures to account for difference in how UFAD and OH systems are sized, to ensure that OH systems are modeled consistently, and that assumptions between OH and UFAD are appropriate.

References

ACM. 2005. “Nonresidential Alternative Calculation Method (ACM) Approval Manual.” California Energy Commission, Sacramento CA, October

ACM 2008. “Nonresidential Alternative Calculation Method (ACM) Approval Manual.” California Energy Commission, Sacramento CA, December.

CEC. 2008. “Building Energy Efficiency Standards for Residential and Nonresidential Buildings.” California Energy Commission, Sacramento CA, January.

Webster, T., K. H. Lee, F. Bauman, S. Schiavon and A. Daly, 2010. “Influence of Supply Air Temperature on Underfloor Air Distribution (UFAD) System Performance.” *Proceedings, SimBuild 2010: IBPSA-USA, Building Simulation.* New York, USA.

Bauman, F., T. Webster, P. Linden, F. Buhl. 2007. “Energy Performance of Underfloor Air Distribution Systems." Final Project Report submitted to California Energy Commission (CEC) Public Interest Energy Research (PIER) Program (500-2007-050). Center for the Built Environment, University of California, Berkeley, CA, April

Bauman, F., T. Webster, D. Dickerhoff, S. Schiavon, K. Lee. 2011 (In press). “Advanced Design and Commissioning Tools for Energy Efficient Building Technologies.” Final report submitted to California Energy Commission (CEC) Public Interest Energy Research (PIER) Program (500-06-049). Center for the Built Environment, University of California, Berkeley, CA

Schiavon, S., K.H. Lee, F. Bauman, T. Webster, 2010. “Simplified calculation method for design cooling loads in Underfloor Air Distribution (UFAD) systems.” Center for the Built Environment, University of California, Berkeley, CA (accepted for publication in Energy and Buildings)

Appendix A - Walnut laboratory

The Walnut laboratory is at 16’ by 16’ UFAD test chamber in Kansas City, KN owned by Walnut Manufacturing. It was designed to conduct full scale testing of UFAD systems for typical office environments. It can be configured for testing perimeter UFAD zones by use of a solar simulator, or interior zones by placing insulating panels over the windows. Different diffusers can be located in the floor including locations adjacent to the window. The drop ceiling can be configured in several modes including an open overhead supply, open overhead return (used with UFAD testing), and traditional overhead ducted supply. Above the ceiling is a “slab simulator” which can be temperature controlled to simulate the effects of a multi-story building. The slab simulator was not used in these tests as its effect has minimal impact on the results for this study. Figure A1 shows the test chamber configured for three workstations.

Figure A1a: Test chamber view to West, with solar simulator in background behind window

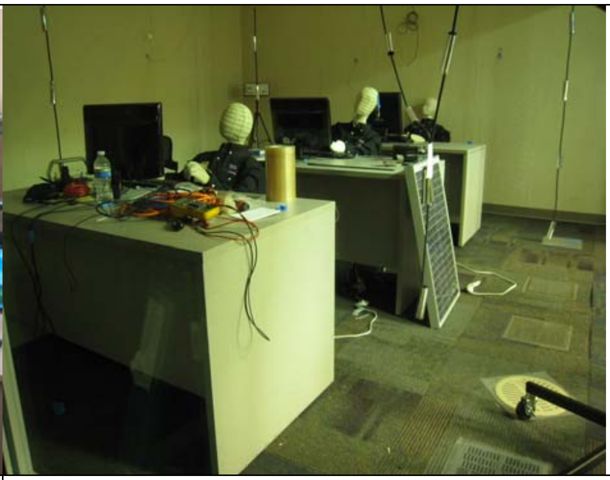


Figure A1b: View to South East corner, various VAV diffusers seen in the floor

Figure A2 shows the solar simulator which consists of three rows of four 2' round 1000 W high intensity discharge (HID) lamps which are angled down at 30 degrees to simulate summer afternoon sun incident on west facing windows. These are located in an environmental chamber (EC), where the air temperature can be controlled. The air in the EC is blown at the window via a slot at its bottom in an attempt to keep the window surface temperature uniform and to simulate effects of wind. With all the lights on the temperature can be as low as about 85 ^{o}F, with the lights off the temperature can be below freezing.

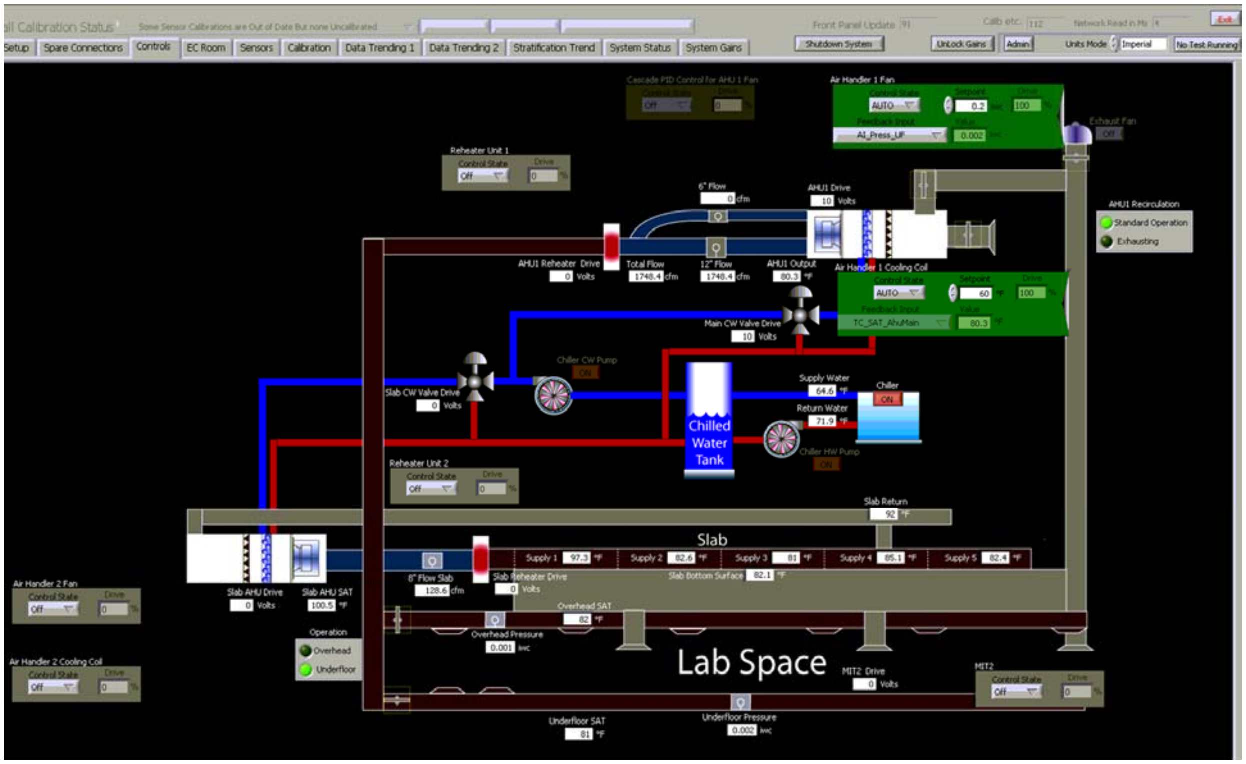
HID lamps are used because they were relatively efficient and have approximately the same visual/IR distribution as the sun. By adjusting the air temperature the surface temperature of the window can be made to be similar to the naturally occurring window surface temperature thus providing the same perimeter load to the test chamber. Solar intensities can be adjusted by placing different screens in front of the lamps. This increases the load in the environmental chamber which limits experiments to an air temperature of about 90 ^{o}F.

FigureA2: solar simulator in the EC

Hot or cold air is provided by an HVAC system that services the test chamber and slab simulator equipment located in the adjoining warehouse on top of a storage area, as seen in Figure A3. The measurement of air flow delivered to the test chamber is located here as well. It consists of two flow meters that can measure the flow from about 25 to 1500 cfm. A direct expansion (DX) unit provides cooling to the EC; the condensing unit is located at the upper left in Figure A3.

Figure A3: HVAC equipment for the test chamber and EC

Measurement and control of all points is provided by a National Instruments hardware with a LabView based frontend monitoring and control software. Over two hundred points are recorded, mostly temperatures, but also air flow, pressures, configuration status, internal loads, lighting and HVAC parameters. Time series plots of all values are available in real time as is the stratification profile from each of five “trees” as well as a user defined “average” trees. Each measurement sample results in an ascii file of all these values. Figure A4 shows the user interface of the HVAC control screen where control settings such as supply air temperature, air flow, and HVAC parameters can be set. Some controls can have multiple control values, for example the air flow can be set to maintain a constant air flow or be controlled by the thermostat reading to achieve setpoint. The thermostat can be a single sensor or the average of a list of sensors.



Front Panel Update 91
Calib etc. 112
Network Read in Ms 4
Units Mode Imperial

Cascade PID Control for AHU 1 Fan: Control State Off, Drive 0 %

Air Handler 1 Fan: Control State AUTO, Setpoint 0.2 iwc, Drive 100 %, Feedback Input AI_Press_UF, Value 0.002 iwc

Exhaust Fan Off

Reheater Unit 1: Control State Off, Drive 0 %

6" Flow 0 cfm
AHU1 Drive 10 Volts
AHU1 Reheater Drive 0 Volts
Total Flow 1748.4 cfm
12" Flow 1748.4 cfm
AHU1 Output 80.3 °F

Air Handler 1 Cooling Coil: Control State AUTO, Setpoint 60 °F, Drive 100 %, Feedback Input TC_SAT_AhuMain, Value 80.3 °F

Main CW Valve Drive 10 Volts
Slab CW Valve Drive 0 Volts
Chiller CW Pump ON
Supply Water 64.6 °F
Return Water 71.9 °F
Chiller ON
Chiller HW Pump ON

Reheater Unit 2: Control State Off, Drive 0 %

Slab Return 92 °F
Supply 1 97.3 °F
Supply 2 82.6 °F
Supply 3 81 °F
Supply 4 85.1 °F
Supply 5 82.4 °F
Slab Bottom Surface 82.1 °F
Slab AHU Drive 0 Volts
Slab AHU SAT 100.5 °F
8" Flow Slab 128.6 cfm
Slab Reheater Drive 0 Volts
Overhead SAT 82 °F
Overhead Pressure 0.001 iwc

Air Handler 2 Fan: Control State Off, Drive 0 %

Air Handler 2 Cooling Coil: Control State Off, Drive 0 %

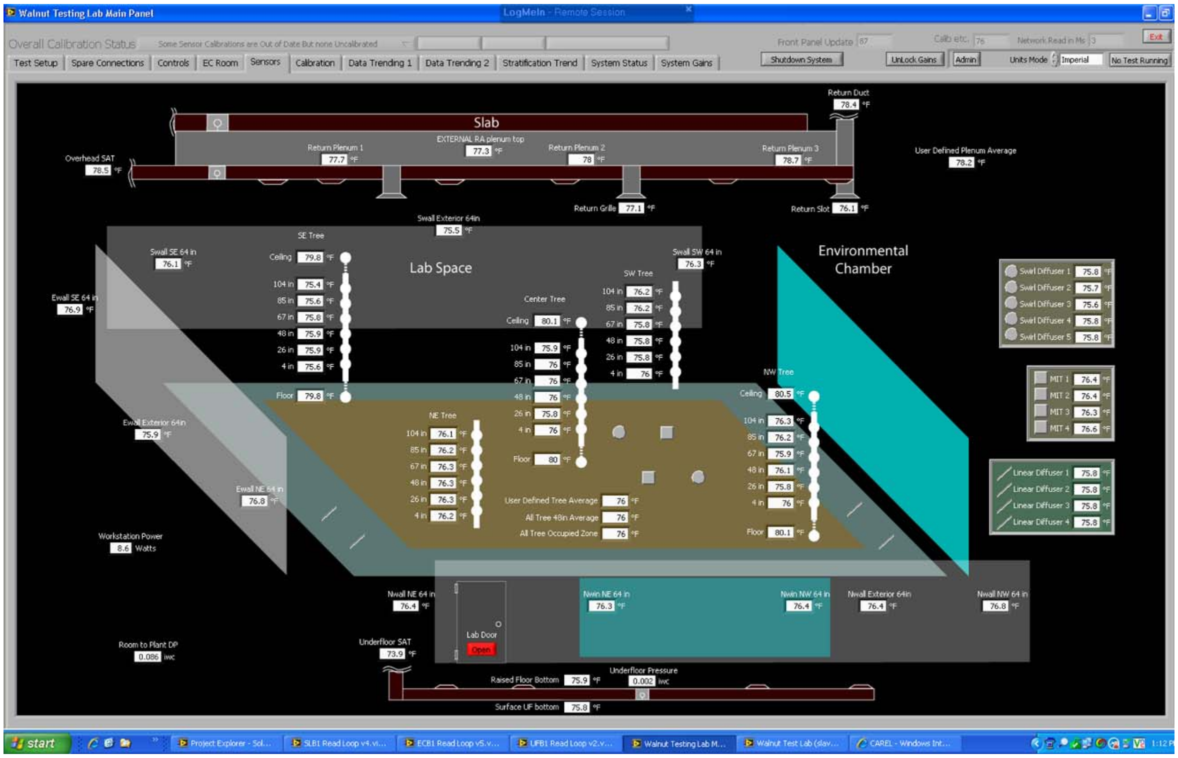
MIT2 Drive 0 Volts

MIT2: Control State Off, Drive 0 %

Underfloor SAT 81 °F
Underfloor Pressure 0.002 iwc

Figure A4: HVAC system controls

Figure A5 shows the user interface depicting the test chamber. Current values for “tree” temperature, supply and return plenums and surfaces are displayed.



Overall Calibration Status: Some Sensor Calibrations are Out of Date But none Uncalibrated
Front Panel Update 87
Calib etc. 76
Network Read in Ms 3
Units Mode Imperial

Return Duct 78.4 °F
Overhead SAT 78.5 °F
Return Plenum 1 77.7 °F
EXTERNAL RA plenum top 77.3 °F
Return Plenum 2 78 °F
Return Plenum 3 78.7 °F
User Defined Plenum Average 78.2 °F
Return Grille 77.1 °F
Return Slot 76.1 °F
Swall Exterior 64in 75.5 °F
Swall SE 64 in 76.1 °F
Swall SW 64 in 76.3 °F
Ewall SE 64 in 76.9 °F
Ewall Exterior 64in 75.9 °F
Ewall NE 64 in 76.8 °F
Workstation Power 8.6 Watts
Room to Plant DP 0.086 iwc

SE Tree: Ceiling 79.8 °F; 104 in 75.4 °F; 85 in 75.6 °F; 67 in 75.8 °F; 48 in 75.9 °F; 26 in 75.9 °F; 4 in 75.6 °F; Floor 79.8 °F

Center Tree: Ceiling 80.1 °F; 104 in 75.9 °F; 85 in 76 °F; 67 in 76 °F; 48 in 76 °F; 26 in 75.8 °F; 4 in 76 °F; Floor 80 °F

SW Tree: 104 in 76.2 °F; 85 in 76.2 °F; 67 in 75.8 °F; 48 in 75.8 °F; 26 in 75.8 °F; 4 in 76 °F

NE Tree: 104 in 76.1 °F; 85 in 76.2 °F; 67 in 76.3 °F; 48 in 76.3 °F; 26 in 76.3 °F; 4 in 76.2 °F

NW Tree: Ceiling 80.5 °F; 104 in 76.3 °F; 85 in 76.2 °F; 67 in 75.9 °F; 48 in 76.1 °F; 26 in 75.8 °F; 4 in 76 °F; Floor 80.1 °F

User Defined Tree Average 76 °F
All Tree 48in Average 76 °F
All Tree Occupied Zone 76 °F

Swirl Diffuser 1 75.8 °F
Swirl Diffuser 2 75.7 °F
Swirl Diffuser 3 75.6 °F
Swirl Diffuser 4 75.8 °F
Swirl Diffuser 5 75.8 °F

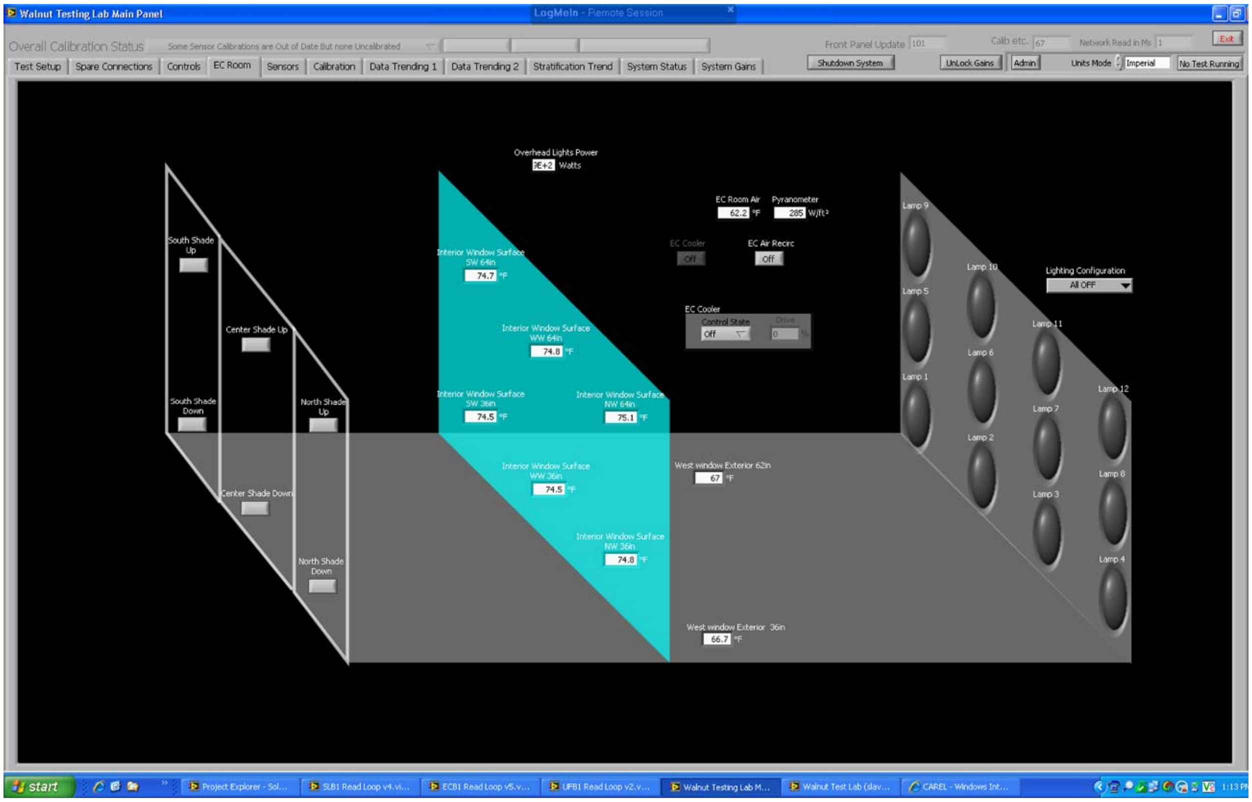
MIT 1 76.4 °F
MIT 2 76.4 °F
MIT 3 76.3 °F
MIT 4 76.6 °F

Linear Diffuser 1 75.8 °F
Linear Diffuser 2 75.8 °F
Linear Diffuser 3 75.8 °F
Linear Diffuser 4 75.8 °F

Nwall NE 64 in 76.4 °F
Lab Door Open
Nwin NE 64 in 76.3 °F
Nwin NW 64 in 76.4 °F
Nwall Exterior 64in 76.4 °F
Nwall NW 64 in 76.8 °F
Underfloor SAT 73.9 °F
Underfloor Pressure 0.002 iwc
Raised Floor Bottom 75.9 °F
Surface UF bottom 75.8 °F

Figure A5: Test chamber user interface

The control of the solar simulator and status of the EC is illustrated in another user interface screen as shown in Figure A6. Stratification “trees” and trend values are illustrated in another screen, Figure A7, where an average “tree” can be defined. Other values can be shown on a total of four trend series plots, each capable of multiple y-axis values to show varying units. A predefined set of values can be plotted together or one can select any value of interest to display.



Overall Calibration Status: Some Sensor Calibrations are Out of Date But none Uncalibrated

Front Panel Update: 101

Calib etc.: 67

Network Read in Ms: 1

Units Mode: Imperial

Overhead Lights Power: 0E+2 Watts

EC Room Air: 62.2 °F

Pyranometer: 285 W/ft²

EC Cooler: Off

EC Air Recirc: Off

EC Cooler Control State: Off; Drive: 0 %

Lighting Configuration: All OFF

Interior Window Surface SW 64in: 74.7 °F

Interior Window Surface WW 64in: 74.8 °F

Interior Window Surface SW 36in: 74.5 °F

Interior Window Surface NW 64in: 75.1 °F

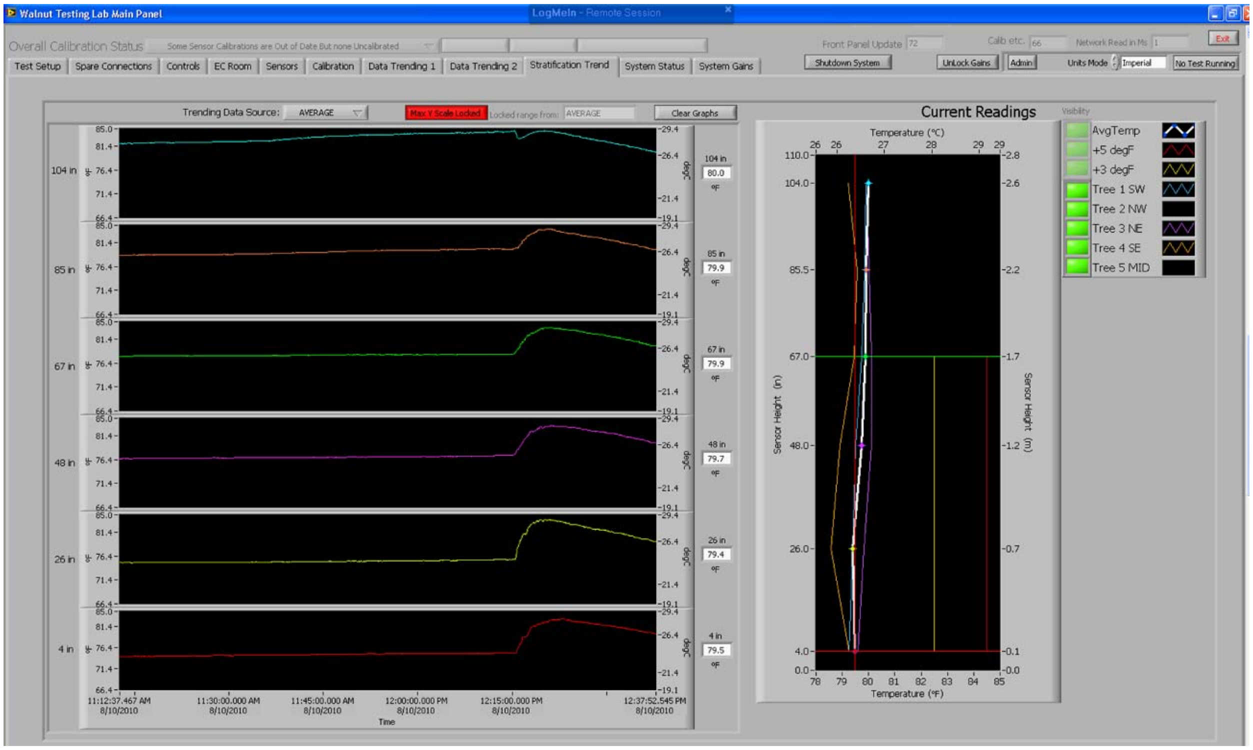
Interior Window Surface WW 36in: 74.5 °F

Interior Window Surface NW 36in: 74.8 °F

West window Exterior 62in: 67 °F

West window Exterior 36in: 66.7 °F

Figure A6: Solar similar control and window surface temperatures



Overall Calibration Status: Some Sensor Calibrations are Out of Date But none Uncalibrated

Front Panel Update: 72

Calib etc.: 66

Network Read in Ms: 1

Units Mode: Imperial

Trending Data Source: AVERAGE

Locked range from: AVERAGE

104 in: 80.0 °F

85 in: 79.9 °F

67 in: 79.9 °F

48 in: 79.7 °F

26 in: 79.4 °F

4 in: 79.5 °F

Figure A7: Stratification trend values and current “tree” values

Appendix B – Simulation specifications

Table B1- Simulation specifications for Energyplus correlation engine

S = sensitivity variable

Y = AND load variation parameter (i.e., exercise to get wide load ranges for regression models and to test significance/influence)

Bold values indicate default baseline values

Item	OH baseline[2] *Standard design*	(real) UFAD Baseline *Proposed design*	EP UFAD/OH para runs	Remarks
General				Specs conform to T-24-2008
Total area	varies	varies		
Stories	**3**	3		
Floorplate size, 1000 ft^2	**20**, 40 60	Same	**Y**	
Aspect	**1.5**	Same		
Service core	no	no		
Climates				Cities with largest number of buildings
Oakland (San Francisco)	Yes	same	**Y**	CZ3: (Oak AP?)
Sacramento	Yes	same	**Y**	CZ12: SAC
Los Angeles	Yes	same	**Y**	CZ6: LA AP
Sizing specs,				
Outdoor, Htg/Clg temps	WME/0.5%	same		
Indoor, Htg/Clg temps, °F	70/73			
Heating design loads	Ltg = std sched, other loads = 0% peak	same		
Cooling design loads, internal (L,P,E)	90% of peak	Same		Interior zones can be oversized up to 33% (reduces chance of messing up load based SAT reset)
Building envelop				
Walls, roof	Climate dependent	Same		
Windows	Fake	Fake		
WWR	**35**,50,65	same	**Y**	

---

[2] This column shows specifications based on ACM for standard design intended for DOE2; for our correlation runs we model as shown to simulate the DOE2 specifications as closely as possible. When we run eQUEST base inputs are values shown here.

Item	OH baseline[2] *Standard design*	(real) UFAD Baseline *Proposed design*	EP UFAD/OH para runs	Remarks
SHGC	Climate dependent	Same		Prescribed by Tables 143
U-value	Climate dependent	Same		Prescribed by Tables 143
Infiltration, cfm/ ft^2-wall	.038	Same		Schedules prescribed by ACM Table N2-8
Internal loads				
Occupancy, ft^2/person (per)	100	Same		
Occupancy, ft^2/person (Int)	100	"		
Occupant, radiant fraction	0.6	"		
Occupant, sensible gain [Btu]	250	"		
Clo, summer	0.5	"		
Clo, Winter	1.0	"		
Activity, W/person	120	"		
Air velocity	0.2	"		
Lighting, load (Per) (W/ ft^2)	0.5,**0.85**,1.2	Same	**Y**	ACM shows lighting inputs in this range depending on compliance methods; tradeoffs could allow even wider range
Lighting, load (Int) (W/ ft^2)[1]	0.5,**0.85**,1.2	Same	**Y**	Table N2-5; (bold indicates baseline)
Lights, Return air fraction	0.0	Same		
Lights, Radiant fraction (LW)	0.32	Same		
Lights, Visible fraction (SW)	0.24	Same		
Equipment, load (Per) (W/ ft^2)	1.0,**1.34**,1.7	Same		Table N2-5: Fixed, no tradeoffs
Equipment, load (Int) (W/ ft^2)[1]	1.0,**1.34**,1.7	Same		"
Equipment, Radiant fraction	0.50	Same		Seems reasonable, middle of HOF recommendations
Thermostat setpoints				
Day/night, cooling, °F	73/77	same		Not sure what happened here: ACM shows 73/77 but standards say deadband must be 5F; those setting just don't make sense so lets use these.

Item	OH baseline[2] *Standard design*	(real) UFAD Baseline *Proposed design*	EP UFAD/OH para runs	Remarks
Day/night, heating, °F	70/65	same		
Internal mass (furniture)				
Mass, #/ ft^2	80	Same		Use CWF for eQuest; wood properties for both ; EP use Full Exterior solar distribution; standard internal mass object
Area coverage	85%	Same		
Interzone walls				
Type	Air walls	same		
U-Factor, Btuh/ ft^2-F	1.0	same		
HVAC				
System type	System 4	UFAD: VSFC		Central chilled water VAV/RH
VAV boxes, Per	VAV reheat	VSFC		
VAV boxes, Int	VAV reheat	VAV reheat		
VAV minimum air flow	20%	20%		For VSFC use dual minimums, 6%, for leakage flows when VSFC is off in deadband, otherwise use 20%
Reheat delta T,° F	40	50		Design discharge to supply air temperature difference
VAV heating, heating max	50%	100% (clg max)		EnergyPlus does not currently support max heating setpoints
AHU, Supply	1	1		
Night cycle	No	No		ACM Chapter 5 for software testing does not use night cycle
Economizer: (type/hi limit)				
Oakland (San Francisco)	DDB: Integrated	Same		Differential dry bulb (DDB) is allowed to be user selected (ref ACM Section 3) therefore not limit is needed.
Sacramento	DDB: Integrated	Same		
Los Angeles	DDB: Integrated	Same		

Item	**OH baseline**^{2} ***Standard design***	**(real) UFAD Baseline** ***Proposed design***	**EP UFAD/OH para runs**	**Remarks**
OSA method	Fixed	Same		
AHU Min OSA, cfm (cfm/ ft^2)	0.15	Same		
Zone min OSA method	Per person	Same		
Zone min OSA, cfm/person	15	Same		
Fan design efficiency, %	0.63	Same		
Fan design static, iwc	4.5	Same		For regression equations only; see below for adjustments
Motor efficiency, %	0.9	Same		
SAT , °F	**55,** 60, 65	55, **60**, 65	**Y**	Reset is prescribed by Section 144, either load based or OSA. Do not assume reset. Run correlations with constant and check later for offsets due to OSA reset.
Supply static pressure reset	Yes	Yes		Accomplished by specifying part load curves representative of static pressure reset performance
AHU Return, Return/relief/none	Relief	Same		Appears to be a choice of the designer
Fan design static, iwc	0.6	"		
Efficiency	0.37	"		
Fan coil Terminal units	VAV/Reheat	VAV/noRH (INT), VSFC (PER)		Modeled as…
Terminal unit static, iwc	NA	0.50		
Terminal unit fan efficiency	NA	15%		Standards require ECMs on all large system terminal units. ACM section 2.5.3.5 require ECMs to be modeled at 50% of full load power. This might be ok for the motor but not for the fan! Use our combined efficiency for now. Its W/cfm = .37 so its low due to low SP
Terminal unit part load	NA	cubic		
Exhaust fans	None	None		

Item	**OH baseline**[2] ***Standard design***	**(real) UFAD Baseline** ***Proposed design***	**EP UFAD/OH para runs**	**Remarks**
Chillers	2@50%	Same		
Type	Cent	Same		
COP	5.5	Same		
Leaving water reset OSA sched,°F	44/80-54/60	same		
Chiller staging	90%			
Pumps	P/S, S-VS	same		
Cooling tower	2-speed	same		
Leaving temp setpoint	70	same		
Boilers	2@50% ea	same		
Type	Forced draft	same		
Efficiency, nominal	0.78	same		
Part load curve	Yes	same		
Supply temp, °F	varies	same		
HW reset OSA sched, °F	20/180-150/50	same		
Staging	90%	Same		
Schedules	Tables N2-8	same		
Adjustments - TBD				Impacts on end use TBD later
Design fan static pressure	**4.5**	2.5, **3.5**, 4.5		Depends on plenum configuration; will study for adjustments after correlation runs; use bold for correlation runs
Stratification	NA	HI RAS		Use 2x diffusers for Hi RAS
System type – York system	NA	Yes		Use min = min ventilation
Economizer type	High limit	Differential DB		Approved option: Included in standard design if applicable. Differential econo for UFAD, high limit for Standard
SAT reset	Yes	Yes		T-24 specifies reset with load based or OSA; ACM says can used fixed as well; both specify use the same for baseline and proposed; load based does not work well and OSA does not save energy in CA climates