Energy Research and Development Division
FINAL PROJECT REPORT

ADVANCED INTEGRATED SYSTEMS TECHNOLOGY DEVELOPMENT

Appendix

Prepared for: California Energy Commission
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University of California, Berkeley

JULY 2013
CEC-500-2014-074-AP

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Additional thanks to the LBNL team of Eleanor Lee, Brian Coffey, Luis Fernandes, and Andrew McNeil for their tremendous contributions on the detailed modeling of the New York Times Building.

PREFACE

The California Energy Commission Energy Research and Development Division supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

The Energy Research and Development Division conducts public interest research, development, and demonstration (RD&D) projects to benefit California.

The Energy Research and Development Division strives to conduct the most promising public interest energy research by partnering with RD&D entities, including individuals, businesses, utilities, and public or private research institutions.

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- Energy Innovations Small Grants
- Energy-Related Environmental Research
- Energy Systems Integration
- Environmentally Preferred Advanced Generation
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- Renewable Energy Technologies
- Transportation

The *Advanced Integrated Systems Technology Development* is the final report for contract number 500-08-044, conducted by The Center for the Built Environment, University of California, Berkeley. The information from this project contributes to Energy Research and Development Division's Buildings End-Use Energy Efficiency Program.

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ABSTRACT

To achieve the radical improvements in building energy efficiency being called for by the State of California, it will be necessary to apply an integrated approach involving new designs, new technologies, new ways of operating buildings, new tools for design, commissioning and monitoring, and new understanding of what comprises a comfortable and productive indoor environment. All of these themes define important goals that have guided the broad and comprehensive research effort described in this report. Research methods have included field studies, laboratory studies, energy and thermal comfort modeling, and technology transfer through participation on American Society of Heating, Refrigerating and Air Conditioning Engineers standards and technical committees. Center for the Built Environment research is also guided by its 40 industry partners, who serve as the project advisory board for the project.

The work done under this project has advanced the understanding of new and innovative approaches to space conditioning in buildings featuring integrated design with combined low-energy systems. The research has generated the following findings, new tools and modeling capabilities, and recommendations: (1) lessons learned from three case studies of advanced integrated systems, (2) new guidelines for design, performance, and control of underfloor air distribution, radiant, and personal comfort systems from simulation studies, (3) updated software and improved guidance for simulation of underfloor air distribution, radiant and personal comfort systems in EnergyPlus, (4) advancement of personal comfort system technology to the field demonstration stage through the development and fabrication of several prototype personal comfort system devices, (5) a building performance evaluation toolkit based on wireless sensing and web-based analysis applications and data archiving, (6) guidelines for the development of building performance feedback systems (energy dashboards) that encourage building operators and occupants to reduce energy use, (7) an updated advanced Berkeley thermal comfort model, and (8) important updates to American Society of Heating, Refrigerating and Air Conditioning Engineers Standard 55 that support advanced integrated systems and significant contributions to other American Society of Heating, Refrigerating and Air Conditioning Engineers guideline documents.

Keywords: field studies, laboratory studies, energy simulation, building energy use, thermal comfort, thermal comfort modeling, integrated systems, underfloor air distribution (UFAD), radiant systems, thermally activated building systems (TABS), personal comfort systems (PCS), energy dashboards, wireless sensing, building standards

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RADIANT COOLING SLAB SYSTEM DESIGN, CONTROL AND APPLICATION

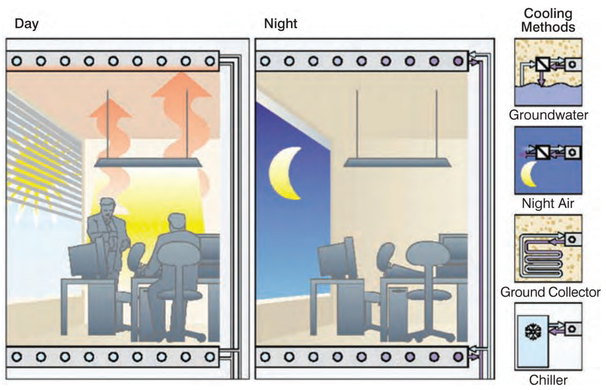
Student: *Jingjuan (Dove) Feng,* Advisors: *Stefano Schiavon, Fred Bauman, Gail Brager*

Radiant cooling slab systems are specifically designed to integrate building elements in the overall energy strategies, where building structure is cooled down by circulating water through plastic tubing embedded in the ceiling or floor slabs. To improve understanding of applications, design, and control optimization of the systems, the Center for the Built Environment (CBE) is conducting an ongoing study of the David Brower Center, Berkeley, which has an exposed radiant ceiling slab.

Radiant Cooling Slab Systems

Concept

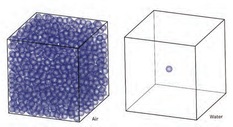
Radiant cooling slab systems are an embeded water based surface cooling system.



(Source: Olesen, B. (Feb, 2012))

Advantages

- Energy efficiency
- Occupant thermal comfort
- Integrated architectural design



Challenges

- Limited cooling capacity
- Acoustic quality
- Condensation concerns
- Unfamiliarity with design and control

Example Buildings

Zollverein School of Management and Design

- Kazuyo Sejima + Ryue Nishizawa, SANAA

Kunsthaus Bregenz

- Peter Zumthor

Sudwestmetall Regional Headquarters

- Dominik Dreiner Architekt

Case Study: David Brower Center

Design Team

Solomon E.T.C. – WRT
Integral Group
Tipping Mar + associates

Loisos and Ubbelohde
Siegel & Strain Architects

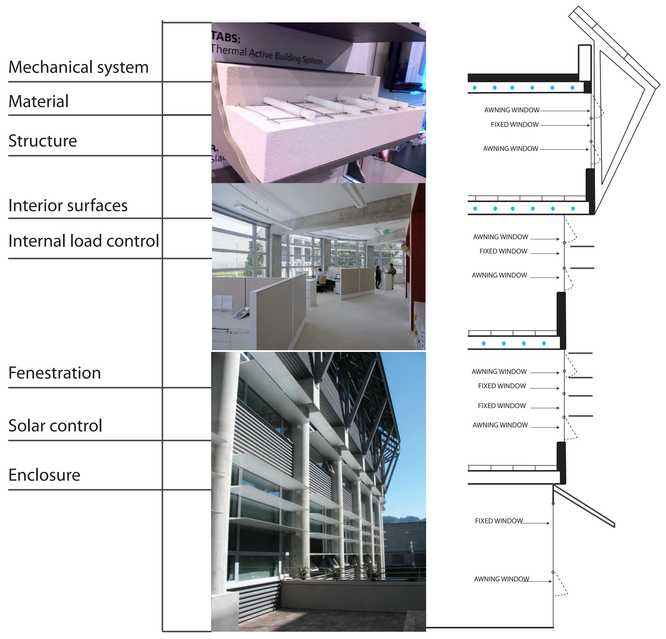
Building

New, 4-story, 43,000 ft2 office and public space

Features

LEED platinum building
Radiant slab integrated with UFAD

Integrated Design

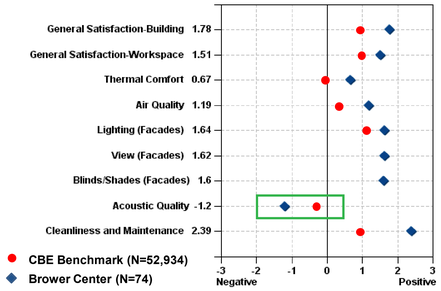


Field Study

- To assess occupant satisfaction with the building
- To analyze the energy consumption
- To evaluate and improve the controls and operation

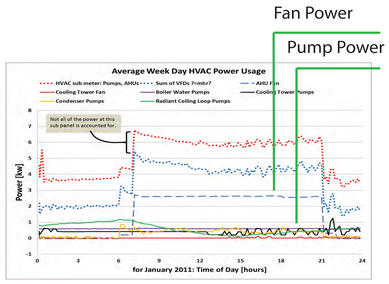
Post Occupancy Survey

- Positive feedback for most categories
- Dissatisfaction with noise privacy

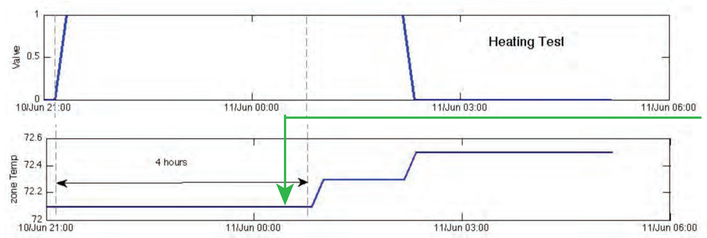


Energy Performance

Performance Metrics	Current (Ending 6/30/2010)	ENERGY STAR Label	National Average
ENERGY STAR Rating	99	75	50
Energy Use Intensity			
Site (kBtu/ft^2)	47	109	147
Source (kBtu/ft^2)	68	157	212



Control Performance



Slow response: after the valve open for 4 hours, zone temperature start to increase

Reference

1. Olesen, B. (Feb, 2012) Using building mass to heat and cool. Ashrae Journal, Feb. 2012: 19-24.
2. Kiel, M. (2010). Thermally activated Surfaces in Architecture. Princeton Architectural Press
3. Olesen, B. (July, 2001). "Radiant Floor Heating in Theory and Practice." Ashrae Journal: 19-24.
4. Watson, R. D. and K. S. Chapman (2002). Radiant Heating and Cooling Handbook, McGraw-Hill.

DAVID BROWER CENTER: THERMAL COMFORT AND RADIANT PERFORMANCE STUDY

July 2011

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Darryl Dickerhoff: CBE/LBL
David Heinzerling: CBE
Dove Feng: CBE
Frank Chuang: UC Berkeley
Ryan Miller: David Brower Center

Center for the Built Environment

CBE

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BACKGROUND

Over the course of four weeks (May 24, 2011 – June 24, 2011), researchers from CBE worked at the David Brower Center to study the thermal performance of the building. There were three main goals in this study: (1) help verify and assist with known thermal comfort problems in the building, (2) pilot the portable wireless measurement system, primarily the ICMs and mobile UFAD cart, and (3) collect data on radiant slab surfaces for use in a modeling study. This report highlights findings related to the first goal from the fourth floor measurements, primarily for use by the Brower Center facilities team.

RESULTS

The northwest side of the 4th floor was monitored from 6/14 – 6/22. There is a known thermal comfort problem on the fourth floor caused by the fact that a single thermostat is controlling the entire north-facing perimeter zone (radiant ceiling slab) that includes several separate tenant spaces with different heat loads and cooling requirements. This report looks at this issue from two related angles—temperature and pressure.

Underfloor plenum pressure

A known issue in the Brower center is the low plenum pressure in the east zones which are served by AHU2. For comparison, we did measurements on both east and west parts of the building, and the west is served by AHU1 that runs normally at 50% of its full capacity. Figure 1 shows an overview of the pressure measurements that were taken in the various zones on each floor. The pressures at the east end are in a range of 0.002 to 0.004 in. wc, while for the west end the pressures are from 0.01-0.02 in.wc, which are in the normal range for an Underfloor Air Distribution (UFAD) system. Note that the plenum pressures in the east end of the building are nearly one order of magnitude less than those in the west end.

Low pressure indicates low fresh air supply to the spaces. The east zones shared fresh air supply from AHU2 with restaurant Gather, and with the kitchen hood running in the restaurant, a large amount of air goes to the restaurant instead of the underfloor plenum of the offices. In addition, controlled by CO_2 level, the underfloor air dampers controlling the volume of air delivered to the underfloor plenum are mostly at minimum position, thus further increasing the resistance of air going to the office spaces. So even though AHU2 is running at its maximum 100% speed, the east zones are still starved for air.

The consequences of reduced fresh air supply from the UFAD system include degraded indoor air quality and potential thermal comfort issues. We measured the CO_2 level in an office to be around 700-800 ppm, and we also heard about stagnant air experience from some tenants during our visit.

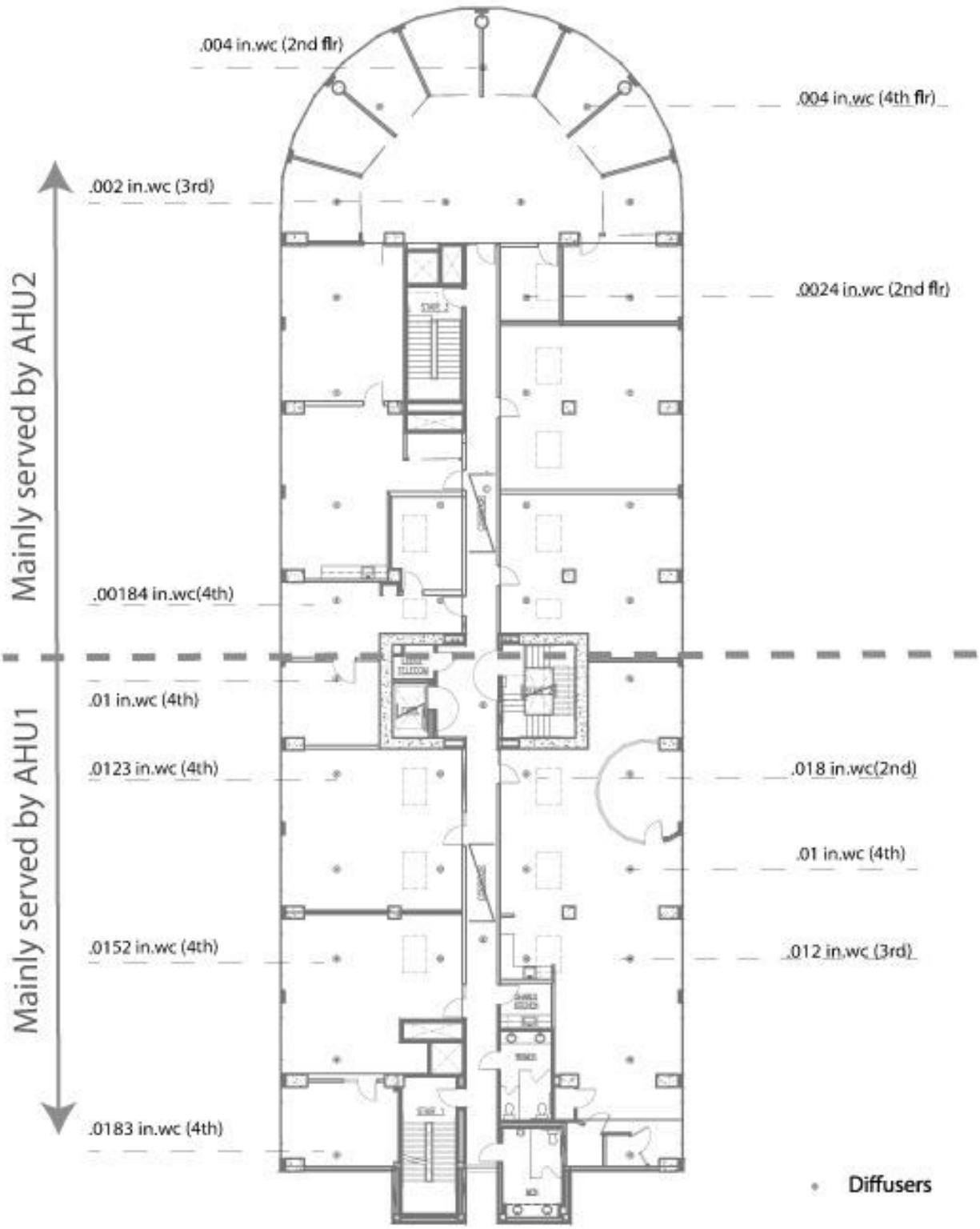

Figure 1: Map of a typical floor plan for floors 2-4 showing pressure measurements taken at various locations on each floor

Underfloor plenum temperature

Figure 2 is an illustrated thermal contour map of the fourth floor underfloor plenum. The white dots represent wireless mote locations where temperatures were taken. The data shown represent a snapshot of the plenum temperatures at a moment in time. Temperatures were linearly interpolated to provide a contour map.

Thermal decay in the underfloor plenum depends on airflow rate and temperature distribution caused by how the supply airflow is delivered to the plenum. Temperatures are typically dictated by how long the air has to travel (while picking up heat) to a particular location. There was limited variation in temperature in the underfloor plenum for the fourth floor, which is consistent with the fact that the air is largely unconditioned. The results do indicate that warmer plenum temperatures occur toward the eastern end of the building. This is consistent with the ongoing issue with AHU2 (the eastern air handler), which is currently providing a lower amount of air into the underfloor plenum due to shared resources with the restaurant Gather. Lower airflows will tend to produce higher plenum temperatures.

Temperature trending

One way to understand the thermal comfort in a particular zone is to look at the dry-bulb temperature (DBT) in conjunction with the active slab surface temperature over an extended period of time. We used the indoor climate monitor (ICM) to measure DBT and surface mounted temperature sensors to measure the surface temperature of the radiant ceiling slab in various zones.

A gridded map of the fourth floor is shown in Figure 3. The charts in the rest of this report refer to measurement locations indicated with letter/number combinations that can be found on this map. The outlined zones represent the radiant control zones. The highlighted squares are color-coded to correspond to the line colors in Figures 4-5.

Figure 4 and Figure 5 display ceiling surface and DBT trends in three different areas of the fourth floor north side spaces in which there were known thermal comfort issues. The results confirm that there exists a large difference in air temperature (3-4°F) between the warmer east space (C18) and the cooler west space (A4). Complaints of overly warm conditions from the east space (C18) are consistent with the temperature data taken. However, the other two zones measured (C12 and B4) fell within a reasonable comfort range for this testing period. While the west space (B4) has previously complained of overly cold temperatures, it is hard to validate these concerns for this week of testing, during which weather conditions were considerably warmer than in previous months.

Figure 5 highlights two days in which pre-cooling was enabled in the radiant ceiling. When the ceiling slabs are left to float, the surface temperatures between the east and west side of the same radiant zone is significant (~4°F); however, while the slab is active, the temperatures align within ~1-2°F of one another as expected. All three tenant spaces are controlled by the same single zone of the radiant slab, which results in poor control for highly differentiated spaces (occupant density, plug loads, etc.).

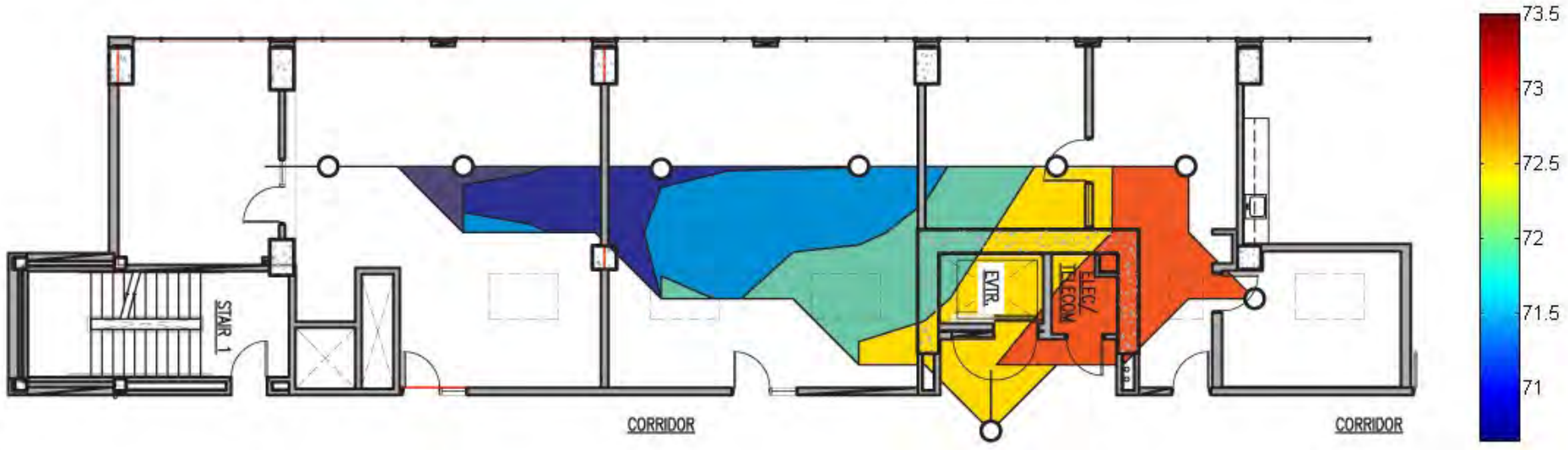

Figure 2: Northwest 4th floor underfloor plenum temperature distribution (6/15/2011 2:00 PM)

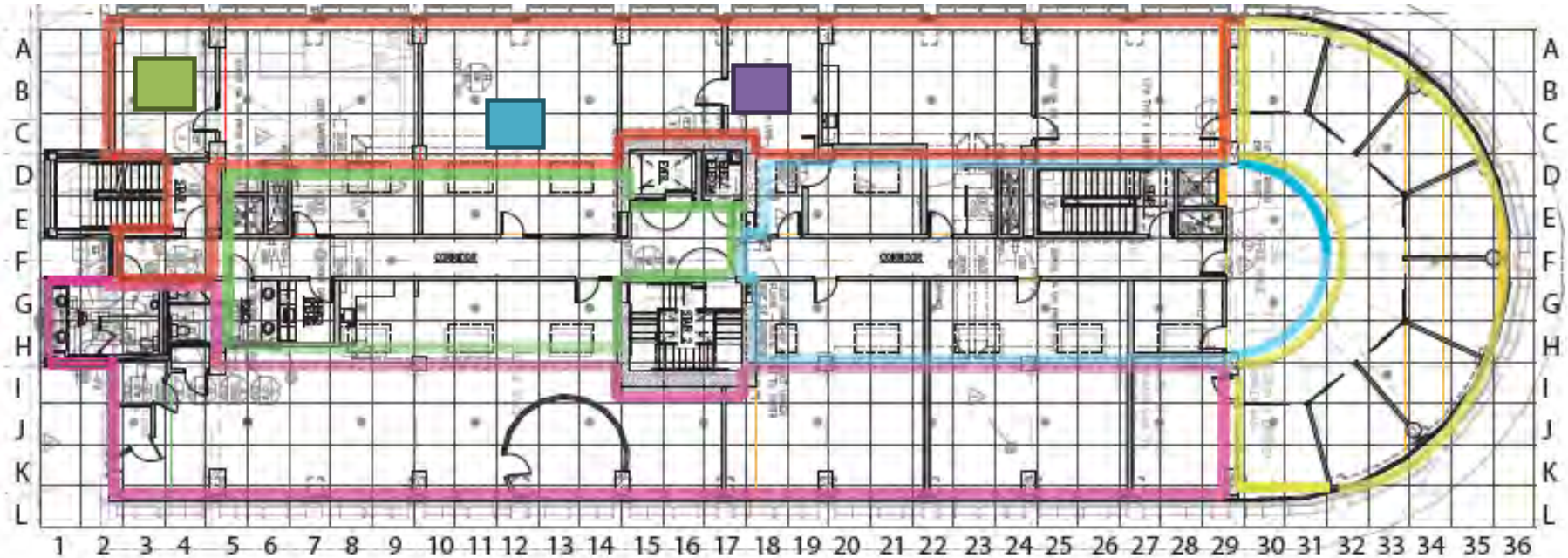

Figure 3: Map of 4th floor – grid locations, radiant slab zones, and color-coded zones of interest that refer to Figures 5-6

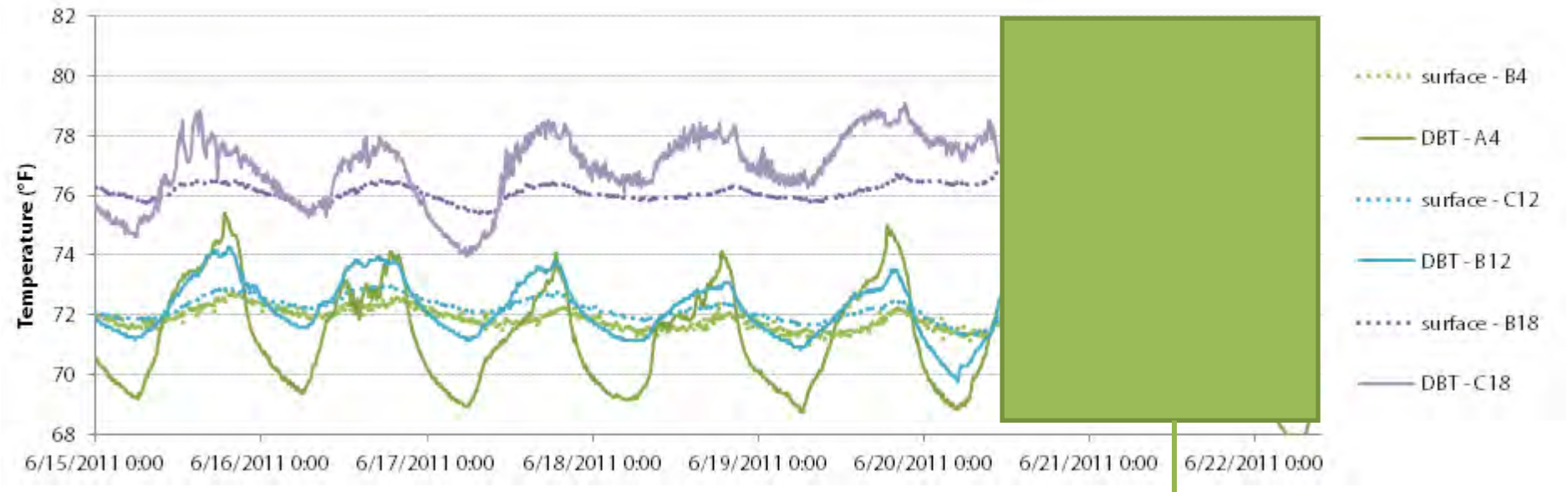

Figure 4: One week of ICM/surface readings for three areas in the northeast side of the 4th floor

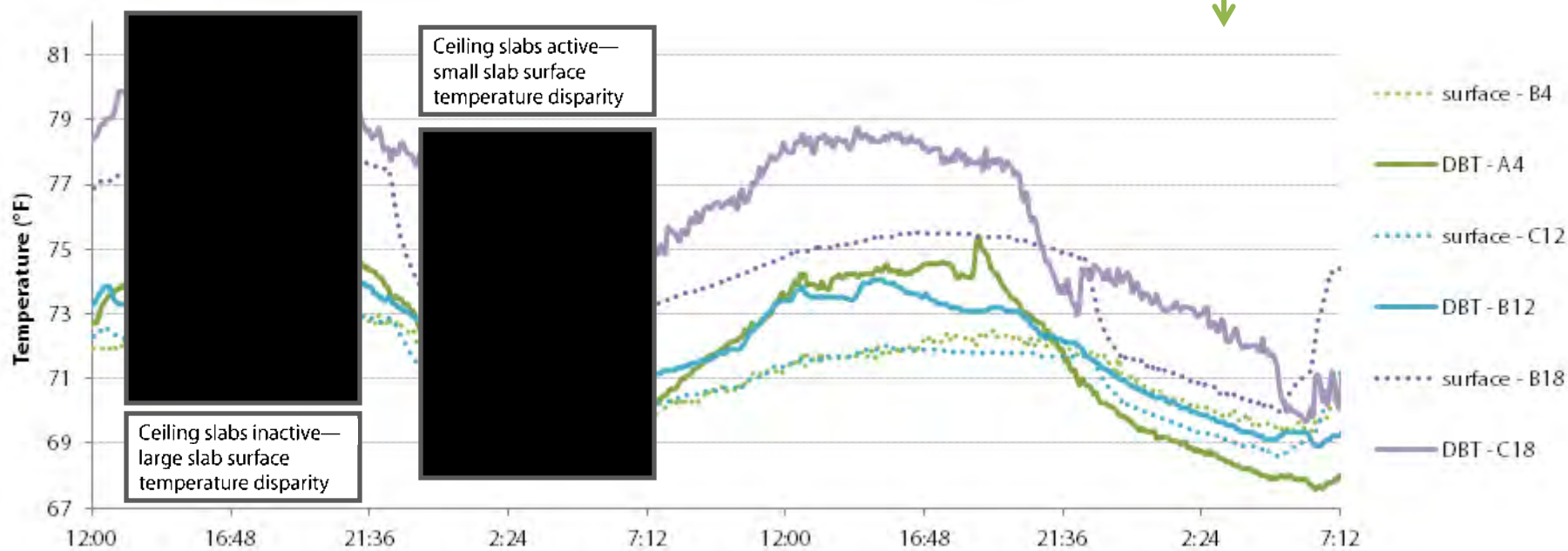

Figure 5: One day of ICM/surface readings for three areas in the northeast side of the 4th floor

Stratification and occupied zone temperature

UFAD systems are commonly controlled using a thermostat mounted at a 4 ft height. If significant stratification develops, the occupied zone ((OZ) defined as the vertical region from 4 inches to 67 inches from the floor) may end up being too cool and cause discomfort. The degree of stratification is gauged by the difference between the temperatures at 67 and 4 inches (namely the head-foot temperature difference). We generally consider stratification optimal when it is in the range of 2 to 4°F. As a single value, stratification serves as a good indicator of UFAD performance, though the full stratification profile (air temperature at eight room heights) provides a clearer picture of UFAD temperature performance. Because the Brower Center's UFAD system is used primarily for ventilation rather than cooling, typical stratification profiles are not necessarily expected as a measure of system efficiency and performance.

The north side of the 4th floor was monitored from 6/14 – 6/22. Figure 6 and Figure 7 summarize the thermal stratification of the measured areas in the fourth floor. While the shapes of the profiles are similar, the profile in the east space (C18) is shifted significantly to the warmer side with an average occupied zone temperature above 75°F for this particular measurement time (2:40 PM). These results demonstrate again the significant temperature difference that exists between the eastern space (C18) and the western-most space (A4 and B7) on the 4th floor. To improve the system's ability to respond to these observed differences in room temperature, it may be advisable to install a second thermostat in the west space (A4 and B7) on the 4th floor.

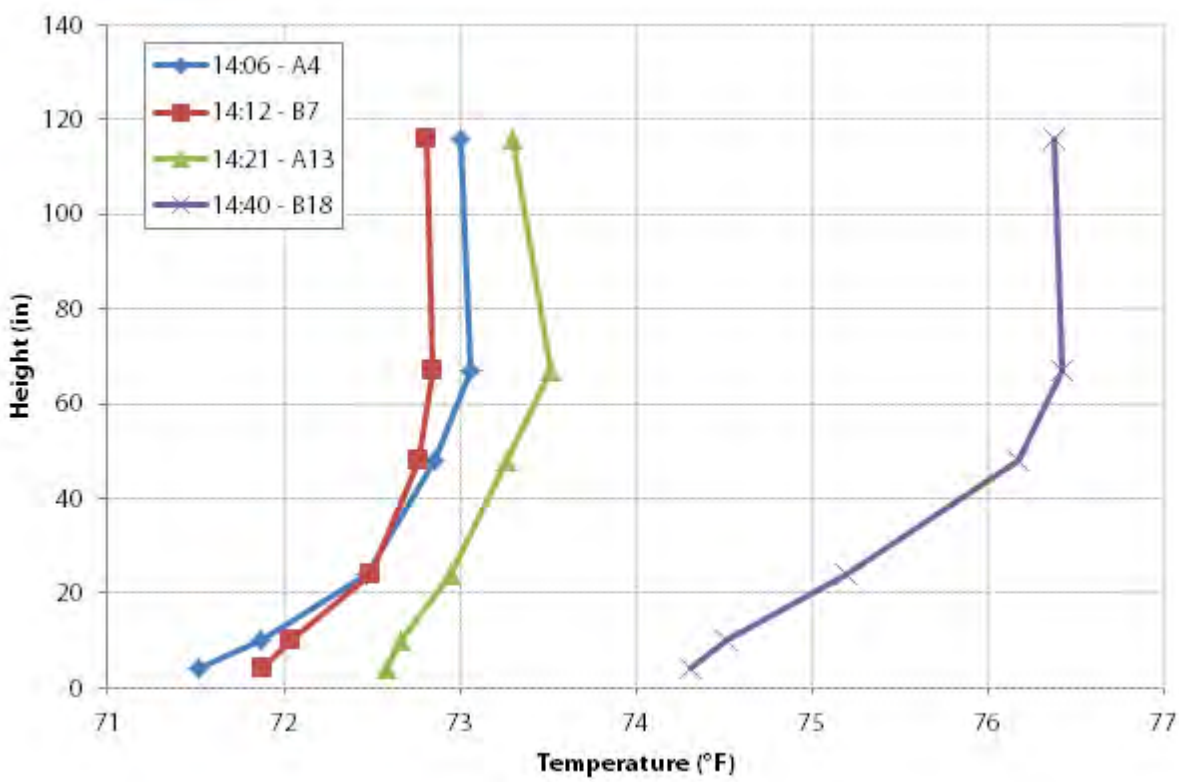

Figure 6: Room air stratification profiles for various locations on 4th floor

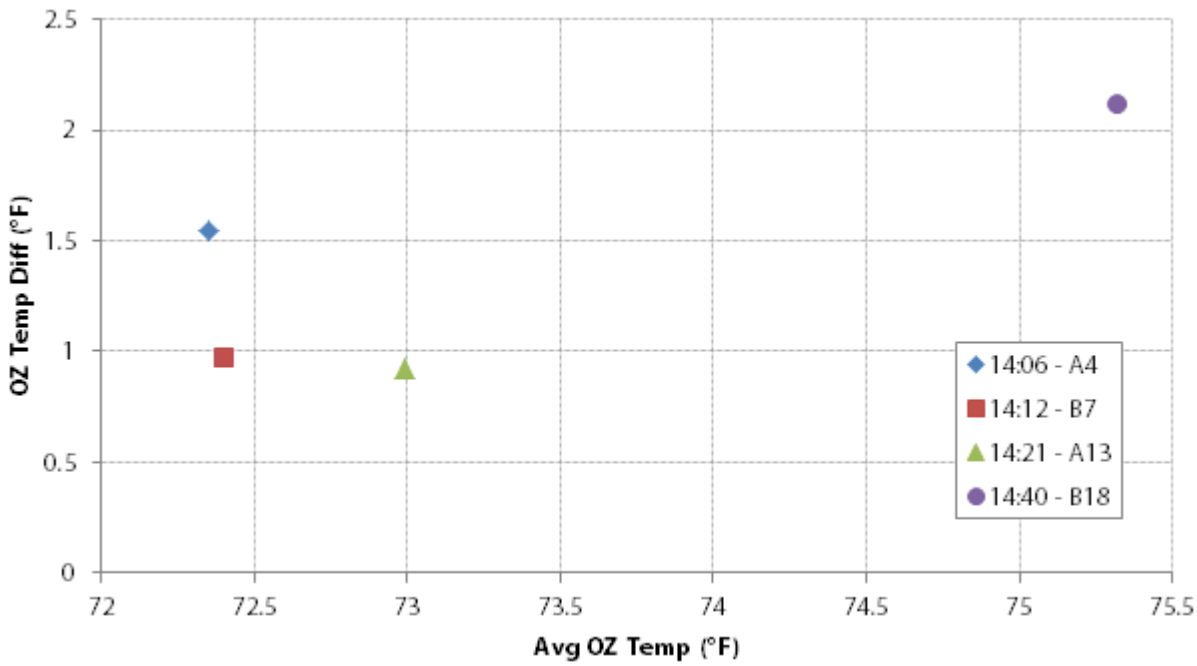

Figure 7: Occupied zone stratification vs. average occupied zone temperature on 4th floor

Stratification trending

In addition to the spot measurements taken with the mobile UFAD cart, a long term stratification profile trend was generated for one location over the period of 6/14 – 6/22 for the fourth floor.

These trends show the spread between the floor and ceiling surface temperatures and how the occupied zone temperatures vary over the course of a week and a day. Figure 8 and Figure 9 show that the range of occupied zone temperatures for this particular location on the fourth floor was ~3°F over the course of a day, showing rather stable space temperatures, albeit on the colder side.

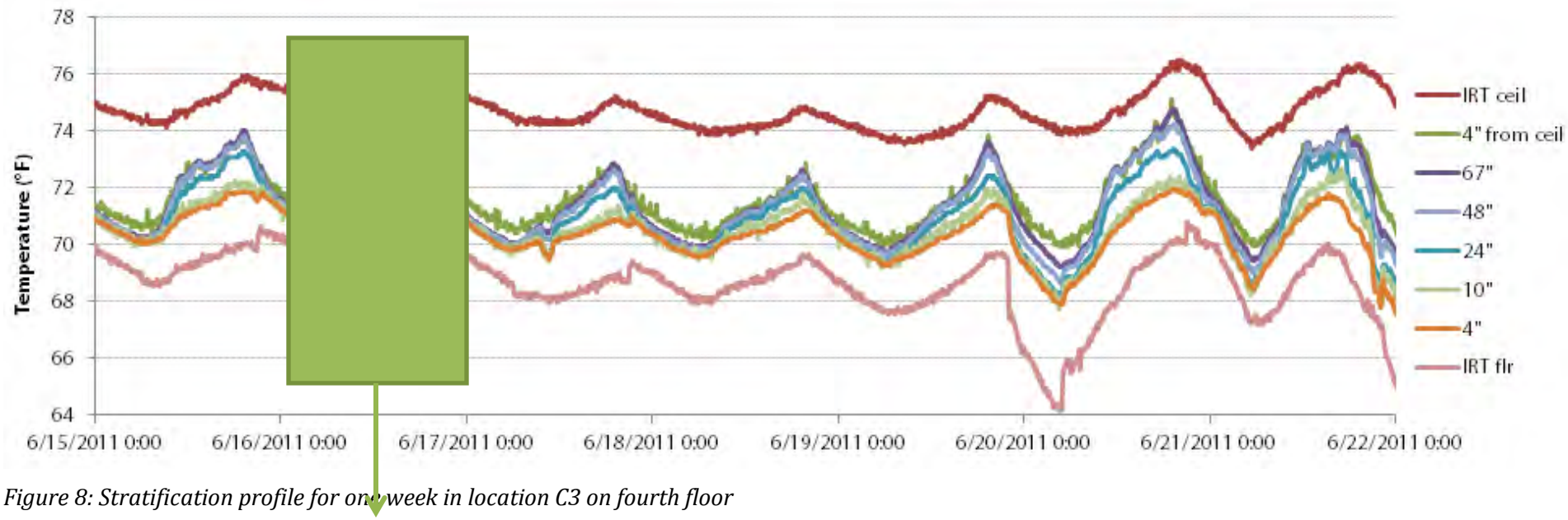

Figure 8: Stratification profile for one week in location C3 on fourth floor

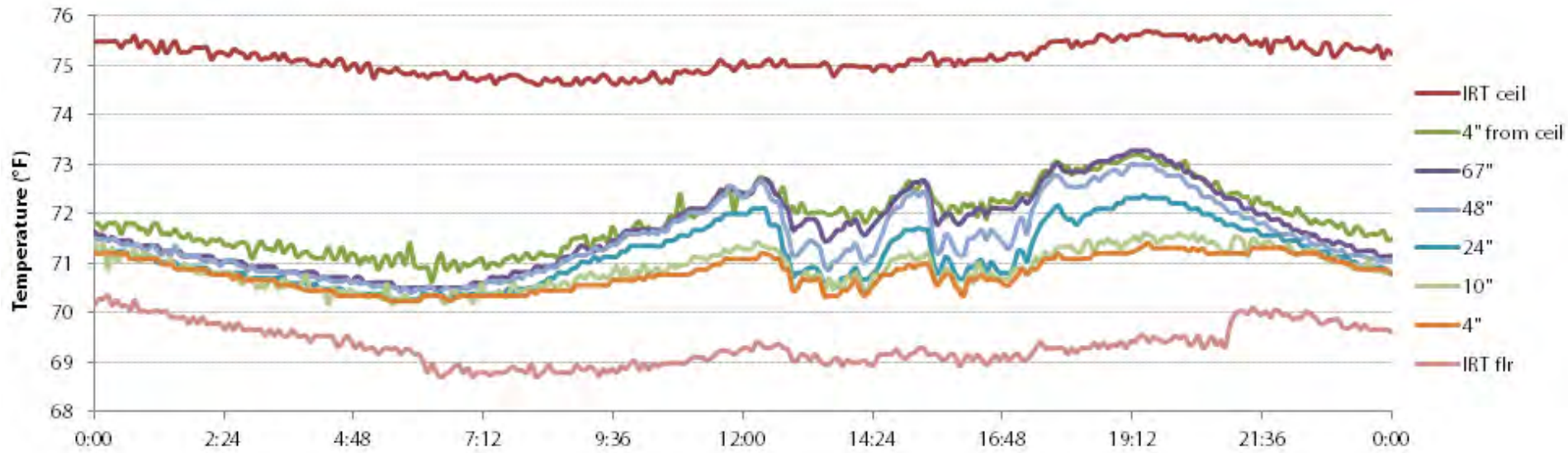

Figure 9: Stratification profile for one day (6/16) in location C3 on fourth floor

CONCLUSIONS AND RECOMMENDATIONS

The measurements taken on the fourth floor confirm the known temperature and pressure issues and provide a detailed look at the thermal comfort conditions for the tenant spaces on the northwest side of the building. We offer the following recommendations:

- Install a new thermostat in the perimeter area (within about 10 feet of the exterior windows) of Suite 420 (B7) on the fourth floor and connect to the building management system (BMS). This will provide a direct measure of the room air temperature in that tenant space and will also allow increased flexibility for controlling the temperature of the north-facing perimeter radiant slab that spans the entire length of the building (see Figure 3). Modifications will also need to be made to the Metasys programming that controls the radiant slab temperature. With two thermostats available to control the north perimeter zone on the 4th floor, options include controlling the slab temperature based on (1) thermostat in Suite 460 (existing controls), (2) new thermostat in Suite 420, or (3) average of two thermostats. It may take some trial and error to determine the best control solution for this radiant slab zone.
- Fix the make-up air issue with AHU2 in order to provide more air to the eastern half of the building. Apparently, work is now underway on this issue through negotiations with the restaurant Gather. By increasing the amount of outside air delivered by AHU2 into the Brower Center offices on floors 2-4 (and coincidentally reducing the amount of air delivered by AHU2 to the restaurant Gather) the following benefits can be realized in the eastern half of the building: (1) improved indoor air quality and reduced symptoms of stagnant or stale air, (2) potential to improve the warm thermal comfort issues experienced by tenants in some areas of the eastern end of the building, and (3) reduced energy consumption by AHU2, which is currently running at 100% speed during most of its operational time.

Internal Report August 2011

Technical Report on California State Teachers Retirement System Building: UFAD Performance and Blinds Study

Tom Webster, Fred Bauman, David Heinzerling, Stefano Schiavon, Darryl Dickerhoff,
Center for the Built Environment (CBE)
University of California, Berkeley

Dave Troup
HOK – Mechanical Engineer

David Hill
Able Services – Building Manager

August 1, 2011

1. BACKGROUND

For the week of July 19-23, 2010 members from CBE traveled to the California State Teachers Retirement System (CalSTRS) building in Sacramento with two main goals in mind: (1) help validate the CBE UFAD design tool with measured building data and (2) evaluate the performance of the CalSTRS UFAD system during different blinds setting scenarios (open, horizontal, and closed). During certain times of year, CalSTRS building engineers have had trouble meeting setpoints because of solar loading. Two spaces were monitored: (1) a conference room on the 3rd floor and (2) an open plan space on the 11th floor.

This project was funded by CBE and the Public Interest Energy Research program.

2. METHODS

Third floor conference room

In the 3rd floor conference room, pressure measurements were taken to determine the leakage between the isolated plenum space of the conference room and adjacent rooms and plenums. Swirl diffusers were covered and bar grille diffusers were disconnected from ducts to simulate actual plenum environment. Air flow was measured from each bar grille diffuser and correlated with plenum pressure. Wireless temperature sensing motes (wireless devices associated with the CBE portable measurement cart) were deployed in three diffusers, as well as the supply air entrance into the conference room and the supply air exit from a duct into the plenum space. Temperature stratification and plenum pressure were recorded each day and

evening by the portable measurement cart. One pyranometer was setup on the bottom window sill in the southwest corner of the conference room to record transmitted solar radiation. Another pyranometer was setup on the roof to record environmental solar radiation for the entire testing period. The rooftop pyranometer malfunctioned, so data is only available from the 3rd floor pyranometer. Two days of testing were done with blinds up and one day of testing was done with blinds closed. An infrared gun was used to measure the temperature of the floor with and without direct sunlight.

11th floor open plan space

In the 11th floor open plan office space, plenum pressure was measured with both the portable cart and a wireless mote connected to a pressure sensor. Wireless motes were placed in swirl diffusers throughout the southeast plenum section of the open plan space and at the two ducts that supply the plenum to evaluate thermal decay in the plenum. One wireless stratification tree was placed on the interior and one was placed near the perimeter to record temperature stratification in addition to the cart measurements. The portable cart was moved to various locations within the open plan space to evaluate the stratification of the space. One day of testing was done with blinds in the horizontal position and one day with the blinds open.

3. RESULTS

UFAD temperature performance

Thermal stratification was measured on both the third and eleventh floors using the portable measurement cart in order to assess the effect of blind position on thermal comfort and the ability of the UFAD system to satisfy the cooling load. UFAD systems are commonly controlled using a thermostat mounted at a 4 ft height. If significant stratification develops, the occupied zone (OZ) defined as the vertical region from 4 inches to 67 inches from the floor) may end up being too cool and cause discomfort. The degree of stratification is gauged by the difference between the temperatures at 67 and 4 inches (namely the head-foot temperature difference). We generally consider stratification optimal when it is in the range of 2 to 4°F. As a single value, stratification serves as a good indicator of UFAD performance, though the full stratification profile (air temperature at thirteen room heights) provides a clearer picture of UFAD temperature performance.

Third floor conference room

Figure 1 shows the room air stratification profiles for the third floor conference room during two different times (8:00am and 11:00am) and blinds positions (open and closed). The solid lines represent the 8:00am measurements and the dotted lines represent the 11:00am measurements. The red lines refer to the “blinds closed” scenario and the blue lines refer to the “blinds open” scenario.

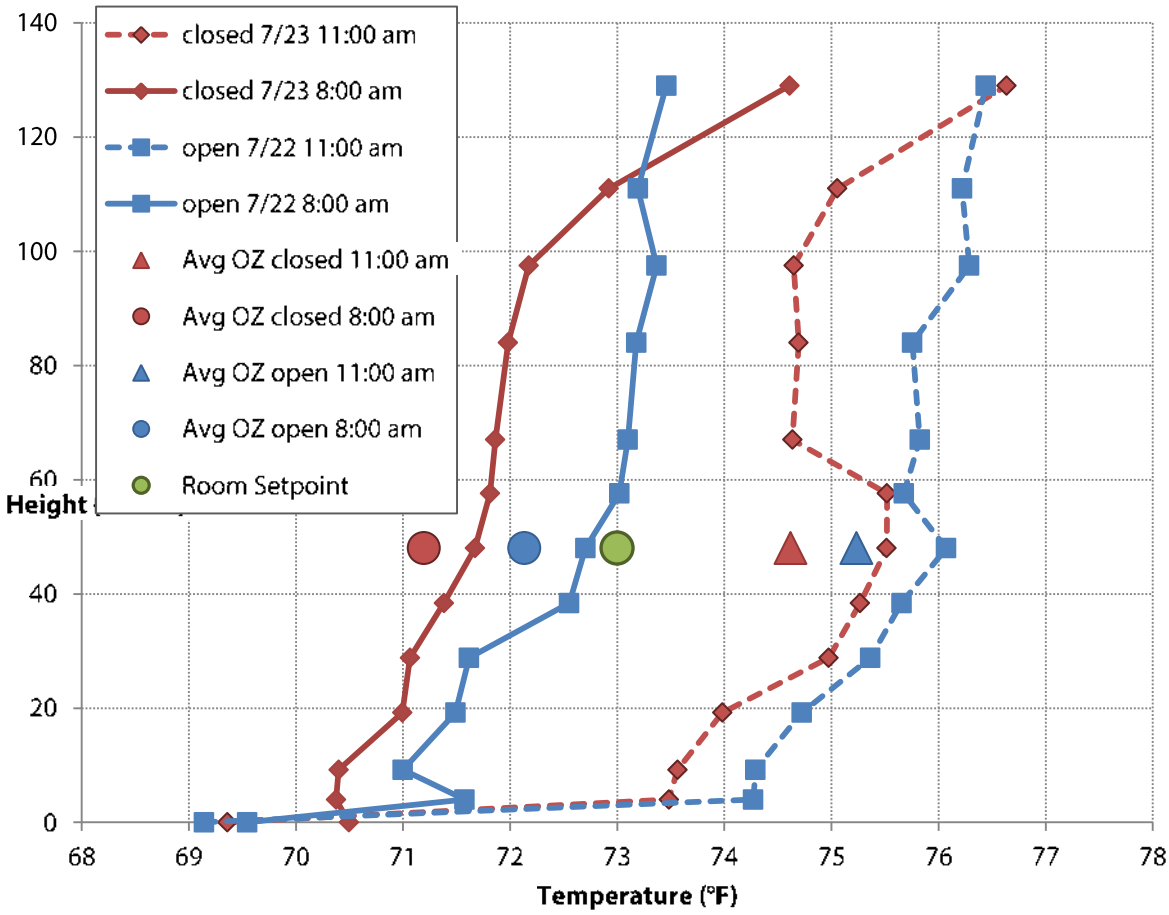

Figure 1: Thermal stratification profiles for blinds scenarios in third floor conference room

There are multiple points of interest in this chart that are discussed in the sections that follow.

Blinds position effect on room temperature

The stratification profiles for the "blinds open" scenario are warmer than the stratification profiles for the "blinds closed" scenario, indicating a small cooling effect of the blinds (~0.6 – 1.2°F). Figure 2 shows the solar load entering the room as measured by a pyranometer that was placed on a southeast window sill (unaffected by blinds position). The solar load on the space was nearly identical on the two days of interest (7/22 and 7/23); the small difference at the peak we assume has negligible impact on the results. This suggests that the cooling effect observed was related to the blinds position. The peak solar load occurred at ~9:30am on both days but there was little direct solar gain due to the orientation of the glazed walls.

The blue/red circles and triangles in Figure 1 represent the average occupied zone temperatures for the corresponding scenarios, and the green circle represents the space setpoint (73°F). The measurements indicate that the space is cooler than setpoint at 8:00am and that by 11:00am the system is unable to meet setpoint. As noted by the facilities managers, the evening purge needed adjustment in order to prevent the necessity of a morning heating period.

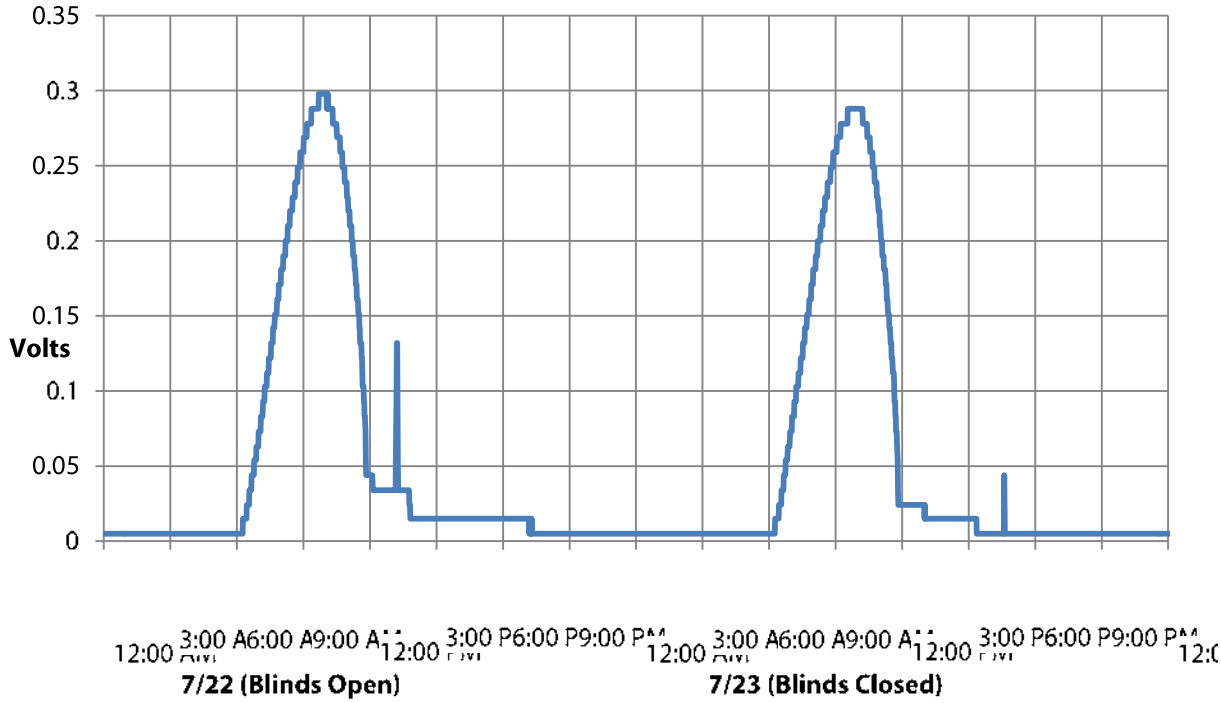

Figure 2: Solar radiation measurements

Blinds position effect on fan terminal unit airflow

Figure 3 shows building management system (BMS) data (fan terminal unit command airflow, space temperature, setpoint, and supply air temperature) for the corresponding two-day time period. Supply air temperature (SAT) (FTU discharge temperature in BMS) and setpoint are only shown for the occupied time in order to reduce clutter on the chart and enhance clarity. The full two day trends of SAT and setpoint are shown in Figure 4.

Despite the slight cooling effect associated with closed blinds, we do not see an associated reduction in fan energy during the occupied hours of the day when the system is operational. The fan terminal unit operates the same amount of time above its minimum setting during the closed blinds scenario and the open blinds scenario (8.5 hours). However, the relationship between fan terminal unit energy and blinds position is confounded by the differing supply air temperature behaviors between the two days. During the blinds open day, the supply air temperature drops from 69°F to 65°F starting at 3:00PM while the same drop in supply air temperature happens at 4:15PM on the blinds closed day. This appears to be the result of AHU SAT resetting.

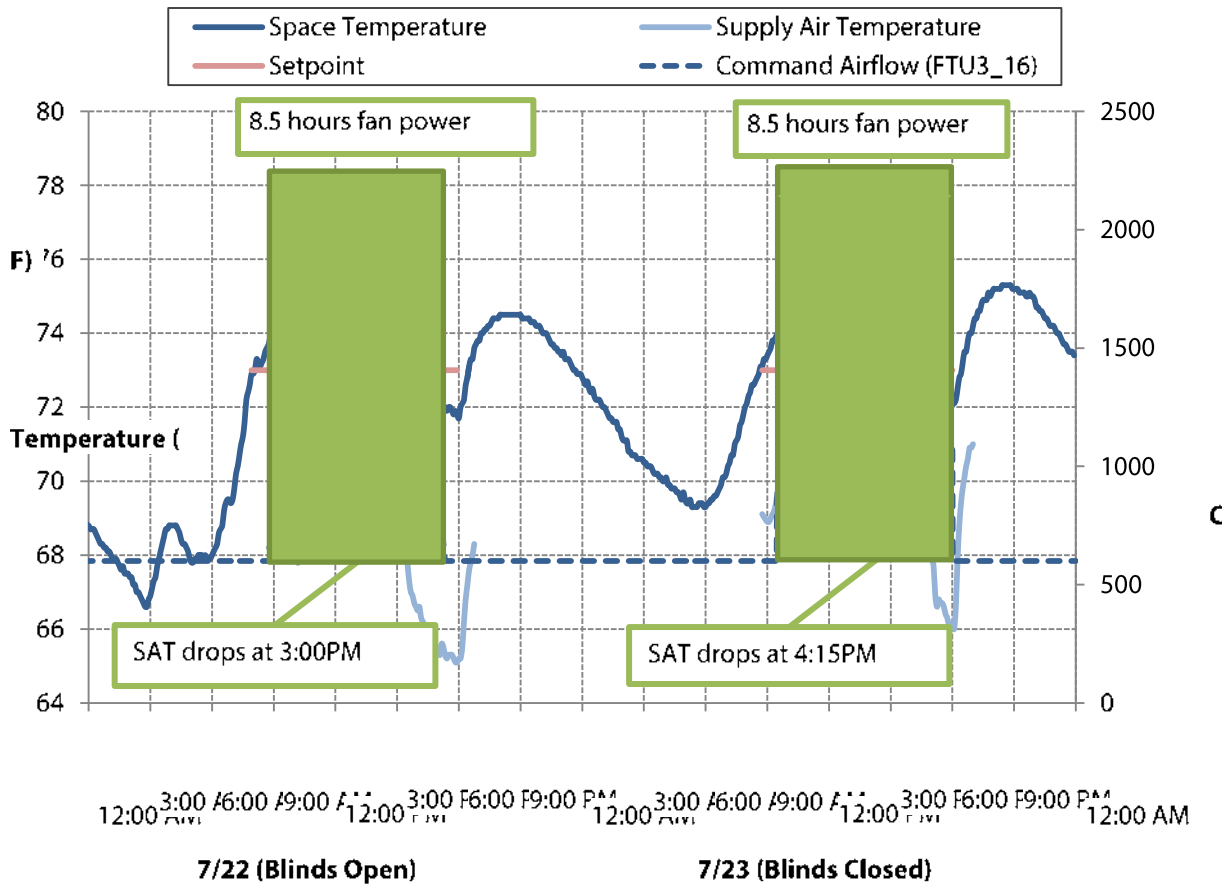

Figure 3: Airflow and space temperature BMS data for third floor conference room

Figure 4 shows space, supply, and setpoint temperatures for the corresponding two day period. The early morning of 7/22 is characterized by spikes in the supply air temperature that correspond to working out kinks in the morning warmup routine. Because the conference room is located in the southeast corner of the building, the solar load peaks in morning and there is no direct sun in the afternoon. The supply air temperature drops from 69°F to 65°F between 3:00-6:00PM, which in combination with the high airflow, leads to overcooling in the afternoon, at least for this zone (the supply air reset could be activated by other factors outside of this zone). This whole sequence is complicated by multiple changes occurring at the same time and the fact that the box is undersized (or maximum flow settings are too low) resulting in loss of room temperature control in the early part of the day. Then in the latter part of the day when the room temperature drops, the box airflow does not reduce fast enough to account for the rapidly decreasing room temperature that is occurring due to the combined effect of the lower SAT and box high (but decreasing) airflow. This situation could be mitigated by better sizing of the boxes and tuning of the controls to respond more rapidly to effects such as the SAT reset. Perhaps the AHU SAT reset is also too aggressive. The setpoint temperature (red-dotted line) currently follows a traditional setback routine (73°F occupied times and 75°F unoccupied times). We recommend raising the setpoint to 75°F along with the previous suggestions to gain control of the zone. Additionally, the room setpoint steps down to 73°F from 1:40AM – 2:15AM, which is likely a programming mistake, which is accompanied by an unnecessary drop in supply air temperature.

Due to the inconsistency in supply air temperature behavior, and the fact the zone is not being controlled well for much of the day, we are unable to assess the effect of blinds position on fan box energy consumption over the course of the day. However, it is clear from the data that when the blinds are closed the load is lower in the room resulting in lower room temperatures. This is consistent with our hypothesis that the solar gain is concentrated at the windows and doesn't enter the space itself. Simulations have shown that for mixed systems (normal overhead VAV) the overall load is the same when low-e windows are used, but for UFAD systems there is greater stratification near the windows so the solar load is "siphoned" up to the ceiling thus reducing the impact on the zone load. This behavior is clearly shown by the large increase in temperature at the top of the profile when blinds are closed. The reason for the odd shape of the 11:00am profiles is not clear. It suggests that the measurement tree is being impacted by some factor; this sometimes occurs when airflow from the diffusers bounces off the ceiling, lowering the temperature in the upper part. (See below).

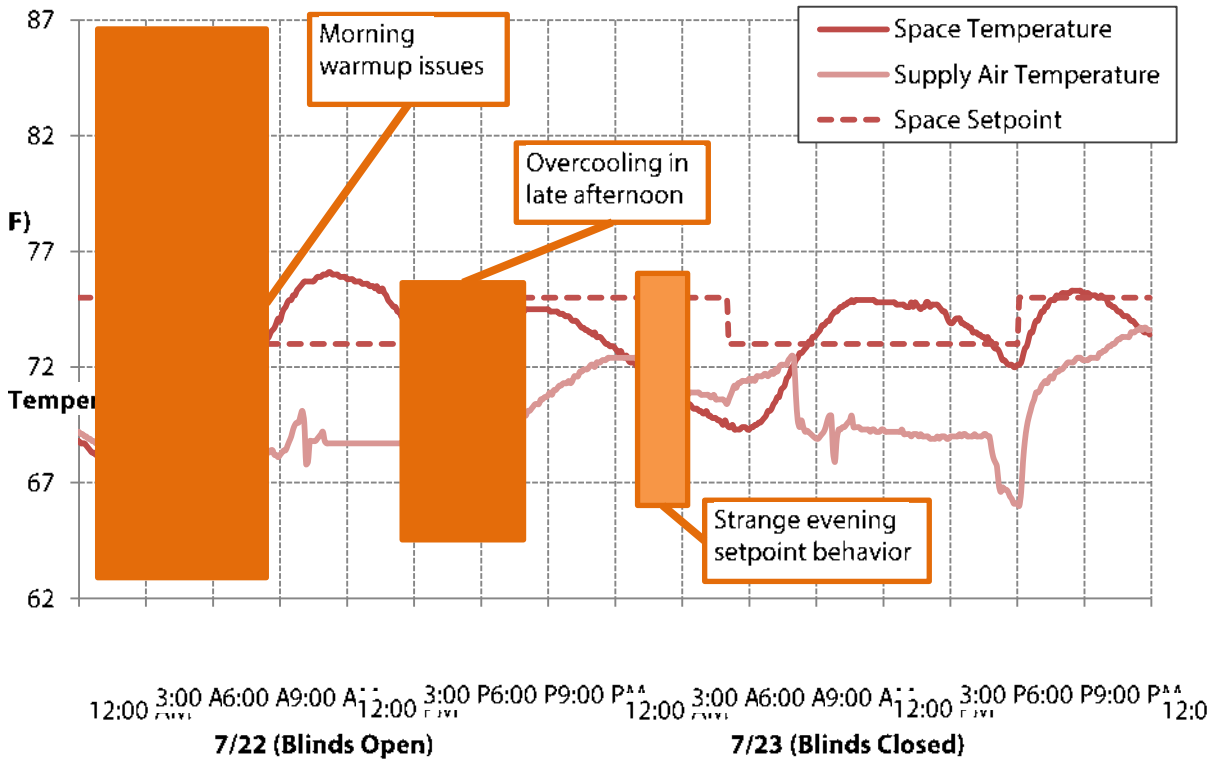

Figure 4: Space, supply, and setpoint temperature data for third floor conference room

Blinds position effect on stratification profile shape

The stratification profiles in Figure 1 follow a fairly normal shape during the 8:00am measurements, but follow a less ideal stratification profile during the 11:00am measurements, with the stratification deteriorating near the top of the occupied zone (57"). Figure 5 shows 11:00am measurements compared to 12:00pm measurements. While the "blinds open" profiles are very similar, the "blinds closed" 12:00pm profile shows a further deterioration of stratification. The air temperature four inches from the ceiling is colder than the air temperature in the occupied zone. This profile suggests that cold air from the perimeter diffusers is being thrown with sufficient velocity to reach the ceiling, creating a mixed-air environment rather than a stratified environment. The deterioration of stratification related to airflow can clearly be seen

in Figure 6, which displays trends for thermal stratification and the corresponding command airflow. Reducing the maximum airflows of the perimeter fan terminal units would help prevent this stratification degradation, but may also affect the ability of the system to control to setpoint. Additionally, raising the setpoint would reduce the amount of time the fan terminal unit is at maximum airflow, which would improve stratification.

The behavior demonstrated in these tests highlights the dilemma with UFAD systems. Unless the units are sized for high SATs they will not be able to meet the load and the high airflow compromises the stratification further aggravating control of the space load. These affects can be ameliorated by either installing more (or different, low throw) diffusers or by lowering AHU SAT. However, lowering AHU SAT tends to end up overcooling the interior spaces. Raising the space setpoint would help all of these issues.

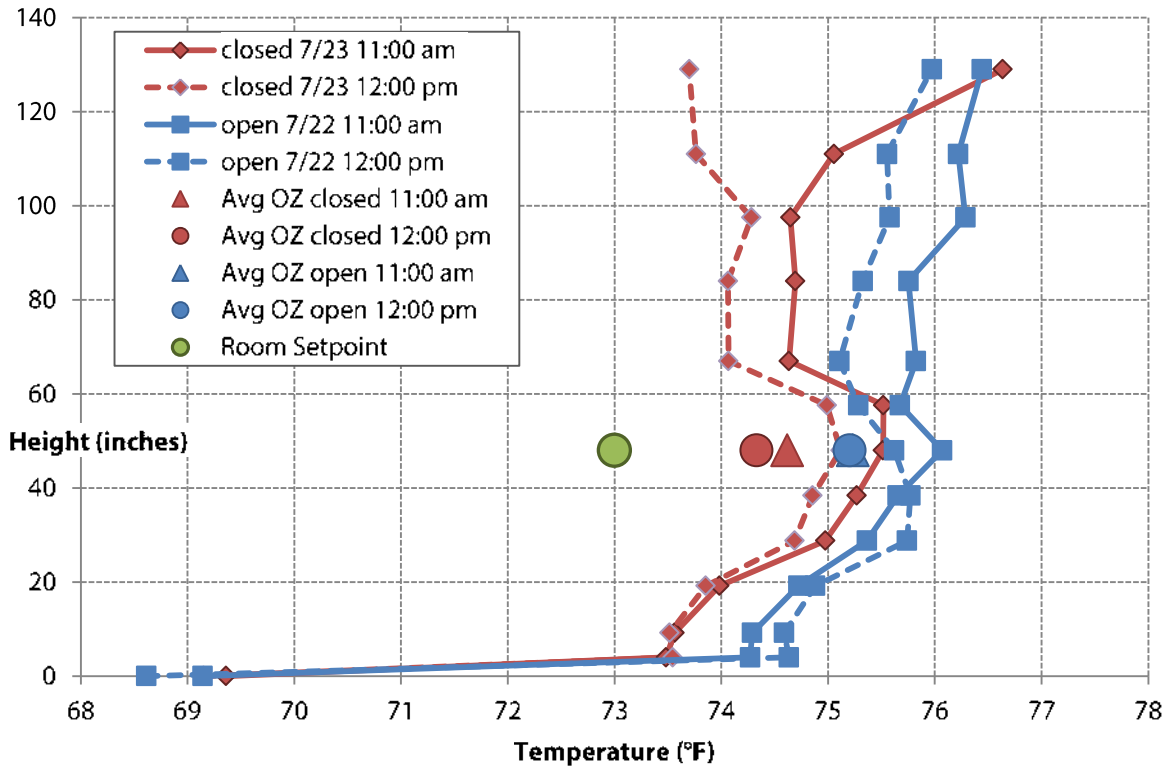

Figure 5: Thermal stratification profiles for blinds scenarios in third floor conference room

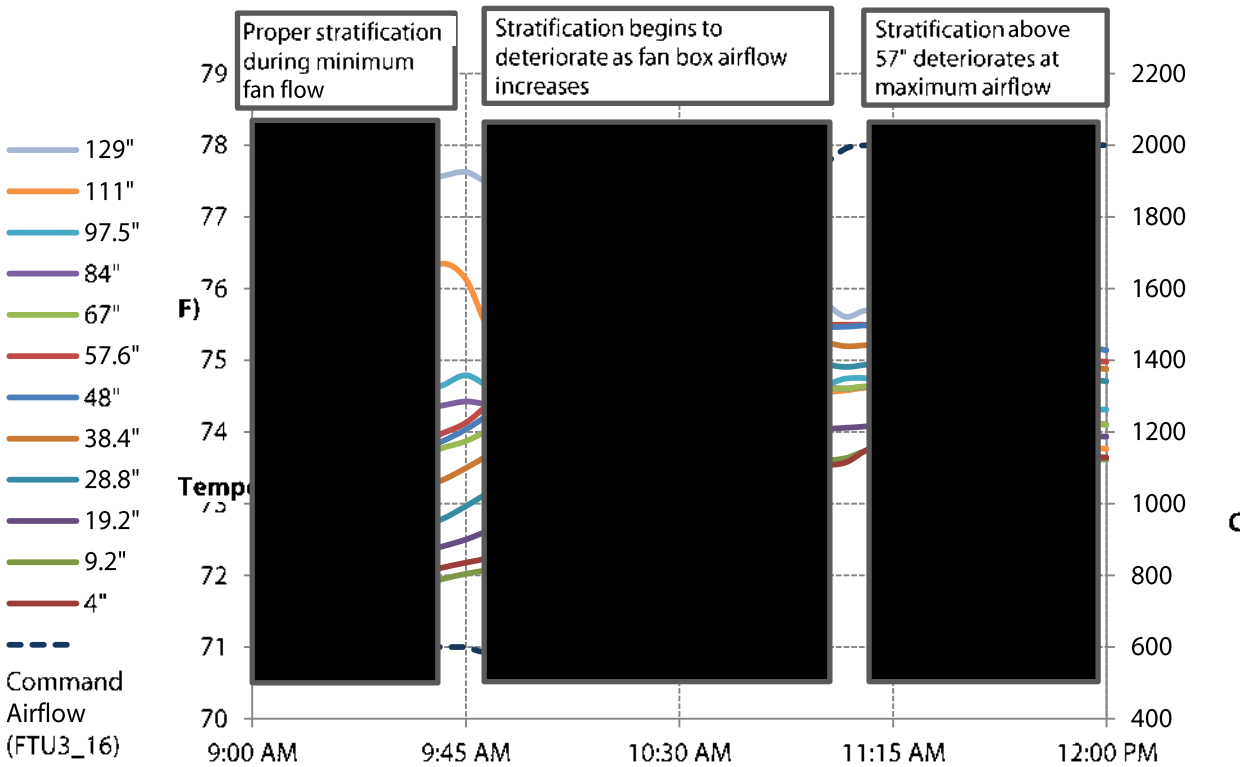

Figure 6: Thermal stratification trends and command airflow for blinds closed scenario

11th floor open plan space

Blinds position effect on room temperature

Figure 7 shows the average stratification profiles for the hour-long measurement period on each of the two blinds case scenarios (horizontal and open). The profiles are very close to each other, indicating that the system is controlling well and the blinds position is not affecting the temperature in the space. Further discussion of this finding can be found in the section:

Blinds position effect on fan terminal unit airflow.

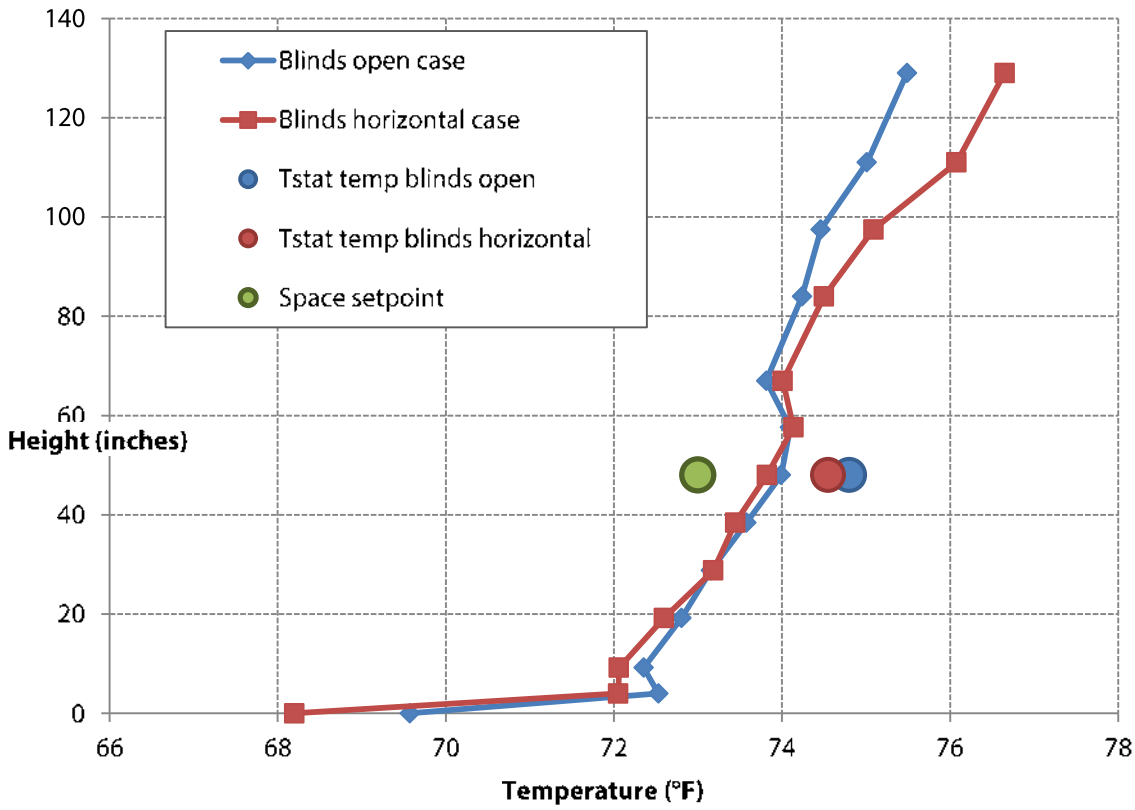

Figure 7: Thermal stratification profiles for blinds scenarios in 11th floor open plan space

The stratification profiles for this space were consistent with expectations of a well-performing UFAD system. Figure 8 provides a graphical summary of the performance of the UFAD system in the 11th floor open plan space. The chart plots average room air temperature stratification against the average occupied zone air temperature for each location measured on the 11th floor on two days of testing (blinds open and blinds horizontal). There are two shaded areas representing different "comfort zones." The beige shaded area represents the comfort zone as determined by the CBE comfort modeling study.[1] The pink shaded area represents the comfort zone as determined by ASHRAE Standard 55. The ASHRAE comfort zone was determined from operational and occupant parameters based on ASHRAE standard comfort zone models. In this case, we assumed a **Met**abolic value = 1.2, **Clo**thing value = 0.6, **R**elative **H**umidity = 50%, and **V**elocity at the occupant <50 fpm.

All points fall within the two comfort zones and most points fall within the CBE Comfort Model zone, which we feel provides a more accurate picture of comfortable conditions. There is little variability in both stratification and occupied zone temperature, indicating good control of the open plan space to setpoint (73°F). The occupied zone temperatures fall on the colder side of the comfort zone, indicating potential to raise the setpoint for energy savings.

[1] H. Zhang, C. Huizenga, E. Arens, and T. Yu, "Modeling Thermal Comfort in Stratified Environments," 2005, *Proceedings of Indoor Air 2005*, *Beijing*, 133 – 137.

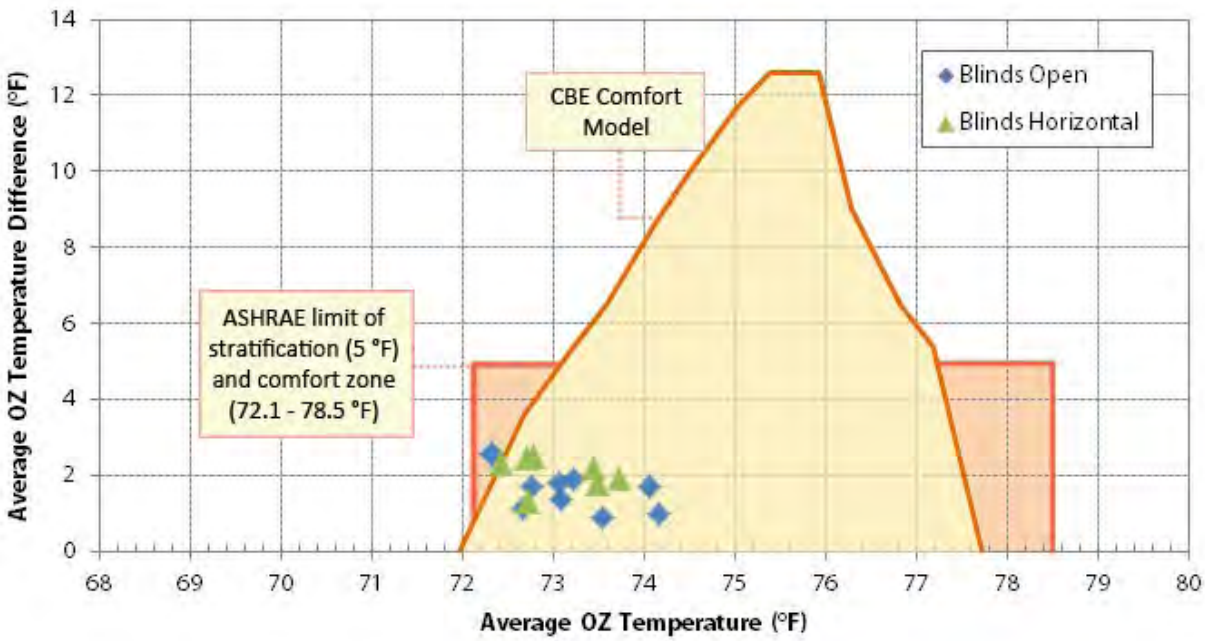

Figure 8: UFAD performance summary for 11th floor open plan office

Blinds position effect on fan terminal unit airflow

Figure 9 shows the BMS data for fan terminal unit airflow and space temperature in the southeast corner of the 11th floor open plan space. The chart plots two days of data, corresponding to the open and horizontal blinds scenarios. The chart shows a 1.5 hour difference in time the fan terminal unit spent above its minimum airflow rate between the blinds horizontal (9.5 hours) and blinds open (11 hours) scenarios. As in the third floor results however, the supply air temperature inconsistencies between the two days of measurement prevent us from drawing any conclusion concerning blinds position and fan terminal unit energy. It is likely the difference in fan operating hours results from the difference in time when the SAT is reset; later reduction (blinds open) results in longer operating time for the fan coil unit. As is the case for the 3rd floor, the units seem undersized for the AHU SAT being used and the room setpoint seems too high.

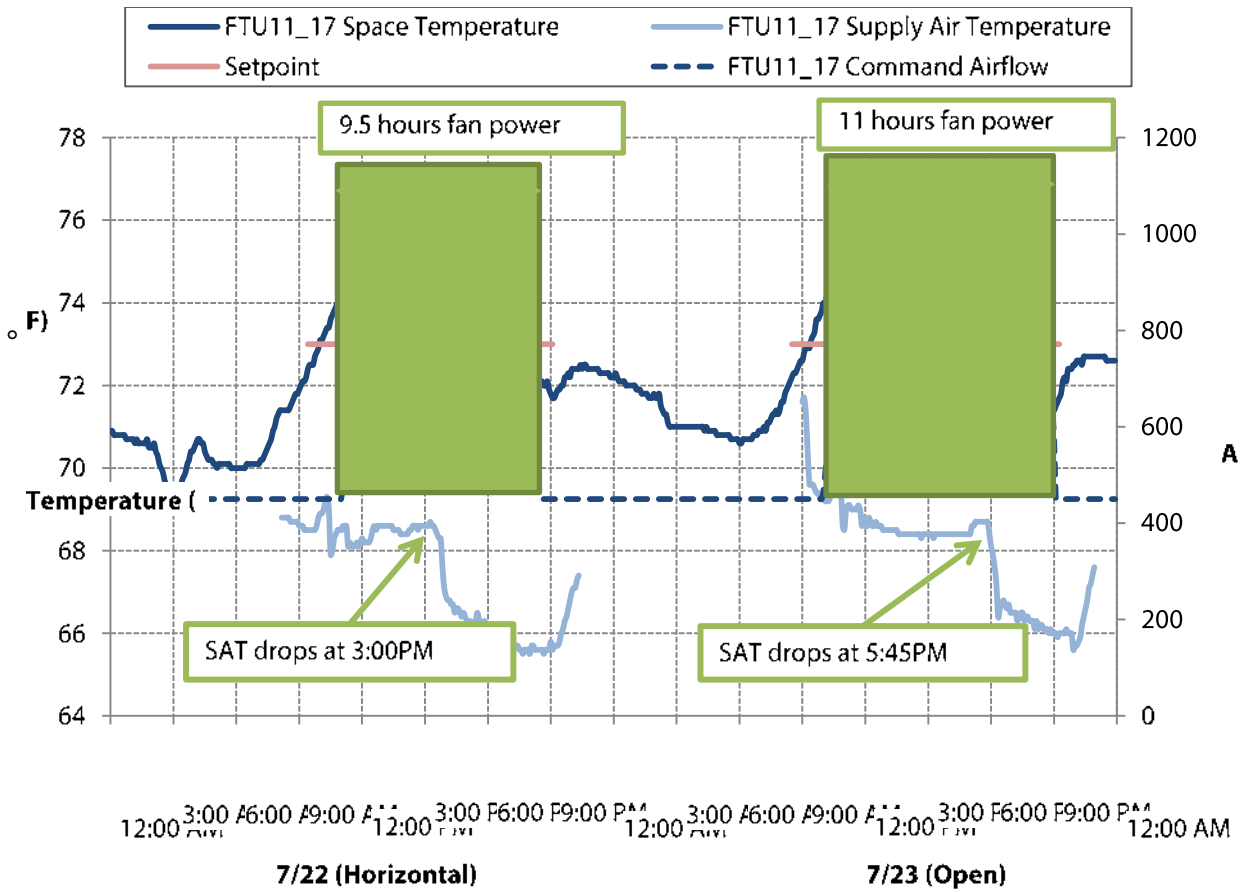

Figure 9: Building management system data for 11th floor open plan space

Thermal decay

Figure 10 is an illustrated thermal contour map for the eleventh-floor southern zone underfloor plenum. The white dots represent wireless mote locations where temperatures were taken. The temperatures shown in Figure 10 represent the average temperature over the course of a half-hour measurement period. Temperatures were linearly interpolated to provide a contour map.

Thermal decay depends on airflow rate and temperature distribution caused by how the supply airflow is delivered to the plenum. Temperatures are typically dictated by how long the air has to travel (while picking up heat) to a particular location.

Unfortunately, a software bug led to data collection problems with the majority of the wireless motes, leading to a very limited data set. Without data to fill out the larger measurement area, we are unable to draw conclusions regarding thermal decay in the plenum. However, the small dataset does provide a picture of underfloor decay for this section of the 11th floor and suggests a maximum of 4°F decay in this small area. Further study is warranted to analyze how thermal decay may factor into the ability of the system to meet setpoint in the space.

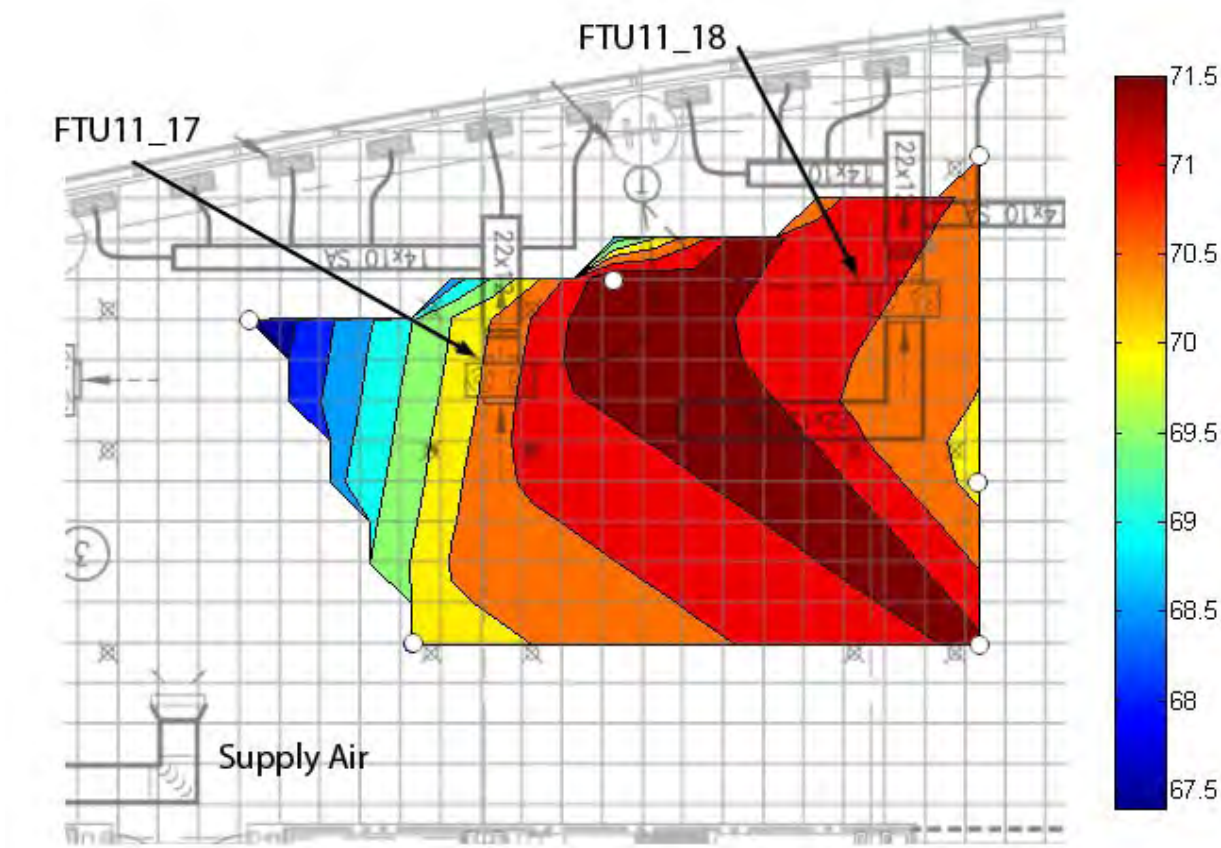

Figure 10: Underfloor thermal decay in southeast section of 11th floor open plan area

Multi-path leakage test

In the 3rd floor conference room, air leakage was measured to areas adjacent to the underfloor plenum, including to the room (Category 2 leakage), to the adjacent underfloor plenums, to the floor below, directly to the return plenum, and to outside the building. The purpose of this multi-path leakage test is to simultaneously characterize airflow rates through all major leakage pathways from the underfloor plenum. The accuracy of this test method has been demonstrated at a few other buildings during the last few years.

Multi-path leakage test method

In the single zone leakage methods usually employed leakage is determined assuming all the leaks have the same pressure:

$$Q = k\,\Delta P^{n} \quad \textbf{(1)}$$

Where:

Q is the leakage airflow rate (cfm)

ΔP is the pressure across the leakage pathway (iwc, or Pa]

k and n are regression coefficients (0.5<=n<=1)

The Multi-Path approach uses a more realistic model of leakage from the underfloor plenum by measuring and determining the air leakage correlation for each major leakage pathway. Rather than limiting our attention and measurements to only the pressure difference between the plenum and room (leaving other pathways poorly determined), this measurement technique provides greater detail by modeling the total leakage from the underfloor plenum as the sum of several right hand terms from Equation (1) each representing the leakage through a major pathway. In the case of our tests in the CalSTRS building, we used the following equation:

$$Q = k_1 * \Delta P_1^{n1} + k_2\ \Delta P_2^{n2} + k_3\ \Delta P_3^{n3} + k_4\ \Delta P_4^{n4} + k_5\ \Delta P_5^{n5} + k_6\ \Delta P_6^{n6} + k_7\ \Delta P_7^{n7} \quad (2)$$

Where:

$k_i\ \Delta P_i^{ni}$ is the leakage contribution from each of the following four pathways:

i = 1, room through all openings (Category 2 Leakage)

i = 2, adjacent plenum to the North

i = 3, adjacent plenum to the West

i = 4, the floor below (the atrium)

i=5, the return plenum above the room

i=6, the outside

i=7, the perimeter duct system (not a leakage path in normal operating modes)

For Equation (2) to be used successfully, the pressure differences governing the various pathways must be independently changed and sufficient data points taken to yield a good regression. Although it would be desirable to be able to hold one pressure (for example, the plenum/room pressure difference) constant, it is not necessary to do this as long as enough data points are collected. It is also quite difficult to do this in practice using a building's control system. In the CalSTRS building, we used the BMS controls to adjust the pressure in the adjacent underfloors plenums, the supply flow (and thus pressure) to the floor below, and the speed of the return fan to change the return plenum pressure, and a "Blower Door" to vary the pressure in the room (and its return plenum) in various combinations to allow a range of differential pressures to be obtained for each leakage pathway. In the case of the outside, natural fluctuations in wind speed and direction provided the necessary pressure variations.

The advantage of the multi-path leakage test is that by developing a unique correlation for the leakage from the plenum to all openings, it can accurately predict true Category 2 leakage as a function of pressure difference.

In many instances the flow exponent "n" is difficult to determine and a single "n" can be used from the measurement of "total leakage". At CalSTRS, sufficient measurements were taken to be able to independently determine each flow exponent.

Prior to our leakage tests we found and fixed a large hole (about 6 inches in diameter) between the underfloor plenums. There may have been other similar penetrations that could be easily sealed which we did not find.

Leakage results

Initial analysis showed no or very little correlation of underfloor pressure with changes to the pressure of the floor below or to outside, thus these paths were eliminated. Table 1 shows the regression coefficients, normal operating pressures and resulting leakage flows. Note that leakage to/from adjacent underfloor plenums likely still serves an occupied zone, just not the 3rd floor conference room. Only about 3% of the total supply flow does not go to an occupied zone.

Table 1: Leakage and normal operating pressures and flows of leak paths to the 3rd floor conference room underfloor plenum. Normal operating pressures are for 11 AM to 2 PM on July 20-23, 2010. The flow into the 3rd floor conference room UF plenum was 2000 cfm (command by the BMS system).

Leakage Path	Flow coefficient (k) {for flow in cfm}	Flow exponent (n)	Normal operating pressure (Pascals)	Normal operating flow (cfm)	% of Normal operating flow
Room	10.88	.545	15.0	48.2	2.4
North UF Plenum	37.47	.500	13.8	138.9	6.9
West UF Plenum	0.67	1.000	14.8	9.9	0.5
Floor below	0 (forced)	N/A	N/A	N/A	0.0
Return Plenum	11.79	.568	17.3	57.4	2.9
Outside	0 (forced)	N/A	N/A	0	0.0
Perimeter Duct	5.95	.852	N/A	N/A	N/A
Total Leakage					12.7

Figure 11 shows time series data for these flows during normal operating conditions. Negative flows indicate flow into the 3rd floor conference room underfloor plenum; positive flows are leakage from the plenum.

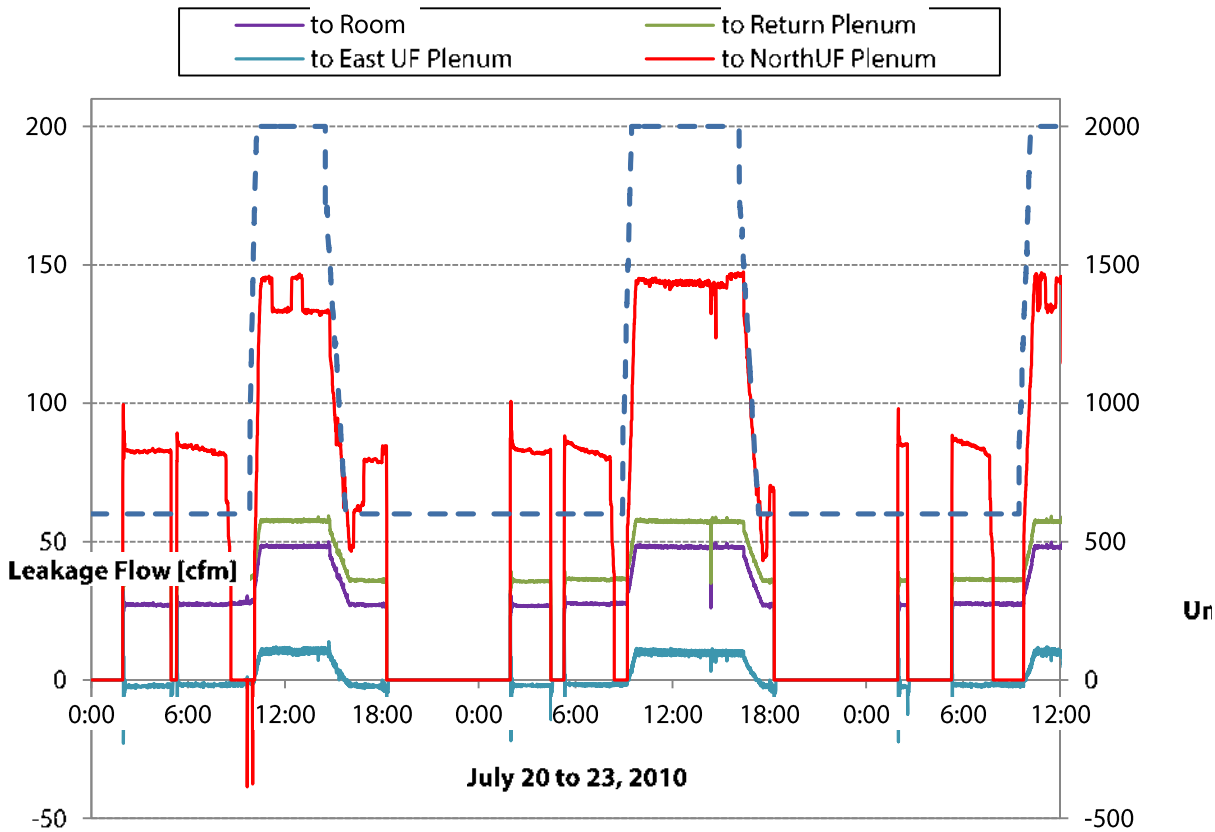

Figure 11: Leakage and normal operating flows. Note that a supply flow of 600 cfm is still reported by the BMS even during periods when the fan must be off (as indicated by the leakage to room going to zero).

Leakage to the conference room is relatively small (2.4% of the delivered air) compared to other measurements of Category II leakage. Leakage between the underfloor plenum and its return plenum in the drop ceiling above the room, presumably via walls and any existing chases has not been measured before and is a bit of a surprise. The primary leakage path is from the underfloor plenum to another underfloor plenum. At about 150 cfm during normal occupied operating conditions, this leakage is unlikely to lead to significant energy inefficiency or comfort problems in the adjacent rooms, but does lower the effective capacity of the cooling delivered to the conference room by about 7%.

4. CONCLUSIONS AND RECOMMENDATIONS

Third floor conference room

Our cart measurement results indicate that having the blinds closed has a slight cooling effect compared to having the blinds all the way open. While this cooler room air temperature was noted throughout the morning of testing, we are unable to draw conclusions from fan terminal unit operation due to inconsistencies in supply air temperature behavior between the two testing days and because the terminal units appear to be undersized for the SAT being used. Stratification profiles in the conference room indicate that the maximum airflow velocity out of the perimeter diffusers is too high, destroying stratification above 57”. More definitive results could come from

future testing where a fixed and lower supply air temperature was used along with an increased room setpoint.

Leakage in the space was measured to be relatively small and is not likely to cause comfort problems. While any leakage results in some inefficiency, our results indicate the leakage paths in the conference room are small compared to other similar scenarios we've tested.

11th floor open plan space

Our cart measurement results indicate that for the measured period, there is no difference in air temperature with the blinds *open* compared to blinds in a *horizontal* position. Despite initial evidence that fan terminal unit energy decreased in the horizontal blinds scenario, we were unable to draw conclusions because of inconsistencies in supply air temperature behavior between testing days.

The UFAD system is performing well, with proper levels of stratification and fairly good control to setpoint. The measured spaces were controlled to the cool end of the comfort zone, indicating potential for increased setpoints.

Future testing would provide a clearer picture of underfloor thermal decay and its effect on overall UFAD system performance.

Internal report April 2013

The New York Times Building Field Measurement Report

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Center for the Built Environment (CBE)
University of California, Berkeley

April 30, 2013

INTRODUCTION

The purpose of this project is to conduct a detailed post-occupancy evaluation (POE) of The New York Times (NYT) Headquarters, a 52-story high performance office building in downtown Manhattan. The Times Building has attracted interest nationally due to its design incorporating innovative technologies, including advanced external shading, automatic internal shading control, dimmable electric lighting, and underfloor air distribution (UFAD). This POE study is sponsored by U.S. DOE's Commercial Buildings Partnerships (CBP) Program, which encourages building owners and operators to collaborate with research staff at national laboratories and universities to explore energy-saving ideas and strategies in retrofit and new construction projects. It is hoped that some of the innovations used in the NYT Building can become a model and prototype for larger scale implementation and replication in new and existing buildings in New York and nationally.

Researchers from the Center for the Built Environment (CBE) at UC Berkeley and Lawrence Berkeley National Laboratory (LBNL) are collaborating on this study; CBE is focusing on evaluating the performance of the UFAD system, while LBNL is studying the shading and lighting performance. The approach being used in this project is to carefully document measured energy savings and comfort performance in The New York Times Buildings. Monitored data will be used to conduct a semi-validation of a specially developed whole-building energy simulation model of one tower floor of The New York Times Building using EnergyPlus. The model will be capable of representing the combined energy performance of the shading, dimmable lighting, and UFAD systems. The semi-validated model will be used to assess potential energy savings in comparison to an ASHRAE Standard 90.1 reference prototype building. The results will have primary applicability to office buildings but will be useful in part to all commercial buildings.

The purpose of this internal report is to describe the measurements and analysis of the UFAD system performance at the New York Times Building, including key lessons learned and recommendations.

BUILDING PERFORMANCE DATA COLLECTION AND ANALYSIS

The goal of work in this area is to monitor and collect a variety of building performance data related to the UFAD system. As mentioned above, data describing the performance of the internal shades and dimmable lighting system will be provided by LBNL researchers. The data will be analyzed to assess the energy performance of a typical tower floor in the building, and to calibrate the EnergyPlus simulation model which will then be used for comparison with the predictions of an EnergyPlus model of the building (described below). Finally, based on our review and analysis of the UFAD performance data, we will make recommendations, if appropriate, to improve overall UFAD system operation at The New York Times Building in terms of energy use and thermal comfort performance.

DESCRIPTION OF THE NEW YORK TIMES BUILDING

The New York Times Building (Figure 1) was completed and occupied in 2007. The 52-story, 1.5 Mft^2 office tower, at 1,046 ft high, is tied with the Chrysler Building as the 4th highest in New York City. Architect Renzo Piano employed a unique double-skinned façade, featuring clear floor-to-ceiling glazing to provide views and transparency in combination with a second external skin made up of horizontal ceramic rods that serves as a sunshade. The building also incorporates an advanced automatic internal shading and dimmable electric lighting system developed in collaboration with Lawrence Berkeley National Laboratory. The internal shades operate to further control solar loads while allowing natural daylight to replace the use of electric lighting as much as possible. The New York Times Company occupies floors 2-27, the first floor is retail and entrance lobby, and the upper floors are occupied by other tenants. Underfloor air distribution (UFAD) was installed on all floors occupied by the Times. As discussed below, CBE researchers installed their detailed wireless monitoring system on the 20th floor, which was selected to represent a typical tower floor of the high-rise NYT Building.

Figure 1: The New York Times Building

DESCRIPTION OF FIELD MONITORING OF UFAD SYSTEM PERFORMANCE

During the first field visit (August 22-26, 2011), an array of wireless sensors was deployed to measure temperature, water flow, and power on the 20th floor of the NYT Building (see Figure 2). These wireless sensors continuously monitored conditions over the entire study period (August 25, 2011 – January 9, 2012) and transferred data to our data server in Berkeley in real time.

Pictures of the monitoring setup are included in the next set of figures:

- Stratification trees – air temperature stratification at perimeter and interior locations of each building exposure (Figure 3)
- Stratification cart – air temperature stratification, floor/ceiling surface temperatures, and underfloor plenum pressure at various locations across the floorplate (Figure 4)
- Power meters – lighting, fan-powered-box, plug-loads, and whole floor consumption (Figure 5)
- Chilled water supply and return temperatures and water flow rate (Figure 6)
- Underfloor plenum temperature at selected locations across the floorplate (Figure 7)
- Fan-powered-box, FPB, flow calibration (Figure 8)

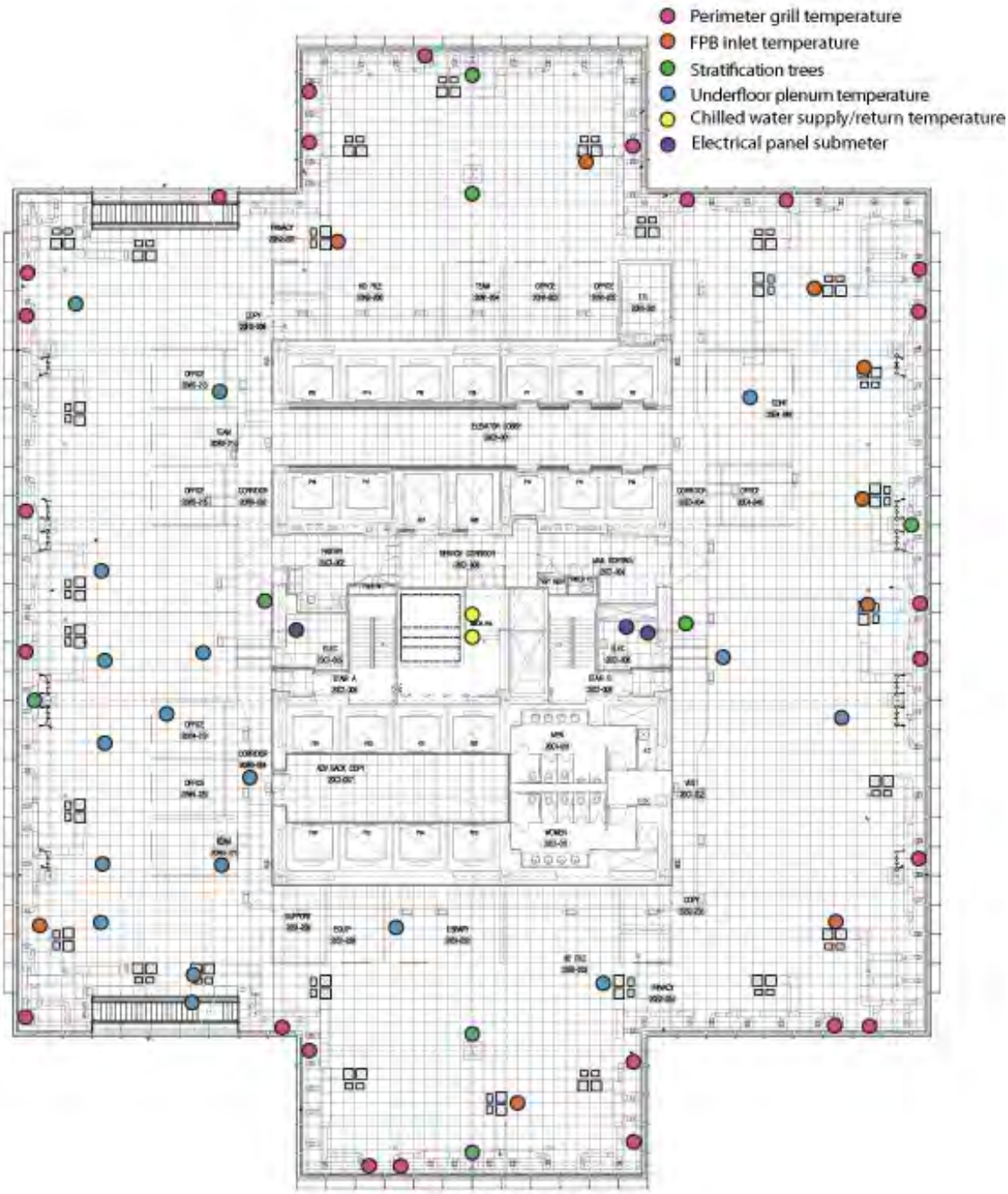


Figure 2: Map of wireless sensors deployed on 20th floor of NYT Building

Figure 3: Stratification tree

Figure 4: Stratification cart

Figure 5: Power metering

Figure 6: Chilled water flow and temperature metering

Figure 7: Underfloor plenum temperature (fan-powered-box inlet temperature)

Figure 8: Fan-powered-box flow meter calibration

PRELIMINARY RESULTS OF UFAD THERMAL COMFORT PERFORMANCE

This section will discuss some of the preliminary findings of UFAD system performance as measured by stratification and ability to keep the building comfortable.

UFAD systems are commonly controlled using a thermostat mounted at a 4 ft height; although for interior zones at the Times, the thermostats are located at 84 inches from the floor. If significant stratification develops, the average temperature in the occupied zone ((OZ) defined as the vertical region from 4 inches to 67 inches from the floor) may end up being too cool and cause discomfort. The degree of stratification is gauged by the difference between the temperatures at 67 and 4 inches (namely the head-ankle temperature difference). We generally consider stratification optimal when it is in the range of 3 to 4°F. As a single value, stratification serves as a

good indicator of UFAD performance, though the full stratification profile (air temperature at eight room heights) provides a clearer picture of UFAD temperature performance.

As a preliminary look at stratification data from the study period, the next series of figures shows how the UFAD system performed on a relatively warm day—September 14th (high outdoor temperature of 84°F). Figure 9 displays trends of relevant BMS data for the day of interest. FPB13 refers to the fan-powered-box that controls the area of the East perimeter zone where our stratification tree was located. The AOV signal represents the signal that was being sent to the fan-powered-box and is a proxy for airflow rate. The dotted red and blue lines correspond to data collected by our temperature sensors whereas the solid red and blue lines represent thermostat readings derived from the BMS data. The differences between the thermostat readings and our temperature sensors could indicate a calibration issue or may be due to differences in location of the respective sensors, but requires further investigation. The setpoint for this zone was set rather low at 72°F and the system has a difficult time controlling to such a low setpoint. This is likely due to the relatively high AHU supply air temperature (SAT) combined with thermal decay in the plenum.

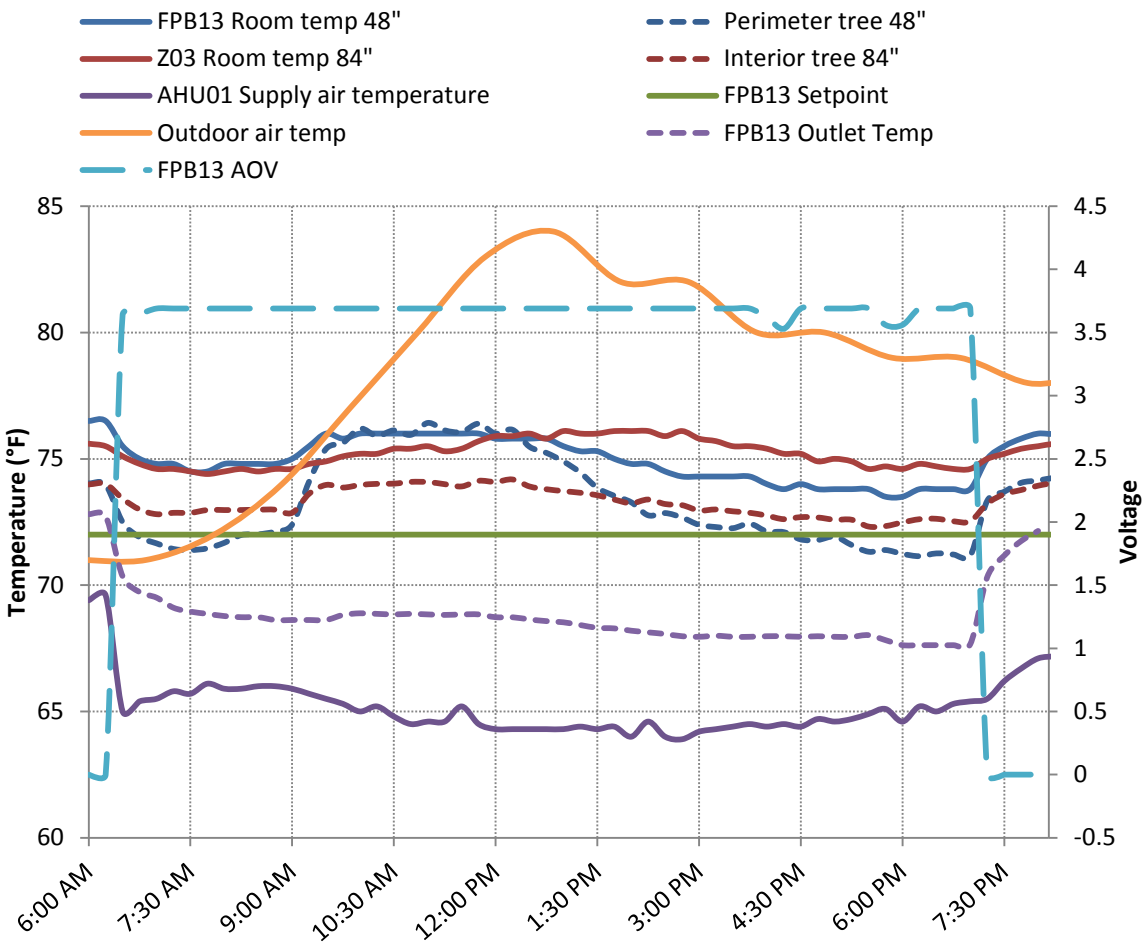
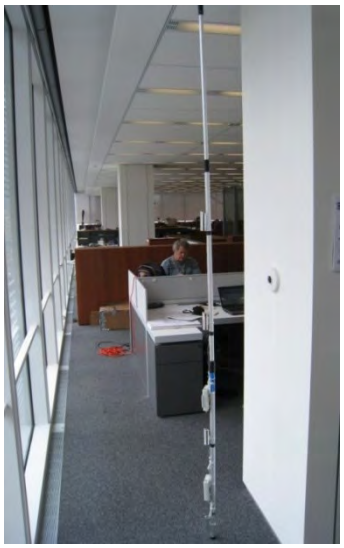


Figure 9: BMS, outdoor temperature, and wireless temperature data for September 14th

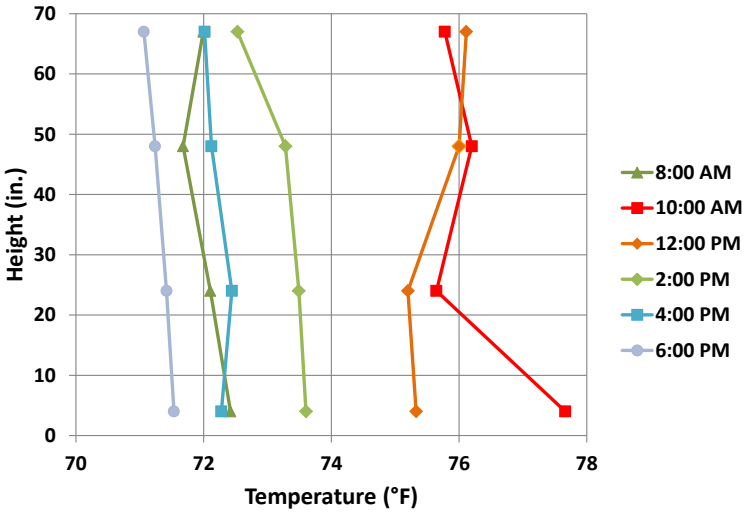
Stratification profiles

Figure 10. East interior zone stratification pole and example hourly temperature profiles, 9/14/2011.

Figure 11 presents measured temperatures in the perimeter zone, which show a different temperature profile compared to the interior location. At this pole location, quite close to the exterior window with linear bar grilles in the floor along the base of the windows, virtually no stratification is observed, indicating a mixed room air distribution pattern. In all likelihood, higher supply airflow rates from the perimeter bar grilles are blowing cool air upwards into the space near the window, creating the mixing and in some cases negative stratification. This pattern of increased mixing with little stratification has been frequently observed in UFAD installations with open plan offices. Note that the higher measured temperature at the lowest (4-in.) height at 10 am is due to direct solar radiation, both incident on the shielded temperature sensor and creating a warmer carpet surface temperature. These perimeter grilles also provide heat during the winter (from the ducted underfloor variable speed fan-coil units) that serves as a warm air curtain next to the windows, thus preventing uncomfortable cold air downdrafts. When compared to the ASHRAE comfort zone in Figure 12, higher average occupied zone temperatures (~76°F) are seen at 10 AM and 12 PM noon in this east perimeter zone, but at all other times during the day average temperatures are closer to the low end of the comfort zone.



a) Photo of pole

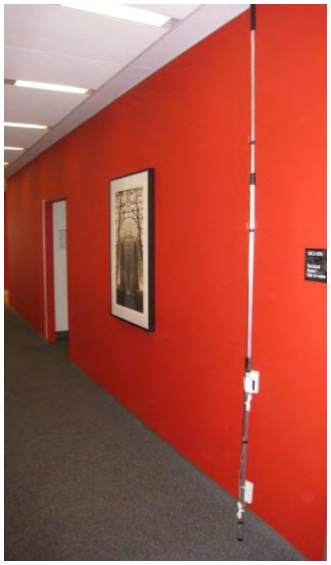


b) Temperature vs. height

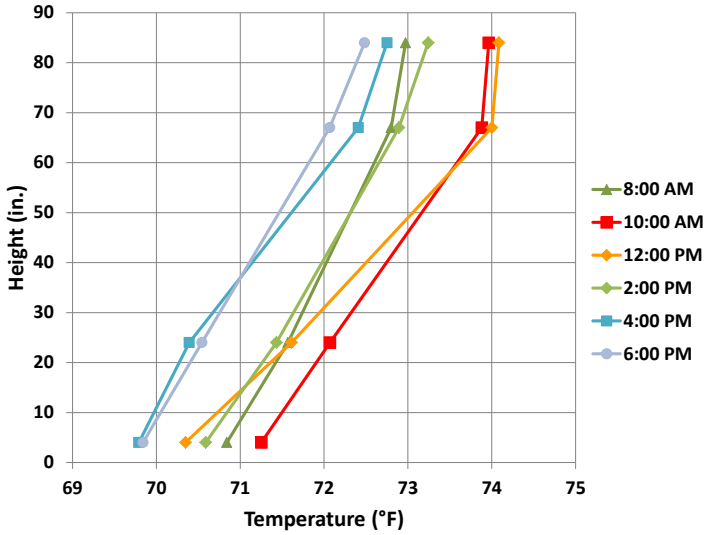
and Figure 10 present photos and representative measured hourly vertical temperature profiles from the stratification poles located in the east interior zone and east perimeter zone, respectively. Data are shown for one day, 9/14/2011, during warm weather.

Results for the interior zone (Figure 10) indicate that a reasonable amount of stratification (2-3°F) is achieved in the occupied zone, as indicated by the difference in temperature between standing head height (67 in.) and ankle height (4 in.). Note that the warmest temperatures occurred at 10 AM and 12 PM noon, matching the expected peak solar load for this east-facing zone. Stratification is one indicator of good UFAD cooling performance because it demonstrates how comfortable conditions can be maintained in the lower occupied zone, while allowing warmer and less comfortable conditions to exist in the higher space elevations; in general, energy can be

saved by increasing the setpoints (thus lowering the airflow and reducing cooling) while still allowing the average conditions in the occupied zone to remain comfortable. In the case of The Times building, the interior zone thermostats are all located at a height of 84 in. (for architectural reasons). The Times Company has implemented an interior zone setpoint temperature offset to account for the effects of stratification.



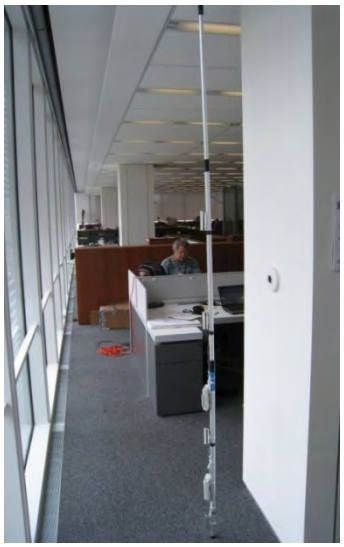
a) Photo of pole



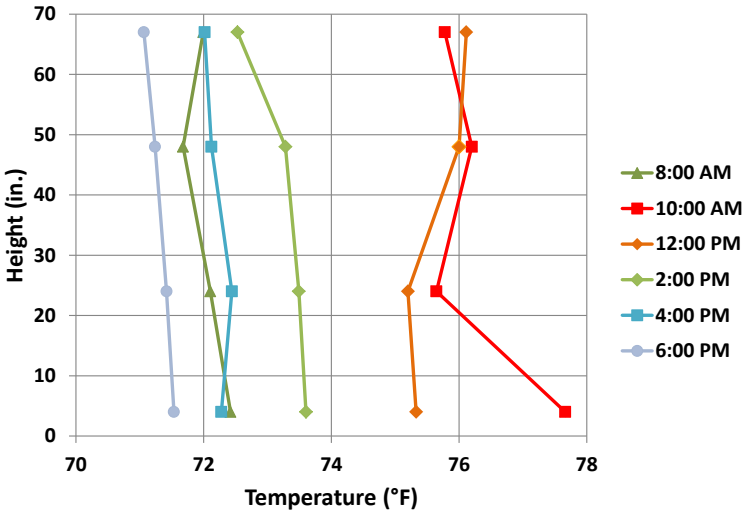
b) Temperature vs. height

Figure 10. East interior zone stratification pole and example hourly temperature profiles, 9/14/2011.

Figure 11 presents measured temperatures in the perimeter zone, which show a different temperature profile compared to the interior location. At this pole location, quite close to the exterior window with linear bar grilles in the floor along the base of the windows, virtually no stratification is observed, indicating a mixed room air distribution pattern. In all likelihood, higher supply airflow rates from the perimeter bar grilles are blowing cool air upwards into the space near the window, creating the mixing and in some cases negative stratification. This pattern of increased mixing with little stratification has been frequently observed in UFAD installations with open plan offices. Note that the higher measured temperature at the lowest (4-in.) height at 10 am is due to direct solar radiation, both incident on the shielded temperature sensor and creating a warmer carpet surface temperature. These perimeter grilles also provide heat during the winter (from the ducted underfloor variable speed fan-coil units) that serves as a warm air curtain next to the windows, thus preventing uncomfortable cold air downdrafts. When compared to the ASHRAE comfort zone in Figure 12, higher average occupied zone temperatures (~76°F) are seen at 10 AM and 12 PM noon in this east perimeter zone, but at all other times during the day average temperatures are closer to the low end of the comfort zone.



a) Photo of pole



b) Temperature vs. height

Figure 10: East perimeter zone stratification pole and example hourly temperature profiles, 9/14/2011.

Thermal comfort performance

To investigate comfort conditions in the interior zone, Figures 12 and 13 show how the average hourly occupied zone temperature conditions compare with an estimated comfort zone based on ASHRAE Standard 55-2010 (assumptions: met = 1.2, clo = 0.6, relative humidity = 50%, velocity near occupant < 50 fpm) [ASHRAE 2010]. Each point represents the average of all temperatures measured within the occupied zone (4-67 in.) for the same hourly profiles in Figures 10b and 11b, respectively. The amount of stratification in the occupied zone is shown on the y-axis; ASHRAE Standard 55 specifies the maximum acceptable stratification as 5°F. The comfort results for the east interior zone shown in Figure 12 indicate that all average temperatures are very near or slightly below the lower (coolest) boundary of the comfort zone. When compared to the ASHRAE comfort zone in Figure 13, higher average occupied zone temperatures (~76°F) are seen at 10 am and 12 noon in this east perimeter zone, but at all other times during the day average temperatures are closer to the low end of the comfort zone.

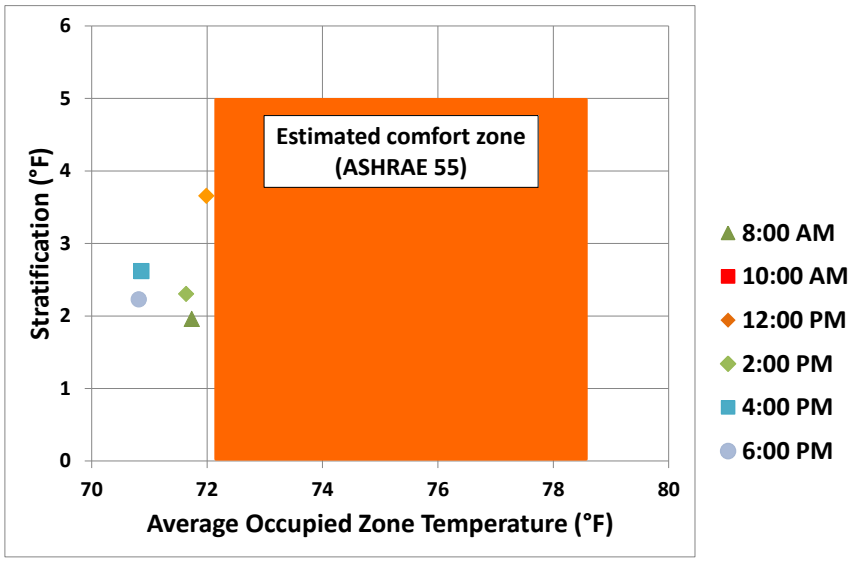


Figure 12: Occupied zone temperature and stratification of East interior zone, 9/14/2011.

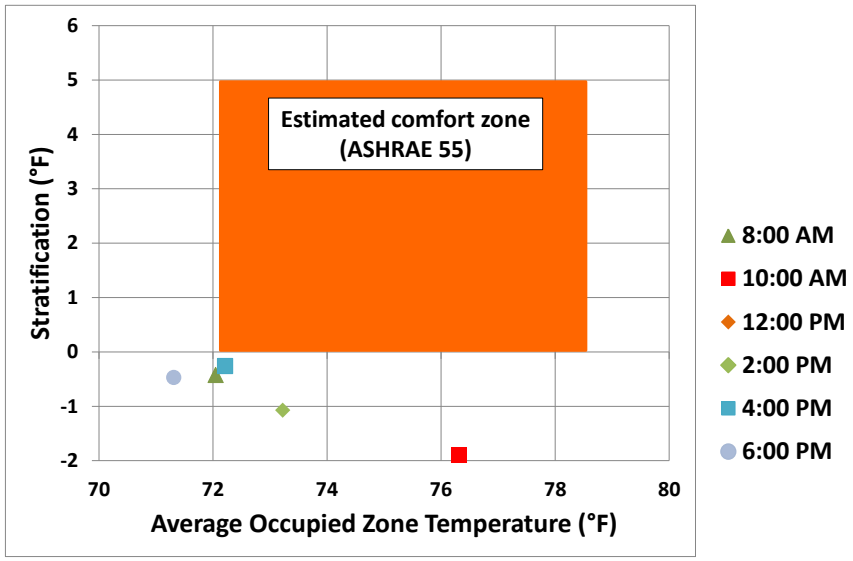


Figure13: Occupied zone temperature and stratification of East perimeter zone, 9/14/2011

SUPPLY PLENUM PERFORMANCE

The underfloor plenum temperatures were measured at the perimeter diffusers of each of the 27 fan-powered-boxes and at the inlet of eight selected fan-powered-boxes. Temperatures were measured at the outlet of each of the six sections of the air highway (representing supply air temperature entering the plenum), as well as in fourteen diffusers. In the figures below these locations are shown as different colors on a graduated temperature scale. The underfloor pressures are also shown for each controlled plenum zone. These are located on each figure in the core area of the building near their corresponding zone.

Figure 11 shows the data for September 3rd, one of the hottest days during the measurement period, for 5 PM.

Figure 12 shows the data for December 22nd, a cold day when the heating coils in several of the fan-powered boxes were operating, as indicated by the warm perimeter diffuser supply air temperatures.

In these figures the measured perimeter diffuser temperature is displayed for the entire area served, as are the temperatures associated with a full section of the air highway. Individual diffusers and fan-powered-box inlet temperatures are displayed as a single point.

Note the increase in underfloor pressure of the south zone. This indicates increased airflow to this area.

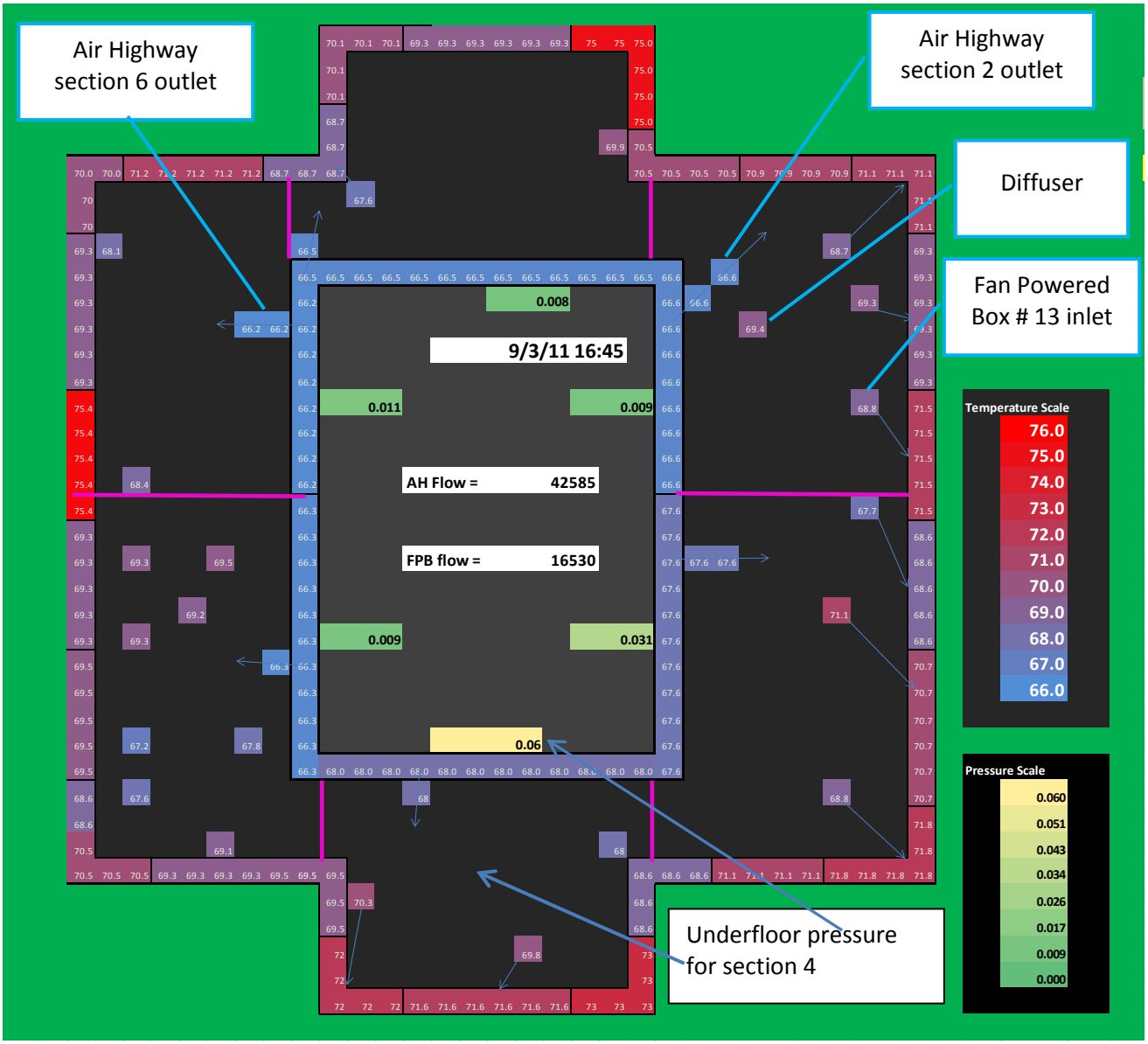


Figure 11: Hot day, late afternoon underfloor plenum temperature (°F) and pressure (iwc) distribution.

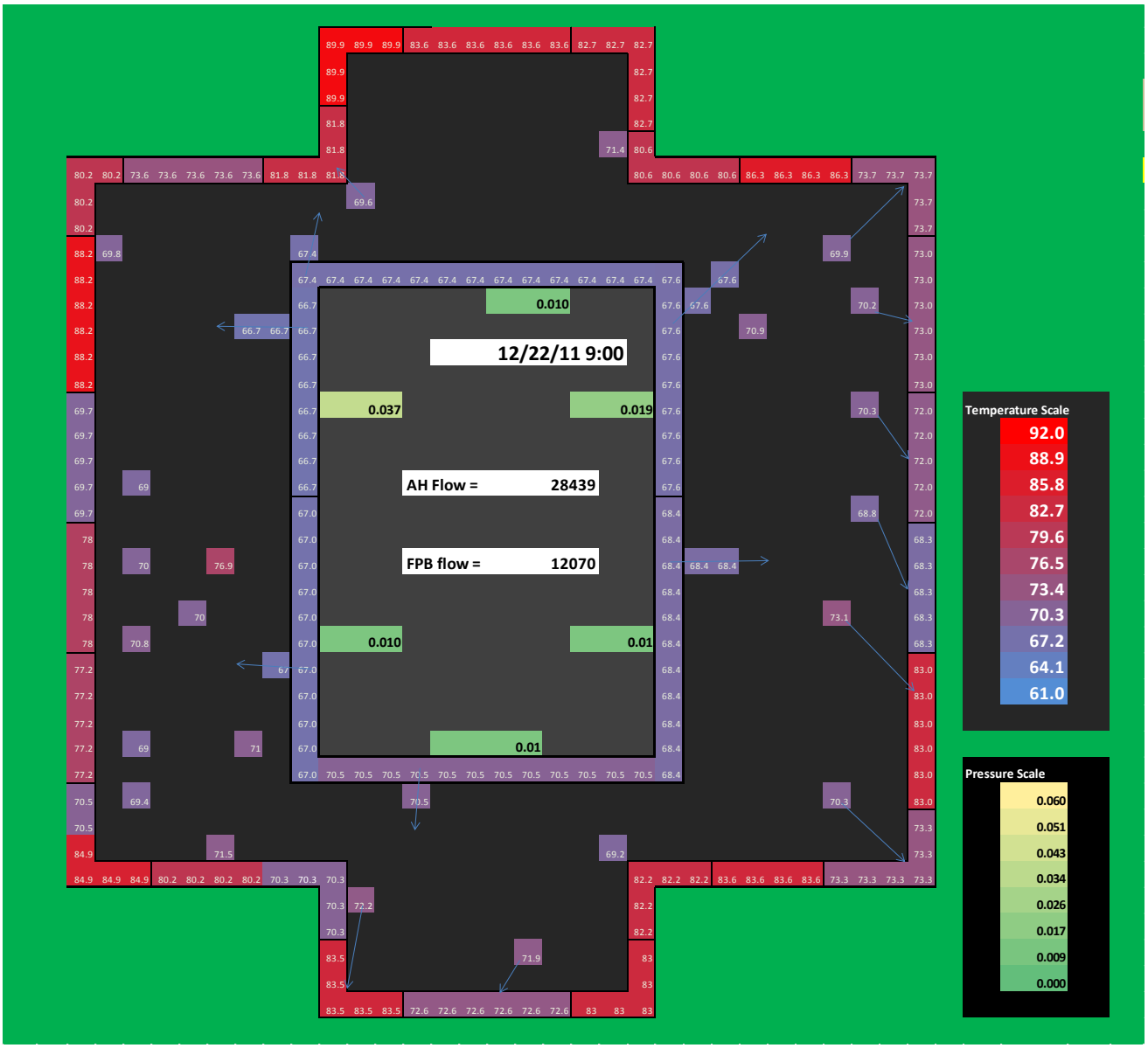


Figure 12: Cold day, morning underfloor plenum temperature (^{o}F) and pressure (iwc) distribution

This day starts with most perimeter zones calling for heat. This diminishes during the day.

INTERNAL LOADS AND HVAC PERFORMANCE

Internal loads

Weekday and weekend data from each month were used to make average daily profiles of lighting and plug loads for each month of measurements. Note that the August profile is not considered truly representative because it included only a few days.

Figure 13 presents lighting loads, which show remarkable similarity in the morning hours for weekdays, very close to 0.5 W/ft^2. There are slight differences in summer vs. winter behavior in the afternoons as the average light power increases to about 0.6 W/ft^2. Weekend loads indicate an irregular occupancy, but still substantially lower than for weekdays, as expected; ~ 0.1 – 0.2 W/ft^2.

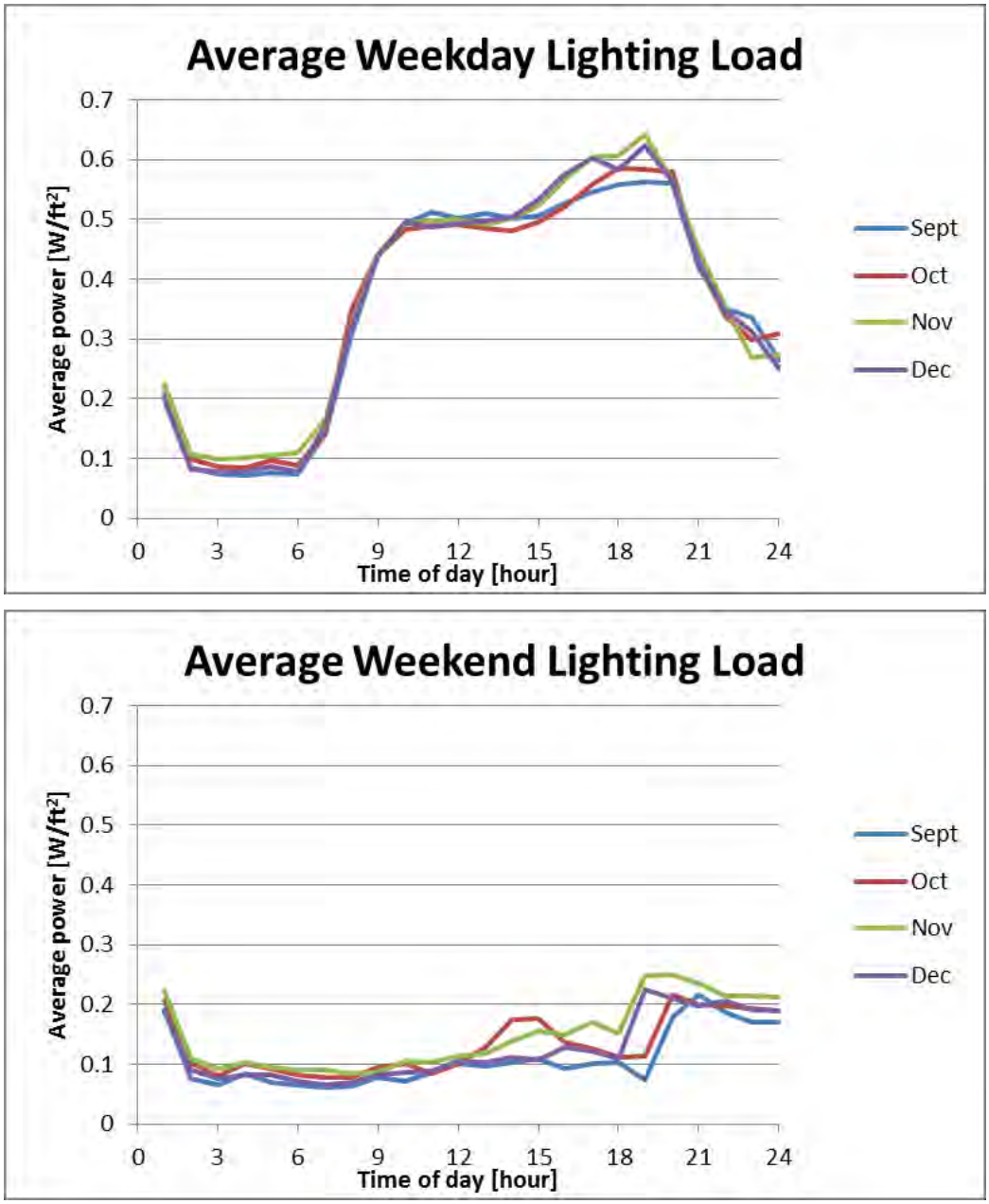
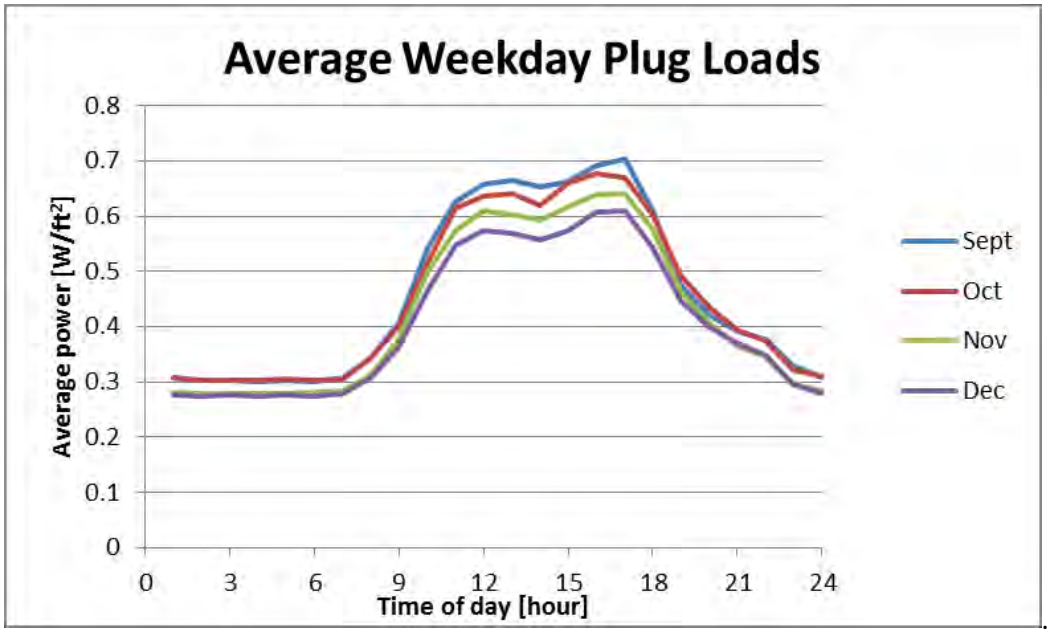


Figure 13: Monthly lighting load profiles

Plug load profiles shown in Figure 14 reveal a slight seasonal dependence with somewhat reduced loads in the winter; ~0.6 – 0.7 W/ ft^2 at peak, and 0.3 W/ ft^2 at nights and weekends.



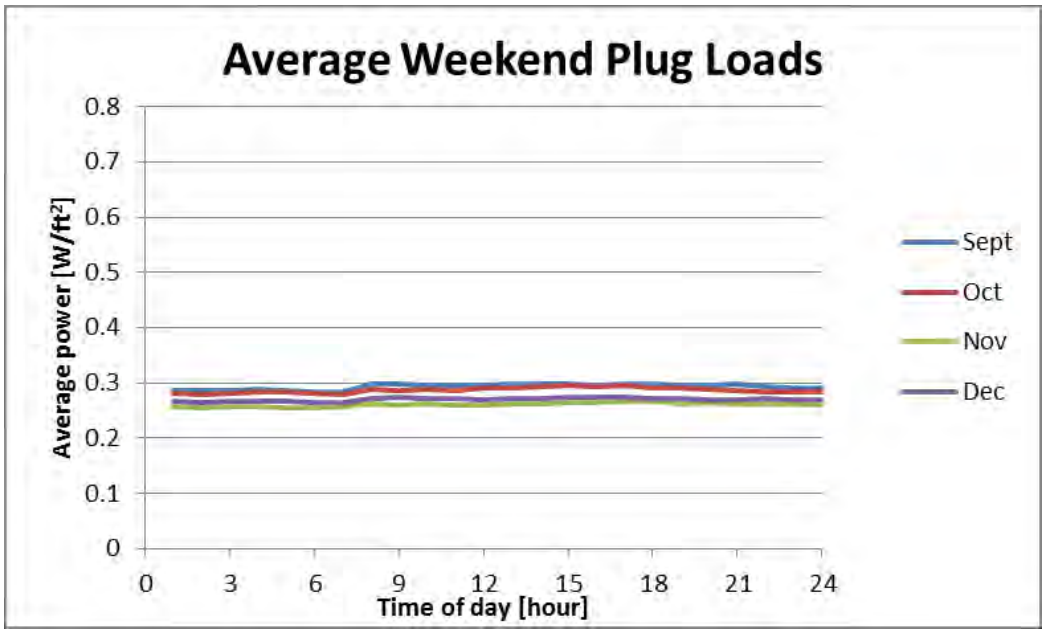
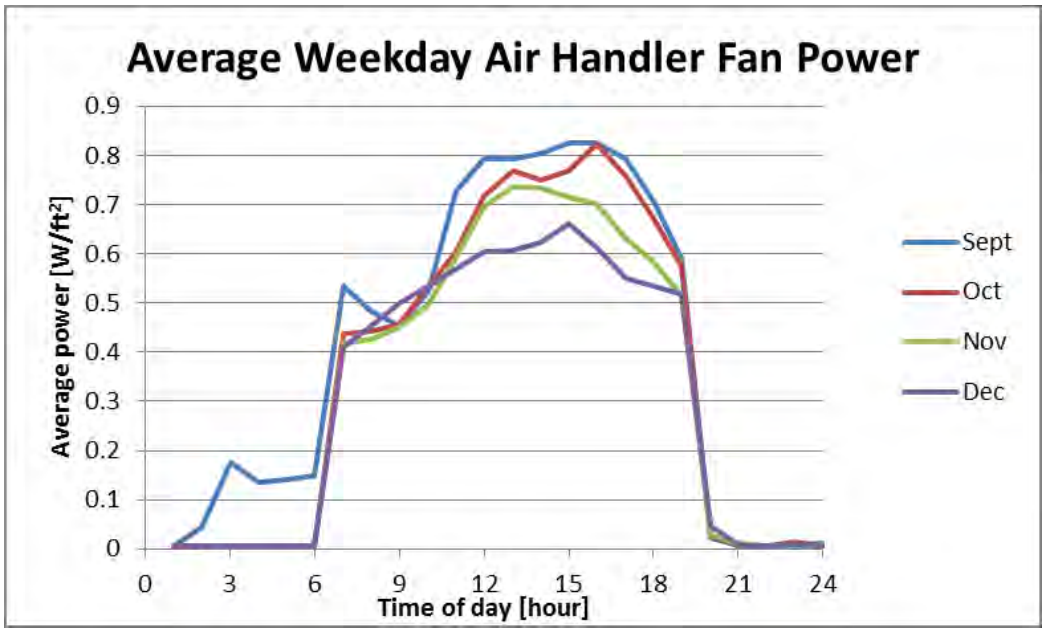


Figure 14: Monthly plug load profiles.

HVAC system - AHU

Figure 15 shows load profiles for the dedicated air handler on the 20th floor where the weekend peak load is about 25% lower than weekday.



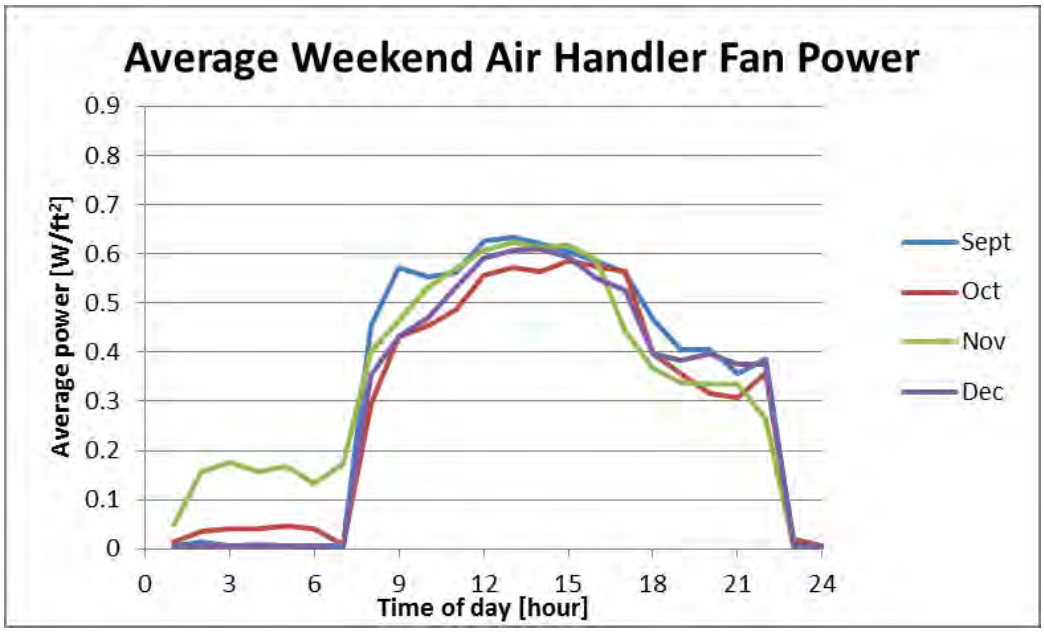


Figure 15: -Monthly air handler load profiles.

Figure 16 shows heat extracted by the chilled water system (cooling energy) and shows little change between seasons. There is only ~15% difference between weekday and weekend demand.

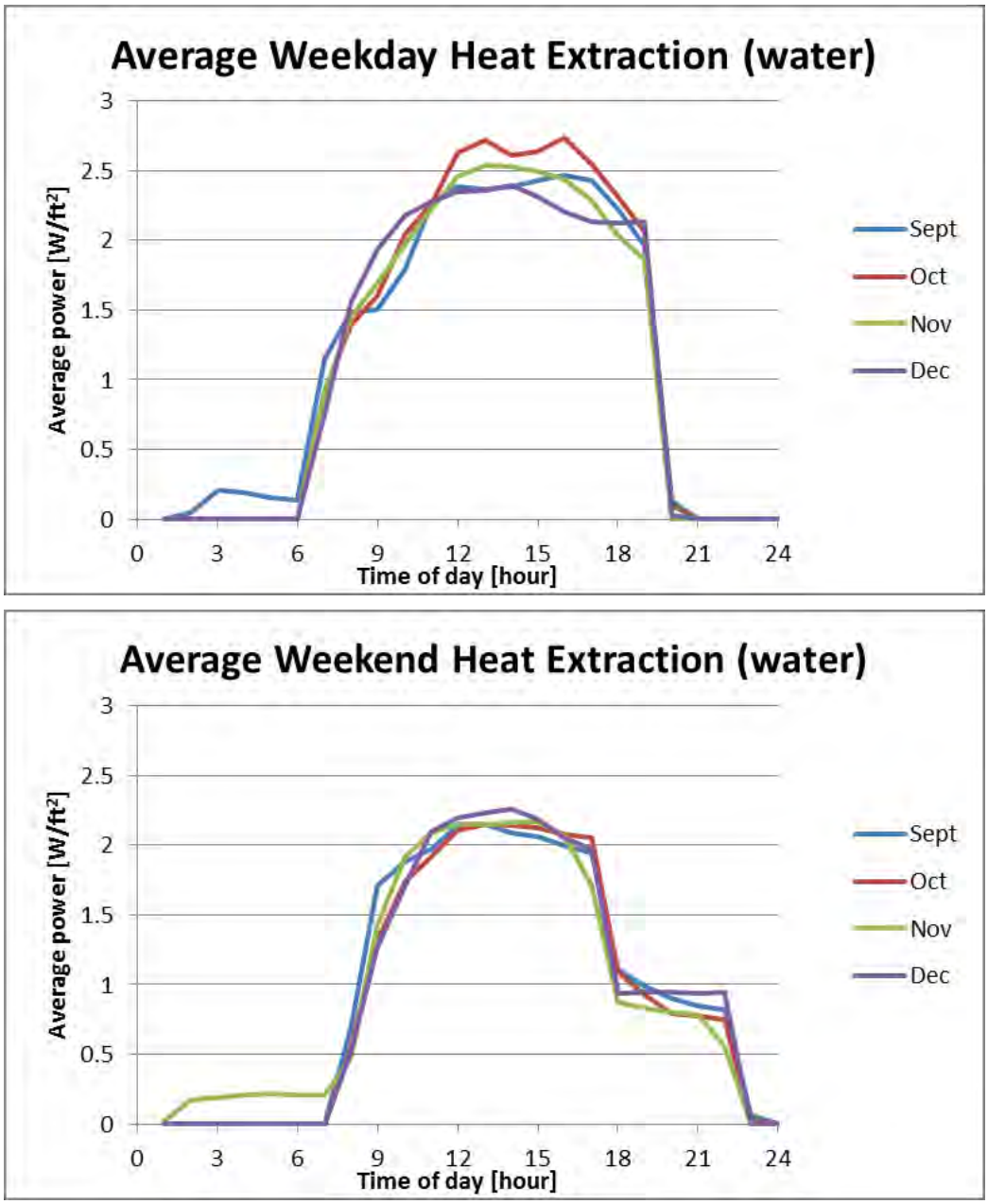
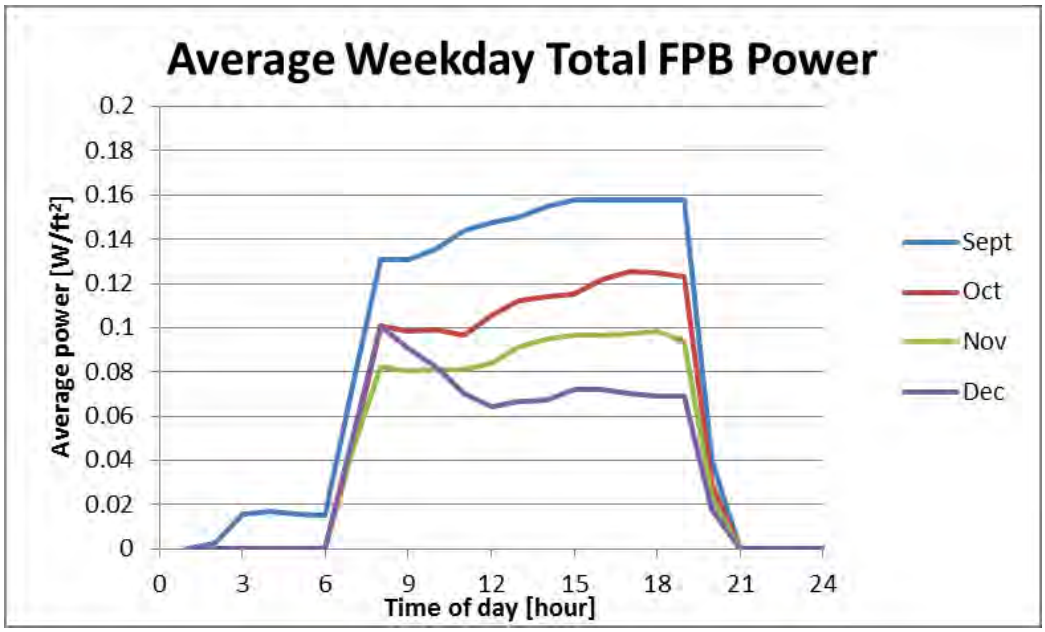


Figure 16: Monthly profile of the heat extraction.

HVAC – Terminal units

Figure 17 shows that the perimeter fan powered boxes (FPB) total power decreases in colder months since the fans run a minimum volume during heating. Weekend usage is almost as high as in weekdays but with reduced hours.



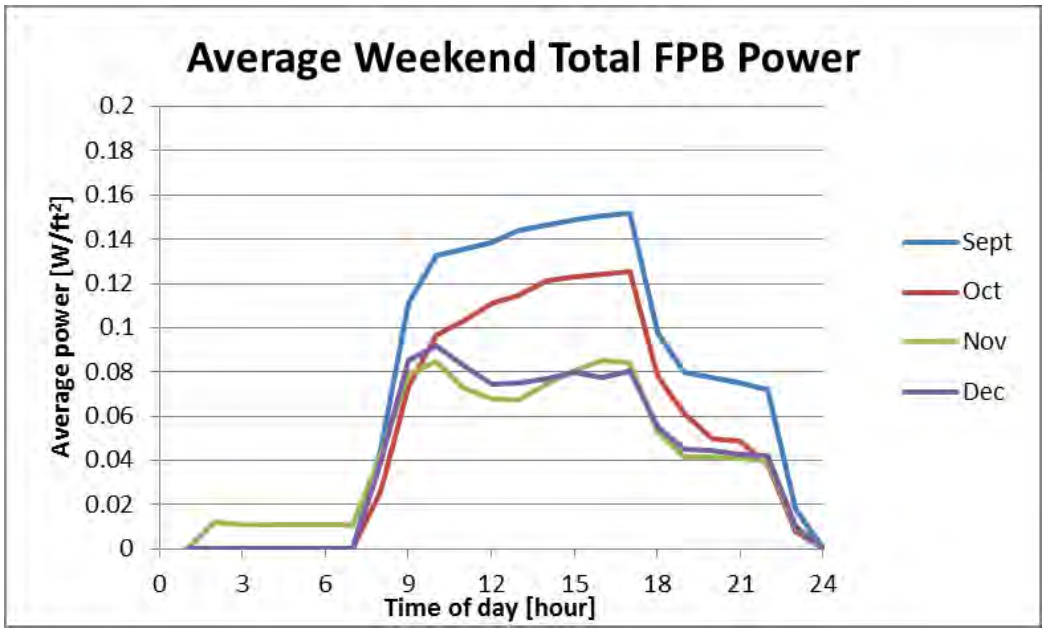


Figure 17: Mouthly FPB power usage profiles.

The flow grid calibrations for six FPBs were checked. These showed an error of up to 20%, but some boxes were low and others high. The average correction was less than 1% so no correction attempt was made to any FPB in the calculations of flow, power, and heat gains.

Figure 18 shows the power/flow relationship measured for one box at two different outlet configurations. Measurements were conducted initially with the perimeter diffusers fully open,

then were repeated with about 60% of the diffuser area blocked. There was very little difference in the power flow relationship for these two conditions. This relationship was used for every FPB in subsequent calculations of FPB energy usage. This relationship will be used in the calibrated EnergyPlus simulation to model the FPB power consumption.

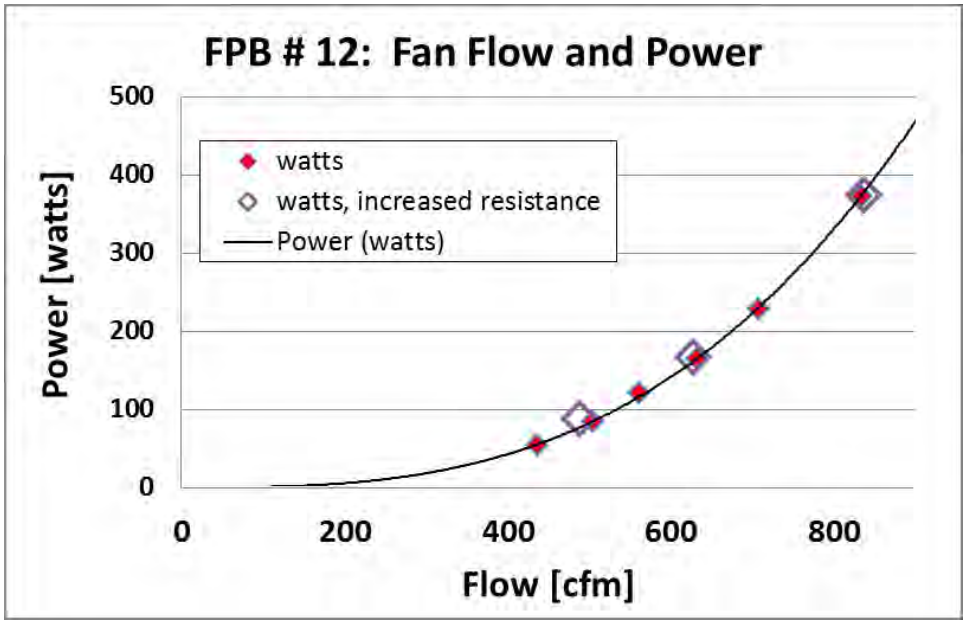


Figure 18: The power/flow relationship for FPB12; with and without increased resistance to flow.

LESSONS LEARNED AND RECOMMENDATIONS

Setpoint control to optimize thermal comfort and energy use

A review of measured space temperatures indicates that comfort conditions are being maintained close to the low (coolest) end of the comfort zone (72°F) specified by ASHRAE Standard 55-2010 for the large majority of time. This finding is consistent with results from the occupant survey conducted by the Times Company in which the large majority of people expressed satisfaction with thermal comfort, although among those expressing dissatisfaction, a slightly larger percentage of respondents indicated that conditions were too cool. Since the "comfort zone" represents a range of temperatures within which most people (80%) will be satisfied, these results suggest that cooling energy savings could be realized by raising the setpoint temperature while still maintaining similar or possibly more comfortable conditions. Previous research has demonstrated that increasing the space setpoint temperature by 2°F (1°C) can reduce cooling energy by 7-10%, depending on climate [Hoyt et al. 2009]. A similar increase in setpoint temperature is recommended for the Times Company, which would move the average comfort conditions closer to the middle of the ASHRAE comfort zone while achieving important energy savings. It is also recommended that the setpoint control logic, including the existence of a zone temperature deadband in the range of at least 2-3°F between cooling and heating setpoints for the perimeter fan-coil units (typically prescribed by building standards), within the building management system (BMS) be carefully reviewed to ensure that space setpoint control is operating to deliver efficient heating and cooling to the occupants.

Interior zone

Due to the 84-inch height of interior zone thermostats, setpoint temperatures should be at least

1-2°F higher than the desired temperature in the occupied zone to account for stratification. As described above and shown in Figure 10, the Times Company has already implemented a strategy of maintaining the setpoint temperature at the 84-inch height 1-2°F higher than the temperature at a typical (4-ft height) thermostat location. However, given the stratification that is maintained in the occupied zone for the majority of the time, it is recommended that the Times Company consider increasing the amount of this offset, so that average temperatures in the occupied zone can be increased above the low end of the thermal comfort zone.

Perimeter zone

The observed well-mixed conditions at the perimeter combined with the measured supply air temperature at the perimeter bar grilles of about 70°F suggest that relatively high airflow rates are being delivered to the space. The cooling setpoint temperature has been adjusted to be 73°F (typically, it would be 74°F), and the heating setpoint temperature has been increased to 74°F to offset the effects of the cold window frames in winter conditions (typically, it would be 70°F). These revised setpoint temperatures have been selected by the building operators to improve the overall comfort experience in the perimeter zones of the building. However, these adjustments will tend to increase both the cooling and heating energy use in the perimeter zones. It is recommended that these revised setpoint temperatures be carefully reviewed and changed, if possible, to reduce energy consumption while minimizing the impact on the employees. For example, it may be possible to reduce airflow rates (saving fan energy) by raising perimeter cooling setpoint temperatures, and thereby helping to promote more stratification while maintaining acceptable comfort conditions. Furthermore, it would be desirable to set the cooling setpoint equal or higher than the heating setpoint so that the system is not constantly trying to heat or cool when the operation switches between heating and cooling mode (sometimes several times per day).

Review of other control operations

The following additional control issues were observed during the review of monitored data from the Times Building. These lead to the following considerations for professionals designing, commissioning, and operating such systems, but would require deeper analysis to fully resolve.

1. Setpoint switching: Under some conditions in winter, setpoint switching occurs over short periods of time from 74°F heating to 73°F cooling. Operators have attempted to manage this by using supervisory commands for current setpoint in an attempt to lock the boxes into either heating or cooling mode, so the data may reflect isolated instances. This situation could be improved by implementing a deadband between heating and cooling setpoints in the fan-coil unit controllers instead of using a single setpoint imposed by the supervisory system.
2. Zone setpoint control issues: There were a number of instances observed in perimeter zones under cooling operation where the setpoints were not met (for example, see Figure 22), indicating inadequate airflow rates, too high of supply air temperatures, or both. Possible corrective actions that could be considered include: (1) check maximum airflow setting of fan-coil unit serving that zone (in some cases, settings were less than the specified design airflow capacity for the unit), (2) use a higher zone cooling setpoint temperature (see previous discussion) if acceptable comfort can still be maintained, and (3) reduce the supply air temperature at the air handler (see below).
3. AHU SAT: The AHU leaving temperature setpoint is relativity high at 66°F. This was

established by Building Operations to provide optimal comfort, and in particular avoiding overcooling in the interior zone. Lowering this AHU setpoint temperature and/or using a reset strategy, may allow the perimeter fan-coil units to control to setpoint better (see above). However, if a lower AHU SAT is used, care must be taken to reduce airflow rates to the interior zone accordingly to avoid overcooling. One approach to reduce airflow would be to review and possibly reduce the minimum volume setpoint for the zebra dampers controlling air delivery into the underfloor plenum. These strategies, along with raising the cooling setpoints (described above), and implementing a 1-3°F deadband in perimeter zones could improve overall control operation of the UFAD system.

4. Weekend and evening operation: The Times Building Operations has taken the wise approach of reviewing off-hour and weekend occupancy schedules for its various departments and implementing matching UFAD operating schedules. In the case of the Times Building, the underfloor plenum is divided into six separate low-pressure control zones per floor. This allows the UFAD system to be more efficiently operated to provide conditioning to one or more zones based on occupancy.

Overall, the operators have done an excellent job of navigating among these issues to provide an acceptable comfort level (see survey results). Without these constraints it appears that average comfort could be moved in a more positive direction.

Lessons learned drawing from the analysis of the Times Building, as well as experiences of the authors with other UFAD systems are as follows:

1. Cooling setpoints: As described above, cooling setpoints should be set higher than conventional practice with overhead systems to account for stratification and combat overcooling (a pervasive problem in the industry).
2. Heating setpoints may need to be set higher when large window to wall ratios are used (as in the Times Building).
3. It is recommended that a deadband of 2-3°F minimum be used between heating and cooling setpoints.
4. AHU supply air temperature settings (and reset) should be decided on the basis of the impact on interior zone comfort, minimum ventilation rates, and terminal unit sizing and potential cooling setpoints.
5. Linear bar grilles in the perimeter, while good for heating, provide challenges for desirable cooling (stratified) performance due to the increased mixing caused by the discharged air into the space.

The implementation of an underfloor air distribution (UFAD) system in The Times Building proved to be very successful for both The Times management and workforce. Based on the high level of satisfaction reported in the survey responses, the thermal comfort and indoor environmental quality provided by the UFAD system in combination with the other energy efficiency measures were well received. As discussed previously, whole-building energy modeling results indicate that the building also performed very efficiently compared to a code-compliant baseline building. Beyond the careful selection of complementary advanced energy efficient technologies, the overriding reason behind the high quality environment achieved at The Times was the commitment and attention paid to installing and commissioning the various systems over time. In the case of the UFAD system, operators were able to fine-tune control strategies and thermostat setpoints to provide a thermal environment that was stable and comfortable across the entire

floorplate. Due to this consistent operation and reduced occupant complaints, the operators expressed strong support for the choice of a UFAD system.

Additional lessons learned from the current operation of the UFAD system in The Times are still under consideration for further tuning-up of the building and can also serve as guidelines for improving the application of UFAD in new and retrofit buildings.

- Pay attention to the placement of floor diffusers in the vicinity of building occupants. Although most occupants will adjust their nearby floor diffuser very infrequently, the availability of having some amount of personal control is very important to occupant satisfaction and should always be provided if possible.
- Temperature stratification is an important characteristic of a well-operated and controlled UFAD system. At The Times, good stratification (2-4°F in the occupied zone) was maintained in interior zones. However, vertical mixing provided by the linear bar grilles at the perimeter tended to eliminate almost all stratification near the windows. Efforts to increase stratification in the perimeter could reduce energy consumption, but care must be paid to maintain acceptable comfort levels.
- In stratified environments, thermostat temperature setpoints (at 4-ft level) should be adjusted to account for the cooler temperatures near the floor. The goal is to maintain the average temperature in the occupied zone (up to 6-ft height) at the desired setpoint temperature. For example, in a stratified space, it may be necessary to raise the thermostat setpoint by 1-2°F. In fact, in The Times Building, temperature measurements demonstrated that most average occupied zone temperatures in the stratified interior zones were near the low end of the thermal comfort zone defined by ASHRAE Standard 55-2010. Raising the thermostat setpoints in these areas could provide an opportunity to reduce energy use (by reducing airflow quantities) without compromising comfort, and possibly improving comfort.
- A commonly observed operating condition in UFAD buildings is a tendency to "overair" the conditioned space. It is recommended to conduct a careful review of airflow quantities, including checking the minimum ventilation rates that are set in the underfloor VAV terminal units in the perimeter, as well as the primary VAV controls serving the underfloor plenum.
- Another option for control adjustment that can be considered in terms of improving comfort and/or energy performance is to change the supply air temperature leaving the air handler.

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Hoyt, T., K.H. Lee, H. Zhang, E. Arens, and T. Webster. 2009. Energy savings from extended air temperature setpoints and reductions in room air mixing. *Proceedings from International Conference on Environmental Ergonomics*, Boston, August 2 – 7.

Appendix 2.2.1: Building Performance Evaluation Toolkit Specifications

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Center for the Built Environment (CBE)
University of California, Berkeley

April 2, 2013

INTRODUCTION

This report summarizes system specifications for the Building Performance Evaluation (BPE) toolkit that CBE developed under both the PIER "Advanced Integrated Systems Technology Development" (Project No. 500-08-044) and CIEE/CEC PEIR "Wireless Measurement Tools for Better Indoor Environments" (project No. 500-10-048). Toolkit hardware details are contained in Appendix 2.2.2: BPE Toolkit Wireless Hardware Device Report. A full report of this project is pending publication (Webster et al. 2013)

This report provides an overview of the system specifications for the BPE Toolkit. It consists of three main sections: *System architecture* provides an overview of the entire systems and the interrelation between the various components as well as specifications for the main infrastructure components; the *Toolkit hardware* section outlines the set of IEQ measurement tools that use a wireless mesh networking system; the *Toolkit software* section outlines an open-source, web-based analysis and reporting tool for evaluating IEQ performance data.

SYSTEM ARCHITECTURE

Wireless mesh networks provide a communication connection between sensors and allow a single point of data storage. Figure 1 provides an overview of how system components link together to achieve this single data collection location.

At the building level, a set of sensors/devices is connected to wireless mesh nodes (also named motes) that transmit data to a local buffering database. This buffering database (Postgres SQL running on an

Advantech ARK 1503 fan less embedded PC) is connected to the Internet via either a building network connection or a cellular broadband connection. Data is sent through this Internet connection to an application server located outside of the building. Because the data is accessible through the Internet, data access is possible from inside and outside of the building network.

In addition to the set of sensors and devices in the Toolkit, an optional connection between the Building Management System (BMS) and the Internet can be made to facilitate read-access of BMS data from the same location as Toolkit data. Both the BMS and Toolkit data connections are made via a secure connection using the Simple Mapping and Actuation Profile (sMAP) (Stephen Dawson-Haggerty et al. 2010) that is detailed in the toolkit software section below. Drivers for Johnson Controls Metasys, Siemens Apogee, and Automated Logic Controls have been used successfully to import BMS data in real-time.

On-site access by practitioners is provided by either a netbook (Asus Eee PC Seashell Series) computer or tablet (I-pad, 32 mb). Remote access is available from any computer connected to the Internet.

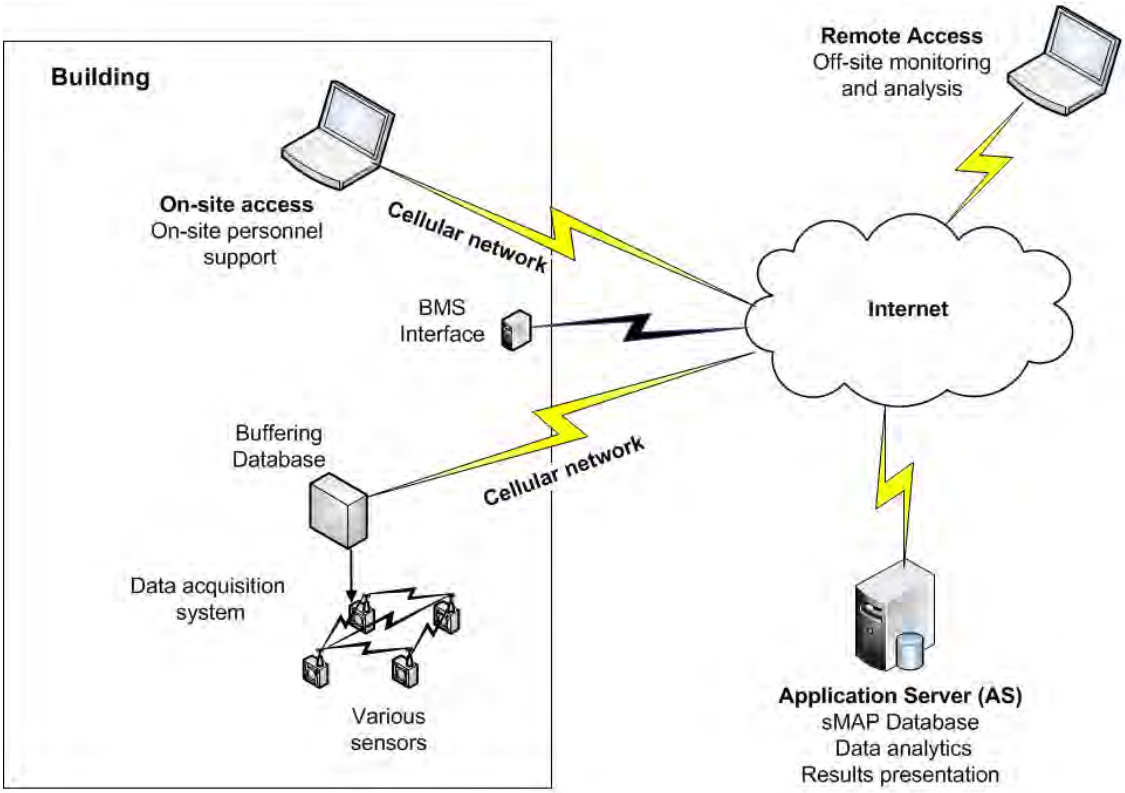

Figure 1: System architecture

TOOLKIT HARDWARE

The hardware components of the Toolkit include a wireless mesh networking system, sensors, and custom devices designed to house multiple sensors. Table 1 provides an overview of the Toolkit instrumentation for a system including 20 Indoor Climate Monitors (ICMs) and one underfloor air distribution commissioning cart (PUCC). A full description of these devices and their detailed specifications can be found in Appendix 2.2.2.

Table 1: Toolkit instrumentation summary

Basic Level	Sensor/Instrument	Accuracy (±)	Quantity
Acoustics	Sound level meter	-	1
Indoor Air Quality	CO_2	30 ppm + 3% measured value	20
Lighting/Daylighting	Illuminance	5%	20
	Camera (HDR photography for luminance)	-	1
Thermal Comfort	Infrared temperature (surface temperature)	2 ˚C or 1.5% of reading	2
	Thermistor (air and globe temperature)	0.056 ˚C	50
	Anemometer (air speed)	0.075 m/s	20
	Relative humidity	2%	20
	Differential pressure	1%	1
Wireless System	Mote + IO Board	-	20
	Embedded computer	-	1
	Tablet w/ 4G cellnet	-	1
	Wireless router	-	1
Total Cost (as implemented in the project described herein)			

TOOLKIT DEVICES OVERVIEW

The Toolkit includes several single and multiple-sensor devices that simplify the process of collecting IEQ data in buildings. This section details the design and implementation of those devices.

Indoor Climate Monitor (ICM) – Indoor air quality, lighting, and thermal comfort

The ICM is a wireless device that is capable of sensing PMP-suggested thermal comfort, lighting, and indoor air quality parameters. This device is designed to be placed on an occupant's desk and to measure dry-bulb temperature, globe temperature, air speed, relative humidity, horizontal illuminance, and CO_2 concentration. Continuous measurement of sound levels was not deemed necessary (or recommended in the PMP) and thus an acoustics meter was not included on the ICM (see below).

Portable UFAD Commissioning Cart (PUCC) – Advanced thermal comfort

The Portable UFAD Commissioning Cart (PUCC) was designed to be a portable and wireless alternative to a previously CBE designed UFAD commissioning cart (T Webster, Bauman, and Anwar 2007). Underfloor

air distribution (UFAD) is a type of air distribution system in which air is delivered in the occupied space from an underfloor plenum which results in a stratified room environment. The PUCC measures temperature at 0.1 m, 0.25 m, 0.6 m, 1.2 m, 1.7 m, and 0.1 m from the ceiling to capture the stratification as well as floor and ceiling surface temperatures using infrared temperature sensors (IRTs). Underfloor plenum temperature and pressure are also measured.

Acoustics measurement

The Toolkit includes a Larson Davis LxT sound level pressure meter connected to a wireless mote. At the basic level, the PMP requires A-weighted sound pressure level measurements in representative spaces.

TOOLKIT SOFTWARE

The Toolkit software consists of the data management backend and the analysis and visualization web frontend. The open-source code for this frontend is hosted at http://code.google.com/p/cbesmap. The documentation for the frontend is hosted at http://smap.cbe.berkeley.edu/static/doc/_build/html/.

BACKEND DETAILS

There are two backends that support the web frontend of the Toolkit: the Simple Mapping and Actuation Profile (sMAP) system and Django (PostgreSQL). sMAP handles the collection and retrieval of all time-series data. Django handles the relational aspects of the backend: metadata, users, groups, security, and project information. In the context of the Toolkit, metadata refers to descriptive data that is tied to the sensor data. Metadata is primarily composed of spatial and temporal information, but also includes other information that is detailed later in this section. Django is a python-based web development framework designed for rapid development of database driven websites (Anon.). The Toolkit uses Django, coupled with a PostgreSQL database, to allow for simple python-based interaction with sMAP.

sMAP is a set of tools to enable simple and efficient exchange of time-series data through web-enabled applications (Stephen Dawson-Haggerty et al. 2010). sMAP has three major components which are shown in Figure 2.

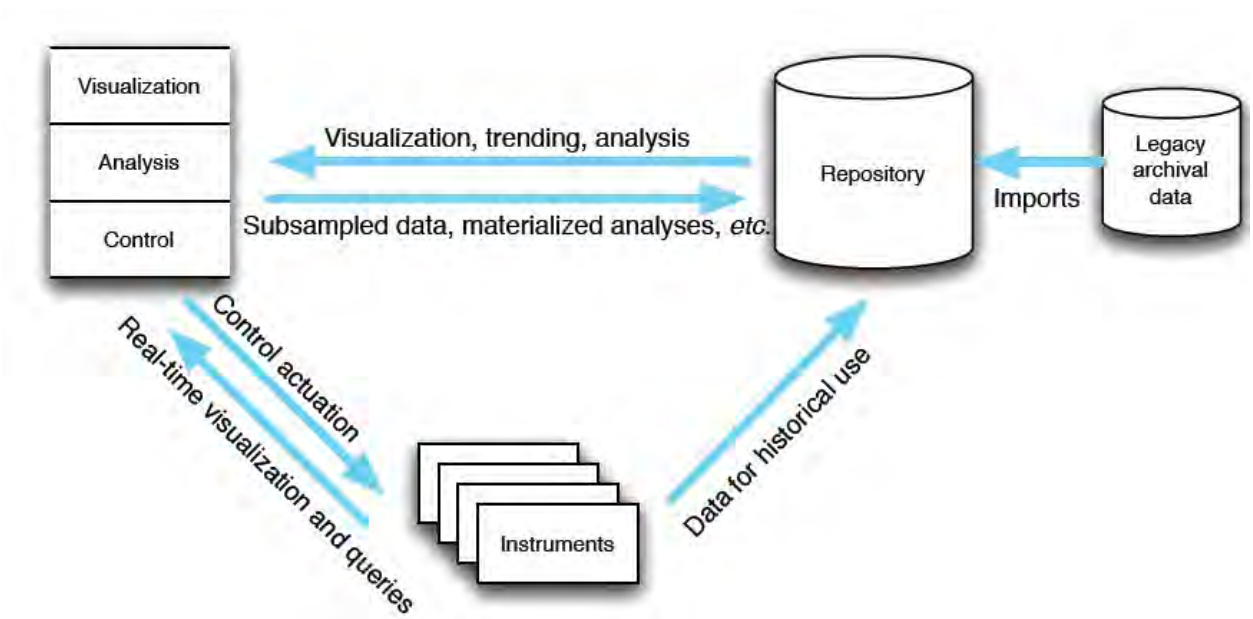

Figure 2: sMAP components and data exchange paths (S. Dawson-Haggerty)

1. Instrument drivers: A library of instrument drivers is available to enable the connection of devices to sMAP through HTTP. There are drivers for wireless devices, for BMS systems (Johnson Controls, Siemens, and Automated Logic Controls), weather services, power meters, and others. Additionally, new drivers are easily written in Python based on the existing example drivers.
2. Repository: The sMAP repository (Archiver) is a database system optimized for time series data (fast-retrieval, efficient compressible storage). The repository also includes a querying language that allows simple retrieval and manipulation of data based on metadata filtering.
3. Web frontend: sMAP comes with an example web-frontend that is a full-featured trend viewer. This frontend example served as the model for the Toolkit frontend.

sMAP simplifies the handling of time-series data. The sMAP querying language allows fast retrieval of data based on user-defined metadata. This querying language also allows on-the-fly manipulation of data streams, allowing users to apply mathematical functions to streams of data (e.g. resample, average).

FRONTEND OVERVIEW

The web frontend of the Toolkit is built on top of the backend using HTML and Javascript. The frontend is used for three main tasks: (1) setup and collection of sensor data and metadata, (2) real-time analysis of data, and (3) scorecard and report generation.

Toolkit setup and data collection procedure

An overview of the toolkit setup and data collection procedure is provided in Figure 3.

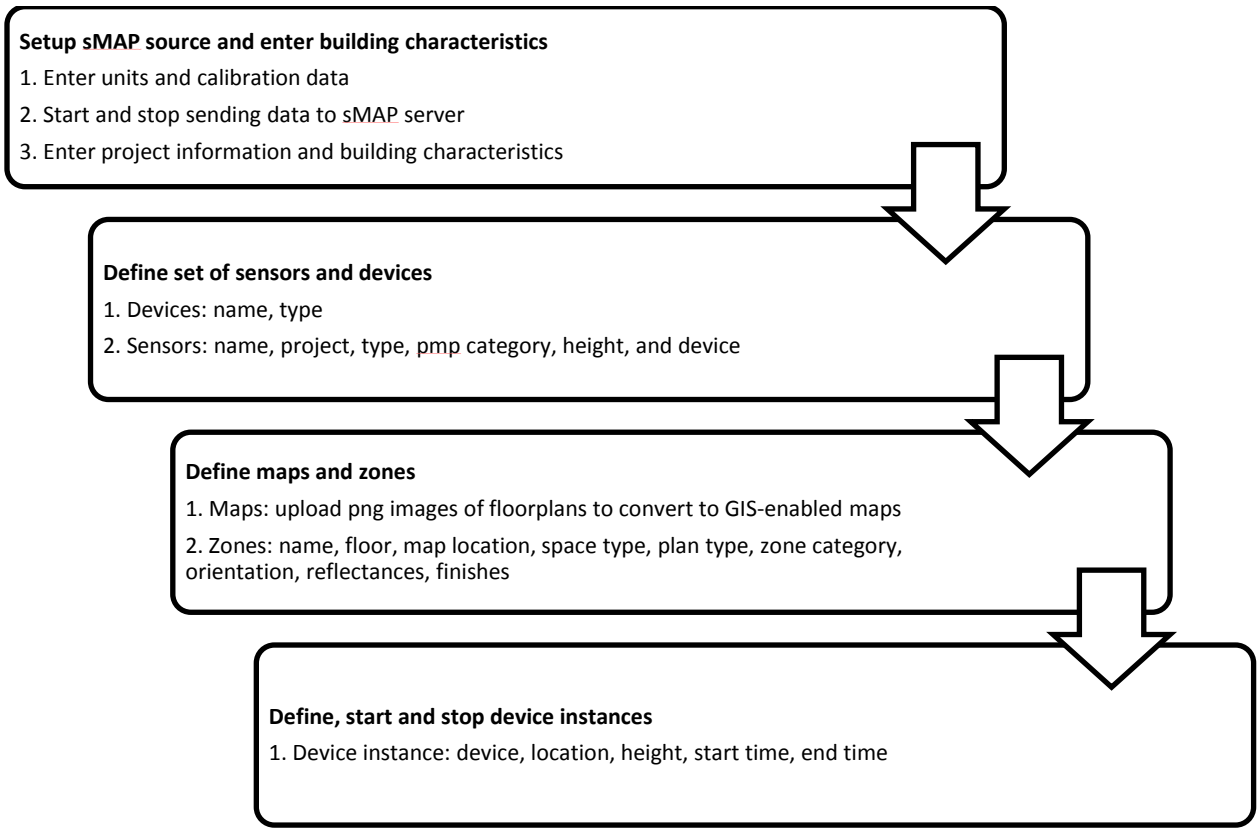

Figure 3: Overview of project setup

The use of GIS-enabled floor plan maps are generated using MapTiler (Anon.) with the Olwidget

framework for OpenLayers (Anon.). These tools allow users to draw zones (Figure 4) and specify testing locations on a GIS-enabled floor plan map (Figure 5), which is also used for data analysis.

Figure 4: Users can draw and edit zones on a GIS-enabled floor plan

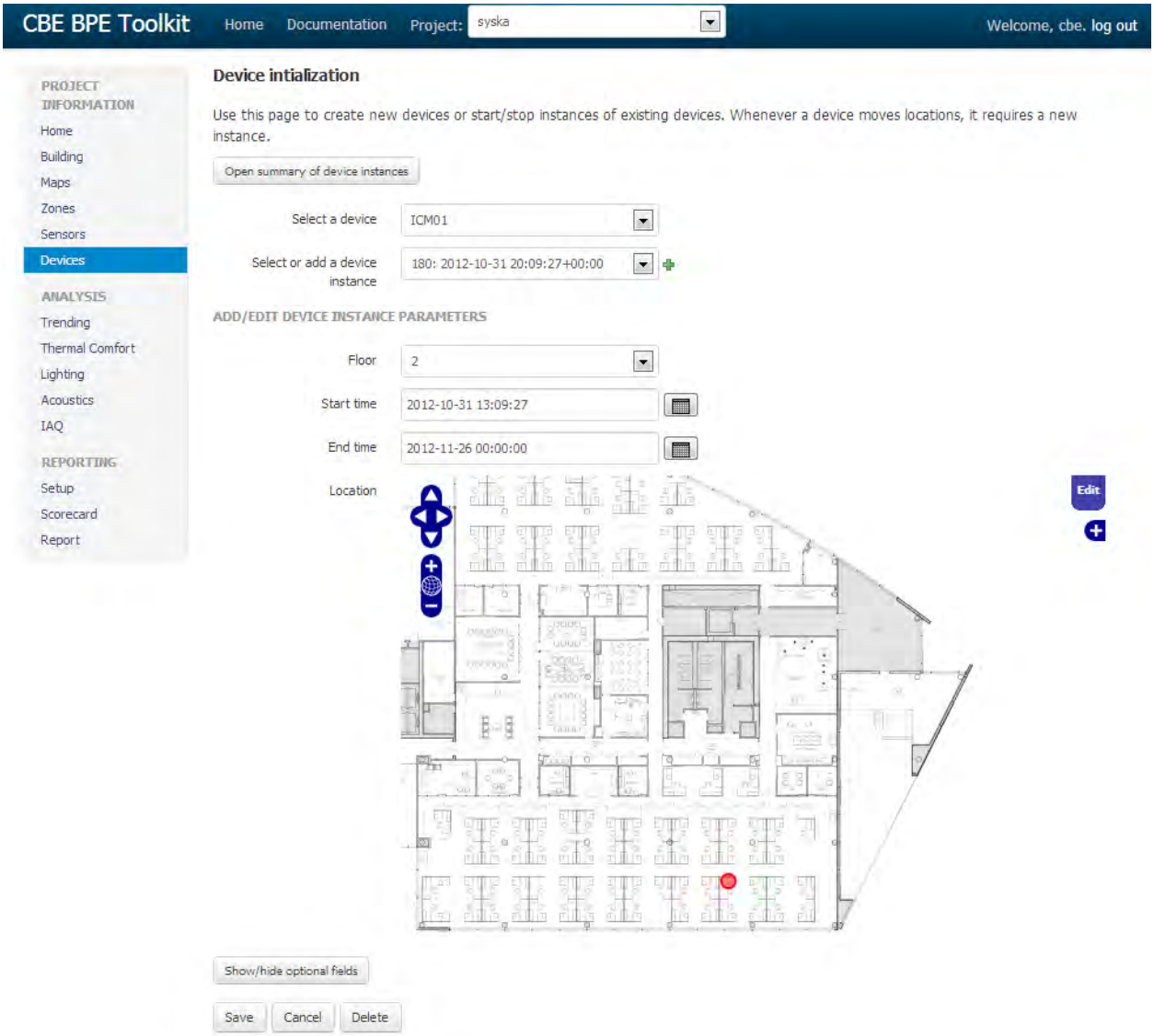

Figure 5: Toolkit webpage for adding a new device instance (measurement)

Real-time analysis of sensor data

The analysis capabilities of the web-based frontend are summarized in Table 2. All analyses are available on real-time data, helping users to catch instrumentation problems and arrive at actionable results faster, potentially shortening the data collection period. The data can be both temporally and spatially filtered to drill down into specific spaces or time-periods. Additionally, the user can aggregate data spatially to provide summary charts for spatial groups (e.g. orientation or space-type).

Table 2: Summary of frontend analysis capabilities

Analysis method	IEQ categories	Description
Trending	All	The trending application allows real-time trend analysis of any data stream.
Setpoint	Thermal	Setpoint analysis is designed to compare any two parameters, though

analysis	comfort	typically it is used to assess how well a device is controlling to its setpoint, such as zone temperature compared to zone setpoint or air handler static pressure and static pressure setpoint. This particular analysis is most useful when BMS data are available, though Toolkit device data can also be used. For example, this analysis could be used to check the accuracy of zone thermostats by comparing BMS thermostat readings to Toolkit readings of calibrated devices placed next to the thermostats.
Comfort zone analysis	Thermal comfort	The Toolkit comfort zone analysis webpage allows users to analyze the comfort data from the Toolkit Indoor Climate Monitors (ICMs). The setup for the comfort zone chart is similar to the setpoint analysis chart except instead of defining setpoint pairs, the user can choose to filter according to spatial parameters, such as orientation or zone type. Users have multiple chart types to choose from, including: relative humidity vs. operative temperature, CBE Thermal Comfort Tool for ASHRAE 55, time-series charts and map-based charts for PMV and operative temperature.
Stratified systems analysis	Thermal comfort	Stratification refers to the increasing temperature gradient of the air in a space that is conditioned by a stratified system: an underfloor air distribution system (UFAD) or displacement ventilation system (DV). The Toolkit includes two analysis types for analyzing stratification data: room-air stratification and comfort zone analysis – stratification.
Whole test-period analysis	Acoustics	The whole test period analysis is used for short-term tests. There are three chart types to choose from: column, boxplot, and map. The column chart shows the average over the whole test period for each test that matches the filtering. The boxplot chart will take the length of the test and split it into 1-minute chunks which are then summarized as a boxplot. The map chart will show the mean value over the whole test period for the zone in which the measurement was taken colored according to a user-definable scale.
Time-slices analysis	Lighting, IAQ	The time slices analysis is used for long term tests and is dependent on the resample/average rate chosen by the user. For example, a user could look at hourly lighting values from an ICM with this analysis. There are three chart type options: line, boxplot, and map.
Performance summary model analysis	All	The performance summary model analysis is used to determine what percentage of measured values fall within certain classes. See (Heinzerling et al. 2013) for more details.

Scorecard and report generation

The Toolkit scorecard and report generation features enable users to summarize and communicate performance results quickly. The scorecard is based on both subjective and objective measurements and is detailed in (Heinzerling et al. 2013). Report generation is a tool that pulls together user-saved analysis charts and places them in a PDF-exportable document that includes documentation on how the data were collected and tips on appropriate interpretation of the charts.

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Appendix 2.2.2:
BPE Toolkit Wireless Hardware Device Report

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University of California, Berkeley

April 2, 2013

INTRODUCTION

This report summarizes the hardware aspects of the Building Performance Evaluation (BPE) toolkit that CBE developed under both the PIER "Advanced Integrated Systems Technology Development" (Project No. 500-08-044) and CIEE/CEC PEIR "Wireless Measurement Tools for Better Indoor Environments" (project No. 500-10-048). Specifications for this system are contained in Appendix 2.2.1: BPE toolkit specifications. This report provides a description of the semi-final version toolkit hardware devices currently based on older versions of Dust Networks technology as offered by Vigilent. Yet to be completed under the CIEE/CEC contract is the final version of the wireless (mote) hardware based on the latest offerings from Dust Networks that will be integrated into the devices discussed herein. See the CIEE/CEC contract final report for details of these devices. (Tom Webster et al. 2013)

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Initially this hardware was developed as a part of the BPE Toolkit that was closely aligned with ASHRAE's Performance Monitoring Protocol (PMP) guide. However, we have departed from the PMP in several respects and added a performance evaluation component which has evolved and expanded the original scope. The hardware described here is just one component of the BPE Toolkit system that is fully described in Webster et al. [2013 In Press]

TOOLKIT HARDWARE

The hardware components of the Toolkit include a wireless mesh networking system, sensors, and custom devices designed to house multiple sensors. Usability and accuracy were the major objectives behind the Toolkit hardware design. Cost also played an important role, though costs were assumed to be high for a prototype design. The word "usability" masks a broad set of design parameters that

together achieve an intuitive and usable system. The following sections will highlight where decisions were made to achieve greater usability within the target group of commissioning agents, mechanical/electrical/plumbing (MEP) consultants, and building operators.

TOOLKIT DEVICES

The Toolkit includes several single and multiple-sensor devices that simplify the process of collecting indoor environmental quality (IEQ) data in buildings. This section details the design and implementation of those devices.

Indoor Climate Monitor (ICM) – Indoor air quality, lighting, and thermal comfort

The Indoor Climate Monitor (ICM) was developed as part of a previous CBE research project involving occupant comfort in buildings with operable windows (Paliaga 2004; Brager, Paliaga, and de Dear 2004). While the primary shells of the original ICMs were reused for the Toolkit, temperature and relative humidity sensors were replaced (for compatibility and increased accuracy), an illuminance sensor and a CO_2 sensor were added to the device, and all sensors were wired to two Vigilent motes, see Figure 2. . The new ICM is a wireless device that is capable of sensing PMP-suggested thermal comfort, lighting, and indoor air quality parameters. This device is designed to be placed on an occupant's desk and to measure dry-bulb temperature, globe temperature, air speed, relative humidity, horizontal illuminance, and CO_2 concentration. Continuous measurement of sound levels was not deemed necessary (or recommended in the PMP) and thus an acoustics meter was not included on the ICM. Figure 1 and Figure 2 show the outside and inside of the ICM device respectively.

The set of sensors chosen for the ICM represent a compromise in cost and accuracy, though all the sensors were chosen with accuracy and interchangeability as primary factors. The following three sections discuss the different hardware and applications that were developed for the ICM.

Figure 1: ICM device with CO_2, illuminance, globe, air velocity, dry bulb temperature, and relative humidity

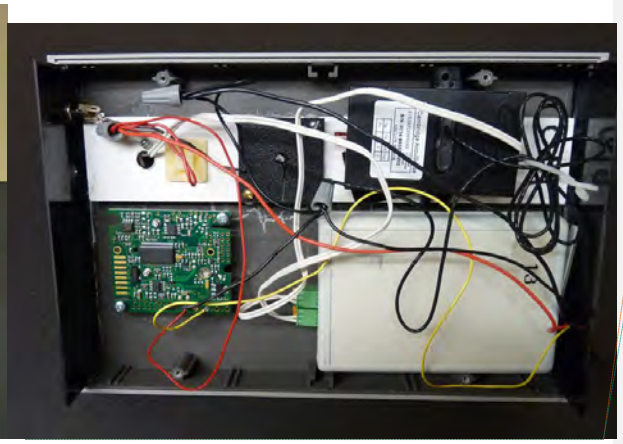

Figure 2: Inside view of the ICM device (showing one of two motes)

Comment [SSP2]: If you remove the background (it should be easy) the figures would improve.

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Thermal comfort sensors

The ICM measures dry-bulb temperature using a radiation-shielded thermistor. Both the globe

temperature sensor and the dry-bulb temperature sensor use thermistors that are accurate to 0.1°C with 1% interchangeability. Each thermistor was calibrated using a dry-well temperature calibration unit while connected to the wireless input-output board that computes temperature based on a 10,000-ohm reference resistor. The details of the radiation shielding and ICM globe temperature sensor are available in Paliaga (Paliaga 2004). Additional theory behind the globe temperature sensors are available in (Benton, Bauman, and Fountain 1990; Humphreys 1977; Fountain 1987). Currently, operative temperature is computed as the average of mean-radiant temperature (assumed here to be equal to globe temperature) and dry-bulb temperatures. Future work will add the ability to compute other thermal comfort parameters including Standard Effective Temperature (SET).

Lighting sensor

The basic level of the PMP suggests measurements of horizontal illuminance in areas that were deemed problematic in an occupant survey. At the intermediate level, the PMP suggests full grid measurement of horizontal illuminance and luminance measurements of areas with potentially problematic glare. While a hand-held Licor illuminance meter may be used to obtain full-grid illuminance measurements as suggested by the PMP, such a procedure is impractical and overkill for the purposes of IEQ evaluation. The ICM is capable of measuring horizontal illuminance, but not luminance. The Toolkit can use HDR photography coupled with the lighting simulation program Radiance to evaluate luminance information, though analysis is not currently integrated with the web-based software application that supports the Toolkit. Future implementations of the Toolkit web application aim to include luminance analysis methods similar to those provided by (Konis 2012).

Horizontal illuminance is measured using a Licor Photometric sensor which has cosine correction and is accurate to ±5%. An amplification circuit was built to convert the μA signal from the sensor into a 0-10V signal that the mote can interpret. The Licor sensors were compared against a recently calibrated Minolta T-1H illuminance meter which is accurate to ±2% to obtain calibration coefficients of reasonable relative accuracy.

Indoor air quality sensor

Indoor air quality is a complex science and accurate measurement techniques are typically difficult and expensive. For typical commercial buildings that do not have specific outdoor air quality problems (PM_{10}, $PM_{2.5}$, ozone, or air-toxics non-attainment problems), the primary method for managing IAQ is to guarantee an adequate outside airflow rate (ANSI/ASHRAE 2010). For this reason, the basic and intermediate levels of the PMP require verification of outdoor air flow rates to ensure compliance with ASHRAE Standard 62.1 (ANSI/ASHRAE 2010). The Toolkit deviates from the PMP and does not include a tool for the measurement and analysis of outdoor airflow rates, though such a tool could be added in the future. Methods for accurately measuring outdoor airflow rates can be complex and require access to multiple mechanical spaces in a building.

For these reasons, CO_2 measurement was chosen as the parameter to indicate indoor air quality because of its prevalent use in buildings for demand-controlled ventilation and as an effective proxy for occupant generated pollutants. The European Standard 15251 (CEN 2007) allows the control of outdoor airflow rate as a function of the CO_2 concentration. Ozone, volatile organic compounds (VOCs), and particulate matter ($PM_{2.5}$ and PM_{10}) were also considered, though reasonably priced sensors were deemed to be too inaccurate to provide valuable IAQ performance evaluation. The PMP suggests that CO_2 measurement is a highly inaccurate, but nevertheless potentially useful tool for diagnosing ventilation issues. Persily (Persily 1997) provides details on the connection between CO_2 measurement and IAQ and how to appropriately interpret CO_2 measurement as an indicator of IAQ.

The ICMs provide the opportunity for making multiple local CO_2 measurements in one zone, whereas

most buildings with CO_2 sensors have only one sensor per zone. A K-30 CO_2 module with 1% repeatability and 3% accuracy was selected for the ICM as a balance between accuracy and cost. The sensor uses the automated baseline calibration (ABC) method for self-correction. This method assumes that the lowest CO_2 measurement in a building will be 400 ppm (baseline outdoor level). The sensors were spot checked against an EGM-4 CO_2 sensor by PP Systems that has an accuracy of <1% and found to be within 50 ppm.

Portable UFAD Commissioning Cart (PUCC) – Advanced thermal comfort

The Portable UFAD Commissioning Cart (PUCC) was designed to be a portable and wireless alternative to a previously CBE designed UFAD commissioning cart (T Webster, Bauman, and Anwar 2007). Underfloor air distribution (UFAD) is a type of air distribution system in which air is delivered in the occupied space from an underfloor plenum which results in a stratified room environment. The PUCC measures temperature at 0.1 m, 0.25 m, 0.6 m, 1.2 m, 1.7 m, and 0.1 m from the ceiling to capture the stratification as well as floor and ceiling surface temperatures using infrared temperature sensors (IRTs). Underfloor plenum temperature and pressure are also measured. Figure 1 is a photograph of the PUCC.

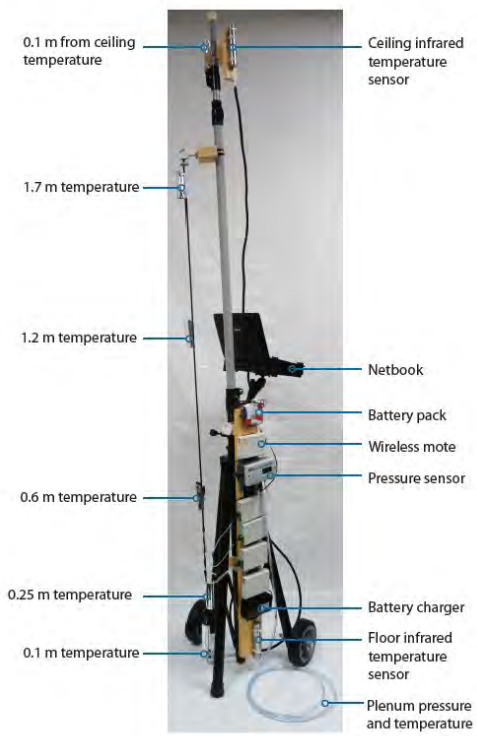

Figure 1: Portable UFAD commissioning cart

Acoustics measurement

The Toolkit includes a Larson Davis LxT sound level pressure meter connected to a wireless mote. At the basic level, the PMP requires A-weighted sound pressure level measurements in representative spaces.

At the intermediate level, the PMP requires octave band analysis to be performed by an acoustics consultant. The Toolkit deviates from the PMP in this regard and does not include a tool or analysis method for completing octave band analysis, though the LxT meter has the add-on capability if such analysis were deemed appropriate in the future.

Hardware costs

Table 1 provides an overview of the Toolkit instrumentation and cost based on off-the-shelf pricing in low volumes (unless otherwise noted) for a system including 20 Indoor Climate Monitors (ICMs – see section 0) and one underfloor air distribution commissioning cart as described above.

Table 1: Toolkit instrumentation summary

Basic Level	Sensor/Instrument	Accuracy (±)	Quantity	Cost (per sensor)
Acoustics	Sound level meter	-	1	$1495
Indoor Air Quality	CO_2	30 ppm + 3% measured value	20	$65
Lighting/Daylighting	Illuminance	5%	20	$440
	Camera (HDR photography for luminance)	-	1	$200
Thermal Comfort	Infrared temperature (surface temperature)	2 ˚C or 1.5% of reading	2	$345
	Thermistor (air and globe temperature)	0.056 ˚C	50	$9
	Anemometer (air speed)	0.075 m/s	20	$385
	Relative humidity	2%	20	$45
	Differential pressure	1%	1	$273
Wireless System	Mote + IO Board	-	20	$650[1]
	Embedded computer	-	1	$1000
	Tablet w/ 4G cellnet	-	1	$700
	Wireless router	-	1	$200
Total Cost (as implemented in the project described herein)				$36,708

[1] Research level pricing at low volume

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Visualizing Energy Information in Commercial Buildings: A Study of Tools, Expert Users, and Building Occupants

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Center for the Built Environment (CBE)
University of California, Berkeley

November 2011

EXECUTIVE SUMMARY

The overarching goal of this research is to identify the optimal methods for visualizing building performance information in commercial buildings. The current phase consists of three research activities: (1) product reviews of building visualization software products; (2) a study of expert users regarding attitudes and practices regarding visualization of energy and other building metrics; and (3) a survey of workplace occupants to understand how they use building energy information, and whether improved access to this information will encourage energy conservation behaviors in the workplace.

Building Visualization Product Reviews

The authors reviewed seven software products available for visualizing energy consumption information in commercial buildings. Companies and product offerings in this space are evolving rapidly, so this report provides a snapshot of trends at the time of this investigation, which was largely conducted in the spring of 2010. The review found that these products show promise for providing feedback about energy and other metrics, and have been reported anecdotally to encourage energy conservation behaviors. These products support multiple users and provide a standard set of features that cater to the needs of both expert users and typical building occupants. Almost all tools surveyed also support tailoring of interfaces to meet specific user needs, however the degree of configurability varies between products. Many of the providers of these products report growing customer bases and claim that the use of these products has promoted energy conservation behaviors. However, more research is needed not only to assess the usefulness of these products but also to validate claims on positive outcomes of use.

Expert User Needs Assessment

The second primary task of this research was to understand and document the energy information practices, needs and preferences of an important category of users — industry professionals who are experts in energy monitoring and analysis. A combination of surveys and "contextual inquiries" was used to understand how these individuals use building energy information and what tools they currently use to visualize it. The purpose of the study was to understand their experiences as users, to learn how useful and usable current tools are, and to identify needs currently lacking from these tools.

The study revealed that access to reliable energy and performance data varies considerably between firms and individuals, and that current tools have numerous shortcomings. For many building managers and design professionals, the process of visualizing building information for analysis, benchmarking and diagnostics, remains a time intensive, do-it-yourself undertaking. Many people interviewed, including those with access to energy visualization tools, still rely on data exported from building management systems (BMS), and manipulated in spreadsheet programs. A desire for more end-use and historical energy data, and a nearly unanimous desire for better methods of communicating with building occupants were observed. Based on these observations, the research team concludes that industry professionals would be well served by software tools that conform to standard conventions described in human-computer interaction literature as "overview first, zoom and filter, then details-on-demand."

Workplace Occupant Needs Assessment

The third primary task of this research involved a survey of workplace occupants to understand how they typically use energy information and whether access to this information influences their energy attitudes and behaviors. Commercial products for visualizing energy consumption for occupants are rapidly being adopted, however access to these products is limited, and occupants for the most part continue to receive building energy and operation information anecdotally and through other means. A majority of the survey respondents report that they already take actions to save energy, and that they would take more energy conserving actions if they got feedback on either the amount or cost of energy used. The primary motivations for saving energy at work are environmental and ethical concerns ("doing the right thing"), followed by saving money for the company.

Finally, this study offers a few suggestions for methods to display building information to occupants and expert users. Based on these findings, the research team is currently studying the potential benefits of using a web-based social network application to promote energy awareness, to improve communications between operators and occupants, and to positively influence energy-saving behavior of typical office workers. Results of this study will be reported as the second and final phase of this research.

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1. PROJECT BACKGROUND AND INTRODUCTION

Commercial buildings account for 19 percent of all energy consumed in the United States,[1] and are seen as an important opportunity for reducing energy consumption and associated greenhouse gas emissions. A number of factors are bringing into effect policies and technologies that will lead to improved monitoring of energy and other building metrics, and that may also encourage energy conserving behavior.

The United States Green Building Council (USGBC) established the Leadership in Energy and Environmental Design (LEED) rating system in 2000 to promote energy efficient design and operation of commercial buildings. LEED has become an important driver towards this goal; however, a significant shortcoming is that as currently applied to new construction and major renovations, compliance with energy performance is based largely on modeled energy use. Since modeled energy use may be an unreliable predictor of actual performance, this lack of performance data undermines the reliability of this rating system (Diamond et. al., 2006). In addition, the last few years has seen aggressive energy policies that will require far higher building performance standards than currently required by LEED. For example, the U.S. Energy Independence and Security Act of 2007 (EISA 2007) requires new and renovated federal buildings to achieve carbon neutrality and net-zero energy for all new commercial buildings by 2030. The state of California too has adopted policies requiring new commercial construction to meet zero-energy targets by 2030. To meet these goals, ongoing monitoring and commissioning of commercial buildings will be necessary.

These new goals and policies provide market opportunities for providers of energy efficient products and services, as well as for "greentech" or "cleantech" startups. Numerous software companies have developed new data acquisition and information visualization products that provide interesting ways to monitor and display building performance data to various building stakeholders. These products offer the possibility of simplifying complex energy monitoring tasks for expert users, as well as the ability to display energy consumption information to non-technical audiences such as commercial building occupants. The growing popularity of these technologies in recent years may be linked to previous research that points to the prevalence of a so-called "Prius effect" which describes the positive behavioral impacts of making energy consumption information visible to users (Darby 2006). The authors' discussions with several providers of building software products reveal similar findings. Although this evidence is largely anecdotal, many of their customers report energy savings of 5-15%.

A number of intervention studies have shown that effective energy feedback can influence energy saving behavior among building occupants (Darby 2006). However, few studies have focused on determining what types of energy information are of value to building users, or on the optimal methods for the display of such information. Furthermore, most of these intervention studies have been conducted in residential buildings and consequently, may not be applicable to commercial buildings. Unlike residential buildings, commercial buildings have highly complex mechanical systems and house a variety of users — managers, operators and occupants — some of whom have little or no control over their energy impacts (Lehrer 2009). Also, the energy performance in commercial buildings is highly dependent on building systems, envelope, and other pre-established characteristics, and less so on decisions and actions of occupants and operators. It is unclear from the current literature how these

[1] US Energy Information Administration. 2008. "Annual Energy Review." http://www.eia.doe.gov/emeu/aer/consump.html (accessed December 30, 2009)

challenges impact the role of information interventions in producing energy savings. The motivation for this research stems from this gap.

1.1 Objectives and Method

This study is being carried out in two phases: (1) the discovery phase, and (2) the design and subject test phase. The discovery phase consists of three related research tasks. The first involves product reviews of commercial energy dashboards to understand their features and capabilities. The second is an expert user study to assess the information needs, preferences and practices of expert users such as facility managers, energy professionals, HVAC design engineers and architects, using a combination of surveys, contextual inquiries and interviews. The final research task involves an energy information survey of workplace occupants.

The objectives for the "discovery" phase of this research include:

- Understanding the range of features available in software tools currently available for displaying building energy information.
- Studying and documenting the energy information needs and preferences of expert users such as facility managers, architects and design engineers that are involved in monitoring and analyzing building performance and energy consumption.
- Learning what energy information is currently available to these professionals and the tools they use to access the building's energy information.
- Understanding the energy attitudes and behaviors of typical building occupants, and whether access to information will encourage energy conserving behaviors.

The "design and subject testing" phase involves consolidating the results of the discovery phase findings to identify optimal methods for displaying building performance information to these stakeholders. For this work we will explore how a social media network may facilitate communication between workplace occupants and building managers. We will investigate how this platform:

- Makes the workplace occupants more inclined to report problems, give feedback on their comfort levels, satisfaction, etc.
- Enables workplace occupants to monitor, share, and compare their personal energy use.
- Encourages energy-conserving behaviors, particularly when social and emotional rewards for eco-behavior are built into its design.
- Provides greater opportunities to building managers to improve buildings' energy performance by offering insights into occupants' attitudes and behaviors.

This cross-disciplinary study considers diverse viewpoints —including building energy engineering, human-computer interaction, and psychosocial behavioral factors — to study the role of energy feedback in commercial buildings. It focuses on the kinds of energy information that are useful to various categories of building stakeholders, and investigates preferred methods for its display. Based on preliminary building user research, two broad categories of building stakeholders that would benefit from energy information feedback were identified. These groups have diverse information needs and practices and include: (1) expert energy information users, and (2) general building occupants. Table 1 shows the types of building information users included in each of these categories. Expert users are typically involved in energy monitoring and analysis. For these users, feedback on energy consumption patterns in buildings provides increased opportunities to assess and improve overall energy performance. In contrast, the general building occupants are not involved in energy management roles. They are typically consumers of energy in a building and have limited ability to influence the building

energy usage patterns. While they may impact energy use to a certain extent through their control of lights, thermostats, computers and/or other equipment, these users otherwise have no control over a building's major energy performance. The research, design and test schema for this study will focus on the information needs of these two categories of users.

Table 1: User Group Demographics

Expert Energy Information Users	Architects, design engineers, facility managers, building operators, building managers, building owners, green building consultants, commissioning agents, academics and researchers
General Building Occupants	Office employees, educators, visitors

This report documents the findings from the first phase of this research. Section 1 of this report has provided the background of this problem. Section 2 includes a review of several leading information visualization software products. Section 3 describes the findings from the expert user survey and contextual inquiries. Section 4 reports the findings from the survey of building occupants. Finally, Section 5 suggests ideas for effective visualization methods in commercial buildings, and describes the second phase of this work now underway.

2. DATA VISUALIZATION TOOLS FOR BUILDING PERFORMANCE

New information visualization methods and tools are being applied in business, science and academia for understanding trends and relationships in large, complex data sets. These innovations provide users with interactive capability for filtering, sorting, and visualizing information and take advantage of what Ben Shneiderman describes as the powerful "bandwidth of human vision."[2]

In the building industry, a growing number of companies are providing products that enable the visualization of building performance data, primarily related to energy and water use and renewable power generation. Many of these companies have been started within the last few years, and we can expect additional companies to enter this market in coming years. In addition, many established IT companies such as Google, IBM, Cisco, Microsoft and Intel have developed energy software products to managing commercial building energy, though there are indicators that some of these products have not been commercial successes[3]. There are several drivers contributing to the adoption of information displays in commercial buildings. Owners of green buildings want to exhibit their accomplishments and gain innovation points under the LEED rating system. Design teams share such goals and also want the ability to easily view building performance, to compare against predicted performance and for benchmarking. Increased public scrutiny of green building results, combined with controversy in the research arena, has increased interest in measured energy performance of buildings (Diamond 2006, Turner & Frankel 2008, Brown 2009).

Further, in addition to the typical visualizations provided by many commercially available energy products, many firms are experimenting with the design of novel visualizations that often described as *tangible* and *ubiquitous*. A case in point is the work of a global engineering design firm, Arup, which is exploring innovative visualizations of energy use in commercial buildings (Figs. 1-2). Such efforts illustrate the growing interest in not only measuring energy performance, but also in using this information in interesting ways to instigate energy conserving behaviors among building occupants.

[2] Ben Shneiderman lecture at Center for Information Technology Research in the Interest of Society (CITRIS), UC Berkeley, 3/3/10.

[3] Retrieved 9/2/11 from http://www.greentechmedia.com/articles/read/as-predicted-cisco-getting-out-of-building-management/.

Figure 1: Conceptual Design for a Screen Showing Energy Performance of a Single Zone

Source: ARUP

Figure 2: An Interactive Wall Showing the Carbon Emissions of an Average Person's Activities.

Source: ARUP

A preliminary phase of this research reviewed several commercial building information products (Lehrer 2009). These products are designed for monitoring energy and water use, for educating occupants about building features, and may also have some usefulness for macro-scale diagnostics. They include features and interfaces that are tailored to the needs of different categories of users such as building occupants, managers, and operators. To make the data more accessible to non-technical users, they often include graphical displays that translate energy information into equivalent units such as the cost of energy or CO_2 produced, or into familiar concepts such as hours of light bulb use. In contrast to conventional Building Management Systems (BMSs) or Energy Management and Control Systems (EMCSs), they primarily offer visual representations of real-time and historic energy and water use, with no control capabilities.

The complexity and cost of such systems vary considerably. At one end of the spectrum are products such as the "Building Dashboard" from Lucid Design Group, that can be purchased as an entry hardware and software package for two buildings for under $10,000.[4] On the other end, a major building controls supplier offered a potential customer a proposal to provide a user-friendly energy monitoring system for a large mixed-use building at a cost of over $170,000 (the building already had an installed BMS).[5]

This section reviews seven building visualization products that are being adopted by a growing number of commercial building owners. These products were selected for review because they had established customer bases and because they provide visually interesting displays of information and/or offered user interfaces appropriate for a range of users. The companies and product offerings in this space are evolving rapidly, so this report provides a snapshot of trends at the time of this investigation, which was largely conducted in the spring of 2010. For a more comprehensive review of energy information systems, see Granderson et. al. 2009.

[4] Retrieved 3/4/10 from http://www.luciddesigngroup.com/starter.php

[5] Personal correspondence with ZGF Architects, 2/24/10

2.1 Pulse Energy

Pulse Energy is a software company located in Vancouver, B.C. that develops energy management software.[6] Pulse Energy provides a suite of three applications that include: (1) the Pulse Dashboard view, an educational tool for general building occupants; (2) the Pulse Management view, which provides detailed trend data for facility managers; and (3) the Pulse Reporting view, for portfolio managers to compare building performance, calculate GHG emissions or financial data, and demonstrate the savings as a result of conservation or efficiency projects (Figs. 3-4).

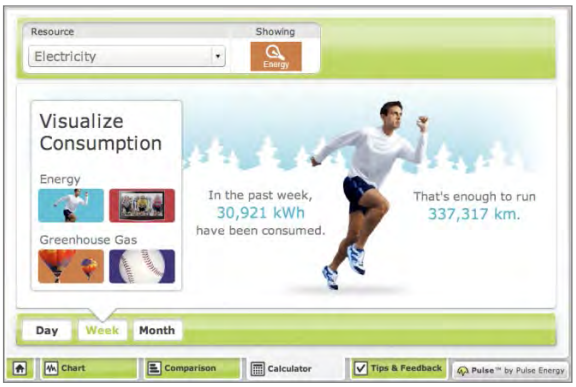
The Pulse Dashboard view translates energy consumption and greenhouse gas (GHG) emissions into units intended to be understandable to a general audience, for example, a distance to be run, the weight of baseballs, or the number of balloons that can be filled with the equivalent amount of GHG. In addition, the dashboard offers the users tips on energy conservation and allows them to send feedback on what additional information might motivate them to reduce resource use. The Pulse Management view is a more detailed interface that allows expert users to set alerts and identify operational improvements through comprehensive energy trend analysis. Expert users also have the ability to customize the look-and-feel of the Dashboard viewed by occupants.

The Dashboard and Management tools include current load profiles and baseline comparison profiles. The baseline profiles normalize for weather and other variables so that occupants can accurately determine the impact of energy conservation measures. Further, these tools allow trend data to be configured to display the energy profiles of electricity and water in varied temporal granularities.

Pulse Energy management reports that its installation base has increased by a factor of ten during a recent year of operation. Some of the firm's prominent clients include the Government of British Columbia, the University of British Columbia (UBC) in Vancouver, and a number of 2010 Olympic venues. At the UBC, the company staff played a key role in organizing competitions among buildings and facility managers to promote the use of the software and reduce energy use. As a result of these initiatives, the UBC campus estimates energy savings of 10-15%, which they attribute to the use of the software. Pulse Energy's online energy management software supported the Venue Energy Tracker that was set up at the Vancouver Olympic Games. The Venue Energy Tracker recorded total energy savings of 906 MWh during the games and marked the first time that Olympic venues monitored and publicly reported energy consumption during the events. As a result of continued success in demonstrating energy savings in the UBC campus and other projects, Pulse Energy in 2010 received a grant of $2.6 million by Sustainable Development Technology Canada, to develop and demonstrate an intelligent energy management platform to improve the energy efficiency of commercial and industrial buildings.

[6] This section is drawn from a combination of personal correspondence during 2010 with Pulse management and the company's website (http://www.pulseenergy.com).

Figure 3: Pulse Energy Dashboard Visualizing Energy Consumed in terms of Distance Run



Source: http://www.pulseenergy.com/

Figure 4: Historical Trending of Electricity Use at Wurster Hall, UC Berkeley

Source: http://www.pulseenergy.com/

2.2 Quality Attributes

Quality Attributes Software is headquartered in Ames, Iowa, and provides software solutions intended to reduce energy cost, energy consumption, and carbon emissions in commercial buildings.[7] Quality Attributes provides a suite of two products for energy monitoring and verification: (1) iBEnergy Software suite geared towards facility managers, architects, executives, building owners and operators; and (2) GreenTouchScreen for general building occupants and visitors (Figs. 5-8).

The iBEnergy suite consists of three products: (1) iBBuilding Application to monitor, measure and manage a single building's operations; (2) iBCampus Application to monitor a group of buildings in the same geographical location; and (3) iBEnterprise Application to track and assess a portfolio of buildings that may be widely dispersed geographically. Each of these applications graphically visualizes both real-time and historical energy data in a single-screen dashboard format. They offer a variety of interactive views and data reporting capabilities that can be tailored to meet users' needs. The iBCampus application also supports ranking buildings by their total and normalized consumption making it easy to spot the best and worst performers in a campus. To facilitate behavioral changes and long-term energy reduction, the application encompasses a competition module between buildings. The iBEnterprise is similar in functionality to the iBCampus but is designed to function on a larger scale.

GreenTouchScreen is an educational tool designed to promote awareness about a building's sustainability measures. It is a web-based interactive kiosk software that displays real-time energy data, and enables historical and normative comparisons of buildings' performance. The product has been customized and installed in multiple locations, including noteworthy installations at the Cold Climate Housing and Research Center in Fairbanks, Alaska; Building B3 in the Department of Energy in Las Vegas, Nevada; the King County Office Building in Seattle, Washington; Great River Energy in Elk River,

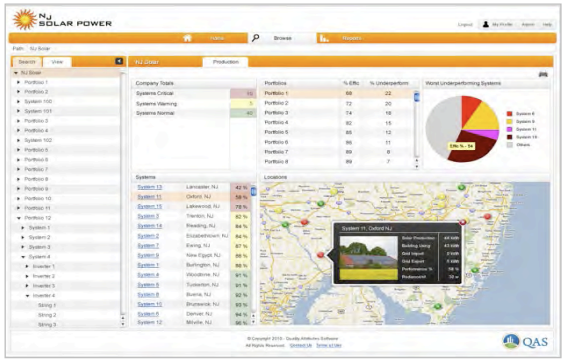
[7] This section is drawn from a combination of personal correspondence during 2010 with Quality Attributes staff and the company's website: (http://www.qualityattributes.com)

Minnesota; Allsteel Inc., San Francisco Showroom in San Francisco, California; and the Broward County South Regional Library in Penbroke Pines, Florida.[8]

Figure 5: iBBuilding Application Used at Park Place, Seattle WA

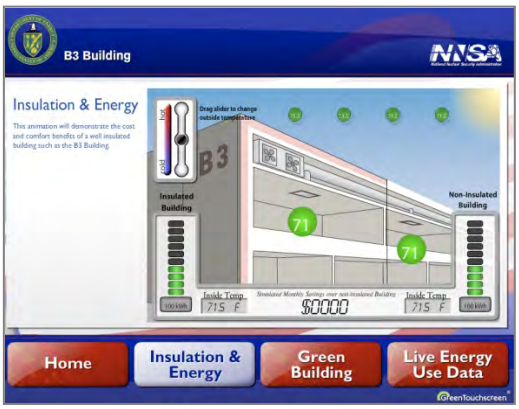
Source: http://parkplace.ibplatform.com/

Figure 6: iBEnterprise Application Used at NJ Solar Power



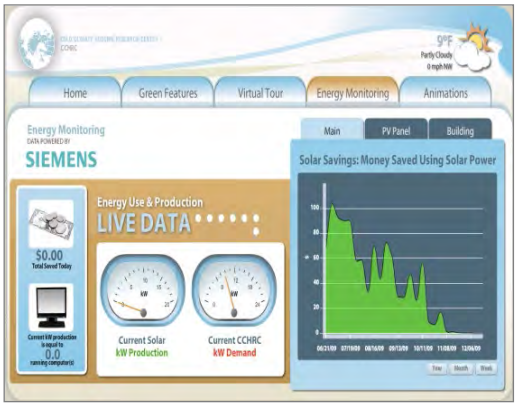
Source:http://www.qualityattributes.com/aspx/products/iBEnterprise.aspx

Figure 7: Animation Showing Green Features in Dept. of Energy Bldg B3, Las Vegas, NV



Source: http://lasvegasb3.doe.greentouchscreen.com/

Figure 8: GreenTouchScreen at Climate Housing and Research Center in Fairbanks, AL



Source: http://cchrc.greentouchscreen.com/

2.2 Noveda

Noveda Technologies of Branchburg, New Jersey, provides a range of energy-monitoring software products geared for facility managers and general occupants.[9] These include: (1) the Sun Flow Monitor that allows for real-time monitoring, diagnostics and reporting of the building's solar energy performance; (2) the Energy Flow Monitor, which monitors a building's energy and natural resource use; and (3) the Facilitmetrix, which combines the functionalities of the Sun Flow Monitor and the Energy Flow Monitor to track renewable and conventional energy use, as well as HVAC system performance; and (4) the Carbon Footprint Monitor, which monitors a building's impact on the environment, and can

[8] Information on the total number of installations was unavailable at the time of this report.

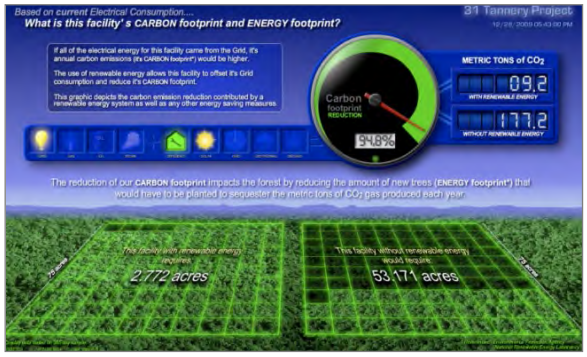
[9] This section is drawn from a combination of personal correspondence during 2010 with Noveda management and the company's website: (http://noveda.com/en/).

evaluate the savings due to energy conserving measures. Besides allowing the users to monitor consumption by end-use in real-time, all these tools store historical data on building energy consumption and facilitate comparative benchmarking against previous utility bills or energy reduction targets (Figs. 9-10).

Figure 9: Noveda Sun Flow Monitor

Source: http://noveda.com/en/page/87?l1=3&l2=4

Figure 10: Noveda Carbon Flow Monitor



Source: http://noveda.com/en/page/87?l1=3&l2=4

At the time of this study, Noveda software had been installed in approximately 60 commercial buildings. Noveda management points to the 31 Tannery Project in New Jersey, which the company claims to be operating with net-zero electrical purchase. The project utilizes Noveda's tools for monitoring and visualizing the building's energy performance, and reports that these monitoring and diagnostic tools promoted positive behavioral changes among building occupants and has contributed to reductions in energy use estimated at 5-15% and a significant decrease in greenhouse gas emissions (RealComm Edge 2008).

2.4 Lucid

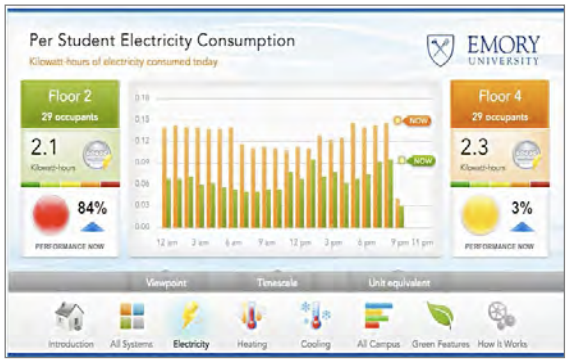
Lucid of Oakland, California, was one of the earliest providers of energy visualization products for residential and commercial buildings.[10] Lucid Design Group employs the name *Building Dashboard* for its products, which include interactive websites and web-based touchscreen kiosks to display building performance information mainly to occupants and visitors. The Building Dashboard is available in versions for use in companies, schools and homes. All versions provide real-time, historical and normative comparatives of energy use data. Some versions also represent consumption data in commonly understandable units such as usage in dollars, equivalent hours of light bulb use and CO_2 emissions. Although the Building Dashboard is primarily an educational tool geared to non-technical audiences, add-on applications such as the Data Downloader for data archival are available for expert energy monitoring and analysis (Figs. 11-12).

At the time of this study, Lucid reported having approximately 50 installations of the Building Dashboard in a range of institutional, commercial and residential settings. Driven by the goal to change the way

[10] This section is drawn from personal correspondence 2009-2010 with Lucid management and the company's website: (http://www.luciddesigngroup.com).

building occupants think, act and consume, Lucid cites examples of installations including college dormitory energy completions that have resulted in energy savings of 10% to 56%.

Figure 11: Per Person Consumption – Lucid Building Dashboard for Students



Source: http://www.luciddesigngroup.com

Figure 12: Normative Comparisons – Lucid Building Dashboard for Companies

Source: http://www.luciddesigngroup.com

2.5 Agilewaves

Agilewaves is a software solutions company located in Menlo Park, California.[11] The company offers two types of energy monitoring and reporting software: (1) the Building Optimization System (BOS) for facility managers, owners and building operators; and (2) the Resource Monitor for residential consumers. The BOS is a web-based system that monitors a building's electric, gas, and water usage in real time. It also enables trend analysis, historical and normative comparisons, and cost and carbon footprint projections. Both these systems receive data feeds on energy use in varying granularity from the building's existing BMS. The BOS also includes alerts that can be configured to suit individual users' needs. Besides energy diagnostics, the Resource Monitor also enables tracking of temperature and humidity, output from solar PVs, performance of solar or geo-thermal systems, and indoor air quality (Figs. 13-14).

A panel of energy efficiency experts at the California Energy Commission evaluated the Agilewaves proposal for "Performance Measurement & Benchmarking for Net-Zero Energy Buildings."[12] Included in these findings, the Energy Commission noted that the Agilewaves Building Optimization System "has the potential of reducing energy use by 30% in retrofitted and newly constructed buildings." One of Agilewaves' clients, Evergreen Partners LLC, reported that the Agilewaves interface deployed at its Johnston Square Apartments in Baltimore provides a central point for command-and-control for building performance and by doing so, offers opportunities to cut operating costs. The installed based of this product was not available at the time of this study.[13]

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[11] This section is drawn from a combination of personal correspondences during 2010 with AgileWaves management and the company's website: (http://www.agilewaves.com).

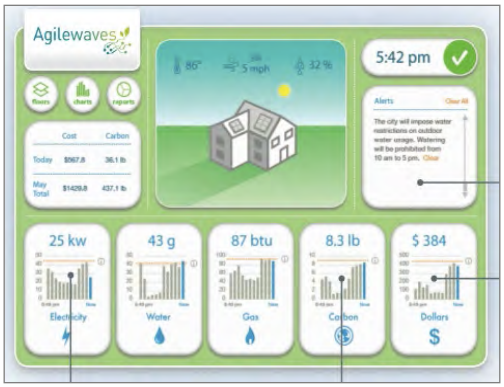
[12] Personal correspondence with Carol Morrison of Agilewaves, January 2010.

[13] Agilewaves declined to share information on individual projects and the number of installations due to non-disclosure agreements with clients.

Figure 13: Agilewaves Building Optimization System

Source: http://www.agilewaves.com/

Figure 14: Agilewaves Resource Monitor



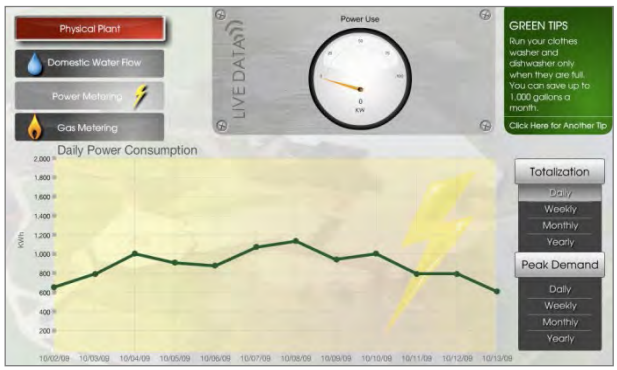
Source: http://www.agilewaves.com/

2.6 Quality Automation Graphics

Quality Automation Graphics of Ankeny, Iowa provides design of custom graphical user interfaces and interactive products.[14] The company's Energy Efficient Education Dashboard (EEED) is primarily an educational tool that is targeted to building occupants and visitors. It displays real-time and historical energy data and also educational animations that highlight a building's energy conserving features. In addition, the system supports a "competition module" that ranks buildings within or outside a campus based on normative comparisons. Unlike the other products reviewed here, the EEED is not a hosted solution. Instead the software is typically installed on the customer's servers and is managed by the clients' IT staff. This allows customers to decide how the system is integrated into an enterprise network. The customer retains control of the EEED and configures not only some aspects of the look and feel of the dashboard, but also its functionality and how it accesses data from the building automation system (Figs. 15-16).

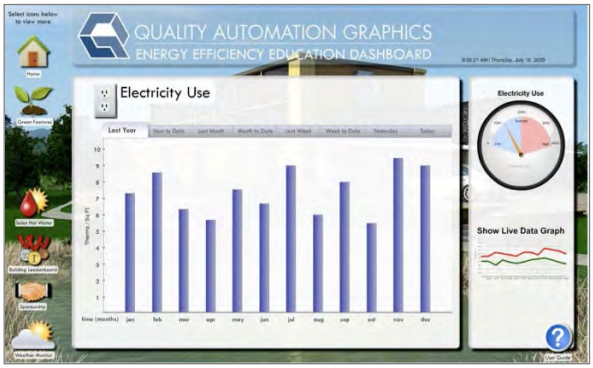
[14] This section is drawn from a combination of personal correspondences during 2010 with QA Graphics employees and the company's website: http://www.qagraphics.com.

Figure 15: EEED Showing Live Energy Data, Daily Power Consumption and Energy Tips



Source: http://www.qagraphics.com/energy-efficiency-education-dashboard/energy-dashboard.html

Figure 16: Monitoring Electricity Use with EEED



Source: http://www.qagraphics.com/energy-efficiency-education-dashboard/energy-dashboard.html

At the time of this study, QA graphics had customized and installed over 85 EEEDs for schools, universities, restaurants, government buildings, commercial buildings, and libraries. Although the company is not able to provide direct energy saving statistics, company representatives report having received positive feedback from customers.

2.7 Prophet Suite

Prophet is a software visualization product that offers customers a holistic view of building performance and allows users to create customized screens and visualizations using a Flex interface.[15] It can be configured for both expert users and building occupants, and to be functional across multiple performance metrics including electricity, water, and gas. The software is available with coordinated hardware including for monitoring and control. The system is compatible with several BMSs including Siemens Apogee, Tridium, and JCI Metasys, and is based on the Niagara development platform (Figs. 17-18).

Prophet was developed with the goal of giving customers control over the design of screens and also data analysis. The product suite allows administrators to configure user accounts and manage access rights and privileges. In addition, users can organize building data in a hierarchy that meets users' individual needs, by campus, building, floor, room, equipment load, etc. The system also allows for data aggregation and normalization can be combined to provide an overall view of building metrics.

Prophet also provides an extensive "widget" library that simplifies the creation of graphical building dashboards and interactive components such as HVAC equipment, floor plans, server room layouts, etc. This allows users to "drag and drop" the elements needed to create screens tailored to specific uses. The company also provides standard screen templates and conducts an annual competition for the most useful and innovative interface designs created by users. One unusual capability not found elsewhere in the review is the ability for users to include live video feeds into screen views if desired.

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[15] This section is drawn from a combination of personal correspondences with Controlco employees during 2010 and the company's website: http://www.controlco.com/software-it.html#prophetSuite.

Figure 17: Prophet Suite Dashboard Builder

Source: http://prophetsuite.com

Figure 18: Interactive Components to Build Graphical Representations of HVAC Equipment

Source: http://prophetsuite.com

2.8 Summary of Product Reviews

All of the energy visualization tools that were reviewed support multiple users and offer a standard set of features to suit the needs of both expert users and the typical commercial building populations. Products such as Lucid's Building Dashboard, Quality Attributes' GreenTouchScreen, and QA Graphics' EEED are primarily marketed as educational tools for informing about building resource use, carbon footprint reduction and other broad sustainability measures. These tools are targeted to diverse audiences including building occupants and visitors, and are intended to raise awareness of building performance and to encourage energy conserving behaviors.

Products such as Noveda's Facilimetrix, iBEnergy from Quality Attributes, BOS and Resource Monitor from Agilewaves, and Prophet are designed for expert users who may have advanced knowledge about energy monitoring and analysis. Most of these tools support complex energy data modeling and analytics. Single screen "dashboards" such as the Pulse Dashboard are designed to support both sets of users, and combine educational and data visualization elements.

Almost all tools surveyed support tailoring of interfaces to meet specific user needs. The degree of configurability, however, varies across tools. While most tools provide the basic interface framework with which users interact through simple on-screen selections, newer tools such as Prophet allow users to build highly customized interfaces using a library of widget interface elements.

In general, the products reviewed allow users with administrative privileges to control both the content and the look and feel of the information presented. They determine the level of information that is displayed to the users lower in the access hierarchy. In most cases, users with minimal access rights typically see only a public version of the dashboard; administrators and users higher on the hierarchy can both visualize the data and with some systems also perform analyses.

Many of these product providers report expanding customer bases and claim that the use of these products has promoted significant energy conserving results. However, additional research is needed to verify these claims, and to understand exactly how these savings may be due to the use of these data visualization tools.

3. A STUDY OF EXPERT USERS

The second primary task of this research is to understand and document the energy information practices, needs and preferences of an important category of users—industry professionals who are experts in energy monitoring and analysis. A combination of surveys and contextual inquiries was used to investigate how these experts use building energy information and what tools they currently use to visualize this data. This study sought to understand their user experiences, to learn how useful and usable commonly used tools are, and what information needs are not being met by these tools.

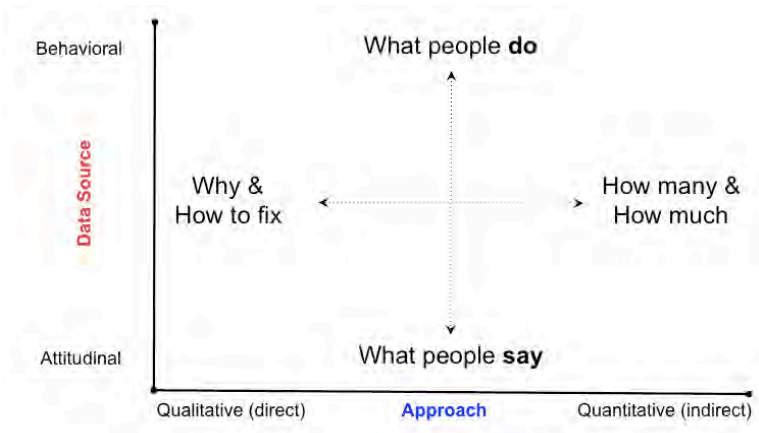
This section describes the research methods and findings from the expert user study. Section 3.1 provides an overview of user experience research methods. Section 3.2 describes the expert user questionnaire. Sections 3.3 through 3.5 describe the findings of the expert survey. The findings reported in this section were presented at the American Council for an Energy-Efficient Economy (ACEEE) 2010 Summer Study on Energy Efficiency in Buildings and are included in the conference proceedings (Lehrer & Vasudev 2010).

3.1 Background on User Experience Research

The goal of user experience research is to provide insight into the users—who they are, what they do, and what they want (Kuniavsky, 2003). User experience (UX) research methods can provide both qualitative and quantitative measures of users' attitudinal and behavioral responses to the product or service in question. While attitudinal UX research methods may be described as "what people say" and documenting their stated beliefs, its behavioral counterpart focuses on "what people do" with minimal interference from the method itself (Rohrer 2008). Both these techniques can be carried out in either a qualitative or a quantitative manner. Qualitative UX studies gather data directly, whereas quantitative studies typically gather data indirectly through an instrument such as a survey or data mining. The choice of a particular UX research technique depends on the purpose and motivations of the study. A diagram of these considerations is shown in Figure 19.

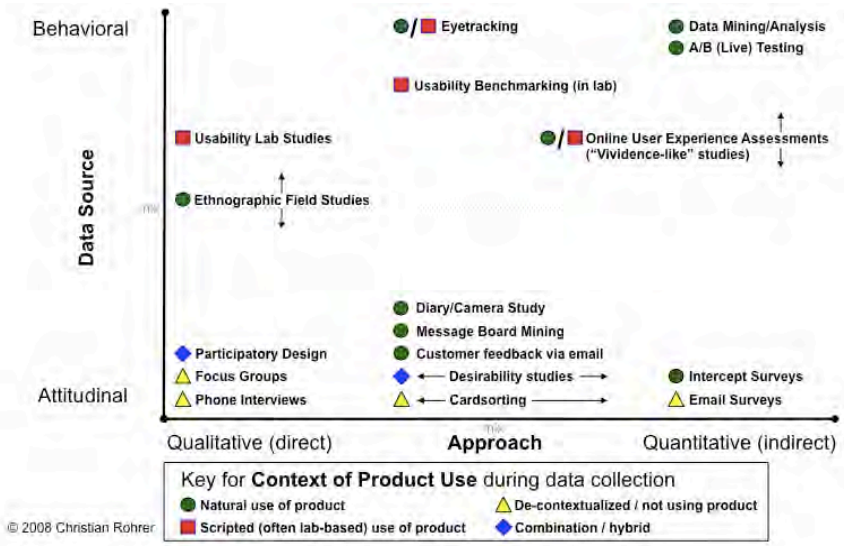
Figure 20 illustrates how numerous UX research techniques fit into this diagram. The chart includes a third variable, "context of product use," that describes how participants use a product or service during the study, and may influence the outcomes of UX research (Rohrer 2008).

Figure 19: UX Research Methods Spectrum



Source: Christian Rohrer, http://www.useit.com/alertbox/user-research-methods.html

Figure 20: Matrix of UX Research Techniques



Source: Christian Rohrer, http://www.useit.com/alertbox/user-research-methods.html

In this study the research team was interested in both attitudinal and behavioral responses, and therefore used a combination of surveys, phone interviews and contextual inquiries to capture a mix of quantitative and qualitative perspectives to understand users' practices and information needs concerning viewing and analyzing building performance information. The context of product use in the contextual inquiries (noted as ethnographic field studies in the chart above) was *near natural*, that is, the subjects were asked to demonstrate the tools they used commonly in their work. The following section describes the research methods and the findings from the expert user study.

3.2 Expert User Survey Development

An expert user survey was developed to gather information from user groups who are highly familiar with energy monitoring and analysis, and who may be able to influence (to varying degrees) energy performance in commercial buildings. Such groups typically include building managers and operators, architects, engineers, commissioning agents, green building consultants, and others. The survey questionnaire consisted of multiple-choice and open-ended questions, divided into three sections. The first section asked users about sources and types of building performance information currently available to them, and the frequency of their use of this information. The questionnaire used conditionally branching pages so that users of energy management systems (described in the questionnaire as having control capability) and/or energy visualization tools (having no control capability) were asked additional questions specific to their use of these tools. The second section asked respondents to rate the usefulness of several types of energy information. The final section included background questions on the test subjects' demographic information and computer use patterns. This questionnaire is included in Appendix A.

3.3 Respondent Demographics and Energy Information Sources

Via email invitations a group of building industry professionals was invited to participate in the survey, using email lists that had been compiled by UC Berkeley's Center for the Built Environment for communications and outreach purposes. The survey results are viewed as a convenience sample, not statistically representative of a larger population, however they are still useful for qualitatively assessing information practices and preferences of expert users. A total of 70 complete responses to this survey was received; the distribution of job titles is shown in Figure 21. Respondents who indicated "other" (19%) included design and construction managers, project managers, utility program managers, and individuals with building or sustainability responsibilities. The ages of respondents were roughly evenly split between the four survey choices: < 30, 30-40, 40-50, and 50-60.

Figure 21: Primary Job Function of Respondents. (N=70)

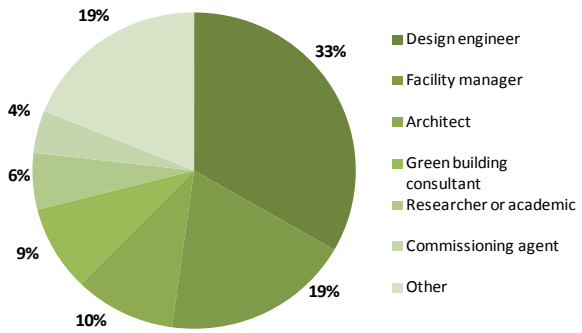
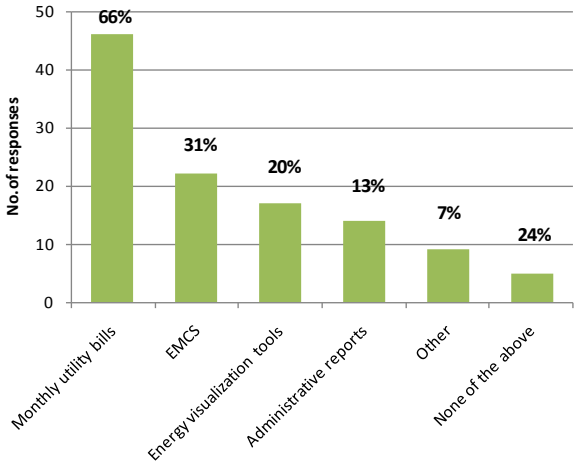


Figure 22: Information Sources Currently Available to Users. (N=70)



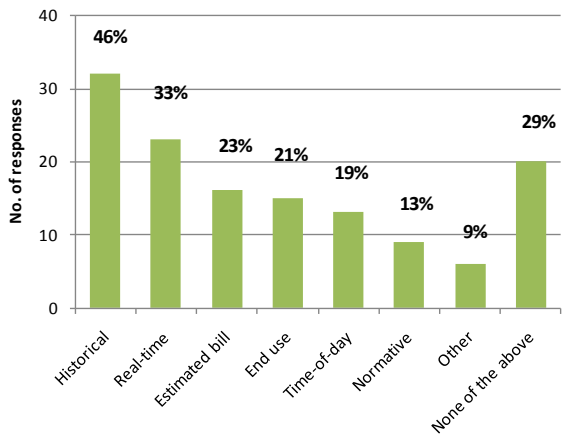
Multiple checkboxes, responses do not add to 100%.

Approximately three-fourths of the respondents currently have access to some building energy information (Fig. 22). The most common source, cited by 66% of respondents, is from monthly utility bills, followed by energy management systems, energy visualization tools, and administrative reports. Respondents were asked about the types of information available to them, for example historical, normative, end use, real time, or time of day (Fig. 23). Close to half of the respondents have access to historical energy use, and 33% have real-time energy data. In addition, respondents were asked how often they view building energy or performance data, and learned that they do so infrequently. Of the 51 respondents that have access to energy information, 39 people (76%) view this information once a month or less, six people (12%) view it a few times a month, and only six people (12%) view it one or more times a week.

Branching pages in the survey inquired further about the use of EMCSs and energy visualization tools. Respondents were asked about the types of systems used (by text box entry), frequency of use, most useful aspects, and shortcomings. Of the 23 respondents that use EMCSs, close to half indicated that they use multiple products. The most commonly cited products include those from Siemens, Johnson Controls, and Automated Logic; included also were Alerton, Invensys, Adura, Tridium, and Barrington. The useful features that respondents listed most frequently include trending, real-time information,

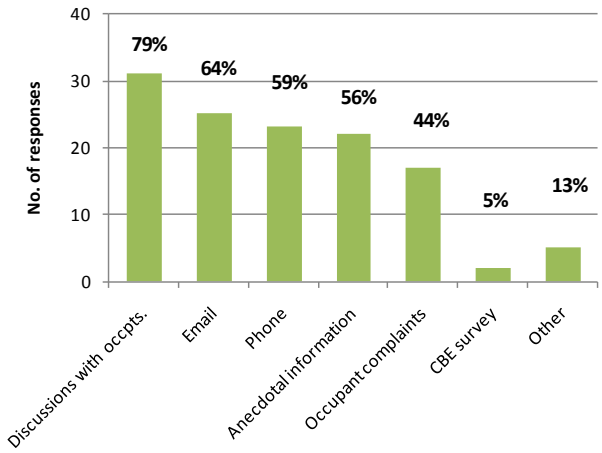
information on individual HVAC units or devices, and graphical displays such as charting and views of zones and floor layouts.

Figure 23: Information Types Currently Available to Users. (N=70).



Multiple checkboxes, responses do not add to 100%.

Figure 24: Occupant Feedback Sources. (N=38).



Multiple checkboxes, responses do not add to 100%.

Complaints about EMCSs include the inability to get usable performance data, inadequate integration with meters, lack of metering of end uses or devices that are not included in the BMS (such as total electrical use, lighting and plug loads), lack of effective historical comparisons, and the inability to format and save charts. Interestingly, many features listed as useful by some respondents are cited as lacking by others, so the features of the various products must vary considerably. Frequency of use of the EMCS information is varied — 14 users (54%) view it once a month or less, while eight (31%) view it once a day or more. As expected, facility managers and commissioning agents reported more frequent use of EMCSs.

Of the 18 respondents that use energy visualization tools (having no control capability), seven (39%) indicated that they use Excel or a spreadsheet program to manually manage data for visualization, something that was also observed during the contextual inquiries. Five users indicated that they use a building monitoring product such as Agilewaves, Fat Spaniel (PV monitoring software), or Obvious. The Energy Star Portfolio Manager was also listed by two respondents. The useful features cited by users include historical information, the ability to compare between selected time periods, the ability to make comparisons between buildings, estimated costs, graphing capability, and energy use intensity.

The list of shortcomings is similar to those noted for EMCS products and includes, for example, lack of integration with other systems, inability to combine multiple energy sources (e.g., electricity, gas, steam and/or chilled water), lack of real-time data, no access to raw data, lack of benchmarking capability, lack of end-use data, and the inability to identify anomalies. The frequency of use for visualization tools is somewhat less than that of the EMCS users, with 50% reporting that they use them once a month of less, and only one person using the system daily.

The fact that a significant overlap between the sets of comments is seen, both pro and con, regarding EMCS and energy visualization tools, seems to indicate that people in this sample are using both sets of tools in much the same way, and that many of these respondents may not make a great distinction between tools with control capability and those that only provide information.

3.4 User Preferences for Energy and Performance Information

To understand the gaps between the desired and currently available energy information, respondents were asked to rank six types of energy information for usefulness in saving energy, on a 5-point scale ranging from "very useful" to "not at all useful." The information deemed most useful is end-use energy consumption cited by 71%, closely followed by historical energy consumption, and time-of-day use (Fig. 25). This finding reveals an information gap: even though end-use energy information had a high utility ranking, only 21% currently have access to it. Another interesting finding is the lower value users put on estimated energy bills. Ironically, many commercially available energy visualization tools stress the benefit of this feature in their advertising.

A prior pilot survey had revealed a high level of interest in end-use energy information. Consequently, this survey included a detailed question regarding the types of end-use data that would be useful (Fig. 26). Within this category, 91% of respondents felt that lighting load data would be most useful, closely followed by plug and process loads, cited as useful by 84% of respondents. In general, all types of end-use data were viewed as valuable.

Respondents were also asked about the usefulness of other types of information that are included in some newer energy visualization products (Fig. 27). The results show a high level of interest in building dashboard tools or simplified building report cards. As many systems are complex, and most users have little time to view this data, such a need seems obvious.

Figure 25: Comparative Usefulness of Types of Information. (N=70)

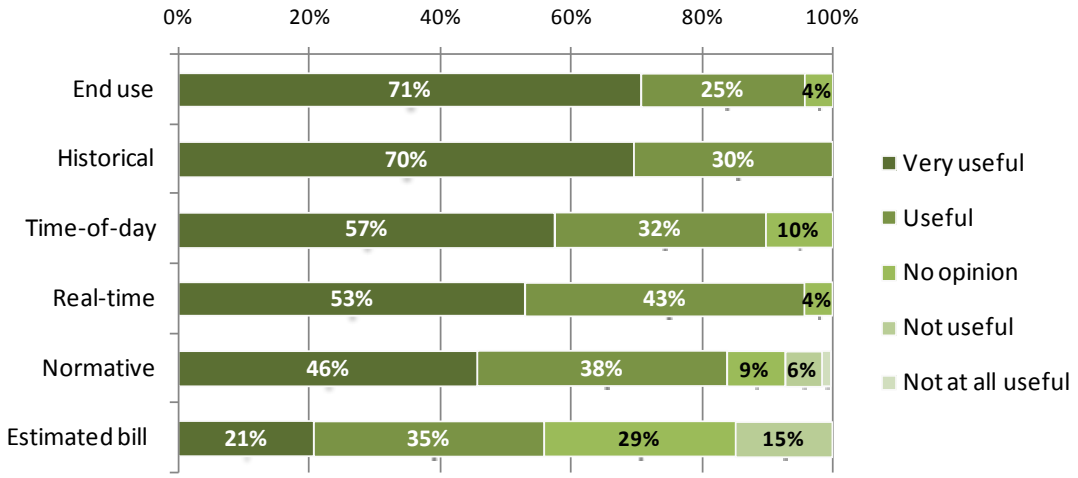
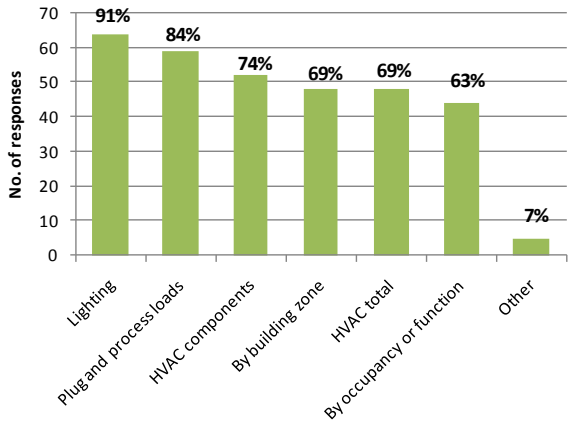
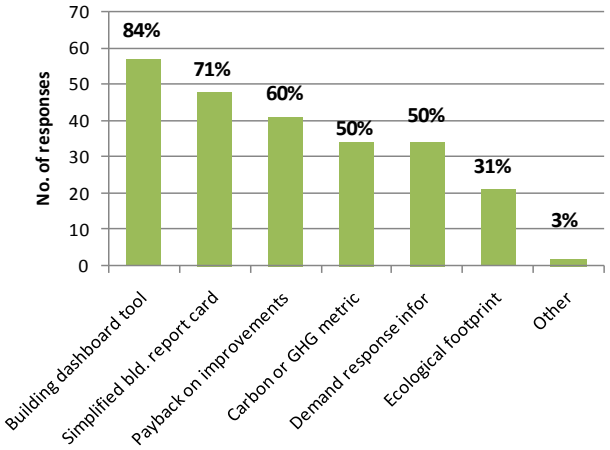


Figure 26: Usefulness of Types of End-Use Data. (N=70)



Multiple checkboxes, responses do not add to 100%.

Figure 27: Usefulness of Other Information. (N=70)



Multiple checkboxes, responses do not add to 100%.

3.5 Information Feedback from Occupants

As building occupants provide a valuable source of information about building performance (Zagreus et al. 2004), respondents were asked whether they get information from occupants regarding "occupants' satisfaction, problems, or general building performance." The results were evenly divided between those that do (38 responses; 54%) and those that don't get such feedback (32 responses; 46%). Of the 38 people that do get occupant feedback, the most common source is from discussions with occupants or tenant representatives (79%), followed by email, phone, anecdotal information, and via complaints logged in a building management system (Fig 24). The survey also showed that 90% of respondents would like to have a more systematic way of communicating with building occupants. While there are a few tools available for this purpose (for example the CBE Occupant IEQ Survey) and some research has been done to test new approaches to occupant feedback (Federspiel & Villafana 2003), this points to an ongoing information need that is generally being overlooked. To help address this need, the authors have designed and will test a prototype social media application intended to facilitate communications between occupants and building operators (Lehrer and Vasudev, 2011).

3.6 Contextual Inquiry of Expert Users

Contextual inquiry is a method for understanding users by interviewing them in their workplaces and observing how they use interactive products. Previous research has used case study methods to document how facility managers use energy information systems to manage energy use in commercial buildings (Motegi et al. 2003, Granderson et al. 2009). For this study interviews were conducted with six subjects having a variety of information needs and practices. As there are many diverse users of building performance information, these case studies included design team members and other building industry professionals.

Each inquiry included a semi-structured interview to learn about the user's background in energy management, current sources of energy information, frequency of use, and interactions with occupants. During the interviews, the subjects were also asked about shortcomings with the current sources of information, and additional building information that might be useful. In addition, the subjects were

asked to demonstrate the tools they used to view building information, with attention to the commonly used features.

The subjects selected for the contextual inquiry were asked to provide a broad range of perspectives on visualizing building performance data. Participants included:

- Engineering team members for a small office building with a zero-energy goal
- Architects in a firm that designs a large number of green and LEED-certified projects
- A mechanical engineer and principal with a mechanical engineering firm known for its high-performance building design
- A project manager for a major university campus
- A facility manager for a single building on a university campus
- Energy and resource consultants with an engineering firm who work with data for buildings, campuses, and entire communities

The following sections summarize the findings of these interviews in terms of the subjects' tools and practices, common limitations cited by these users, and a summary of key information needs that are not met by the tools they currently use.

3.7 Tools and Practices of Contextual Inquiry Subjects

Most of the interview subjects rely primarily on BMSs such as Metasys, Automated Logic, Barrington, and Obvius for energy monitoring. Although the expert survey findings indicate that many people have access to monthly utility bills, none of the interview subjects mentioned using monthly bills. The interviews confirmed the survey findings, which show that many people use multiple BMS and/or energy visualization products.

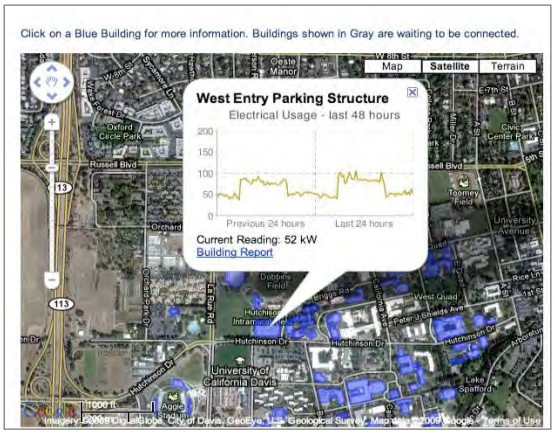
One example of the multiplicity of energy tool use is seen in the practices of the engineering team for a small commercial zero-energy building (ZEB). This team has monitored the building's performance to reach its net-zero goal since the initial occupancy. The project uses control and data systems that the design team describes as state-of-the-art, including a BMS system for HVAC control and monitoring, a web-based software tool to monitor electrical production of the rooftop PV array (required for rebate program compliance), a wireless lighting control system, and an additional product for monitoring overall electrical use.

Consistent with the survey findings, several interview subjects utilize web-based energy dashboards that provide simplified visualizations of energy profiles and trends. Although these tools have limitations (see below), some users describe them as useful for spotting anomalies in whole building energy use and for identifying high base loads. For example, the campus project manager interviewed uses a campus-wide "utilities consumption dashboard," which is publicly available online. Users can select a building from a campus map to view a pop-up window with a trend line of electrical use over the past 48 hours, along with the current power use (Fig. 28). Users can also view more detailed information, including the current demand compared to the previous day, averages, maximums, monthly utilization, and cost data. This subject uses this tool on a monthly basis to understand the energy profile of buildings, however his frequency of use may increase when he has a large number of renovation projects underway.

Of all the subjects interviewed, the architects are the most interested in comparing the overall performance of their projects against other benchmarks. They are using free, publicly available tools, including the Energy Star Portfolio Manager, and a pilot version of the web-based "Energy IQ" benchmarking tool now under development by Lawrence Berkeley National Laboratory. This promising tool allows users to benchmark existing or design-phase buildings against a wide array of energy metrics

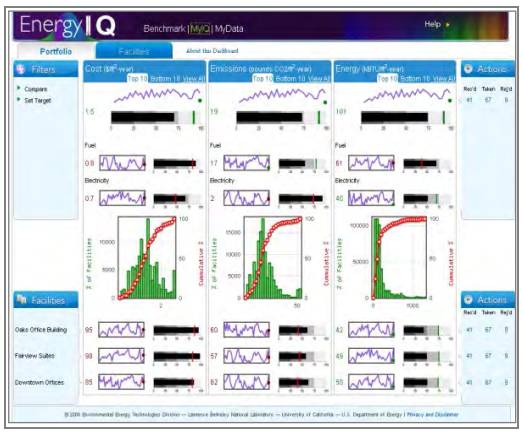
for other buildings (Fig. 29). In addition to these energy-benchmarking tools, they also use the CBE Occupancy IEQ Survey to obtain information on occupant satisfaction in their completed projects.[16]

Figure 28: Campus Utilities Consumption Dashboard



Source: http://facilities.ucdavis.edu/dashboard/

Figure 29: Sample Interface From EnergyIQ



Source: http://energybenchmarking.lbl.gov/

3.8 Limitations of Current Tools

The interview subjects noted several shortcomings of the energy information tools they are currently using. The number of different systems in use, and the lack of integration between them were cited as significant problems. For example, none of the tools used by this group of subjects could combine multiple energy sources such as electrical, steam, and/or gas. This is well illustrated by the large number of systems required for the relatively small (6500 ft^2) ZEB project described above. Another common limitation is the lack of effective visualization of end-use energy data. BMSs are the only source of end-use data, and most do not allow users to view cumulative energy use in a meaningful way.

Few of the energy tools used by this group of users provide capability for data analysis within the tool. For analysis and visualization, many users must download data from a BMS, and use spreadsheet programs, sometimes in conjunction with the Universal Translator tool,[17] to create visually appealing visualizations and presentations (Fig. 30). This is a time-consuming task, and is typically only performed in special cases, such as for diagnostics or reporting building performance.

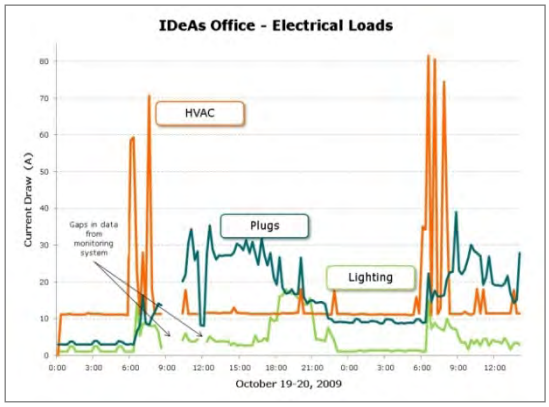
Several interview subjects also noted that simplified building dashboard products have significant limitations. Although they have included such dashboards on several projects, they feel that the energy data from these tools are typically too simple to be useful for building designers wishing to track their buildings' performance. Additionally, they find that these dashboards are rarely available online and do little to benchmark performance against comparable buildings. The university project manager interviewed noted several shortcomings with the dashboard-style campus utilities tool described above. Data is not normalized, so building-to-building comparisons can be misleading to non-expert users. The system has also been known to report erroneous data, and on several occasions failed to reflect major

[16] Information on this survey resource is available at http://www.cbe.berkeley.edu/research/survey.htm

[17] A free software tool designed for the management and analysis of data from building management systems available at http://utonline.org/cms/

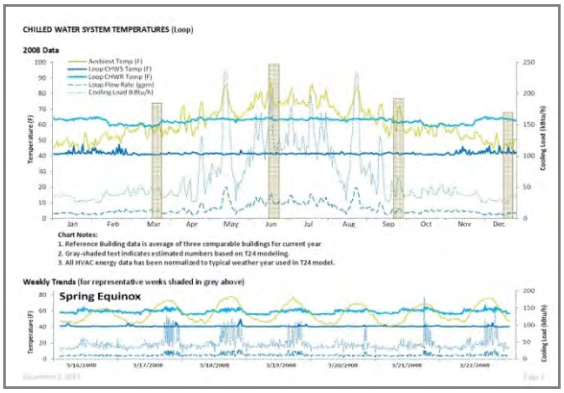
changes in occupancy. Such occurrences have raised concerns among users about the credibility of the tool.

Figure 30: Elec. Load Visualization Produced using Universal Translator



Source: IDeAs

Figure 31: Custom Data Visualization Proposed for Campus Buildings



Source: Bill Starr, UC Davis

3.9 Summary of Key Information Needs Noted in Contextual Inquiries

Although the interview subjects varied in terms of information needs and access to energy information, they reported a common number of unmet information needs that are summarized below. As part of the interview process, the subjects were asked to imagine and describe their ideal energy visualization tools; the responses to this question are revealing and some are included in this summary.

High-level overview with drill-down capabilities, including visualization of end-use energy information including lighting, plug loads, and HVAC components: The interview subjects report that they use energy information tools infrequently, confirming the survey findings. They require a visualization that provides a quick overview, with an ability to drill down for detailed information when needed. One group of subjects described an ideal visualization tool as a cross between a dashboard-style product with overviews of daily and weekly energy use, combined with the capability of the BMS system, including alarms to identify anomalies. In pursuit of an effective summary of building information, one interview subject had even charted out a proposal for a tiered building report that ranges from general building information to system-level detail (Fig. 31).

Integration of energy visualization features with data analysis: Many users rely on data downloaded from BMSs and manipulated in spreadsheet programs. The ability to filter and generate energy analyses in tabular or graphical form directly from the energy monitoring system would be a great time saver for these users.

Support for normalization and energy benchmarking: Several interview subjects cited the need to accurately benchmark between buildings, including normalized values and energy use intensity.

Compatibility with existing BMSs: The multiplicity of systems, proprietary BMS protocols, and lack of interoperability were common complaints. Some interview subjects described an ideal tool that would be based on open source products, and that could be built in a modular fashion with the flexibility of web-enabled tools.

Support for occupant interaction capability: Several of the interview subjects stated a desire to have better interaction with building occupants, a finding that supports the survey results. One group of

subjects believed that that they would benefit from an ability to record occupant discomfort with time and location data that could be compared to BMS data for diagnostics. (Such a system was piloted in a U.S. General Services Administration building, reported in Federspiel & Villafana 2003.)

3.10 Summary from Expert User Study Directions for Future Research

Although this research deals with a relatively small sample, one can begin to generalize about information preferences of expert users of building information. The survey research revealed that many users have access to monthly utility data, and many also have access to EMCS or data visualization data. Many users do not make a great distinction between information systems with control capability and those without, and that for most users viewing this information is an infrequent activity. Building operators and commissioning agents are more frequent users of this information (assuming they will not be ambivalent about the value of control capability.) A preference for historical energy data, and a desire for end-use data of many types, with lighting and plug/process loads ranking highest were also observed. The survey revealed a nearly unanimous desire for better methods of communicating with building occupants, and this finding was supported by the interview research.

The expert user study found that access to reliable energy and performance data varies considerably between firms and individuals, and that the current tools have numerous shortcomings. Many people cited the lack of integration of energy sources and control systems, and the inability to modify and save views. Both the surveys and interviews showed that many users, including those with access to building information tools, still use data exported from BMSs, and manipulated in spreadsheet programs for analysis, visualization, and presentation. For many people, the serious analysis of managing building data remains a time intensive, do-it-yourself undertaking.

Many users would also like access to an information dashboard overview, or simplified building report card, while many users need to drill down for detailed information. These professionals would be well served by software tools that conform to a convention described in human-computer interaction literature as “overview first, zoom and filter, then details-on-demand” (Shneiderman 1996).

Finally, the contextual inquiry research revealed that there is a range of reasons why these expert users seek building performance information. The facility managers and engineering professionals are interested in viewing current and cumulative performance and fine-tuning operations. The architects interviewed are less likely to be involved in operational details but are seeking lessons-learned for future design and general performance data that can inform the design of future projects.

4. ENERGY INFORMATION SURVEY OF TYPICAL OFFICE OCCUPANTS

A building information survey for workplace occupants was developed and implemented to investigate energy attitudes and behaviors of occupants in commercial buildings, and to evaluate the potential for energy feedback to promote energy conserving behaviors. The survey questionnaire consisted of multiple-choice and open-ended questions, divided into two sections. The first section asked about people's energy use in their workplace environments: specifically, about their current sources of energy information, energy use sensibilities, and efforts to energy conservation. People were also asked about the kinds of energy information that might be useful for saving energy and the preferred methods for display. The second section asked about respondents' demographics, type of workspace, and energy awareness at home. (The questionnaire is included in Appendix B.)

4.1 Respondent Demographics and Current Sources of Energy Information

Past users of CBE's Occupant IEQ Survey were contacted by email to identify potential occupant survey sites. Five buildings were identified and a person familiar to people within the organization was employed to forward the survey link to occupants. Table 2 summarizes the buildings and survey responses.

A total of 170 complete responses to this survey were received. Approximately half the respondents were in the two youngest age groups: under 30 (26%) and 30-40 (26%). The remaining respondents were divided between the older categories: 40-50 (18%), 50-60 (20%), and over 60 (9%). There was roughly an equal number of male (47%) and female (53%) respondents. Most of the respondents work in enclosed private office spaces (29%), cubicles with partitions (28%), and in workspaces in open offices with no partitions (25%). Approximately three-fourths of the respondents report access to some sort of energy information. The most common source, cited by 39% of respondents is from a co-worker or manager, followed by company and external communications, and graphical displays (Fig. 32).

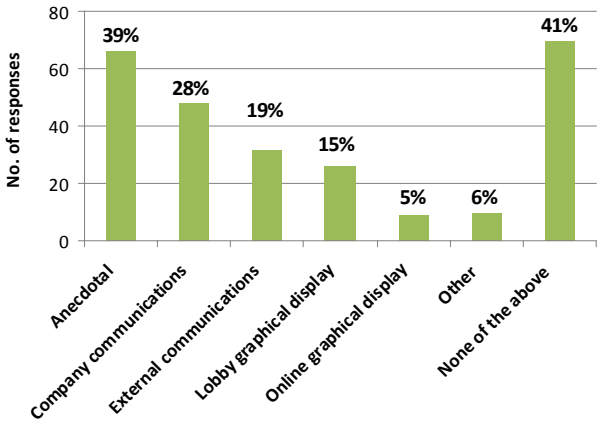
Table 2: Summary of Occupant Survey Participant Buildings

Building description	Location	Type	Responses
Campus design and construction management office	Davis, Calif.	Office	28
Urban design and landscape practice	Philadelphia, Penn.	Office	49
College preparatory K-12 school	Haverford, Penn.	Educational	56
Academic building with offices, classrooms, and studios	Berkeley, Calif.	Educational	35
Department of natural resources building	Madison, Wis.	Office	3

The survey asked users of graphical displays additional questions about the use of these displays, and learned that viewing this information is not a frequent practice. Out of the 35 respondents that view electronic displays, most (57%) access this information once a month or less, 9 people (30%) view it few times a month and only 4 (13%) one or more times a week. They were also asked how this information is useful to them (Fig. 33). Approximately two-thirds responded that they view this information out of a

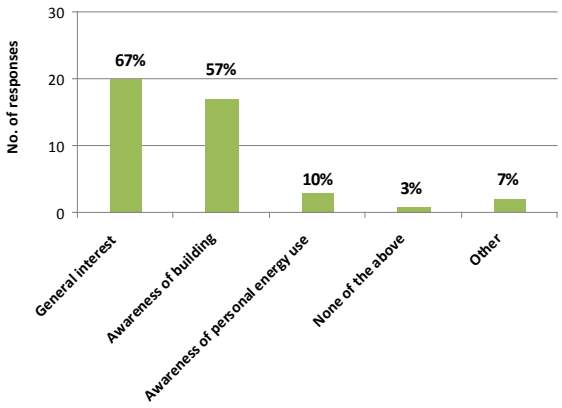
general interest. Some indicated that this information makes them more aware of their building (57%) and a few indicated that it makes them aware of their personal consumption patterns (10%). In terms of other useful information they would like to see in such a display, about three-fourths of respondents (75%) cite the usefulness of viewing building electrical use. Other useful information include building water use (43%), descriptions of green features (25%), carbon footprint (21%) and the amount of renewable energy generated (18%) by the building.

Figure 32: Current Sources of Energy Information (N=169)



Multiple checkboxes, responses do not add to 100%.

Figure 33: Usefulness of Energy Information in Electronic Graphical Displays (N=30)



Multiple checkboxes, responses do not add to 100%.

4.2 Self-Reported Energy Behaviors and Motivations to Conserve

The survey also included several questions about behavior and attitudes towards energy conservation in the workplace. A surprisingly large number of the respondents (91%) strongly agree or agree with the statement that they currently make an effort to save energy at their workplace (Fig 34). They report that they currently take energy-saving actions including turning off the ceiling lights (73%) and desk lights (66%), turning off equipment (62%), using energy-saving settings in equipment (59%), adjusting blinds and/or windows (48%), purchasing energy efficient equipment (18%), and adjusting thermostats (11%) as additional efforts to control energy use in their workplaces. (Although such high levels of self-reported energy saving may not reflect actual behavior, it at least shows the respondents attitudes towards such actions.) A question regarding the motivations for these energy saving actions was also included in the survey (Fig. 37). The most frequently cited reasons are to benefit the environment (92%), “because it is the right thing to do" (86%), and to save money for the company (63%). Also, 40% would take actions in order see the results represented in an energy information display.

Many respondents indicated (strongly agree or agree) that they would make more of an effort to conserve energy if they had knowledge of the amount (80%) and costs (73%) of energy they were consuming (Figs. 35 & 36). This clearly shows that information feedback has great potential value in supporting energy conservation efforts and by doing so can play an important role in lowering their carbon footprints.

Figure 34: Response to: "I currently make an effort to conserve energy in my workplace." (N=171)

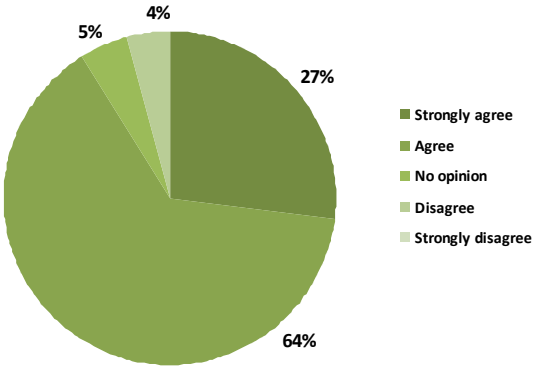


Figure 35: Response to: "If I knew more about the *amount* of energy consumed in my workplace, I would make more of an effort to conserve energy." (N=171)

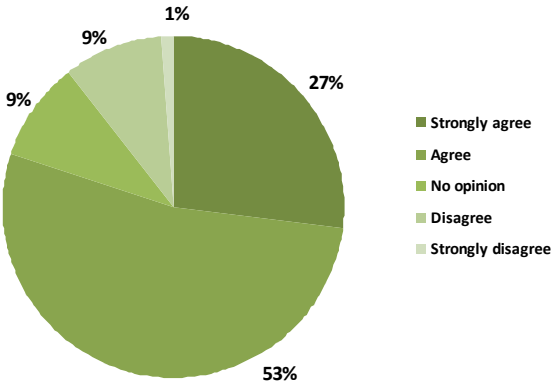
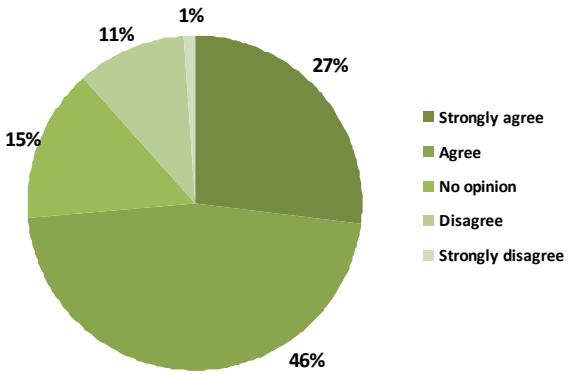


Figure 36: Response to: "If I knew more about the *cost* of energy consumed in my workplace, I would make more of an effort to conserve energy." (N=170)

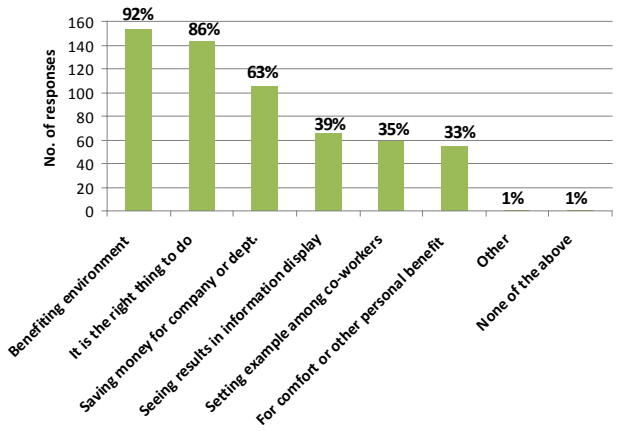


4.3 Preferences for Representing Energy Information

The survey further asked respondents which methods of representing energy use in the workplace would be most useful (Fig. 38). The results highlight a strong need to visualize both the amount (75%) and related costs (70%) of energy use in the workplaces. People also responded that showing energy information in terms of pollution created or prevented (58%) and commonly understandable units such as light bulbs and homes powered (43%) would be useful.

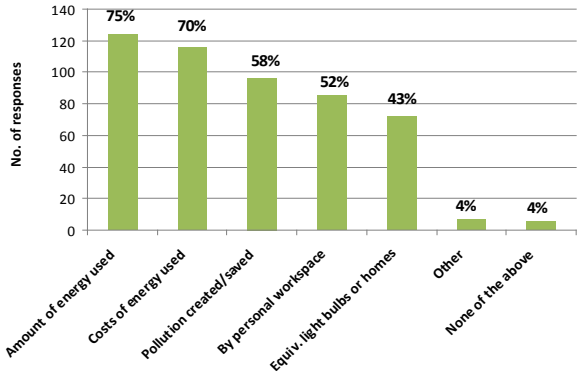
It is interesting to note that people indicate that their main motivations to conserve energy stem from environmental and ethical concerns, however they still seem to express preference for visualizations that highlight the *amount* and *costs* of energy use as opposed to pollution caused or homes powered, parameters that would seem to be more aligned with their underlying motivations to conserve.

Figure 37: Motivations to Save Energy (N=168)



Multiple checkboxes, responses do not add to 100%.

Figure 38: Useful Metrics for Showing Energy Consumption (N=166)



Multiple checkboxes, responses do not add to 100%.

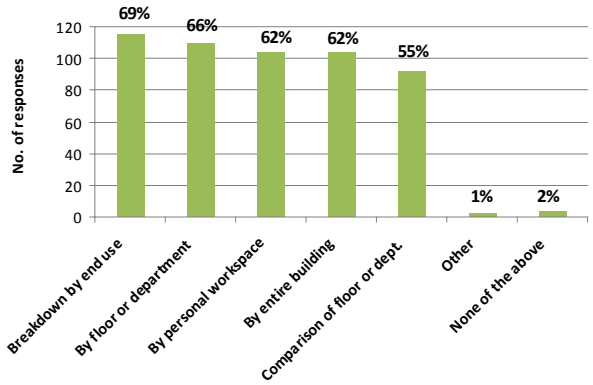
The survey also asked about the kinds of energy details that would be useful (Fig. 39). The results showed that people place a high value on visualizing energy use broken down by end-use (69%) similar to the earlier findings from the expert survey. Other types of information cited as useful include showing energy use by floor, area or department (66%), by personal workspace (62%), and by comparisons of energy use by floors, areas and departments (55%). Finally, there seems to be a strong preference in the method of information display – about three-fourths of the respondents cite the web as being the best medium for visualizing the building's energy information (Fig. 40). [18]

To identify correlations (if any) between people's energy behaviors at their workplace and home, respondents were asked about their actions to conserve energy at home. Even though one would expect a greater sense of care and precaution to minimize energy costs at home, surprisingly, the survey results show that people care about the amount and costs of energy consumption equally at their workplaces and homes. The relationship between self-reported energy conservation at home and at work is charted in Table 3.

Ninety-four percent of the respondents report that they are aware of their energy costs at home, almost all of them (98%) report (strongly agree or agree) that they take steps to reduce their energy use at home, and a only a slightly smaller number (91%) report taking similar steps at work.

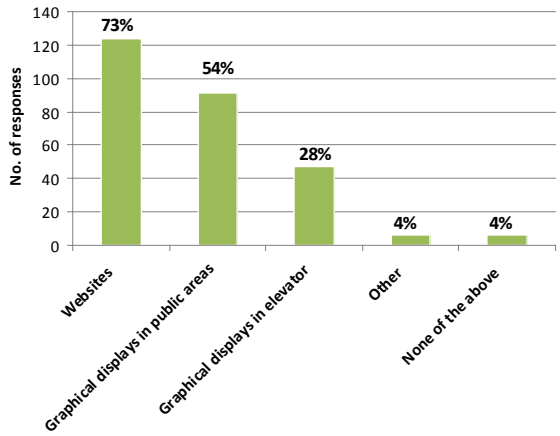
[18] Hand-held devices such as smart phones now present an additional option for energy displays, unfortunately this was not included as one of the check box responses to this question.

Figure 39: Potentially Useful Kinds of Energy Information (N=167)



Multiple checkboxes, responses do not add to 100%.

Figure 40: Preferred Methods for Displaying Energy Information (N=169)



Multiple checkboxes, responses do not add to 100%.

Table 3: Relationship Between Self-Reported Energy Conservation at Home and at Work (N=170)

		Currently takes steps to conserve energy at ***home***				
		Strongly agree	Agree	No opinion	Disagree	Strongly disagree
Currently takes steps to conserve energy at ***work***	Strongly agree	34	11	0	0	0
	Agree	50	57	1	2	0
	No opinion	5	3	0	0	0
	Disagree	2	5	0	0	0
	Strongly disagree	0	0	0	0	0

4.4 Summary of Workplace Energy Survey of Workplace Occupants

The number of survey responses was smaller than desired and included a low response rates in a few of the selected sites. The sample of 170 yields an approximate margin of error of 7.5%; however, a few caveats should be noted. There may be a bias towards higher energy awareness in this sample. This is a self-selecting population, and more energy-aware individuals may be likely to take the time to respond to such a survey. In addition, the buildings included the offices of campus construction and engineering staff, who may be more interested in building energy than other types of office occupants. Accepting

these limitations, the study does provide insight into the energy attitudes of some workplace occupants, and their preferences for viewing energy information that are summarized below.

Slightly more than half of the survey respondents have access to some information about energy use in their workplaces: The most common sources of this information are word-of-mouth from co-workers or managers, followed by company and external communications. The small number of occupants that have access to a graphical energy display view it infrequently, most people indicate that they view it once a month or less.

A large majority of workplace occupants report that they already take actions to save energy: Turning off desk lights is the most commonly cited action, followed by turning off ceiling lights and equipment, using energy-saving settings for computers and monitors, and adjusting windows and/or blinds. Additional research would be valuable to see the degree to which this is true in practice, to what degree such actions are beneficial.

Many respondents indicated that they would take more energy conserving actions if they got feedback on either the amount or cost of energy used: Respondents did not show a strong preference between cost and energy used. People indicated that they would be interested in seeing energy use for the entire building, and also broken down by end use, by floor or department, and/or at the level of individual workspaces. They had a preference for seeing information via a website that they could view at will, or such an interface located in a kiosk in a lobby or public area.

The primary motivations for saving energy at work are environmental and ethical concerns ("doing the right thing"), followed by saving money for the company: Other motivations included an interest in seeing results in an information display, setting an example for co-workers, and for reasons of comfort or personal benefit.

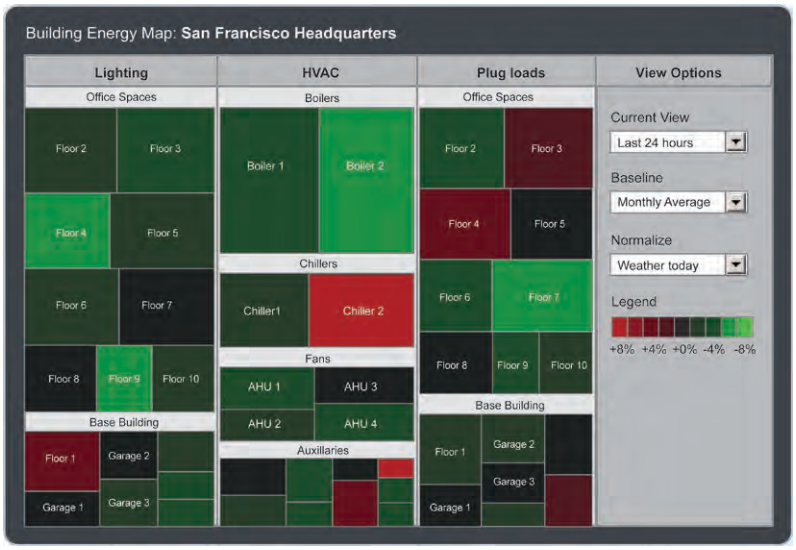
5. DISCUSSION AND DIRECTIONS FOR FUTURE WORK

This report summarizes the first phase of this research project. Results from the expert user study (section 3) were presented at ACEEE 2010 Summer Study on Energy Efficiency in Buildings and is included in the conference proceedings (Lehrer & Vasudev 2010). The study identified a number of information needs that are not being met by current visualization tools, although a growing number of companies are working on innovative new concepts to address these needs.

One type of visualization that appears promising for viewing trends in buildings, but that has not been generally adopted by building software providers, is the "treemap" visualization. Treemaps can display multivariate data and allow users to quickly understand trends, along with their relative importance. Clicking on squares on the treemap allow users to drill down into deeper levels of information without navigating multiple menus. A highly successful application of this method is the Map of the Market, a powerful data visualization of stock market trends.[19] The research team hypothesizes that applied to building operations, a treemap could allow a building manager to quickly identify trends or anomalies and drill down for more information.

[19] Map of the Market is viewable at http://www.smartmoney.com/map-of-the-market/

Figure 41: Prototype Building Treemap Visualization



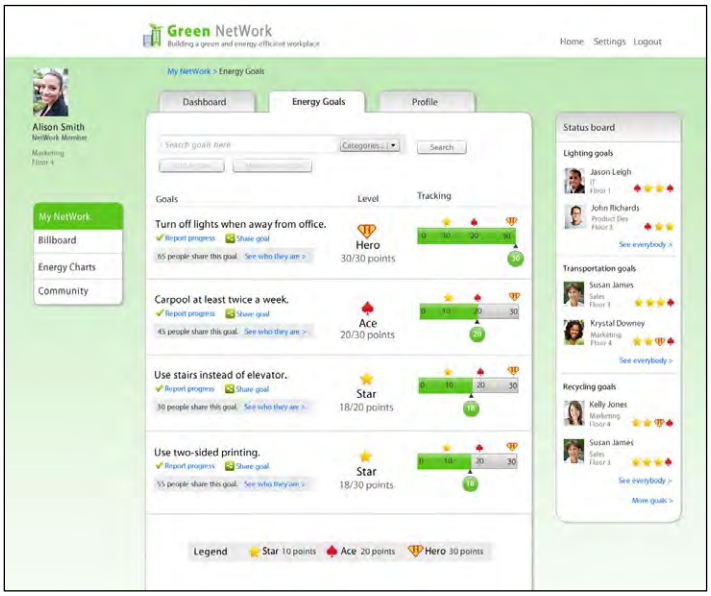
Source: Center for the Built Environment (CBE)

A prototype treemap visualization that might be applied to a commercial building is shown in Fig. 41. The rectangles represent both the relative energy use of various end uses (shown by the areas of the rectangles) and the trends of each relative to a benchmark (shown by the colors of the rectangles). In Fig. 41, the large red square in the HVAC section makes it immediately apparent that a chiller is operating at a level well above the norm. Another bright red rectangle below shows that an auxiliary load is operating above the norm, but its small size helps the building operator know that this problem is less urgent than the errant chiller. Although these anomalies might not show up on a trend line of overall building energy use, it is immediately obvious in this treemap visualization. A view options panel (at right in Fig. 41) would allow the user to select the timeframe for the data visualization, select the benchmark data, and control normalization of the data. For more detailed information, a user clicks on a rectangle of interest to view other screens with details such as trend lines or component status.

Results from both the expert user study and the workplace occupant study (section 4) lend weight to the belief that commercial building energy use and operations could be improved through the availability of tools that enable operator-occupant communication, and that show occupants detailed information about building energy use, especially if that information could be made relevant to their individual workspaces or offices.

Based on these findings, the authors hypothesized that a social media application, integrated into the workplace, can enable such communication and also take advantage of social influence to engage and positively affect energy-related behavior of occupants. For the second phase of this research, a prototype of such an application was developed through iterative design, and will be tested with a number of subjects to evaluate the ability of such an application to provide a platform for commercial building operators to better communicate with occupants, and to influence attitudes and behaviors of office workers (Fig. 42). The conceptual approach to this work was presented at the CHI 2011 Conference on Human Factors in Computing Systems as an extended abstract. (Lehrer & Vasudev, 2011). The results of this work are reported in appendix 2.3.2 of PIER .

Figure 42: Prototype Social Media Application for Energy and Building Operations



Source: Center for the Built Environment (CBE)

6. ACKNOWLEDGEMENTS

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APPENDIX A: BUILDING INFORMATION QUESTIONNAIRE FOR EXPERT USERS: ARCHITECTS, ENGINEERS, BUILDING OWNERS AND OPERATORS

Page 1: Please tell us about your current sources of building performance information.

1. What is your current source of energy information in your building(s)? Check all that apply.
   - ☐ Monthly utility bills
   - ☐ Administrative reports
   - ☐ Energy management and control system (EMCS) or building management system (BMS)
   - ☐ Energy visualization tools (energy information only, no control capability)
   - ☐ Other_____
   - ☐ I have no energy information
2. How frequently do you review energy and/or other performance information about your building(s)?
   - o Once a month or less
   - o A few times a month
   - o One or more times a week
   - o Once a day or more
   - o I have no such energy information
3. Which of the following kinds of energy information do you have currently? Check all that apply.
   - ☐ Real-time energy consumption (Current use)
   - ☐ Normative energy consumption (Compared to other buildings)
   - ☐ Historical energy consumption (Compared to previous time periods)
   - ☐ Time-of-day consumption (Energy use at different hours of the day)
   - ☐ End use energy consumption (Separate energy use of heating, cooling, lighting, etc.)
   - ☐ Estimated bill based on energy consumption (Energy cost for current period and beyond)
   - ☐ Other _____
   - ☐ None of the above
4. Do you receive information from occupants in the building, regarding occupants' satisfaction, problems, or general building performance?
   - o Yes
   - o No
5. If you responded yes to question 4 above, how do you receive this information? Check all that apply.
   - o From complaints submitted by occupants (via a CMMS or other logging method)
   - o From discussions with occupants or tenant representatives
   - o Email
   - o Phone
   - o Heard anecdotally from others
   - o Other _____
   - o

   Page 1A: Branching page if respondent selects EMCS/BMS or energy visualization tool in question 1.

   You responded that you use an EMCS/BMS, or an energy visualization tool. Please tell us more about this:
   - o What energy management and control system(s) (EMCS) or building management system(s) (BMS) do you use? _____

- How often do you use it/them?
- Once a month or less
- A few times a month
- One or more times a week
- Once a day or more
- What features of the EMCS or BMS are most useful?_____
- Are there features the EMCS or BMS are lacking? _____
- What energy visualization tools (excluding EMCS or BMS) do you use? ____
- How often do you use it?
    - Once a month or less
    - A few times a month
    - One or more times a week
    - Once a day or more
- What features of the energy visualization tools are most useful? _____
- Are there features the energy visualization tool is lacking? _____
-

Page 2: Please tell us what types of information would be useful to you.

6. How useful to you is the following energy-related information for saving energy? (Scale response: Very Useful, Useful, No opinion, Not Useful, Not at all useful)
    - Real-time energy consumption (Current energy use now)
    - Normative energy consumption (Energy use compared to other buildings)
    - Historical energy consumption (Energy use compared to previous time periods)
    - Time-of-day consumption (Energy use at different hours of the day)
    - End use energy consumption (Separate energy use of heating, cooling, lighting, etc.)
    - Estimated bill based on energy consumption (Energy cost for current period and beyond)
7. Which of the following end use load information might be useful to you? Check all that apply.
    - ☐ Lighting
    - ☐ HVAC total
    - ☐ HVAC components (heating, cooling, fans)
    - ☐ Plug and process loads
    - ☐ By building zone (floor or workspace levels)
    - ☐ By occupancy or function (classrooms, offices, conference rooms)
    - ☐ Other _____
8. Would it be valuable to have better methods of communicating with the occupants in the building in a systematic manner?
    - Yes
    - No
9. What other types of information would be useful to you? Check all that apply.
    - ☐ Simplified building report card
    - ☐ Building dashboard tool (single screen that gives an overview of multiple building metrics)
    - ☐ Carbon emissions or greenhouse gas metric
    - ☐ Ecological footprint metric
    - ☐ Payback period calculator for improvements
    - ☐ Demand response information
    - ☐ Other _____

Page 3: We have some general questions about you and your interest in future research of this type.

10. Which title best describes the work you do?
    - o Facility manager
    - o Building operator
    - o Building owner, developer, or real-estate professional
    - o Architect
    - o Design engineer
    - o Commissioning agent
    - o Researcher or academic
    - o Green building or energy consultant
    - o Other _____
11. What is the average time that you spend on the computer in a day?
    - o Less than 1 hour
    - o 1-4 hours
    - o 4-8 hours
    - o More than 8 hours
    - o I don't use a computer at all
12. What are the most convenient ways to receive information about your building(s)? Check all that apply
    - ☐ Desktop computer in my workplace
    - ☐ Laptop computer that I can take with me
    - ☐ Handheld device
    - ☐ Standalone display or kiosk
    - ☐ Email notifications
13. What is your age?
    - o Less than 30
    - o 30-40
    - o 40-50
    - o 50-60
    - o Over 60
14. Would you be willing to participate in phone interviews, site visits or paid usability tests?
    - ☐ I would be willing to participate in phone interviews.
    - ☐ I would be willing to have to someone visit and interview me.
    - ☐ I would like to participate in paid usability tests.
    - ☐ No thanks, I don't want to participate.
15. Please provide contact information to register for the prize drawing, and if you told us you would like to participate in future phases of this study.

    Name

    Email address

APPENDIX B: BUILDING INFORMATION QUESTIONNAIRE FOR WORKPLACE OCCUPANTS

Page 1: Tell us about your energy use in your workplace:

1. What information do you have about energy use in your workplace, if any? (Check all that apply.)
    - ☐ Company communications (newsletter, articles, posters, email, etc.)
    - ☐ External communications (articles, websites, email, etc.)
    - ☐ A graphical energy display available online [branch to questions 13-16]
    - ☐ A graphical energy display in the lobby [branch to questions 13-16]
    - ☐ Things learned from a co-worker or manager
    - ☐ Other source of information _____
    - ☐ I have no energy information

    Please indicate whether you agree or disagree with the following statements:
2. I currently make an effort to conserve energy in my workplace.
    [Strongly Agree, Agree, No Opinion, Disagree, Strongly Disagree]
3. If I knew more about the ***amount*** of energy consumed in my workplace, I would make more of an effort to conserve energy. [Strongly Agree, Agree, No Opinion, Disagree, Strongly Disagree]
4. If I knew more about the ***cost*** of energy consumed in my workplace, I would make more of an effort to conserve energy. [Strongly Agree, Agree, No Opinion, Disagree, Strongly Disagree]
5.

    Page 1A:Branching page if respondent selects graphical energy displays online or in lobby in question 1.

Tell us more about your use of graphical energy displays:

- o You have said that you have gotten information from a graphical energy display. How often do you look at this display?
- o Once a month or less
- o A few times a month
- o One or more times a week
- o Once a day or more
- o How has this information been useful to you?
- ☐ It is of general interest
- ☐ It makes me more aware of the building
- ☐ It makes me aware of my own energy using actions
- ☐ Other_____
- ☐ It's not very valuable to me.
- o What are the most interesting or useful features of this energy display?
    - o Electrical use
    - o Renewable generation (for example, solar or wind power)
    - o Water use
    - o Carbon footprint
    - o Description of green features
    - o Other _____
- o What additional features would be interesting or useful to you?

Page 2: Tell us about your energy use in your workplace:

6. What actions ***do you currently*** take to save energy in your workplace? (Check all that apply.)
    - ☐ Turning off desk light(s) when not in the room
    - ☐ Turning off ceiling light(s) when not in the room
    - ☐ Turning off equipment when not in use (printers, fans, heaters, etc.)

- ☐ Using energy-saving settings for computer and monitor
- ☐ Purchasing energy efficient equipment
- ☐ Adjusting thermostat
- ☐ Adjusting blinds and/or windows
- ☐ Other _____
- ☐ None of the above

7. What motivation would make you most likely to save energy in your workplace? (Check all that apply.)
    - ☐ Saving money for my company or department
    - ☐ Benefiting the environment
    - ☐ Seeing the results in an information display
    - ☐ Setting an example among co-workers
    - ☐ Because it is the right thing to do
    - ☐ For my comfort of other personal benefit
    - ☐ Other _____
    - ☐ I am not interested in saving energy in my workplace
8. What would be a useful way to display how much energy is being used in your workplace? (Check all that apply.)
    - ☐ A website showing the ***amount*** of energy used
    - ☐ A website showing the ***costs*** of energy used
    - ☐ A website that shows the energy used in my personal workspace
    - ☐ Energy displayed in terms of pollution created or prevented
    - ☐ Energy displayed in terms of the number of light bulbs or homes that could be powered
    - ☐ Other _____
    - ☐ I am not interested in seeing energy use
9. What kind of details about the energy used in your workplace would be most useful to you? (Check all that apply.)
    - ☐ Energy used by my personal workspace
    - ☐ Energy used by my floor, area, or department
    - ☐ Energy used by the entire building
    - ☐ A comparison of energy used by different floors, areas, or department
    - ☐ Energy use broken down by use (lighting, equipment, heating, etc.)
    - ☐ Other _____
    - ☐ I am not interested in seeing energy use
10. What method of displaying energy information in the building would be most useful to you? (Check all that apply.)
    - ☐ A website I can view when I am interested
    - ☐ Graphical energy displays in the building lobby or other public areas
    - ☐ Graphical energy displays in the elevator
    - ☐ Other_____
    - ☐ I am not interested in seeing an energy display
11. Which of the following best describes your personal workspace?
    - o Enclosed office, private
    - o Enclosed office, shared with other people
    - o Cubicles with partitions
    - o Workspace in open office with no partitions (just desks)
    - o Other

Page 3: We have a few additional questions about you and your energy use at home:

12. Are you aware of the utility costs in your home?
    - Yes
    - No
13. Please indicate whether you agree or disagree with the following question:
    I currently take steps to conserve energy in my home.
    [Strongly Agree, Agree, Not Sure, Disagree, Strongly Disagree]
14. What is your age?
    - Less than 30
    - 30-40
    - 40-50
    - 50-60
    - Over 60
15. What is your gender?
    - Male
    - Female
16. Would you be willing to participate in a paid usability test that will take place at the UC Berkeley campus?
    - Yes
    - No
17. Please provide contact information to be entered in the drawing for the prize and/or if you indicated above that you would be willing to participate in future phases of this study.
    Name
    Email address

Evaluating a Social Media Application for Conserving Energy and Improving Operations in Commercial Buildings

David Lehrer, Center for the Built Environment, University of California, Berkeley
Janani Vasudev, Empirical, Inc. User Experience Research and Design
Soazig Kaam, Arts et Métiers ParisTech Bordeaux

September 2012

1. ABSTRACT

Compared to the wealth of studies on residential energy behavior, studies on the energy attitudes and behaviors of commercial building occupants have been few. However, occupants exert significant control and influence over energy use in commercial buildings, and it has been estimated that 20% to 50% of total building energy use is controlled or impacted by occupants. This study explores the potential for using a web-based social network to promote energy awareness and influence energy-conserving behavior in the workplace. The research team developed a social media application prototype and conducted usability testing with 128 subjects to understand the perspectives of typical office building occupants. The key findings presented are: 1) the influence of personalized energy information; (2) the influence of normative energy information; (3) the potential for sharing personal energy goals and energy data; (4) the effects of incentives such as self-selected goals or rewards, and (5) the implications of using social media for improving communications between building occupants and operators.

Findings suggest that highly individualized energy information, at the level or individual workstations or offices, offers benefits for engaging and informing individuals about their energy use, and that the cost of energy is viewed as the most useful energy metric, a finding supported by previous research. Social aspects of sharing energy use information and personal energy goals were also viewed favorably by the usability test participants. Overall the study found considerable potential for using social media to engage commercial building occupants in energy conservation, and to improve communications between occupants and building management. The paper concludes with recommendations for the design of energy feedback systems including those with social media characteristics.

2. BACKGROUND

In both business and academic circles there is a growing interest in better understanding how to influence human behavior to reduce energy use in the built environment. As energy conserving technologies make building systems more efficient, the portion of energy use attributable to occupants increases, and some of the most cost-effective energy efficiency gains may now be made by changing the behavior of energy users (Sullivan and Sullivan 2009). In spite of notable successes, engaging residential energy customers and commercial building stakeholders to consider energy issues and energy related behaviors remains an ongoing challenge.

2.1 Previous research on energy feedback, behavior, and interface design

A body of research spanning close to four decades shows the potential for incentivizing energy saving in residential settings using a variety of interventions such as financial incentives, energy feedback, and social influencers such as social norms and commitments. Noteworthy meta-analyses of energy conservation pilot programs have found that household energy feedback can produce substantial energy savings that is generally persistent during the time that the feedback is available. Direct energy feedback from a meter or display monitor was found to provide savings in the range of 5-15% (Darby, 2006). A meta-analysis of 57 studies found savings of 4-12%, with the greatest savings resulting when direct or real-time feedback was provided, and when such feedback was more specific to the end use (Ehrhardt-Martinez et al. 2010). Most of the energy savings appear to result from simple behavioral changes such as turning off lights and equipment, and less from investments in energy efficient products.

A more recent meta-analysis from 2012 includes additional pilots of real-time energy feedback that displayed energy consumption and cost through in-home displays, web interfaces, and pre-payment meters, in the U.S., U.K., and Ireland, showing savings of 0 - 19.5%, with an average savings of 3.8% (Foster and Mazur-Stommen 2012). While the average savings was found to be modest, the researchers were encouraged by a small set of households that saved over 25%, a group they call the "cybernetically sensitive," who respond readily to energy feedback, either by being predisposed to such feedback, or due to some new type of learning, habit, or motivation that resulted from the energy feedback. The study found no "one-size-fits-all" solution, but rather that effective feedback pilots resulted from effective design and content of the systems, the reliability of the devices, the level of users' engagement, and the degree to which learning and habit forming are influenced by the system (Foster and Mazur-Stommen 2012).

Tailoring energy information to the cybernetically sensitive and others is becoming more possible through the proliferation of web-based feedback systems enabled by the adoption of advanced metering technologies, commonly referred to as "smart meters." Many utilities, acting under state and federal mandates, have deployed energy information programs by providing consumers with individual energy feedback. It has been estimated that as many as half of all residential customers will have smart meters by 2020 (Karlin 2012).

The visual design and usability of energy feedback systems are important considerations that until recently have gotten little attention in the research literature. Few studies have looked at the specific design of energy feedback interfaces, and there is a significant gap between environmental psychology feedback literature — with large subject samples that provide statistically significant results — and human-computer interaction (HCI) studies that generally rely on smaller subject groups and providing more qualitative findings (Froehlich et al. 2012).

A review of residential demand-response and technology pilots launched by a California utility noted that participants found the energy feedback and curtailment devices difficult to use, and reported that there is "substantial room for improvement in the user interfaces of these devices" and that "devices that are difficult to use are less likely to be used." (Moran and Foster 2012). The paper suggested that vendors need to meet current standards for usability, and that larger-scale purchasers of such products request usability reports. In addition, the authors suggest that tailoring information according to the level of awareness or involvement of the energy users is helpful in keeping their interest.

Foster and Mazur-Stommen 2012 found that people had preferences for certain types of information displayed, and that "the cost of electricity was recalled most easily and seen as most relevant, followed by electricity consumption, and in some cases the 'traffic light' feature providing an environmental clue of usage."

2.2 Energy and occupant behavior in commercial buildings

Compared to the wealth of studies on residential energy behavior, studies on the energy attitudes and behavior of commercial building occupants until recently have been few. A prevailing belief has been that in commercial buildings occupants do not have a major influence on energy use, and that energy efficiency programs will be most successful if aimed at facility managers and operators. However, occupants affect energy consumption by controlling office equipment such as computers, copiers, and printers; controlling overhead or task lights; and in some cases through control of the thermal environment via thermostats, personal fans and heaters, shades, and/or operable windows.

As commercial buildings become more energy efficient through improved envelope and system technologies, and building teams striving for net-zero goals, the percentage of building energy subject to influence by occupant behavior increases. Anecdotal examples show that occupant and manager behaviors are difficult to predict, and in some cases may prevent buildings from meeting net-zero or ultra-low energy targets (*Centerline* 2008). Furthermore, simulation software tools are of little help in predicting or modeling the effects of occupant behavior (Fabi et al. 2011).

The amount of energy under occupants' control is significant. Recent studies found that plug loads represent over 20% of commercial building energy use in California (Moorefield et al. 2011) and offer a great potential for energy savings by powering down computers at night and weekends, and by turning off printers and speakers that are left on continuously but infrequently used (Mercier and Moorefield 2011). Large commercial building energy databases show that plug loads make up over 30% of the total electricity use (CEUS 2006, CBECS 2003).

However, recent studies further reveal that plug loads may be much higher in certain buildings; for example, an energy dashboard installed in a research building (non-laboratory) revealed that miscellaneous plug loads made up approximately 60% of the total building energy use, more than the peak value of HVAC and lighting combined (Marini et al. 2011). Energy data collected by Lucid, a maker of energy dashboards for commercial buildings, included three large office buildings in which plug loads represent over 50% of total electricity use (Murray 2011).

A study of energy efficiency programs in five commercial buildings in North America found savings of 4% for a behavior-only program, to 75% for a comprehensive program in which behavior change was one part. The study outlines strategies for successful behavior programs such as the inclusion of five key engagement techniques: feedback, benign peer pressure, competition, rewards, and reference to appropriate social norms (Bin 2012). Bin uses the term "green work styles" to describe attitudes and behavior favorable to conserving energy at work, and cites actions such as turning off lights, monitors, copiers, and other equipment, and using energy-saving computer settings.

An energy efficiency concept developed at MIT proposed the use of community-action based models to address non-financial barriers with a three-part approach on energy feedback, community engagement, and motivation of individuals. The approach suggests the use of a pledge and tracking system, providing social recognition and rewards, and leveraging social networks to induce energy conserving behaviors (Alschuler et al. 2011).

Prompts represent another approach to encouraging energy saving behavior in the workplace. Researchers found that simple and inexpensive measures, such as sending a calendar reminder encouraging employees to turn off equipment at night and on weekends, reduced desktop computer energy use by an average of six percent (Mercier and Moorefield 2011).

2.3 Social influences on environmental behavior

Using social networks and peer influences to impact energy behavior shows significant promise. Past research suggests that social networks are more likely than other channels of communication to way to inform people of innovations. (Darley and Benninger 1981). While energy feedback may be useful, research suggests that feedback alone is not sufficient to maximize savings, and that additional methods combined with feedback are likely to provide better results. This can be done by tailoring information, making the information vivid, and by using social approaches that include making goals and commitments, using social comparisons and norms, and engaging occupants in small, actionable steps (Ehrhardt-Martinez et al. 2010).

Descriptive norms inform us about what is typically done in a given situation, and have been shown to be more effective in changing environmentally responsive behavior than an appeal focused solely on environmental protection (Goldstein et al. 2008). Descriptive norms in the form of neighbors' energy consumption have shown to produce average household energy savings of two percent, and when combined with prompts can mitigate the potential "boomerang effect" in households that already use low amounts of energy (Allcott 2011). (The boomerang effect describes the potential for low energy users to increase their consumption when they learn that they are more energy efficient than their

peers.) Injunctive messages — those that convey social approval or disapproval — have also been shown to eliminate the boomerang effect (Schultz et al. 2008).

2.4 Leveraging social media for behavior change

The rapid growth of social media has created new online destinations for people to engage in social dialogue and interaction, information exchange, and collaborate on a wide variety of topics. The development of a "culture of participation" describes a fundamental shift from a "consumer culture in which finished goods are consumed passively to a culture of participation" in which people are enabled through the use of "Web 2.0" tools to collaborate on personally meaningful activities (Fischer 2009). Social media are also being utilized to bring about significant social and political change. During the "Arab Spring" the world observed as social media tools were used to mobilize protests leading to the eventual overthrow of regimes in Tunisia, Egypt, Libya and Yemen. In Egypt the opposition movement was launched through use of a Facebook campaign that resulted in protests of tens of thousands of demonstrators (Stepanova 2011). Data collected via ubiquitous social media sources are also proving valuable for research, and have been shown to be useful in predicting the outcomes of future events and outperforming market-based predictors (Asur and Huberman 2010).

Social media tools are being increasingly used to drive behavior change for improving health and other desired social outcomes. The term *captology* coined by B.J. Fogg has been used to describe "computers as persuasive technology" (Fogg 2003) and supports the view that social media may drive behavior change by combining techniques of persuasion with the influential power of mass media (Fogg 2008). Fogg notes that many applications for Facebook have been created for the purpose of impacting behavior, whether for simply purchasing products or for engaging people in social or political activities.

Shared personal documents and information may be described as "social objects"– often in the form of photos, videos, and/or text – that can be viewed, linked to, and distributed by others (Bell 2008). A pilot study for a residential energy feedback program allowed users to create social objects by saving "snapshots" of their energy consumption charts and posting them to a virtual community bulletin board. Posting snapshots allowed users to ask "experts" to help interpret energy consumption, to suggest what was occurring, and to identify possible savings. The study included other social strategies, for example allowing people to compare their energy use to similar households, and resulted in overall energy savings of over 9%. The study also found that the level of engagement, as measured by the frequency of logging into the site, was correlated to energy savings. Between 10% and 25% of users shared their own energy information or posted a comment to an energy expert forum, and 35% to 55% viewed social content. A group of highly engaged users took frequent "snapshots" of their energy charts to document the energy use patterns in their homes. (MacLaury et al. 2012).

3. MOTIVATIONS, HYPOTHESES AND METHODS

The body of research that shows how social factors and energy feedback lead to energy savings in residences raises many interesting questions about how such approaches may be used in a workplace setting. The overarching objective of this study was to evaluate the potential benefits of using a web-based social network, integrated into the workplace environment, to promote energy awareness and

positively influence energy-conserving behavior of typical office workers. A secondary objective was to understand the potential for using a social media platform to improve communications between occupants, facility managers, and building operators.

This research builds on previous studies conducted at UC Berkeley on visualizing information in commercial buildings, looking at the information needs and preferences of both occupants and building operators. Researchers found that a high percentage of workplace occupants report taking actions to conserve energy at work. They say that they would make greater efforts to save energy if they had information about the energy they use and its associated cost (Lehrer and Vasudev 2011). In addition, over 90% of commercial building professionals surveyed expressed a desire for a more systematic way of communicating with workplace occupants (Lehrer and Vasudev 2009). A shortcoming of current occupant-operator communications methods is that they primarily provide one-way communication from workplace occupants to building managers, so occupants have little or no feedback on whether their complaints are heard and/or acted upon. A social media platform offers a convenient way to provide two-way communications between operators, and to allow potentially useful peer-to-peer interactions among occupant peers.

The research team developed a simplified energy feedback prototype with social media features in order to test a number of research questions: (1) the influence of having more personalized energy information compared to zone or whole-building energy use; (2) the influence of normative information such as average energy use of other office colleagues, or selected individuals; (3) the potential effects of seeing and sharing personal energy use data with others in the workplace; (4) the potential effects of letting people set and share energy goals, and receiving reward "badges" for meeting such goals; and (5) the potential benefits of using a social media application for improving communications between with building occupants and operators.

3.1 Research Approach

The prototype was developed on a scale that would be suitable for a single building or corporate campus. The prototype was designed with two types of users in mind: (1) typical commercial building occupants, and (2) expert users such as building managers, building design professionals, or commissioning agents. The prototype was tested with subjects intended to represent these user groups: the study used 128 university students and staff subjects to represent the perspectives of typical office building occupants. In addition, one-on-one interviews were conducted with six expert users who interacted with the prototype, including commercial building energy managers, design professionals, and facility managers.

The project team designed the prototype using an iterative design process. Previous research found that web users viewed the "design and look" as the most important factor in determining the credibility of a website, followed by the navigation structure (Fogg et al. 2003). Therefore the research team endeavored to create a prototype with clear navigation and compelling visual design so that subjects would view the prototype as credible, and that the test experience would better approximate interaction with a real working application.

The team identified the key features to be studied, developed wireframe layouts for typical pages, and then fully rendered "comps" to mockup a functional web application. The prototype allowed people to track their own energy-related activities, to share this information, and to view and react to peers' activities, using interface conventions that are familiar to users of leading social media applications such as Facebook and LinkedIn.

The research team created 13 "clickable" pages using the Adobe Fireworks web prototyping tool and posted pages to the web. Simple navigation links allowed participants to browse pages on the site, and all pages were numbered so that they could be easily referenced in the questionnaires. The prototype may be viewed at http://www.cbe.berkeley.edu/prototype/

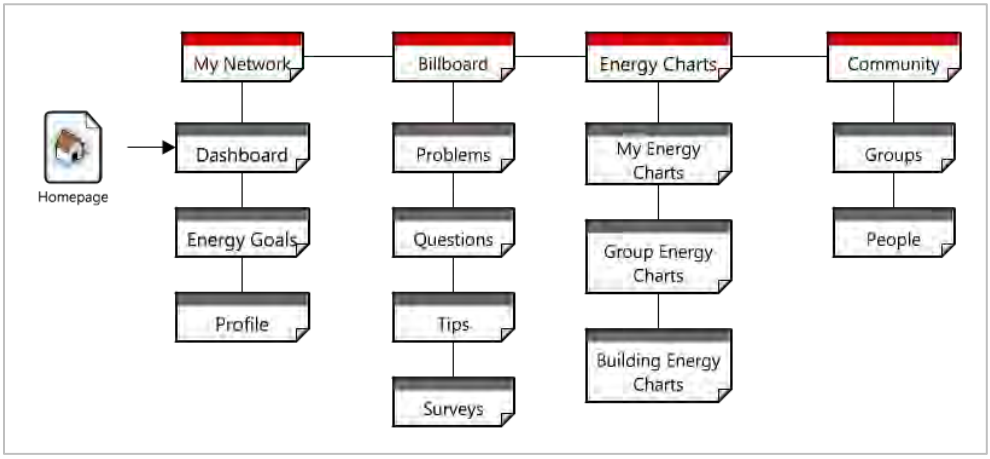

Figure 1. Site map for the social media prototype

The application prototype included four top-level sections: (1) my network, (2) billboard, (3) energy charts, and (4) community (Figure 1). The "my network" section included a dashboard landing page, and a goals feature that conceptually allowed users to set personal energy-related goals, to monitor progress towards these goals, to share this information, and to view and react to peers' activities. A site map of the prototype is shown in Figure 1.

A billboard section was provided to allow occupants to report problems, questions, and energy tips. This feature was designed to resemble typical social media concepts, allowing users to "join," "like," or "comment" on posts, including building managers who can respond to them. A simplified occupant survey function —allowing managers to poll occupants regarding general or specific workplace questions — was also included in this section.

An energy charts section was included to display energy use at various scales, from the individual office or workstation, to floor by floor (represented as a competition) to whole-building energy performance. Various alternates of the energy pages were mocked-up in order to study participants' preferences regarding energy information using various scales and metrics. These prototype features, and corresponding questions, were included as it is becoming increasingly possible to meter end-use energy consumption (e.g., lighting and plug loads) down to the level of individual offices or workspaces, and that costs for such devices are becoming competitive enough that leading companies are investing in

these products[1]. Finally, a community section was included, with a groups page that would allow people to work collaboratively on energy or building related activities.

3.2 Occupant test procedure

The prototype was tested with subjects at UC Berkeley's Experimental Social Science Laboratory (Xlab) using a classroom configuration shown in Figure 2. The Xlab staff recruited subjects by email to attend one of four test sessions lasting 1-1/2 hours. The test protocol included four steps. After signing in and signing consent forms, subjects completed a "pre-demo" online questionnaire with general questions about subjects' demographics and energy attitudes. Next, the research staff provided a ten-minute demonstration of the prototype to introduce the main features. The subjects were then given a hypothetical situation to consider when responding to the paper questionnaire, namely which their employer's management is encouraging people to use a new web-based application to conserve energy, and to help manage the operation of the building.

The subjects were then asked to review the features of the application and to respond to a paper questionnaire with multiple choice and ranking questions about specific aspects of the prototype. Finally, they were asked to answer an online "post-demo" questionnaire with questions about their experience viewing the prototype, focusing on the social aspects of the application. This questionnaire included open-ended questions about aspects of the prototype, including specific likes, dislikes, additional features that could be added, and general comments. The three questionnaires used in the test are included as appendices A, B, and C.

Figure 2. Xlab test setup.

3.3 Energy expert test procedure

To evaluate the potential benefits and barriers of such an application from the perspective of building professionals, we conducted one-on-one interviews with expert users. The objective of this work was to understand whether building managers see potential benefits from using such an application to communicate with building occupants, track and respond to complaints, and survey occupants about buildings management issues. Via email the research team invited building industry professionals to participate in the study, and those selected for the study included facility managers, an energy consultant, and an architect.

Interviews were conducted in person, or remotely using desktop sharing utility, and lasted for about 60 minutes. The interviews followed a semi-structured format, participants were walked through the various features of the application and were asked to reflect on its overall usefulness and user-experience.

[1] Personal correspondence with industry contacts at Adobe Systems and Lucid, May 2012.

4. RESULTS FROM BUILDING OCCUPANT STUDY

The prototype was used in the Xlab study to study the research questions described above. Below we describe the subject sample, and summarize results.

4.1 Demographics, social media use, and environmental attitudes of study participants

This section briefly summarizes the results of the pre-demo questionnaire. The 128 subjects were comprised of 80% undergraduate students, 11% administrative staff, 7% graduate students, and 2% other. The age distribution reflected the Xlab's campus recruitment pool: 24% under age 20, 69% ages 20-29, and 8% age 30 or older. Asked about office experience, 41% of subjects reported having one year or less of office experience, 39% had 1-3 years, and 20% had 4 years or more. As may be expected of such a subject sample, the survey found that 99% of the subjects use social media sites, that 8% report using them on an hourly basis, and 71% use them several times a day.

The subject pool was largely a pro-environmental group. A large number of subjects feel that environmental issues are important to them personally (42% "very important," and 48% "somewhat important"), and that it's important for individuals to reduce their impact on the environment (68% "very important," and 27% "somewhat important"). Asked to express their thoughts about energy conservation, 35% responded that "energy conservation is good for the environment," and 45% feel that it is their "duty as a socially conscious person" to conserve. Regarding their own home energy use, 84% reported that they were "fairly familiar with utility costs" or had a "general idea about them," and 66% reported that they were already taking measures to reduce their household energy use.

With subjects that consist of generally pro-environmental university students, the results may not be representative of the full range of North American office workers. However, the sample may be a reasonable representation for employees in sectors such as technology and new media that have a large percentage of educated younger workers (for example, Facebook, Google, Yahoo!). Such progressive companies may be more likely to adopt energy-savings programs, and more likely to be early adopters of energy feedback and social media tools for company purposes.

Previous research also provides evidence that younger people have been found to be linked to higher energy savings due to energy feedback (Foster Mazur-Strommen 2012). Considering the test subjects' familiarity with social media and their generally pro-environmental attitudes, the research team believes that such a subject pool may provide useful insight about the social aspects and energy features of the prototype.

4.2 Test results regarding granularity of energy information

To investigate the influence of varying granularity of energy displays from whole building, to zones within the building (in this case floor-by-floor), to the level of individual offices or workspaces, subjects were asked to rate these alternatives in terms of their interest and usefulness. The displays showed power trends for a week, energy used and/or cost saved (Figures 2-5). Similar to many pages on the site, the page displays other users that have shared information on the site.

The subjects expressed interest in the energy displays of all three levels of granularity, with a slight preference for the floor and individual levels. However their ranking of the usefulness of the three energy displays increased with the level of granularity, with individual level displays getting the highest score, as shown in the chart in Figure 5. Subjects showed the most interest in their personal energy use, which informs the first of our five research questions. The personalized energy feature was also frequently cited as a favorite aspect of the application, as summarized in the summary of "liked" features (Table 1) and shown in the examples below:

> *Energy charts clearly show the cost of energy that I used. It also compares with average energy usage which gave me more realization of how much I am using.*
>
> *I like that you can monitor your own energy use and so you know how effective you are being in your efforts at using less energy.*
>
> *I like the visualization of my own energy consumption and also having an idea of the cost.*

Figure 2. Whole building power total and by major end use

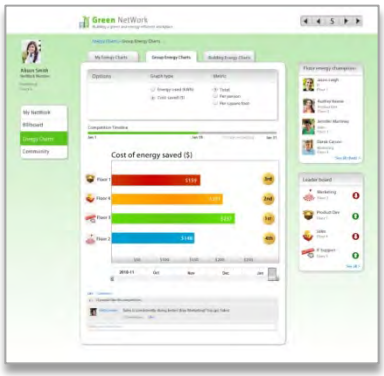

Figure 3. Energy use shown by floor level (energy competition)

Figure 4. Individual energy use

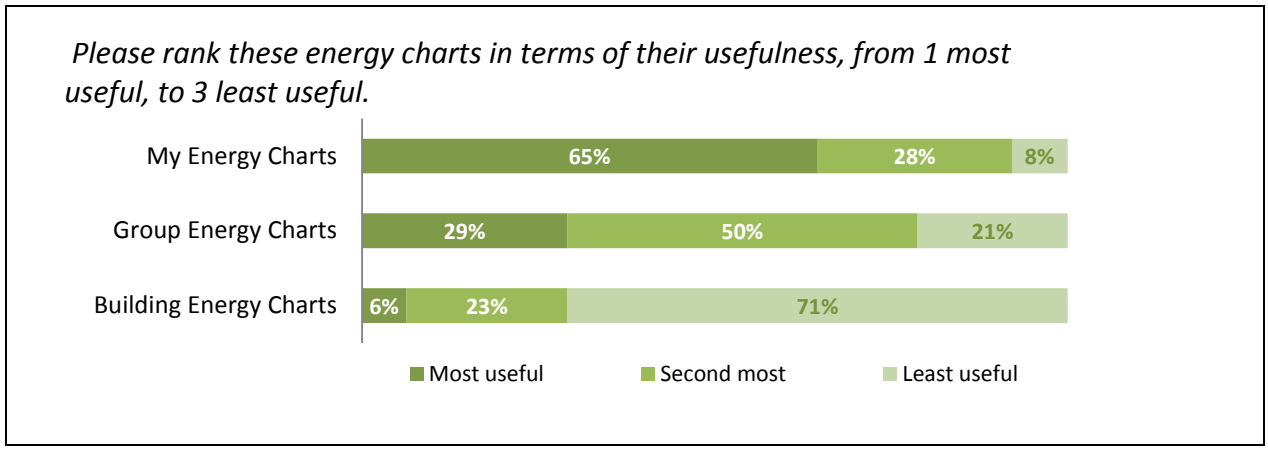

Figure 5. Ranking of usefulness of energy granularity

4.3 Test results regarding energy metric alternatives

Behavior change literature suggests that people are more likely to respond to informational cues if the information is made "vivid" to the audience. As energy use is all but invisible to building occupants, a goal of this study was to identify which types of energy metrics would be considered the most relevant, interesting or useful to potential users of an energy feedback application. Subjects were asked to evaluate four energy metrics that represented energy use at the level of a single occupant. The displays each represented daily energy use for one week, including the weekly total, represented by four metrics: (1) power in watts, represented by a time-series chart; (2) a bar chart showing energy used in kWh; (3) a bar chart showing energy used in "light-bulb equivalent" (the number of 25-watt bulbs in use for one hour); and (4) a bar chart showing the cost of energy used. The prototypes represented values that might be realistic in an office setting, for example, the weekly cost for energy to the individual workspace was shown as $1.43. Subjects were asked to rate how useful and interesting these charts were, and also to rank them from most to least useful.

The responses show that subjects found the cost metric display to be the most interesting and useful, the power (watts) and total energy (kWh) to less so, and the light-bulb equivalent display to be the least interesting and useful. This stated preference for cost as an indicator for energy use is consistent with other recent research (Foster and Mazur-Stormmen 2012) and was preferred in spite of the relatively low total cost of energy displayed. The results of a ranking question are shown in Figure 6. The conventional energy metrics (watts and kWh) were found to be of lesser interest and usefulness to subjects. Although the "light-bulb equivalent" was intended to show energy use using a familiar object, in fact this was viewed as the least interesting and valuable of the three energy metrics.

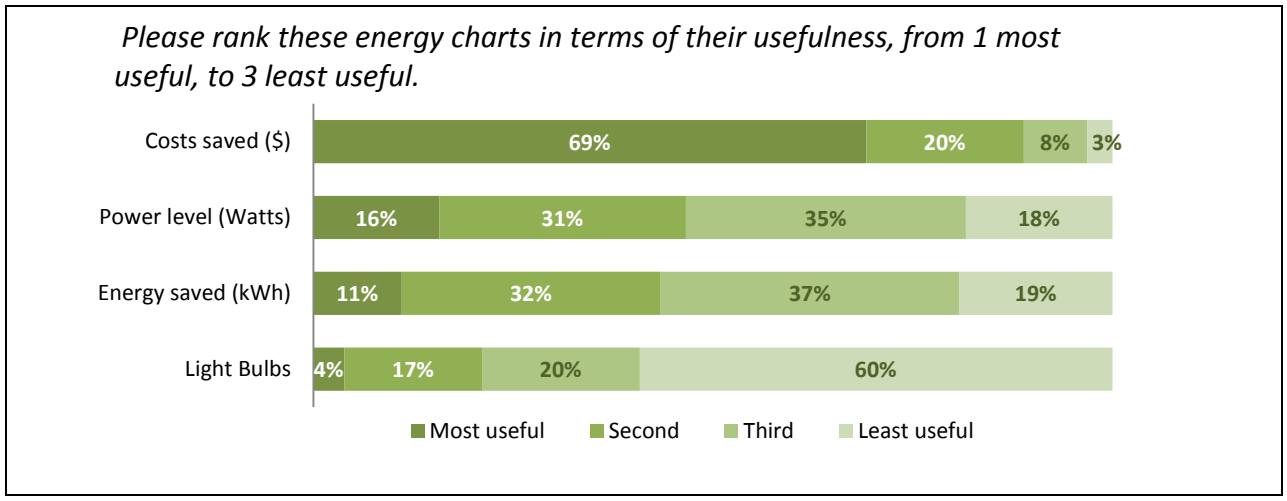

Figure 6. Ranking of usefulness of energy display

The comments in the closing questionnaire showed the cost metric to be one of the prototype features most liked by subjects, and provides insight as to why this metric was preferred:

> *...most people (unless they are familiar with the watt and joules or technical terms like that) will not be concerned about pages like "watts used." I think people will be more concerned about the costs of energy and not the amount of energy spent.*

Seeing the energy use in terms of Kilowatts kind of held my interest a little less I think because it's not a form of measurement that I am familiar with, as currency, like dollars, is much more familiar and is maybe more attractive for a social media type of thing like this.

Cost is probably the biggest driver for someone to save energy and having this comparison between days increases one's awareness of his or her daily energy usage.

4.4 Preferences and attitudes regarding normative energy information

Subjects were asked to consider energy displays showing their personal energy use compared with various types of normative references. The energy display shown was a one-week bar chart, with a weekly total shown in terms of the cost of energy (found to be the most useful metric, as noted above). Subjects were asked to evaluate and rank the following options: (1) no comparison; (2) personal energy use compared to the average use; (3) personal energy use compared to an individual that could be selected by the user.

The participants showed a strong preference for comparison to the average user, with 79% of subjects ranking this at the most useful option, and 13% ranking it as the second most useful (Figure 7). The comparison to a user-selected individual was ranked as most useful by 17% of subjects. A comparison to an average user is the most meaningful from a statistical standpoint, and the subjects seemed to understand this intuitively. However the ability to compare one's energy to peers may be useful for engaging individuals who are not interested in energy data in itself, but who might enjoy the gaming aspect of a friendly competition among peers. In general, the energy chart features were viewed favorably by almost all test subjects, with 98% responding "strongly agree" or "agree" that the energy chart features of the prototype would be useful in saving energy.

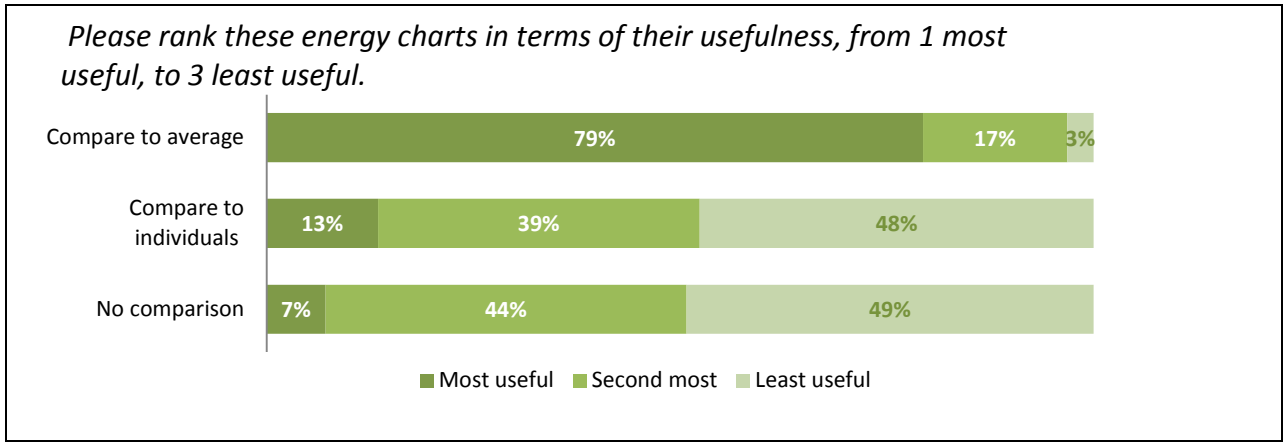

Figure 7. Ranking of usefulness of normative energy information

A number of responses to open-ended questions reveal participants' sentiments regarding comparisons to the average users and selected individuals:

I also like how the user can compare his or her energy usage to that of everyone else's (average). If my usage was higher than the average, looking at the charts will make me realize that and I would make a more conscious effort to conserve energy.

[I liked] seeing my personal energy use as well as my office mates.

I liked the comparisons between individual energy use. It made it competitive and like a game, which is a good thing when it comes to energy conservation (a typically dull conversation topic.)

I like the comparison features that let you see your personal energy consumption compared to the average. Even breaking it down by floor or department is helpful because it keeps others motivated to conform to other energy saving groups.

4.5 Findings regarding sharing personal energy information

After subjects reviewed the three options for normative energy information, they were asked additional questions about the social aspects of this feature — whether they would share their energy information, whether they would be interested in seeing energy charts of others, and whether such sharing of information would be interesting and/or useful for saving energy. (In the short demonstration provided by the research staff, it was explained that users could opt-in to sharing their energy charts with others in their workplace.)

The results from these questions are shown in Figure 8. Two-thirds of the subjects (66%) responded "strongly agree" or "agree" that they would share their energy charts with others. Responses were similarly positive for related questions: whether sharing information would be useful in saving energy (72% "strongly agree" or "agree"), subjects' interest in seeing energy charts of other people (79% "strongly agree" or "agree"), and whether the social aspects of the application would increase subjects' interest in energy (79% "strongly agree" or "agree"). Overall the results suggest that the social media aspects of such an application would be effective for engaging building occupants and increasing interest in their personal energy use.

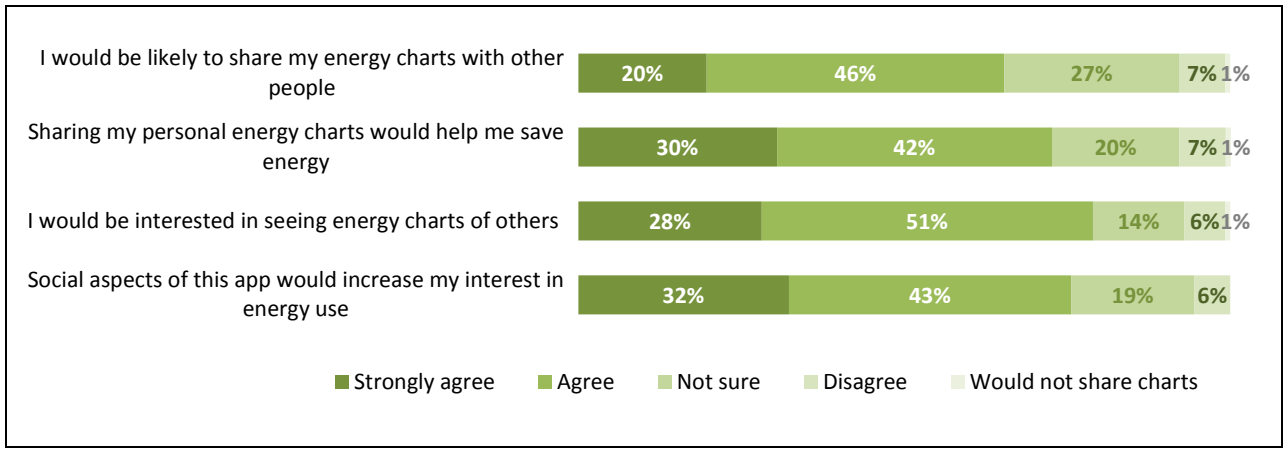

Figure 8. Likert scale questions regarding sharing energy information

Younger individuals tend to be more comfortable sharing information via the web, and this is reflected in the percentage of positive responses from these participants. Comments on sharing personal energy goals were generally positive, as in these examples:

I will be more likely to set higher goals to save energy if I knew people are tracking my usage through this application.

I also like how the user can compare his or her energy usage to that of everyone else's. If my usage was higher than the average, looking at the charts will make me realize that and I would make a more conscious effort to conserve energy

However other comments about sharing personal energy information include concerns that would need to be considered in the design of such an application. For example, some comments included concerns about the use of energy charts by peers or management to view patterns of occupancy, that peers might be judgmental of others who used more energy, and/or that this may contribute to a "less friendly" work environment. Also, as people may have different equipment needs, some participants expressed the opinion that such comparisons are not fair, and that might cause them to opt-out of sharing or using the application:

I think a person versus person feature...could negatively and unfairly single some people out that may need to use more energy for their job than their co-workers.

Sometimes, people may not want others knowing everything about them such as energy use. I can just imagine people going, "Wow, that person is such an energy pig," behind other people's backs.

I don't like the feature where other people could view how much energy I consume, and would likely not choose to share such information. For example, by looking at a co-worker's energy use I can become more judgmental about that person.

In the design recommendations at the end of this report, the authors suggest to potential energy feedback developers that the intended uses of personal energy use be made explicit, for example, that such information will not be used for monitoring employees' schedules, hours in the office, or equipment use.

4.6 Results regarding setting and sharing energy goals

The prototype included an "energy goals" feature that allows users to select individual goals, to report their progress towards these goals, and to see other people who have selected the same goals (Figure 9). As represented in the prototype, users can obtain "badge" levels of "star," "ace," or "hero" by reaching designated point scores for each goal.

Subjects reacted favorably to this feature, with 81% of subjects responding "strongly agree" or "agree" that they would be likely to use such a feature. Subjects also felt that they would be likely to share their energy goals with others (77% "strongly agree" or "agree"), that sharing would help them to meet these goals (73% "strongly agree" or "agree"), and that they would be interested in seeing the energy goals of others (80% "strongly agree" or "agree"). Responses to energy goals questions are shown in Figure 11. The energy goals were cited frequently in the open-ended questions regarding favorite features of the prototype.

The energy goals is [sic] also a great start to allow employees to be conscious of their energy usage.

I also enjoy that users would be able to set goals and have other people join those goals too. It really creates an environment of peer support.

I also like that you can set your own goals, because what may be a realistic energy-saving practice for one person might not be as feasible for another.

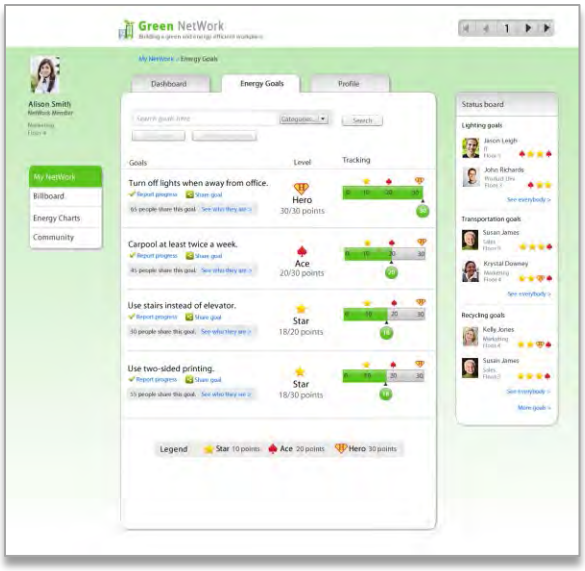

Figure 9. Energy goals feature

Figure 10. Billboard feature

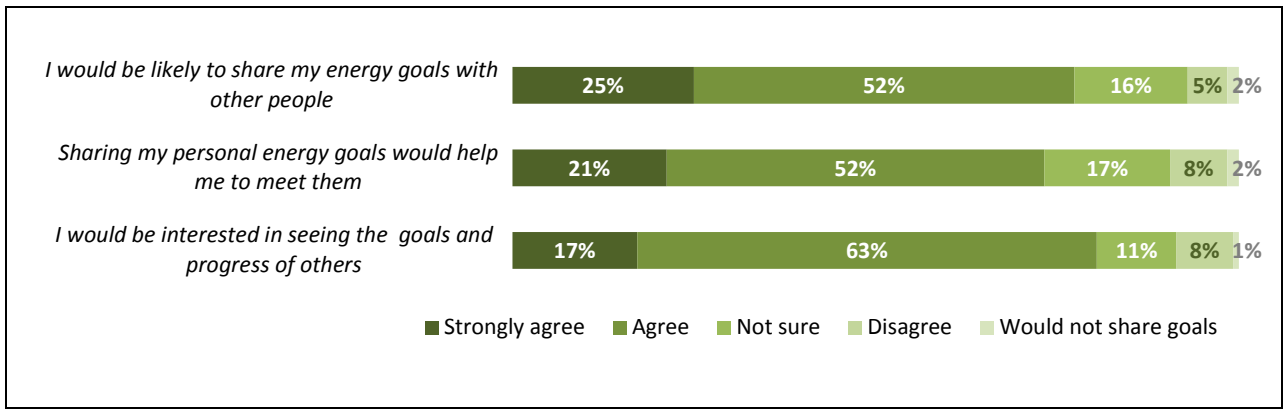

Figure 11. Responses to Likert scale questions regarding energy goals

4.7 Use of the prototype for communicating with building operators and occupants

A billboard feature allows people to report "problems," "questions," and "tips," using a format that is familiar to social media users, such as the "wall" feature of Facebook, allowing people to comment on posts by others, and view building managers' responses to service requests (Figure 10). Study participants responded favorably to this feature, with most all indicating that the billboard feature would be generally useful in the workplace, with 95% responding "very useful" or "somewhat useful". Participants indicated that they would be likely to use the application to report a problem (96% "strongly agree" or "agree") and to ask a question (79% "strongly agree" or "agree") as shown in Figure 12. Many also reported that they would be more likely to use the application than by phone or e-mail, to report a problem (77% "strongly agree" or "agree"). Over half reported that they would be likely to use the application to post an energy tip (58% "strongly agree" or "agree").

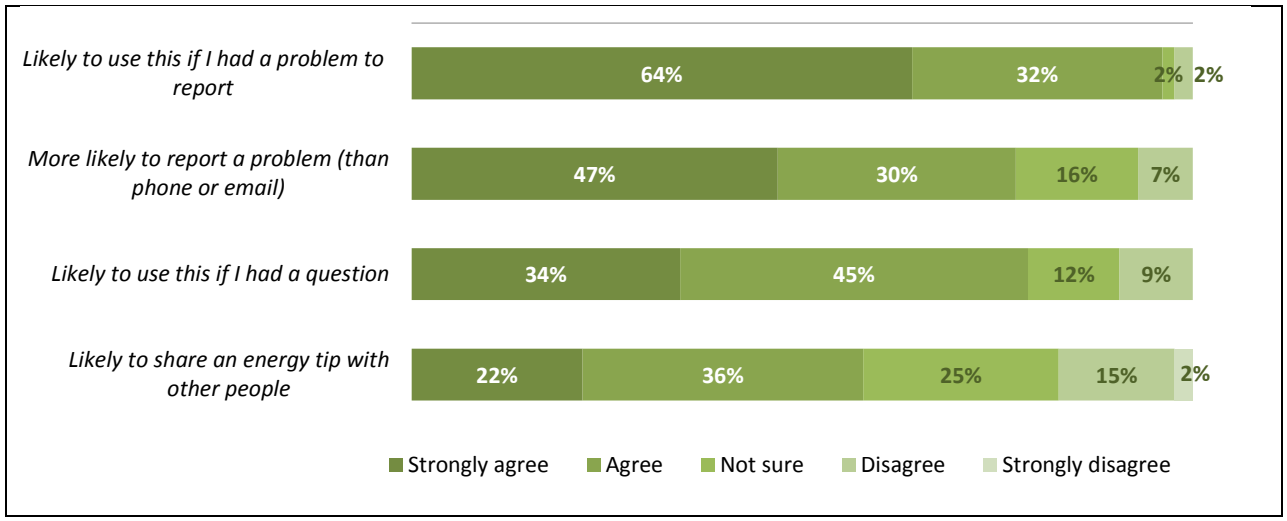

Figure 12. Responses to Likert scale questions regarding billboard feature

Subjects responded that they would be comfortable using a web-based application to report problems or pose questions to facility managers. Compared to making service requests by phone or e-mail, the web app would provide additional benefits by allowing users to see the status of a request, and to see if peers have similar problems. Subjects commented on potential benefits from the collaborative nature of the application:

> *I like how you can personally make complaints or ask questions on the site while also getting feedback from your colleagues--its nice to know you are not the only one experiencing problems and that your complaints will not go unseen.*
>
> *Being able to see other people's complaints about the building is useful because I can see whether my complaints about the building are shared by others.*
>
> *I know people tend to complain about a problem at the office, but don't really know how to get it fixed... so if enough people are complaining in a constructive way, then I see potential for a faster response.*
>
> *Also, I think a board for technical problems with the building is a good idea; currently you call maintenance and try to figure out if they know about your problem, here you can just post it, search if anyone else has the same issue, and maintenance could respond to the board.*
>
> *The Problems page is a very good idea to foster open communication between people and management regarding problems in the workplace.*

4.8 Results of closing questionnaire

After viewing the prototype and completing the paper questionnaire, subjects were asked to complete a closing questionnaire with open-ended questions about the website, and were asked about aspects of the prototype that they liked, things they disliked, features that could be added to improve the application, and general comments. Many the questions inquired about participants' sentiments regarding the social aspects of the application.

4.8.1 Comments about most liked features

In response to the first open-ended question, participants expressed what they most liked about this application. Aspects of the energy charts and billboard features were the most frequently cited features in these responses. The most frequently cited features include viewing/sharing individual energy use (46 like comments), comparing one's personal energy use to the average (45 comments) and being able to report problems and view those reported by others (43 comments). The responses were binned by the research team into generalized categories and summarized in Table 1, below.

The study participants provided many positive comments on the application's potential to increase environmental awareness in the workplace, for example:

> *I like the friendly, social aspects of the application that remind me about the connection between work and the environment. I also like that it makes me think about how our actions affect one another and the environment. It makes me think about saving energy in an easy straightforward way.*

Summary of most commonly cited "liked" features		**Number of comments**
Energy Charts	Viewing, sharing individual energy use	46
	Viewing the energy consumption floor by floor	22
	Other energy charts comments	18
	Seeing personal energy use and cost saved ($)	
	Compare to average cost	45
	Compare to individuals	32
	Other energy use comments	16
	Energy charts: Total "like" comments	**183**
Billboard	Reporting problems and seeing others' complaints	43
	Giving or getting some tips	16
	Asking and answering questions	12
	Other billboard comments	8
	Billboard: Total "like" comments	**82**
Other "likes" comments	Friendly competition	22
	Setting and sharing my energy goals	15
	Interacting with others/social aspect of the website	12
	Clean visual design	8
	Familiarity with other social website	7
	Increase awareness about environment/energy efficiency	6
	Total other "like" comments	**80**

Table 1. Most common responses to question about "liked" features

4.8.2 Comments about disliked features

In the second open-ended question, the participants were asked to note features or aspects of the application they disliked. The most frequently cited disliked features or aspects of the application are summarized in Table 2.

Summary of most commonly cited "disliked" comments		Number of comments
Social media aspect	***Sharing my personal information and energy use***	
	Unfriendly working environment	10
	Other sharing comments	9
	Unfair aspects of energy comparison	
	Different devices used	5
	Other unfairness comments	4
	Social media aspect of the website: Total "dislike" comments	**32**
Posting problems	The effectiveness of posting problems	5
	Other posting comments	8
	Posting problems: Total "dislike" comments	**16**
Other "dislikes" comments	Lacks a strong incentive to save energy	14
	Groups feature	11
	Too much information, overwhelming tracking	9
	Survey feature	6
	Difficult to use the website, and/or find information	5
	Total other "dislike" comments	**49**

Table 2. Most common responses to question about “disliked” features

A number of participants expressed concern that the social aspects of the application could contribute to an unfavorable work environment, and/or make unfair comparisons due to varying equipment needs or work schedules, as noted previously (32 dislike comments). Several subjects noted that more explicit messages about saving energy would be beneficial, and that energy feedback alone would not be a sufficient incentive to induce behavior change (14 dislike comments).

The groups feature drew a number of dislike comments, as participants did not think such a feature would be beneficial. While many thought the billboard feature would be a useful platform for interacting, some negative comments questioned this feature in terms its effectiveness, including the concern for the “potential to create, long, endless threads,” and that seeing the minutia of building related problems would not be valuable and could be tiresome to read through.

4.8.3 Comments about features to add

Finally, the open ended questions asked people for ideas for additional features they would find useful for such an application, and general comments. The responses were highly varied and are summarized in Table 3.

Some participants suggested embedding the application into other social media applications that people already used, and/or make it available on mobile devices. Suggestions included other ways to engage users, such as “more features or applications that reward [the user] for visiting”, ranking the “best commenter”, the “most visited” feature, the “most improved” energy use, or informing people via emails or alerts about a new features.

Summary of features to add comments	Numbers of comments
More educational information about the energy issue	28
More details about one's energy consumption	18
Influence on energy behavior	17
Additional billboard features	10
Extend the social media aspect	10
More options for the "comparison" feature	9
Privacy details settings	6
The website and the user	5
Involvement of the company	5
Total "additional features" comments	**108**
General comments related to additional features	
Explain how simple conservative gestures are effective	10
Financial reward or other prizes	13
Feature to check if a problem has already been posted or resolved	6
Ability to pinpoint which device is the most energy-consuming	5
Application for existing social websites or smart phones	5

Table 3. Comments for additional features

4.9. Summary of occupant usability findings

The Xlab study provided detailed feedback on the prototype, and insights into to the five study questions, which we summarize below:

1. Participants showed a clear preference for more personalized energy feedback, ranking individual energy use first, group or floor energy second, and whole-building energy use third. A display of energy costs was the preferred display format in comparison to conventional energy metrics (kW or kWh) or an energy approximation (in this case, an equivalent number of light bulbs). In responses to open-ended questions, viewing and sharing individual energy use was the most frequently cited beneficial aspect of the prototype.
2. Test participants were interested in normative information and preferred to compare personal energy use to an average user in the building, and this was confirmed in ranking and open-ended question formats. Subjects felt that such information would be valuable for helping them to save energy, and many felt that friendly competition among peers would be useful in an effort to save.
3. Participants responded favorably to a feature for setting, sharing, and tracking personal energy use via an energy charting feature. Responses were generally positive about sharing energy goals and energy use, however a subset of subjects expressed concerns about privacy, how such energy data might be viewed or used, and/or unfair comparisons between users with different equipment needs.

4. Test subjects also responded favorably to an energy goals feature, with 70-80% indicating interest in setting and sharing personal goals, seeing the energy goals of others, and using this feature to help them save energy.
5. Finally, participants were positive about features that allowed reporting of problems, and getting feedback from facility managers about problems, with 96% indicating that they would use such a feature, and 77% saying they would be more likely to use this feature compared to reporting a problem by phone or e-mail. Concerns were raised about the possibility that long and/or irrelevant comment threads might result.

In the final section of this report, the authors provide recommendations for potential developers of energy feedback systems, based on these findings.

5. RESULTS OF ENERGY PROFESSIONALS STUDY

The fifth research question focused on the potential benefits of using social media for improving communications between building occupants and operators. The research team conducted one-on-one interviews with six commercial building experts who would provide a range of perspectives. The objective of this work was to understand how professionals see potential benefits and barriers from using such an application to communicate with building occupants, track and respond to complaints, and survey occupants about buildings management issues. The study participants included:

- A project manager in an energy consulting firm
- An architect in a firm that designs a large number of green and LEED-certified projects
- Facility managers for a single building or multiple buildings on a university campus

Interviews were conducted in person or remotely, using a desktop sharing utility, and lasted approximately 60 minutes, following a semi-structured format. The participants were walked through the various features of the application and were asked to reflect on its overall usefulness and user-experience. Research staff noted subject responses during the interview, and the comments were aggregated by feature or content of the subjects' comments. Comments from the six participants are noted by designations P1-P6.

5.1 Detailed findings from energy professionals study

The interview subjects found the billboard and surveys features to be most interesting of those presented, with the billboard seen as a convenient way to report energy-related problems. Subjects (P2, P3, P4, P6) noted that by design, the billboard ensures that the occupants' concerns capture the attention of more people in the building, thereby overcoming some shortcomings of traditional complaint systems, including response time and lack of acknowledgment that the problem has been received (P3, P4). In cases in which a problem cannot be addressed due to technical or other limitations, the billboard affords a platform for the manager to communicate this to the occupants effectively (P6).

From a building operations perspective, the problem reporting functionality helps a building manager get a sense of where most of the problems in the building are, plan corrective operations and gauge

people's reactions to it (P3, P4). One participant (P3) called the billboard a great platform to post maintenance and other operations-related announcements.

The social components of the application were viewed as useful by both the building managers and the occupants. To the building manager, the number of people that "join" a problem indicates its severity. To an occupant, it may be oddly reassuring that he or she is not the only one inconvenienced by the problem. One participant (P2) noted that while traditional social network terminologies such as "like" build on people's existing knowledge and thus are easily understandable, they might not necessarily be applicable to this scenario and suggested a rethinking of these terms, "*While terms such as 'like' and 'join' are popular and people might understand them better, it is strange that someone might 'like' a problem. 'Agree' or 'strongly agree' might more appropriate*."

Even while most of the participants felt that this public problem-reporting space would make the occupants more aware of energy-related issues in the building and therefore view it favorably, one participant (P1) was wary of its 'social-ness.' "*Some people might be reluctant to post problems they are uncomfortable talking about. They want to keep a low profile. Others, on the other hand, are just more social. This might give a skewed representation of the building's energy climate.*"

Participants also noted a few useful additions to the billboard. One such addition is the ability to filter the problems by priority or date (P2, P5). A search function would also be beneficial particularly when the billboard is tied to a database of documented problems and recommended solutions (P5).

The survey feature was viewed favorably by the energy experts (P2) as well as the building managers (P3, P5, P6). This was seen as a very useful tool particularly to the building manager because its helps them to quantitatively measure occupants' comfort levels (P3, P5). Further, it allows the manager to assess occupants' responses to changes in building controls such as lighting, heat, etc. (P3). Additional useful features would be options to run multiple surveys at the same time, and analytics that track differentials and performance improvements over time (P4, P5).

A number of the participants found energy goals useful (P2 - P6). One participant (P5) described the feature as a "*vehicle to help [an individual] participate in energy conservation*." Participants (P2, P5) noted that people are "*competitive by natur*e" and are "*more likely to do better, if there is someone watching over their shoulders.*" One participant (P1) however, was more skeptical. He pointed out that it's in some peoples' nature to "*be rebellious*" and "*not appear too goody-goody*." Further, participants (P2, P6) recommended exploring visualizing people's energy goals and how they relate to the leaders graphically, and provide a way to indicate the larger impact of goals in order to differentiate between them.

To most participants, the costs of providing individual energy feedback outweighed the benefits, from the perspective of the energy professionals. Participants (P1 - P5) indicated several reasons why individual energy displays were the least useful of the energy charts: (1) the information is too granular to be meaningful on the scale of the whole building, (2) it's difficult to measure energy use at this level of granularity, and (3) the difficulty of establishing individual accountability in a shared workplace. However, one participant (P6) was optimistic. He pointed out that the "*devices to measure personal*

energy use are increasingly becoming easy to use and deploy. As a result, obtaining personal energy use data will become easier in the future."

Participants believed that aggregate numbers of energy use including energy/dollars saved over an extended period of time and its relation to other statistics such as national averages are more likely to have an impact on the user than the information that they "*saved three cents over somebody else"* (P4, P6). The concept of comparing personal energy use with others in the building is, however, a "*fun thing to do"* and "*promotes friendly competition"* (P6). A majority of the participants agree that a useful metric would be an individual's carbon footprint (P1, P2, P4, P5, P6). Some suggested that a person's carbon footprint should comprise not only his or her activities at work, but all of her daily activities including travel to and from work, energy use at home and work-related travel (P1, P3, P4).

5.2 Summary of energy experts study

The generally positive feedback from energy professionals regarding the billboard for reporting and responding to comments, and the social aspects of the prototype application, show that such a system may provide benefits to commercial building operators and professionals. Providing two-way communication between operators and occupants was viewed as highly beneficial, and may help operators to triage and prioritize complaints, and to update occupants on complaint status, and to potentially alleviate occupants' frustrations when solutions to reported problems are not immediately available.

For building operations, the individual-scaled energy was seen as less useful than aggregated energy information, however some of the energy experts recognized the potential for driving behavior change offered by the more granular energy feedback. This finding stands in contrast to the occupants' preference for individual energy information and shows that information needs vary considerably between the two groups. This finding leads the research team to recommend distinct interfaces for these two commercial building stakeholder groups, as outlined below.

6. CONCLUSIONS

This study found considerable potential for using social media to engage commercial building occupants in energy conservation, and to improve communications between occupants and building management. Findings reveal that highly individualized energy information, at the level or individual workstations or offices, has potential benefits for engaging and informing individuals about their energy use; and that cost is considered the most vivid and tangible energy metric, something that is supported by other research.

Sharing energy information using social media tools appears to offer a means for increasing occupant engagement and interest in energy matters, as many subjects expressed interest in sharing energy goals and energy charts with others, and seeing that of peers. However an important caveat to this finding is the concern some test subjects expressed that public display of personalized energy information holds a risk of stigmatizing individuals that use more energy than the average user, which may contribute to ill feelings among peers, and may reduce the potential level of engagement by all participants.

6.1 Design recommendations for social media energy feedback

Based on the results of this paper and related research, the authors offer the following design recommendations to potential developers and customers of social media technologies for energy feedback:

6.1.1. Provide energy information that is highly specific to the individual building occupant.

In this study participants showed an overwhelming preference for energy displays on the scale of the individual workspace. As this requires specialized hardware (smart plug strips, advanced lighting controls, etc.) and associated software with costs that are not negligible, this may only be feasible for highly motivated companies. (However the research team found numerous anecdotal examples of leading companies that have installed such systems, and costs are likely to come down as these are adopted and more product choices are available.) When providing individual energy feedback is not possible, zone or floor level energy information is preferable. If only whole building energy data is available, showing energy in terms of per person use use may be an alternative way to engage occupants.

6.1.2. Display energy information in terms of the cost of energy use as the default.

Subjects had a strong preference for seeing energy use data in terms of costs, in spite of the relatively low cost of electricity used by an individual (less than $2 per week per person in this study). In cases where energy use is low, it may be preferable to show energy use in terms of weekly, monthly, or annual costs. For example, an energy display could show a user the yearly cost if the current power level were continued. While a web interface can easily let users toggle between various energy metrics, and this provides the benefit of letting users explore and interact with data, this study found that cost is considered the relevant energy metric, which is consistent with other recent research.

6.1.3. Provide normative energy comparisons in terms of average energy use, and also show the energy use of an energy efficient user.

Subjects were most interested in comparing their energy use to the average user in the building. To avoid the "boomerang effect" (when low-energy consuming individuals use more energy when they see that they are below average) displaying the energy use of an efficient energy use is a viable approach (as seen on some energy feedback systems such as home energy reports that provide comparison the energy use of top 20th percentile in energy efficiency).

6.1.4. Allow users to share and view personal energy displays as "social objects," and to share and view energy saving goals.

Subjects showed a strong inclination to share their energy use charts and goals with others, and indicated that the social aspects of such sharing may be useful for engaging people in energy conservation. This capability should be an opt-in feature, as some subjects expressed concerns about privacy or competition. Having an option in which people can share their energy use anonymously may allow people to be engaged with a program while not being identified personally, for people who harbor such concerns. Alternately, competition could be shown by comparing individuals to their own personal baseline, and comparing and/or reward savings that individuals make over time.

6.1.5. Focus on positive aspects of energy comparisons, avoiding judgmental feedback.

For the reasons noted above, energy use should be shown in positive terms such as energy saved compared to past use, potential for savings, etc. Obviously terms that reflect poorly on groups or individuals (e.g., energy hogs, wasting energy, etc.) should be avoided, and the program should explicitly recognize that energy use will necessarily vary greatly among individuals as a result of varying usage and equipment needs.

6.1.6. Be explicit about the use of energy information being solely for energy conservation

Due to subjects concerns about privacy and competition in the workplace, the authors suggest that energy feedback programs be explicit about using personal energy use information solely for energy conservation, and not for other purposes such as monitoring employee schedules.

6.1.7. Allow occupants to collaborate with peers and have two-way communications with facility managers regarding building problems and repairs.

Both operators and energy experts found a billboard feature that allowed for reporting problems, seeing peers' problems, and operator responses, to be valuable and indicated that they would use such a feature if it were available. However such a system should be designed so that users can easily search and also filter out irrelevant information. To avoid the possibility that such a system will increase the rate of complaints, the authors suggest using an intelligent complaint reporting approach (perhaps with branching radio-button selections, for example), that inform users if a particular problem has already been reported. Such a feature would benefit by allowing facility managers to respond to complaints and to push announcements to building occupants via the application.

6.1.8. Provide specific page views and features for energy professionals and facility managers

Energy professionals and facility managers showed less interest in the highly granular energy information, and more greater interest in whole-building energy, and in the ability to easily survey building occupants on building features and improvements, with the ability to track performance over time.

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APPENDIX A: PRE-DEMO QUESTIONNAIRE

Test Subject Identification Number _____

Test Subject Identification Icon _____

1. Which of the following best characterize what you do?
   - Undergraduate student
   - Graduate student
   - Research staff
   - Administrative staff
   - Faculty
   - Other (please specify) _____
2. Do you have experience working in an office type environment? (In a company, school, university, or other similar setting.)
   - Yes
   - No
3. If you answered yes to the question above, how many years of experience working in an office environment do you have?
   - Less than one year
   - 1-3 years
   - 4-5 years
   - Over 5 years
   - No office experience
4. What is your age?
   - Under 20
   - 20-29
   - 30-39
   - 40-39
   - 50-59
   - Over 60
5. What is your gender?
   - Male
   - Female
6. Do you regularly use social media sites such as Facebook, Twitter, MySpace, LinkedIn, Flickr, etc.?
   - Yes
   - No
7. If you answered yes to question 6, how many different social networks do you follow or participate in on a regular basis (viewing a few times per month)?

   _____

8. If you answered yes to question 6, how frequently do you view or participate in social network sites?
    - Hourly
    - Several times a day
    - Several times a week
    - Several times a month
    - Less than once a month
    - I don't use these sites
9. How important are environmental issues to you personally? (For example, energy use, air pollution, recycling, climate change, etc.)

    - Very important
    - Somewhat important
    - Neutral
    - Not that important
    - Not important at all
10. How important do you think it is for individual people to change their behavior to reduce their impact on the environment? (For example, turning off lights, saving water, recycling, etc.)

    - Very important
    - Somewhat important
    - Neutral
    - Not that important
    - Not important at all
11. Which of the following best expresses your thoughts about energy conservation?
    - I don't try to conserve energy.
    - I feel that better technology is the best solution to reduce energy use.
    - I feel energy conservation is good for the environment.
    - I feel it is my duty as a socially conscious person to conserve energy.
    - My friends/neighbors conserve energy, so I do too.
    - Other (please specify)
12. What do you know about the utility costs in your home? (For example. gas, electricity, and water.)
    - I am fairly familiar with the utility costs.
    - I have a general idea about them.
    - I do not know what the costs are.
    - Other (please specify)
13. Please indicate the response that best represents your current thinking about your household energy use.
    - I don't think much about it; it is what it is.
    - It is a hassle/too hard to try to change my energy consumption.
    - I don't know how to reduce my energy consumption.
    - The potential for cost savings is not worth the effort.
    - I am already taking measures to reduce energy consumption.
    - Other (please specify)

14. Any other comments about energy use and environment?

APPENDIX B: PROTOTYPE DEMO QUESTIONNAIRE

Test Subject Identification Number _____

Icon _____

For each question below, please circle the response that best represents your answer.

Example :

Response 1	Response 2	Response 3	Response 4	Response 5

View the prototype online at http://cbe.berkeley.edu/prototype

Scenario: As you consider the questions below, imagine that you are working in an office building, and the firm management is encouraging people to use a new website to help people to conserve energy, and to manage the operation of the building.

1. Please go to "**Energy Goals"** on page (1).

   This feature (page) allows you to select personal energy goals, and to track your progress with them. You can share this information with others (either with office acquaintances, or everyone) and to see the goals and progress of other people that have shared their information.

   1.1. I would be likely to use this feature (page) to set one or more personal energy goals.

Strongly agree	Agree	Not sure	Disagree	Strongly disagree

   1.2. If I used this feature, I would be likely to share one or more of my energy goals with other people in my office.

Strongly agree	Agree	Not sure	Disagree	Would not share goals

   1.3. Sharing my personal energy goals would help me to meet the goals I set for myself.

Strongly agree	Agree	Not sure	Disagree	Would not share goals

   1.4. I would be interested in seeing the personal energy goals and progress of other people (such as office acquaintances and others).

Strongly agree	Agree	Not sure	Disagree	Would not share goals

   1.5. In general, would the "Energy goals" features of the website be useful to you for saving energy?

Very useful	Somewhat useful	Not useful	Not at all useful	No opinion

2. Please go to "**Billboard > Problems"** on page (2). This feature allows you to report a problem in your workplace, and allows the building manager to respond to it. It also allows you to ask questions related to your workplace, and to post energy saving tips.

   2.1. I would be likely to use this if I had a problem to report (for example, a problem about temperature in the office).

Strongly agree	Agree	Not sure	Disagree	Strongly disagree

   2.2. Having this tool would make me more likely to report a problem than if I had to call or use email to do so.

Strongly agree	Agree	Not sure	Disagree	Strongly disagree

   2.3. I would be likely to use this if I had a question (for example, a question about operating lights or equipment in the office).

Strongly agree	Agree	Not sure	Disagree	Strongly disagree

   2.4. I would be likely to use this if I had an energy tip to share with other people (for example, an idea about how to put your computer in sleep mode).

Strongly agree	Agree	Not sure	Disagree	Strongly disagree

   2.5. Would you be interested in seeing the problems, questions and tips posted by other people?

Problems	Questions	Tips
Very interesting	Very interesting	Very interesting
Interesting	Interesting	Interesting
Not interesting	Not interesting	Not interesting
Not at all interesting	Not at all interesting	Not at all interesting
No opinion	No opinion	No opinion

   2.6. In general, would the "billboard" features of the website be useful to you in your workplace?

Very useful	Somewhat useful	Not useful	Not at all useful	No opinion

3. Please navigate to **"Billboard >Surveys" on page (3)**. This would be used by the building manager to find out how satisfied people in the building are, and to report the results of these surveys.

   3.1. I would be likely to participate in a survey asking about the conditions in the workplace.

Strongly agree	Agree	Not sure	Disagree	Strongly disagree

   3.2. I would be likely to view the results of such a survey.

Strongly agree	Agree	Not sure	Disagree	Strongly disagree

3.3. Would you be interested in the survey feature of the website?

Very interested	Interested	Not interested	Not at all interested	No opinion

3.4. In general, how useful would the "survey" feature of this website be to you in your workplace?

Very useful	Somewhat useful	Not useful	Not at all useful	No opinion

4. Please go to "**Energy Charts > Building Energy Charts"** on pages (4-6). This shows different views of energy use in the building.

Energy Charts > Building Energy Charts on page 4 shows the energy use of the entire building.
Energy Charts > Group Energy Charts on page 5 shows the energy use by floor.
Energy Charts >My Energy Charts on page 6 shows the energy use of your personal office or workspace.

Again using the scenario outlined above, how interesting would you find this information, and how useful would it be for helping you to keep track of your energy use?

4.1. Building Energy Charts on page 4 (entire building)

Interest	Usefulness
Very interesting	Very useful
Interesting	Somewhat useful
Not interesting	Not useful
Not at all interesting	Not at all useful
No opinion	No opinion

4.2. Group Energy Charts on page 5 (energy use of your floor and that of others)

Interest	Usefulness
Very interesting	Very useful
Interesting	Somewhat useful
Not interesting	Not useful
Not at all interesting	Not at all useful
No opinion	No opinion

4.3. My Energy Charts on page 6 (this shows the energy use of your personal office or workspace)

Interest	Usefulness
Very interesting	Very useful
Interesting	Somewhat useful
Not interesting	Not useful
Not at all interesting	Not at all useful
No opinion	No opinion

4.4. Please rank these energy charts in terms of their usefulness, from 1 most useful, to 3 least useful.

Building Energy Charts, page 4	
Group Energy Charts, page 5	
My Energy Charts, page 6	

5. Please go to "**Energy Charts > My Energy Charts**" on pages (6-9).

Then review the following three pages:
Energy Charts > My Energy Charts > Power on page 6 (power use over time)
Energy Charts > My Energy Charts > Amount on page 7 (your energy use by amount)
Energy Charts > My Energy Charts > Light Bulbs on page 8 (your energy use by light bulb equivalents)
Energy Charts > My Energy Charts > Costs on page 9 (your energy use by cost)

Again using the scenario outlined above, how interesting would you find this information, and how helpful would it be for helping you to keep track of your energy use?

5.1. My Energy Charts > Power on page 6 (in watts over time)

Interest	Usefulness
Very interesting	Very useful
Interesting	Somewhat useful
Not interesting	Not useful
Not at all interesting	Not at all useful
No opinion	No opinion

5.2. My Energy Charts > Amount on page 7 (in kilowatt-hours kWh)

Interest	Usefulness
Very interesting	Very useful
Interesting	Somewhat useful
Not interesting	Not useful
Not at all interesting	Not at all useful
No opinion	No opinion

5.3. My Energy Charts > Light Bulbs on page 8 (light bulb equivalents)

Interest	Usefulness
Very interesting	Very useful
Interesting	Somewhat useful
Not interesting	Not useful

Not at all interesting	Not at all useful
No opinion	No opinion

5.4. My Energy Charts > Cost on page 9 (in $)

Interest	Usefulness
Very interesting	Very useful
Interesting	Somewhat useful
Not interesting	Not useful
Not at all interesting	Not at all useful
No opinion	No opinion

5.5. Please rank these energy charts in terms of their usefulness, from 1 most useful, to 4 least useful.

My Energy Charts > Power, on page 6	
My Energy Charts > Amount , on page 7	
My Energy Charts > Light Bulbs, on page 8	
My Energy Charts > Cost, on page 9	

6. Please go to "**Energy Charts > My Energy Charts"** on pages 9-11.

Then review the following three pages:
Energy Charts > My Energy Charts on page 9 (no comparison)
Energy Charts > My Energy Charts > Compare to average on page 10 (compare your energy use to the office average)
Energy Charts > My Energy Charts > Compare to individuals on page 11 (compare your energy use to individuals you select)

Again using the scenario outlined above, how interesting and useful would this information be to you?

6.1. *My Energy Charts* on page 9 (no comparison)

Interest	Usefulness
Very interesting	Very useful
Interesting	Somewhat useful
Not interesting	Not useful
Not at all interesting	Not at all useful
No opinion	No opinion

6.2. *My Energy Charts > Compare to average* on page 10

Interest	Usefulness
Very interesting	Very useful
Interesting	Somewhat useful
Not interesting	Not useful
Not at all interesting	Not at all useful
No opinion	No opinion

6.3. *My Energy Charts > Compare to individuals* on page 11

Interest	Usefulness
Very interesting	Very useful
Interesting	Somewhat useful
Not interesting	Not useful
Not at all interesting	Not at all useful
No opinion	No opinion

6.4. Please rank these energy charts in terms of their usefulness, from 1 most useful, to 3 least useful.

My Energy Charts (no comparison) on page 9	
My Energy Charts > Compare to average, on page 10	
My Energy Charts > Compare to individuals, page 11	

For the following questions, consider the energy charts with comparisons, pages 10-11

6.5. I would be likely to share my energy charts with other people in my office.

Strongly agree	Agree	Not sure	Disagree	Would not share charts

6.6. Sharing my personal energy charts would be useful for me in terms of saving energy.

Strongly agree	Agree	Not sure	Disagree	Would not share charts

6.7. I would be interested in seeing the personal energy charts of other people in the office.

Strongly agree	Agree	Not sure	Disagree	Would not share charts

6.8. The social aspects of the energy charts (sharing my energy use, seeing that of others) would increase my interest in energy use in my workplace.

Strongly agree	Agree	Not sure	Disagree	Would not share charts

6.9. In general, how useful would the “energy charts” features of the website be for saving energy?

Very useful	Somewhat useful	Not useful	Not at all useful	No opinion

7. Please go to “**Groups”** on page 12.

This feature allows you create or join groups of people that are interested in saving energy at work, or helping operate the building in an effective way (for example, a group of people interested in buying EnergyStar computers for the firm).

7.1. I would be likely to use this feature to join a group.

Strongly agree	Agree	Not sure	Disagree	Strongly disagree

7.2. I would be likely to start a group.

Strongly agree	Agree	Not sure	Disagree	Strongly disagree

7.3. I would be interested in this feature of the website.

Strongly agree	Agree	Not sure	Disagree	Strongly disagree

7.4. In general, would the “groups” feature of this website be useful to you in the workplace?

Very useful	Somewhat useful	Not useful	Not at all useful	No opinion

8. Please go to “**Dashboard” on page 13**.

This page is the “landing page” when you log on to the website, and gives you an overview of the various features of the site.

8.1. I would be likely to use this feature.

Strongly agree	Agree	Not sure	Disagree	Strongly disagree

8.2. In general, how useful would the “dashboard” feature of this website be to you in the workplace?

Very useful	Somewhat useful	Not useful	Not at all useful	No opinion

When you have completed this paper survey, please use the link on the laptop to go to the closing survey.

APPENDIX C: POST-DEMO QUESTIONNAIRE

Test Subject Identification Number _____

Test Subject Identification Icon _____

Header, all pages: Using the scenario we described earlier, please respond to the following questions:

1. If an application like this would be available to you at work, how likely would you be to use it ?
    - Very likely
    - Likely
    - Not sure
    - Not likely
    - Not at all likely
2. If an application like this would be available to you at work, how frequently do you think you would be to use it ?
    - Once a month or less
    - A few times a month
    - One or more time a week
    - Once a day or more
    - I do not think I would use this application
3. My experience seeing this prototype has made me more aware of energy use in an office environment.
    - Strongly agree
    - Agree
    - No opinion
    - Disagree
    - Strongly disagree
4. Having such application in my workplace would create an incentive for me to save energy.
    - Strongly agree
    - Agree
    - No opinion
    - Disagree
    - Strongly disagree
5. The social aspects of this application would increase my interest in energy use in my workplace.
    - Strongly agree
    - Agree
    - No opinion
    - Disagree
    - Strongly disagree
6. Being able to share my personal energy goals and patterns with others would create an incentive for me to save energy.
    - Strongly agree
    - Agree
    - No opinion
    - Disagree
    - Strongly disagree

7. The social aspects of the site would make me more likely to use such an application.
    - Strongly agree
    - Agree
    - No opinion
    - Disagree
    - Strongly disagree
8. The social aspects of this application would increase my interest in the operations of the building in general.
    - Strongly agree
    - Agree
    - No opinion
    - Disagree
    - Strongly disagree
9. What features do you specifically like about the application?
10. What features do you specifically dislike about the application?
11. Please list any additional features that would be useful.
12. Additional comments.

APPENDIX 3.1.1
UFAD MODELING UPGRADE SUMMARY

Underfloor Air Distribution (UFAD) Technology Development project

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Center for the Built Environment
University of California
Berkeley, California 94720-1839

September 29, 2011

OBJECTIVES

Document changes to UFAD modeling objects and simulation procedures for EnergyPlus UFAD for new modeling studies.

BACKGROUND

CBE has made many refinements in the UFAD modeling capability in EnergyPlus. Virtually all of these changes appear in the CBE development version of EPlus V6.0. A number of these changes were migrated from V2.1/UFAD. It is anticipated that NFPs will be written for those changes that can be incorporated into release version of EnergyPlus, but some of the methods discussed here require changes to the standard modeling procedures that require some sort of interface that supports these modeling methods as well elements incorporated into CBE's 3 story medium office building prototype example file.

The table below documents all changes made since the initial release in V2.1 as well as upgardes to the model files to conform to ASHRAE Standard 90.1-2010 as represented by PNNL work on a new set of prototype models.

MODEL UPGRADES

Item	Description
Building model	
Envelope, walls **Date:** 9-2011 **Who:** [TW, DF, TH, KHL] **EP:** V6.0 **Project:** Opto2 **Model**: 90.1-2010 **How**: .imf model files	**Objective**: Upgrade wall specifications to conform to 90.1- 2010 and to create a model that is as closely aligned as possible with the PNNL Prototypes. **Approach**: Change wall specs to those used in the PNNL prototypes for UFAD and OH baseline models **Implementation:** Climate based U-factors were implemented by changing the insulation level in walls and roofs for the medium office building.
Envelope, Windows **Date:** 9-2011 **Who:** [TW, DF, TH, KHL] **EP:** V6.0 **Project:** Opto2 **Model**: 90.1-2010 **How**: .imf model files	**Objective**: Upgrade wall specifications to conform to 90.1- 2010 and to create a model that is as closely aligned as possible with the PNNL Prototypes. **Approach**: Standard 90.1-2010 has dispensed with window to wall ratio (WWR) based SHGC and U-Factors for windows; now all windows with WWR from 0-40% have only climate based WWR specifications. Also, previous DOE reference building prototypes by NREL used "theoretical windows" while the new PNNL prototypes use "real" windows based on specification derived from Window 6 that represent common practice but do not meet the exact letter of the 90.1 specifications. **Implementation:** We implemented the same window specifications as used in the PNNL prototypes that use specular average properties.

Supply plenum convection coefficients **Date:** 2011 **Who:** [FB, WP] **Model**: 90.1-2010 **How:** .imf	**Objective:** Devise method to apply customized surface convection coefficients based on CBE full scale testing and CFD studies. **Approach:** Assign an average value to provide independence of plenum configuration. **Implementation:** Complete
Bottom floor ground temperatures for UFAD **Date**: 2008 **Who**: [TH, KHL] **EP**: 3.1_CBE **Model**: BERG_T-24-2005 **How**: E+ code	**Objective**: Account for impact of supply plenum on ground floor slab in CBE model **Approach**: Ground temperatures over long term are determined by the temperature above the slab; they equilibrate to a value between typical ground temperatures and the room temperature. However, for UFAD the temperature above the slab is the supply plenum temperature which is ~ 10F lower than room temperature. **Implementation**: Based on a study with the EP slab simulator wizard and sensitivity runs in EP, created ground temperature schedules for each of three CA climates.

Zone thermal mass (and solar distribution) **Date: 6-2011** **Who:[TH, TW, SS, KHL]** **EP: V6.0** **Project: Fake UFAD/T-24, Opto2** **Model: 90.1-2010 (and T-24-2008)**	**Objective:** Provide realistic interior thermal mass to account for building furnishings **Approach:** There is no study that we could find that provides a well-researched definitive specification. This is understandable due to the diversely different cases that real buildings present. However, it does appear that thermal mass has a relatively small effect on energy for the types of buildings we are modeling here. Therefore, we have adopted the approach used for Title 24-2008 (apparently derived from work that the DOE2 2.2 team has done in the past) which is to use a mass of wood at 80#/sf covering 85% of the floor area. This results in the following thermal capacities: *Wood:* $\rho = 38\ \#/ft3$ C_p = 0.38 Btu/# °F A = 1 ft3/ft2 Therefore: for 80 #/ft2 mCp = 80 #/ft2 * 0.38Btu/# °F = 30.4 Btu/ft2 °F 80#/ft2 / 38 #/ft3 = 2.1 ft thick *Concrete: 6 inches thick* $\rho = 120\ \#/ft3$ C_p = 0.22 Btu/# °F A = 1 ft3/ft2 Therefore: 120 #/ft3 * 0.5 ft3/ft2 = 60#/ft2 mCp = 60 #/ft2 * 0.22Btu/# °F = 13.2 Btu/ft2°F For comparison purposes, a 2.1 thick slab of wood has ~3x heat capacity of a 6 inch slab of concrete. **Implementation:** *Internal mass:* The EnergyPlus internal mass object is utilized without doubling of surfaces. This may not be completely representative of real furniture since the sides may not be fully represented. However, this may be conservative enough since the heat transfer (radiant exchange) occurs only at the top (inside) surface; in real system the sides would interact in other ways in the radiant exchange than the top. Furthermore, sensitivity studies can be done by changing the area ratio. CONSTRUCTION, **4** Light Furnishings, !- Name G05 25mm wood; !- Outside Layer

<table>
<tr>
<td>Radiant fractions, internal loads</td>
<td>
Objective: Specify accurate radiant fractions for people, lighting, and equipment

Approach: The DOE and PNNL prototypes use unrealistic radiant fractions for people and lighting, for the latter the values developed by Fisher and specified in the ASHRAE HOF are not used.

Implementation:
Radiant fractions: We continue to use the radiant fractions that we have been using in the CBE model as follows:
<table>
<tr><th></th><th>CBE</th><th>Prototypes</th></tr>
<tr><td>Lights, Return air fraction</td><td>0.0</td><td>0.0</td></tr>
<tr><td>Lights, Radiant fraction (LW)</td><td>0.32</td><td>0.7</td></tr>
<tr><td>Lights, Visible fraction (SW)</td><td>0.24</td><td>0.2</td></tr>
<tr><td>Occupant radiant fraction</td><td>0.6</td><td>0.3</td></tr>
</table>

Internal loads: We have upgraded our baseline specifications for occupancy, equipment, lighting to conform to 90.1-2010 and the prototypes.
</td>
</tr>
</table>

HVAC	
HVAC system type **Date**: 9-2011 **Who:** [WP, FB, KHL] **EP:** V6.0 **Project:** Opto2 **Model**: 90.1-2010 **How**: .imf model files	**Objective:** Determine system type for CBE model considering ease of simulation and adherence to 90.1-2010 **Approach**: Standard 90.1 specifics a DX VAV packaged unit system for medium office buildings, with one system per floor. However, the CBE model allows for scaling floor plate sizes so that larger buildings with 3 floors can be simulated. Also, to facilitate comparison analysis, a chilled water system appears easier to implement. **Implementation**: Due to considerations above and simplify model changes and .imf complexity, debugging ease, and comparison analysis, use a chilled water system as before.
New Clo model **Date**: 6-2011 **Who:** [SS, KHL] **EP: V6.0** **Project:** Opto2 **Model**: 90.1-2010 **How**: EMS	**Objective**: More accurate comfort modeling **Approach**: Use clo value schedules based on ASHRAE studies that vary clo value with outdoor temperature **Implementation**: Add EMS function to calculate clo value from new outdoor temperature algorithm.
New Gamma Phi **Date**: 9-2011 **Who:**[TH, TW, SS] **EP: V6.0** **Project:** Opto2 **Model**: 90.1-2010 **How**: E+Code,	**Objective**: Refine perimeter zone model based on recent Walnut testing results with linear bar grille and MIT2 testing. **Approach**: Develop new gamma-phi correlations for both Walnut and Titus linear bar grilles (only). Walnut diffusers are not typically used as a grille only; they are used with PWM modulation like the MIT2 boxes. **Implementation**: Modify code and interface to support the following (see sizing below and a full description of the diffuser parameters and gamma-phi profiles in the CBE UFAD Simulations Guide): • LIN1 diffuser: Old York/JCI testing based. This was found to have a disconnect between the area used and the

CBE interface	design volumes used for sizing; a 1.5 ft long diffuser was the default area, while the design volume was for a 3 ft long diffuser. Also, the York/JCI testing was done with both interior and perimeter diffusers which represents just one of the possible cases. New areas are TBD based on LBL testing… • LIN2 diffuser: Use average of Walnut and Titus results for correlation. They are very close to one another since their designs are not very different. The default values in code for the area are based on an assumed length of 3 ft even though the test unit was 6 ft long. The design volume values in the interface must correspond. Since the LIN2 Walnut diffuser gamma-phi curve is not very different from the MIT2 curve and the operation in a Walnut system results in a constant phi solution, we use the MIT2 for all York/JCI simulations. • MIT2: Use new correlations for refining design point in new design procedure. A first estimate is made for the first design day run, but the gamma-phi curve is accessed for the second pass DD which yields a design phi that is subsequently used in the annual run as a fixed phi.
New supply plenum configurations **Date**: 5-9-2011 **Who:**[KHL, TW] **EP: V6.0** **Project:** Opto2 **Model**: 90.1-2010 **How**: .imf macro code, CBE interface	**Objective**: Offer option for alternatives to standard series and parallel plenums used before. **Approach**: Many UFAD systems use an open plenum design as opposed to a divided plenum. Field studies show that the average plenum temperature in the interior and perimeter are nearly equal despite large variances in temperatures. A single plenum is more reflective of the geometry of these systems, but it may not exactly reflect the many different ways air is distributed in these plenums. Furthermore, newer idealized alternative configuration simulations capability offer an opportunity to bracket the range of possibilities within which many actual designs fall. The following options have been implemented: • Single plenum: Entire plenum is mixed so both perimeter and interior zones see the same mixed air temperature that results from heat gain from both interior and perimeter zones. Should represent average conditions for many plenum distributions schemes used in practice. • Parallel: In this option AHU air is delivered in parallel to each of the interior and perimeter plenum zones so they all see the same supply temperature. However, the thermal decay in each plenum will be different based on their airflow heat transfer to the plenum. In the past this option did not show much difference from the series plenum option. However, we now (9-2011) that the convection coefficients were not calculated correctly for this

configuration. This is not considered to have a large impact.

- Ducted: Ducted to all zones. Represents an idealized case without thermal decay. However, this not considered a practical case since few actual systems would duct all the diffusers.

- Reverse series: AHU airflow is ducted to the perimeter zones first, and then flows back to the interior zone. Requires a new object that Dan Fisher developed that accounts for the nodal coalescing (mixing) of perimeter and interior airflows by correcting the flows and temperatures in a dual pass algorithm. The interior zone airflow receives air from the perimeter zones that has been heated by thermal decay, thus reducing the chances that the zone will be overcooled. This represents a system using fabric (or other) ducts. It also could be used to represent an open plenum configuration where the plenum inlets (HVAC shafts) are located in the perimeter zones (in this case the supply fan design static pressure would be decreased from that of a ducted version). The fundamental difference between this object and others currently available in EPlus is the ability to mix. Airflow paths are dictated in the AirLoop objects in EnergyPlus. This includes zones, terminal equipment and splitters and mixers. Currently (2011) only splitters are allowed for supply plenums and mixers for return plenums; therefore a new object is needed to allow mixing in supply plenums (the core plenum) to make reverse series work. Convection coefficients are derived from the entire airflow thru the entire plenum as is the case for series plenum.

Implementation: All of these options have been implemented in the .imf model files where the flow paths are modified and other details as follows are handled:

- Parallel: still some concern about how to compute convection coefficients correctly

- Ducted to perimeter: Two models have been implemented. In both versions for the VSFCU the FSP is set to zero to simulate a VAV box without having to change the .imf files to insert another object for this case. However, in these cases the dual minimums used for VSFCU must be set to the same value like a regular VAVbox has. This needs to be done during run setup with the interface.

    - Case #1, DuctedPerimeterSingle = A single plenum where the interior zone air from the AHU goes gains heat from the entire single plenum while the perimeter terminal unit receives AHU in parallel. . Convections coefficients….TBD

    - Case #2, DuctPerimeterParllel = Perimeter zone plenums are dead space. Interior airflow receives

	thermal decay form interior zone only. Perimeter terminal units receive AHU air in parallel with interior zone as in case #1 above. • Reverse series: A new object is being crafted by OSU to allow for splitting and mixing in the supply plenum objects.
VSFC unit **Date**: 2010 **Who:**[KHL, TW, TH] **EP: V6.0** **Project:** Opto1, **Model**: 90.1-2004, T-24-2005 **How**: CBE E+ code, interface	**Objective:** Update VSFCU to reflect current practice for minimum airflows and VAV heating. **Approach**: Modify code to add features and improve modeling accuracy. **Implementation:** *Off in Deadband***:** The original implementation did not allow for the VSFCU unit to be turned off in the deadband which is common practice. This violates the letter of 90.1 and 62.1, but leakage flows help mitigate this issue; off in DB would save substantial fan energy. *Dual minimums:* Dual minimums are required to allow the operation of these units either in on/off mode and capture the effect of leakage flows with the unit is off. We have modified our CBE code implementation to all these minimums to be set by the user. This will allow the VSCFU model to be used to in DuctedPerimeter plenum configurations to simulate a VAV box (requires that fan pressure be set to zero). *VAV heating:* The current (v6.0) VAV/reheat models in EnergyPlus do not correctly model VAV heating or implement the dual maximum strategy properly, as follows: • Heating maximum air volume – The heating coil design capacity uses the maximum cooling airflow to size the heating coil water flow. This result in abnormally high discharge temperatures under part load conditions and unrealistically allows VAV/reheat units to accommodate loads in cold climates which is not achievable in practice. • Heating discharge temperature –With the above correction, and proper specification of the design discharge temperature the coil water flow will be properly designed for peak load conditions but because the part load air volume is determined with maximum coil water flow two consequences occur: (1) the discharge temperature increases as load decreases, and (2) the part load airflow will be minimized (i.e., lower than it should be if discharge air temperature is controlled) and therefore will not properly account for reheat. We have (temporarily until forthcoming permanent changes are made to these objects) corrected the model to size the coil air volume based on zone design heating load and then the water flow based on this airflow. This work around also limits the maximum heating volume based on cooling design airflow to account for cold vs warm climates. This is done

	to limit problems of discharge air temperature described above. However, in very cold climates the volume limit may result in loads not met, which if sever enough would require a change to a parallel fan powered box, as is normally done in practice. [It is likely that this is the reason that VAV/reheat boxes appear to work for all climates in the DOE Benchmarks].

<table>
<tr>
<td>Return air bypass

Date: 2019?
Who:[KHL, TW]
EP: V3.1
Project: BERG, Opto1,
Model: 90.1-2004, T-24-2005
How: E+code</td>
<td>Objective: Fix return air bypass (RAB) in humid climates (no economizer) and Return air bypass for non-humid or open economizer operation

Approach: Return air bypass – The return air bypass routine only works when the economizer is off, i.e., closed to minimum. This restricts its ability to provide SAT discharge control without a central a reheating coil to control humidity (i.e., cooling coil discharge low, but AHU SAT high) when economizer is open and OSA is less than AHU SAT setpoint. However, where this is needed most is in very humid climates where (as specified in ASHRAE 90.1) there is no requirement for an economizer; where RAB is most likely to be needed.

Implementation: We have input from design practitioners as to how they implement more general strategies, but changes have yet to be made to improve this object, deferred to later.</td>
</tr>
<tr>
<td>Supply fan parameters

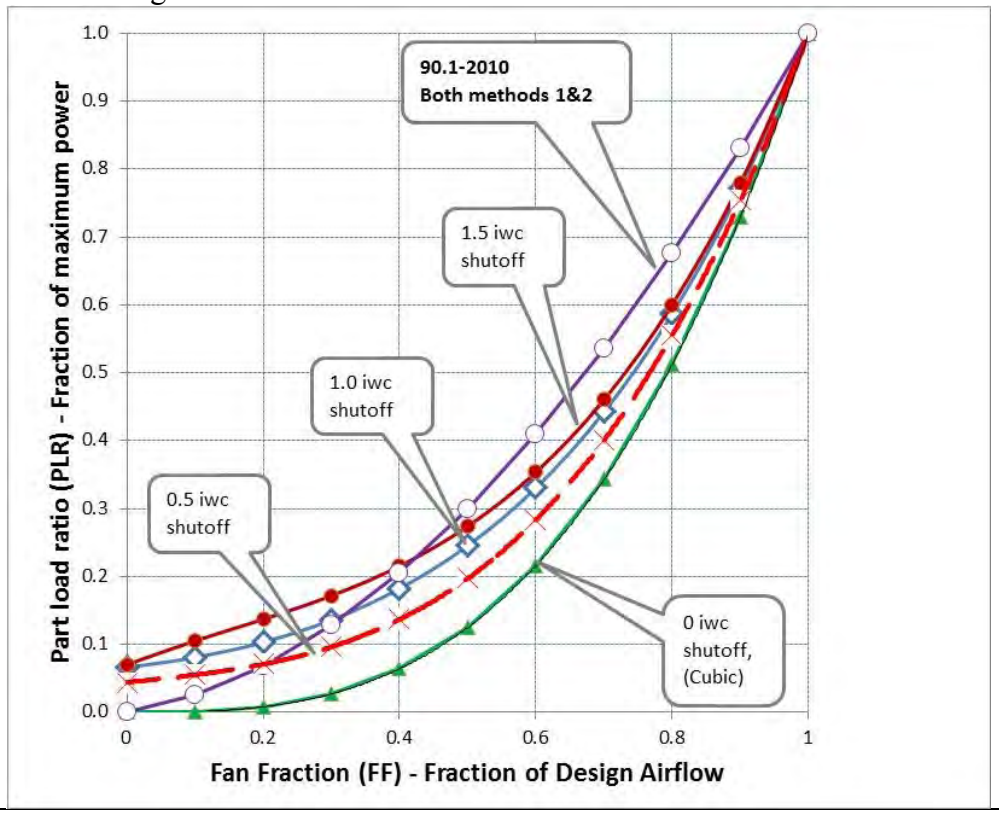
Date: 9-2011
Who: [DF, TW]
EP: V6.0
Project: Opto2,
Model: 90.1-2010
How: .imf, CBE interface</td>
<td>Approach: Provide revised inputs for part load curves and FSP that reflects different plenum configurations. Although PNNL uses the 90.1 fan power limitation specifications, we use a slightly modified version based on our own pressure drop analysis as shown in the CBE UFAD Modeling Guide.

Implementation::
Fan design static pressure: Fan pressures are determined by the plenum configuration as shown in the table below
<table>
<tr><th>Plenum config</th><th>FSP design</th></tr>
<tr><td>OH</td><td>4.3</td></tr>
<tr><td>UFAD – single (open plenum)</td><td>2.8</td></tr>
<tr><td>UFAD - series</td><td>2.8</td></tr>
<tr><td>UFAD – reverse series</td><td>3.1 (add ~0.25 for fabric duct)</td></tr>
<tr><td>UFAD – ducted perimeter1</td><td>4.3</td></tr>
<tr><td>UFAD – ducted perimeter2</td><td>4.3</td></tr>
<tr><td>UFAD - Ducted</td><td>4.3</td></tr>
</table>

Fan Efficiency: Fan efficiency inputs are as shown in the following table based on PNNL/90.1 assumptions for fan efficiency used in the fan power limits.</td>
</tr>
</table>

	Individual	Combined
Fan static efficiency	65%	65%
Motor efficiency	90%	84%
Belt drive efficiency	95%	
VFD efficiency (100%)	98%	
Total Efficiency	54%	54%

Fan part load curves: PNNL uses 90.1-2010 fan throttling curves but these are significantly different than ones we are using from Taylor Engineering and does not allow simulation of static pressure reset adequately as shown in figure below.



Chilled water system specifications **Date**: 9-2011 **Who:** [DF, TH, TW] **EP: V6.0** **Project:** Opto2, **Model**: 90.1-2010 **How**: .imf, CBE interface	**Objective:** Bring chiller modeling specs in line with 90.1-2010 **Approach**: Revise reset schedules and chiller specs **Implementation**: *Chilled water reset*: Revise the reset schedule for chilled water to conform to 90.1 **(G3.1.3.9)** as follows: 44°F at 80°F and above, 54°F (was 50 in CBE prototype) at 60°F and below, and ramped linearly between 44°F and 54°F at temperatures between 80°F and 60°F. *Chiller type and COP*: For small building simulations we will change our chiller specifications to conform to those in the EnergyPlus chiller library under reformulated chillers. One chiller will be used for small (e.g., 20,000 sf) floorplates, and two for large (e.g., 60,000 sf). Use the following chiller type with its associated part load coefficients. ! ReformEIRChiller Trane RTHB 531kW/4.83COP/Valve Screw 531 (151) 4.83 Slide Valve *Pump, cooling tower:* Keep same specifications as CBE models
Outside airflow and VAV box Minimum airflow **Date**: 2011 **Who:**[AD, TW] **EP: V6.0** **Project:** Opto 2 **Model**: 90.1-2010 How: .imf, CBE interface	**Objective:** Revise minimum airflows at vav boxes to more accurately reflect current standards but with ease of research studies in mind. **Approach**: For OH systems, Standard 90.1 **(G3.1.3.13)** specifies VAV box minimums be set at 20% (per addendum h_901.-2007) or the minimum required by standard 62.1 procedures, whichever is larger. For UFAD systems there are other considerations: For VSFCU systems with ECM motors, the minimums are derived from the fact that they only can turn down to about 12% of design flow due to motor speed limitations. CBE uses a strategy where these units can be turned off in the deadband (with is commonly used in practice). However, there is leakage flow through these units of about 6% when the unit is off. This calls for a dual minimum strategy which CBE has devised.

	York/JCI systems with MIT2 diffusers have leakage flows of 10% when the damper is closed. Thus these systems have minimums of at least 10%. Advanced design (e.g., Taylor Engineering) research has shown that minimums below 20% can save energy and do not affect occupant comfort. But they would appear to violate the Std 62.1 and/or 90.1 standards. Standard 62.1 may require higher (critical zone) minimums to support OSA ventilation reset strategies which may lead to higher minimums in some zones. However, other zones may actually require lower minimums than 20% for ventilation purposes (e.g., for 0.15 cfm/sf ventilation the peak airflow would only have to be 0.75 cfm/sf to achieve 20% minimum; therefore a typical 1.0 cfm/sf with 20% minimum will be over aired for ventilation) zones, but standard 90.1 would restrict the zones to 20% it appears. PNNL has implemented this strategy using EnergyPlus objects, but this may create some inconsistencies between models used for comparison. **Implementation**: To simplify simulation studies and reduce the impact if variable rates on sensitivity studies, we will implement a fixed minimum strategy: apply system ventilation efficiency of 0.6 to total building level zone OSA requirements for offices of 0.085 cfm/sf; with the vent eff, this yields ~0.15 cfm/sf. Use VAV box minimums of 20%, which will lead to conservative ventilation except possibly in the interior zone.
Sizing procedures **Date**: 2011 **Who:**[TH, TW] **EP: V6.0** **Project:** Opto 2 **Model**: 90.1-2010	**Objective:** Implement new sizing procedures for terminal unit sizing and number of diffusers that fully account for zone design day variations, thermal decay, and updated diffuser specifications. **Approach**: In progress **Implementation**: See CBE UFAD Simulations Guide
SAT reset **Date**: 2011 **Who:**[AD, TW] **EP: V6.0** **Project:** Opto 2	**Objective:** Identify SAT simulation strategy for OH and UFAD **Approach**: It is unclear as to how effective SAT reset is in overall energy savings, especially load based. The implementation of it appears to be sensitive to VAV box sizing; too large a box drives the SAT to be too high, compromising fan energy; while too small a box drives it to be too low, compromising cooling energy. Also, in real systems it is difficult to implement both SAT and SAP reset reliably. The sizing for OH VAV/Reheat

Model: 90.1-2010 **How**: .imf??	boxes seems to work well. Recently, PNNL has conducted a study that shows saving associated with OSA based SAT reset; in most climates it is worth considering since it is relatively easy and reliable to implement, although the savings are small. UFAD systems on the other hand have special issues to consider. (1) an explicit determination of the thermal decay is needed to properly specify the design inlet air temperature; this has been handled in the CBE sizing procedures by a thermal decay sizing procedure that obviates the need for an explicit design SAT. (2) Since load based sizing is sensitive to box sizing, careful sizing is required; this may be handled by new CBE sizing procedures, that consider thermal decay and exposure based design days. However, it is still unclear as to the savings. On the other hand, OSA based reset may be easier to implement and more reliable. (3) SAT reset may provide extra benefit on comfort in the interior zones, and less reheat in perimeter zones during the winter. (4) Standard 90.1 mandates SAT reset. **Implementation**: the reset schedule will be different for OH and UFAD. In addition, sizing issues must be considered. Reset schedules as follows: • **OH:** SAT = 55 @ OSA >= 70F; SAT = 60 @ OSA <= 50F • **UFAD:** SAT = 60 @ OSA >= 70F; SAT = 65 @ OSA <= 50F

ASHRAE 90.1 Prototype Building Modeling Specificatic

Pacific Northwest National Laboratory, updated on 04-30-2011

	Item	ASHRAE 90.1 Prototype_Medium Office	Simbuild 2010 Paper_SAT study	Optimization study_OPT1 CBE Study	New CBE model -Optimization Study_Opto2	CBE model - ToolKit	BERG model	eQUEST Study
Simulation	Information							
	Standard	PNNL 90.1_2010 scorecard	90.1-2004	90.1-2004	90.1-2010	T-24_2005	T-24_2005	T-24_2008
	Time	N/A	2010	April-June_2010	on-going (Summer 2011)	Aug-Oct_2009		
	EnergyPlus Version	V-6.0	V-3.1	V-3.1	V-6.0	V-3.1	V-3.1	V-6.0
Program								
	Vintage	NEW CONSTRUCTION	NEW CONSTRUCTION	NEW CONSTRUCTION	NEW CONSTRUCTION	NEW CONSTRUCTION	NEW CONSTRUCTION	NEW CONSTRUCTION
	Location	Zone 1A: Miami (very hot, humid) Zone 1B: Riyadh, Saudi Arabia (very hot, dry) Zone 2A: Houston (hot, humid) Zone 2B: Phoenix (hot, dry) Zone 4A: Baltimore (mild, humid) Zone 4B: Albuquerque (mild, dry) Zone 4C: Salem (mild, marine) Zone 5A: Chicago (cold, humid) Zone 5B: Zone 6A: Burlington (cold, humid) Zone 6B: Helena (cold, dry) Zone 7: Duluth (very cold) Zone 8: Fairbanks (subarctic)	CA_San Francisco, CA_Sacramento	Zone 2A: Houston (hot, humid) Zone 2B: Phoenix (hot, dry) Zone 3B: Sacramento (warm, dry) Zone 3C: San Francisco (warm,marine) Zone 4A: Baltimore (mild, humid) Zone 5A: Chicago (cold, humid); Zone 6A: Burlington (cold, humid);	Zone 2A: Houston (hot, humid) Zone 2B: Phoenix (hot, dry) Zone 3B: Sacramento (warm, dry) Zone 3C: San Francisco (warm,marine) Zone 4A: Baltimore (mild, humid) Zone 5A: Chicago (cold, humid); Zone 6A: Burlington (cold, humid);	CA_Los Angeles, CA_San Francisco, CA_Sacramento	CA_Los Angeles, CA_San Francisco, CA_Sacramento	CA_Sacramento
	Available fuel types	gas, electricity	gas, electricity	gas, electricity	gas, electricity	gas, electricity	gas, electricity	gas, electricity
	Building Type (Principal	OFFICE	OFFICE	OFFICE	OFFICE	OFFICE	OFFICE	OFFICE
	Building Prototype	Medium Office	Medium Office	Medium Office	Medium Office	Medium Office	Large Office	Medium Office
Form								
	Total Floor Area (sq feet)	53,600 (163.8 ft x 109.2 ft)	60,000	53,600 (163.8 ft x 109.2 ft)	60,000	60,000	60,000*3 (173.2 ft x 115.5 ft)	60,000 (173.2 ft x 115.5 ft)
	Aspect Ratio	1.5	1.5	1.5	1.5	1.5	1.5	1.5
	Number of Floors	3	3	3	3	3	3	3
	Window Fraction (Window-to-Wall Ratio)	33% (Window Dimensions: 163.8 ft x 4.29 ft on the long side of facade 109.2 ft x 4.29 ft on the short side of	40%	33% (Window Dimensions: 163.8 ft x 4.29 ft on the long side of facade 109.2 ft x 4.29 ft on the short side of the façade)	40%	40%	20%;40%;60%	40%
	Window Locations	even distribution among all four sides	even distribution among all four sides	even distribution among all four sides	even distribution among all four sides	even distribution among all four sides	even distribution among all four sides	even distribution among all four sides
	Shading Geometry	none	none	none	none	none	none	none
	Azimuth	non-directional	non-directional	non-directional	non-directional	non-directional	non-directional	non-directional
	Thermal Zoning	Perimeter zone depth: 15 ft. Each floor has four perimeter zones and one core zone. Percentages of floor area: Perimeter 40%, Core 60%	Perimeter zone depth: 15 ft. Each floor has four perimeter zones and one core zone. Percentages of floor area: Perimeter 39.6%, Interior 44.9%, Service Core	Perimeter zone depth: 15 ft. Each floor has four perimeter zones and one core zone. Percentages of floor area: Perimeter 38.8%, Interior 61.2%,	Perimeter zone depth: 15 ft. Each floor has four perimeter zones and one core zone. Percentages of floor area: Perimeter 38.8%, Interior 61.2%,	Perimeter zone depth: 15 ft. Each floor has four perimeter zones and one core zone. Percentages of floor area: Perimeter 39.6%, Interior 44.9%, Service Core 15.5%	Perimeter zone depth: 15 ft. Each floor has four perimeter zones and one core zone. Percentages of floor area: Perimeter 39.6%, Interior 44.9%, Service Core 15.5%	Perimeter zone depth: 15 ft. Each floor has four perimeter zones and one core zone. Percentages of floor area: Perimeter 38.8%, Interior 61.2%,
	Floor to floor height (feet)	13	13	13	13	13	13	13
	Floor to ceiling height (feet)	9 (4 ft above-ceiling plenum)	9 (4 ft above-ceiling plenum)	9 (4 ft above-ceiling plenum)	9 (4 ft above-ceiling plenum)	9 (4 ft above-ceiling plenum)	9 .75 (3.25 ft above-ceiling plenum)	9 (4 ft above-ceiling plenum)
Architecture								
	Exterior walls							
	Construction	Steel-Frame Walls (2X4 16IN OC) 0.4 in. Stucco+5/8 in. gypsum board + wall Insulation+5/8 in.	1" stone cladding; Wall Insulation; 4" Vertical Air Layer; 5/8" gypsum or plaster board	Steel-Frame Walls (2X4 16IN OC) 0.4 in. Stucco+5/8 in. gypsum board + wall Insulation+5/8 in. See under **CBE_90.1-2010 envelope_Win**	Steel-Frame Walls (2X4 16IN OC) 0.4 in. Stucco+5/8 in. gypsum board + wall Insulation+5/8 in. See under **CBE_90.1-2010 envelope_Win**	1" stone cladding; Wall Insulation; 4" Vertical Air Layer; 5/8" gypsum or plaster board	1" stone cladding; Wall Insulation; 4" Vertical Air Layer; 5/8" gypsum or plaster board	Metal frame building Stone_0.0254m+Insulation_Wall+AirLayer _Vertical+Gypsum_.0128m
	U-factor (Btu / h * ft^2 * °F) and/or R-value (h * ft^2 * °F / Btu)	ASHRAE 90.1 Requirements Nonresidential; Walls, Above-Grade, Steel-Framed	U-0.139; U-0.224, R-2.930	U-0.124; U-0.084; U-0.064	U-0.124; U-0.084; U-0.064	U-0.224, R-2.930; U-0.224, R-2.930; U-0.217, R-3.074	U-0.224, R-2.930; U-0.224, R-2.930; U-0.217, R-3.074	CA_Sacramento (0.061)
	Roof							
	Construction	Built-up Roof: Roof membrane+Roof insulation+metal decking	3/8" built-up roof; Roof Insulation; 8" 140lb concrete;	Built-up Roof: Roof membrane+Roof insulation+metal decking See under **CBE_90.1-2010 envelope_Win**	Built-up Roof: Roof membrane+Roof insulation+metal decking See under **CBE_90.1-2010 envelope_Win**	3/8" built-up roof; Roof Insulation; 8" 140lb concrete;	3/8" built-up roof; 4" 80lb lightweight concrete; Roof Insulation; 4" Horizontal Air Layer; 1/2" acoustic tile	BltupRoofing_0375in+Insulation_Roof+ Concrete_LW80lb_6in;
	U-factor (Btu / h * ft^2 * °F) and/or R-value (h * ft^2 * °F / Btu)	ASHRAE 90.1 Requirements Nonresidential; Roofs, Insulation entirely above deck	U-0.067; U-0.224, R-2.930	U-0.055; U-0.055; Zone 4,5_U:0.055; Zone6_U:0.049	U-0.055; U-0.055; Zone 4,5_U:0.055; Zone6_U:0.049	U-0.076, R-9.102; U-0.051, R-15.494; U-0.051, R-15.494	U-0.076, R-9.102; U-0.051, R-15.494; U-0.051, R-15.494	U-0.065
	Window							

	Glass-Type and frame	Hypothetical window with the exact U-factor and SHGC shown below? (this is not true, verified in the IDF, they use real window, but the U factor and SHGC number cannot match up	Under tab ***BERG Model T_24_2005_Window***		Real window configuration to approximate U-factor and SHGC shown below See under **CBE_90.1-2010 envelope_Win**	Real window configuration to approximate U-factor and SHGC shown below See under **CBE_90.1-2010 envelope_Win**	Under tab ***BERG Model T_24_2005_Window***			Under tab ***BERG Model T_24_2005_Window***			North: 6196+air12.6+103+Al No Break; (Window 5) Non-North: 5286+air7.6+103+Al No Break; (Window 5)
	U-factor (Btu / h * ft^2 * °F)	ASHRAE 90.1 Requirements Nonresidential; Vertical Glazing, 31.1-40%, U_fixed	0.291	U-0.224, R-2.930	Nonresidential; Vertical Glazing, 31.1-40%, U_fixed See under **CBE_90.1-2010 envelope_Win**	Nonresidential; Vertical Glazing, 31.1-40%, U_fixed See under **CBE_90.1-2010 envelope_Win**	0.77 (Title 24) Non-North 0.486, North 0.500 (Window 5)	0.77 (Title 24) Non-North 0.485, North 0.500 (Window 5)	0.47 (Title 24) 0.291, (Window 5)	0.77 (Title 24) Non-North 0.486, North 0.500 (Window 5)	0.77 (Title 24) Non-North 0.485, North 0.500 (Window 5)	0.47 (Title 24) Non-North 0.391, North 0.391 (Window 5)	North: 0.473 (T24-requirement: 0.47) Non-North: 0.47 (T24-requirement: 0.47)
	SHGC (all)		0.389	U-0.224, R-2.930			Non-North 0.34, North 0.61 (Title 24) Non-North 0.314, North 0.553	Non-North 0.41, North 0.61 (Title 24) Non-North 0.390, North 0.553 (Window	Non-North 0.31, North 0.47 (Title 24) Non-North 0.302, North 0.302	Non-North 0.34, North 0.61 (Title 24) Non-North 0.314, North 0.553 (Window 5)	Non-North 0.41, North 0.61 (Title 24) Non-North 0.390, North 0.553 (Window 5)	Non-North 0.31, North 0.47 (Title 24) Non-North 0.291, North 0.369 (Window 5)	North_0.37 (T24-requirement: 0.47) Nonnorth_0.297 (T24-requirement: 0.31)
	Visible transmittance	Hypothetical window with the exact U-factor and SHGC shown above	0.709	U-0.224, R-2.930	See under **CBE_90.1-2010 envelop**	See under **CBE_90.1-2010 envelop**							
	Operable area	0	0		0	0	0			0			0
	Interior Partitions												
	Construction	2 x 4 uninsulated stud wall	5/8" gypsum board+airlayer+5/8"		2 x 4 uninsulated stud wall	Air_Wall:MATERIAL:NoMass (do we want to keep	5/8" gypsum board			5/8" gypsum board			Air wall with U value 1.0 Btu / h * ft2 * °F
	Internal Mass	6 inches standard wood (16.6 lb/ft²)	NO		6 inches standard wood (16.6 lb/ft²)	? (how to model this?)	NO			?			G05 25mm wood cover 85% of the floor area (= 80 lb/ft²)
	internal mass vs. floor												
	Foundation												
	Foundation Type	Slab-on-grade floors (unheated)	Slab-on-grade		Slab-on-grade	Slab-on-grade	Slab-on-grade			Slab-on-grade			Slab-on-grade
	Construction	8" concrete slab poured directly on to the earth	fictitious insulation layer, soil, 8" concrete slab, 140 lbs heavy-weight undried aggregate,		140 lbs heavy-weight undried aggregate, carpet	fictitious insulation layer, soil, 8" concrete slab, 140 lbs heavy-weight undried aggregate, carpet	fictitious insulation layer, soil, 8" concrete slab, 140 lbs heavy-weight undried aggregate, carpet			fictitious insulation layer, soil, 8" concrete slab, 140 lbs heavy-weight undried aggregate, carpet			fictitious insulation layer, soil, 8" concrete slab,
	Thermal properties Perimeter Conduction Factor F-2 (Btu / h * ft * °F)	ASHRAE 90.1 Requirements Nonresidential; Slab-on-Grade Floors, unheated	U-0.379	U-0.051, R-15.494	R-5, F-2 = 0.54	?	U-0.076, R-9.102	U-0.051, R-15.494	U-0.379	R-5, F-2 = 0.54 (R-value from Title 24 2005 Building Energy Efficiency Standards, Table 118-B: Slab Insulation Requirements for Heated Slab-on-grade)			R-5, F-2 = 0.54
	Air Barrier System												
	Infiltration	Peak: 0.2016 cfm/sf of above grade exterior wall surface area (when fans turn off) Off Peak: 25% of peak infiltration rate (when fans turn on)	?			Peak: 0.112 cfm/sf of above grade exterior wall surface area (when fans turn off) Off Peak: 25% of peak infiltration rate (when fans turn on) (note: use DOE-2 method: coefficients is different from old CBE model)	----------			Peak: 0.3 Air-Change/Hour (When fan is on) Off peak:0% of peak rate (When fan turns off)			Peak: 0.038 cfm/sf of above grade exterior wall surface area (when fans turn off) Off Peak: 0% of peak infiltration rate (when fans turn on)
HVAC													
	Heating												
	Type	Gas furnace inside the packaged air	Hot water boiler		Hot water boilers	Hot water boiler	Hot water boiler			Hot water boiler			Forced draft boilers
	Efficiency, nominal	Packaged air conditioning unit	0.8		0.8	0.8	0.8			0.8			0.78
	Part load curve					yes							yes
	Staging					50%							50%
	Cooling												
	Type	Gas furnace inside the packaged air	Electric chiller		Electric chiller	Electric chiller	Electric chiller			Hermetic centrifugal chille			Centrifugal Chiller
	COP	Packaged air conditioning unit	5.26		?	5.26	5.26			5			5.5
	Staging												90%
	Distribution and terminal												
	Terminal type	Gas furnace inside the packaged air	VAV Box with Reheat (Interior and		VAV Box with Reheat (Interior and perimeter)	VAV Box with Reheat (Interior and perimeter)	VAV Box with Reheat (Interior and perimeter)			VAV Box with Reheat (Interior and perimeter)			VAV terminal box with damper and
	Sizing									Zone Cooling Design SAT= AHU SAT+1.8F			
	Minimum air flow		0.15 cfm/sf		Packaged air conditioning unit	?	0.15 cfm/sf			15CFM/Person			20%
	HVAC												
	Outdoor, Htg/Clg temps												WME/0.5%
	Air Conditioning design	autosized to design day	autosized to design day		autosized to design day	autosized to design day	autosized to design day						90% of peak
	Heating design load	autosized to design day	autosized to design day		autosized to design day	autosized to design day	autosized to design day						Ltg = std sched, other loads = 0% peak
	HVAC												
	Thermostat Setpoint	75°F Cooling/70°F Heating	75°F Cooling/70°F Heating		75°F Cooling/70°F Heating	75°F Cooling/70°F Heating	75°F Cooling/70°F Heating			75°F Cooling/70°F Heating			73°F Cooling/70°F Heating (See sheet T-24 2008 schedule)
	Thermostat Setback	80°F Cooling/60°F Heating	90°F Cooling/55°F Heating		80°F Cooling/60°F Heating	80°F Cooling/60°F Heating	90°F Cooling/55°F Heating			90°F Cooling/55°F Heating			77°F Cooling/65°F Heating (See sheet T-24 2008 schedule)
	Supply air temperature	Maximum 104F, Minimum 55F	57F		Maximum 104F, Minimum 55F	Reset!!	57F			57F			55F
	Chilled water supply	NA	45F		NA	45F (reset?)	CWT:45/OAT:80F-CWT50/OAT:60F			45F			44/80-54/60
	Hot water supply	NA	176		NA	176F(reset?)	HWT:150/OAT:50F-HWT:180/OAT:20F			176 F			20/180-150/50
	Economizers	Various by climate location and cooling capacity Control type: differential dry bulb	DifferentialDryBulb		Various by climate location and cooling capacity Control type: differential dry bulb	DifferentialDryBulb	DifferentialDryBulb			yes			Differential dry bulb

	AHU min OSA	ASHRAE Ventilation Standard 62.1 See under **90.1 Outdoor Air**.	0.15 cfm/sf	ASHRAE Ventilation Standard 62.1 See under **Outdoor Air**.	?	0.15 cfm/sf	0.15cfm/sf	0.15 cfm/sf
	Zone min OSA	ASHRAE 90.1 Requirements	No	ASHRAE 90.1 Requirements	?	No	No	15 cfm/person
	Energy Recovery	ASHRAE 90.1 Requirements	No	ASHRAE 90.1 Requirements	No	No	No	No
	Supply Fan							
	Fan schedules	See under 90.1 **Schedules**	See under **Schedule summary_90.1**	See under **Schedules**	See under 90.1 Schedules	See under **Schedule summary_LBL**	See under **BERG T-24 2005 Schedules**	See under **T-24 2008 schedule**
	Supply Fan Total Efficiency (%)	60% to 62% depending on the fan motor size	63.00%	60% to 62% depending on the fan motor size	60% to 62% depending on the fan motor size	63.00%	75.00%	63.00%
	Supply Fan Pressure Drop	Various depending on the fan supply air cfm	3 in.wc	Various depending on the fan supply air cfm	?	3 in.wc	4.5 iwc	4.5
	Pump							
	Pump Power	autosized	HW pump_autosized; CW pump_manual	autosized	Power: autosized; Maxflow:HW_autosized; CW_manual	HW pump_autosized; CW pump_manual	?	autosized
	Cooling							
	Cooling Tower Type	NA	Two Speed	NA	TwoSpeed	Variable speed	?	2-speed
Internal Loads & Schedules								
	Lighting							
	Average power density (W/ft^2)	1	Perimeter zone_1; Core_1.147	ASHRAE 90.1 Lighting Power Densities Using the Building Area Method	1	1	1.00	0.85 (Return air fraction:0; Radiant fraction:0.32; Visible fraction:0.24)
	Schedule	See under 90.1**Schedules**	See under **Schedule summary_90.1**	See under **Schedules**	See under 90.1**Schedules**	See under **Schedule summary_LBL**	See under **BERG T-24 2005 Schedules**	See under **T-24 2008 schedule**
	Daylighting Controls	ASHRAE 90.1 Requirements	NO	ASHRAE 90.1 Requirements	No	NO	?	No
	Plug load							
	Average power density (W/ft^2)	0.75	Perimeter zone_0.8; Core_0.917	See under **Zone Summary**	0.75	0.8	0.8	1.34 (radiant fraction: 0.85)
	Schedule	See under 90.1**Schedules**	See under **Schedule summary_90.1**	See under **Schedules**	See under 90.1**Schedules**	See under **Schedule summary_LBL**	See under **BERG T-24 2005 Schedules**	See under **T-24 2008 schedule**
	Occupancy							
	Average people	200 sf/person	Perimeter zone_ 240sf/person; Core_ 209.3sf/person	See under **Zone Summary**	200 sf/person	240sf/person	Perimeter:107.8 sf/person; Interior: 126.84 sf/person	100sf/person (Radiant fraction:0.367; Sensible heat fraction: 0.61)
	Schedule	See under 90.1**Schedules**	See under **Schedule summary_90.1**	See under **Schedules**	See under 90.1**Schedules**	See under **Schedule summary_LBL**	See under **BERG T-24 2005 Schedules**	See under **T-24 2008 schedule**
Misc.								
	Elevator							N/A
	Quantity	2	N/A	2	N/A	N/A	N/A	N/A
	Motor type	hydraulic	N/A	hydraulic	N/A	N/A	N/A	N/A
	Peak Motor Power (W/elevator)	16,055	N/A	16,055	N/A	N/A	N/A	N/A
	Heat Gain to Building	Interior	N/A	Interior	N/A	N/A	N/A	N/A
	Peak Fan/lights Power (W/elevator)	161.9	N/A	161.9	N/A	N/A	N/A	N/A
	Motor and fan/lights Schedules	See under 90.1**Schedules**	N/A	See under **Schedules**	N/A	N/A	N/A	N/A
	Exterior							
	Peak Power (W)	14,385	N/A	14,385	N/A	N/A	N/A	N/A
	Schedule	See under 90.1**Schedules**	N/A	See under **Schedules**	N/A	N/A	N/A	N/A
Simulation Results								
		See under **SimulationResults**	See under **SimulationResults**	See under **SimulationResults**	See under **SimulationResults**	See under **SimulationResults**	See under **SimulationResults**	See under **SimulationResults**

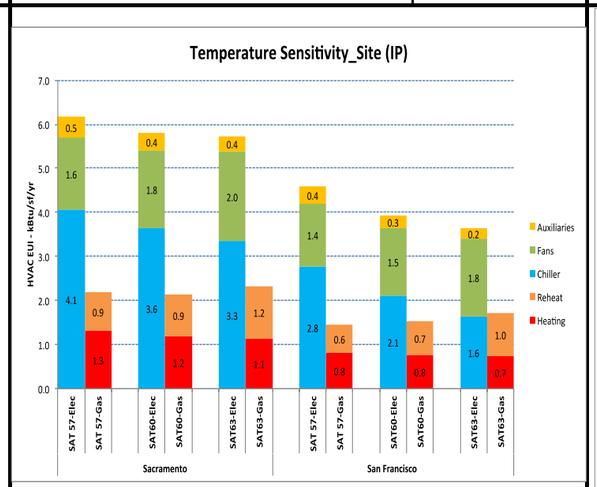
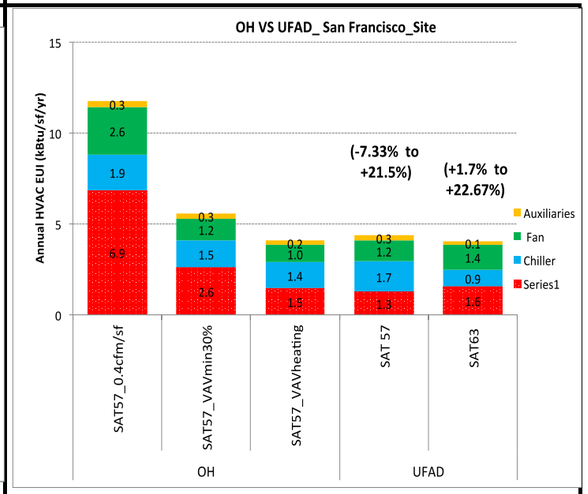
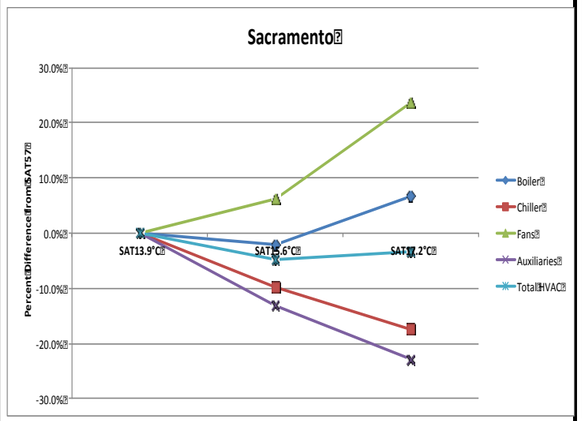
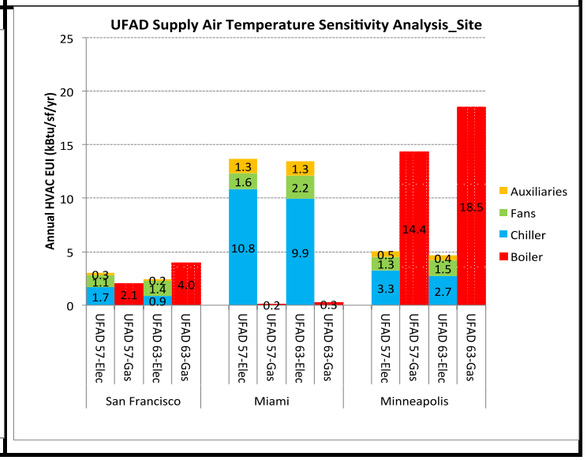
References

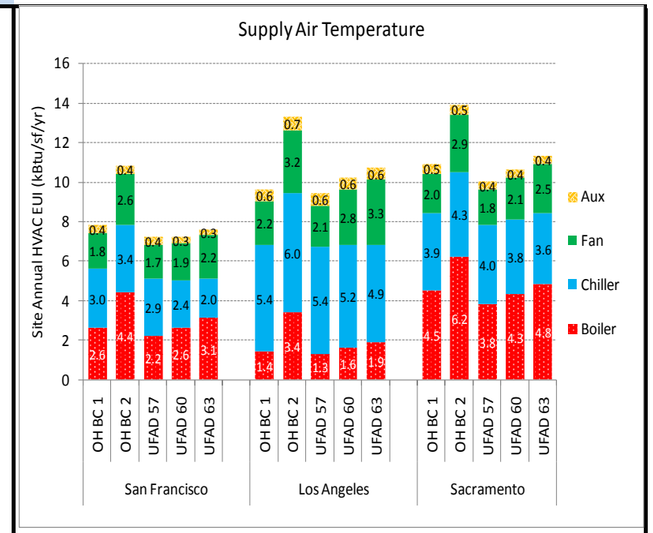
Briggs, R.S., R.G. Lucas, and Z.T. Taylor. 2003. Climate Classification for Building Energy Codes and Standards: Part 2—Zone Definitions, Maps, and Comparisons. ASHRAE Transactions 109(2).

PNNL's CBECS Study. 2007. *Analysis of Building Envelope Construction in 2003 CBECS Buildings.* Dave Winiarski, Mark Halverson, and Wei Jiang. Pacific Northwest National Laboratory. March 2007.

PNNL's CBECS Study. 2006. *Review of Pre- and Post-1980 Buildings in CBECS – HVAC Equipment.* Dave Winiarski, Wei Jiang and Mark Halverson. Pacific Northwest National Laboratory. December 2006.

Gowri K, DW Winiarski, and RE Jarnagin. 2009. Infiltration modeling guidelines for commercial building energy analysis . PNNL-18898, Pacific Northwest National Laboratory, Richland, WA. http://www.pnl.gov/main/publications/external/technical_reports/PNNL-18898.pdf

	Item	ASHRAE 90.1 Prototype_Medium Office	Simbuild 2010 Paper_SAT study	Optimization study_OPT1 CBE Study
SimulationInformation				
	Standard	PNNL 90.1_2010 scorecard	90.1-2004	90.1-2004
	Time	N/A	2010	April-June_2010
	EnergyPlus Version	V-6.0	V-3.1	V-3.1
Simulation Results				
	Detailed results link		Link to stacked results DATA%20ANALYSIS.xlsx - 'Simbuild 2010'!U1 Link to second chart DATA%20ANALYSIS.xlsx - 'Simbuild 2010'!AE46 Link to all other resuts x - 'Simbuild 2010 stacked'!X1	Link to OH VS UFAD optimization results DATA%20ANALYSIS.xlsx - 'CBE opto Charts'!A1 Link to UFAD sensitivity results DATA%20ANALYSIS.xlsx - 'Sensitivity study Charts'!A1
	Quik view			
	Quik view			

New CBE model -Optimization Study	CBE model - ToolKit	BERG model
90.1-2010	T-24_2005	T-24_2005
on-going (Summer 2011)	Aug-Oct_2009	
V-6.0	V-3.1	V-3.1
		Link to BERG results DATA%20ANALYSIS.xlsx - 'Compare to BERG (2)'!N12
		Where I can find more BERG results?
Coming	See under **SimulationResults**	
See under **SimulationResults**	See under **SimulationResults**	See under **SimulationResults**

eQUEST Study

T-24_2008

V-6.0

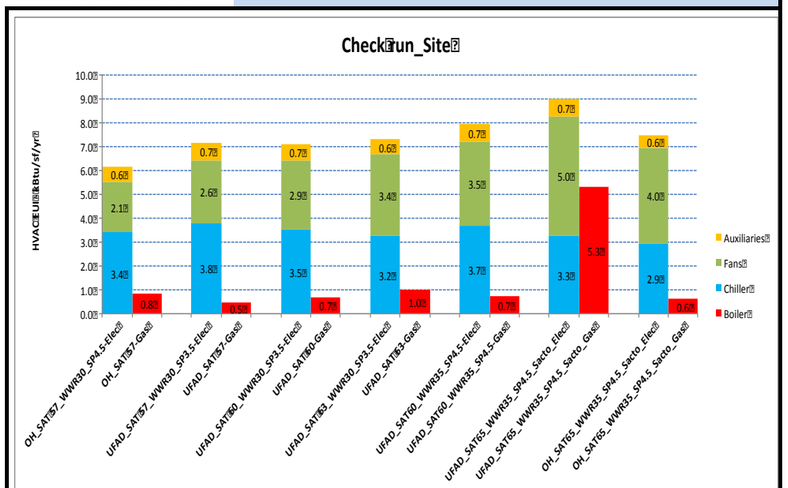
Link to check run results

DATA%20ANALYSIS.xlsx - 'Compare to BERG [2]'!N12

Link to heating sensitivitystudy

eQUEST_sensitivityHeatingStudy!B22

Link to other results



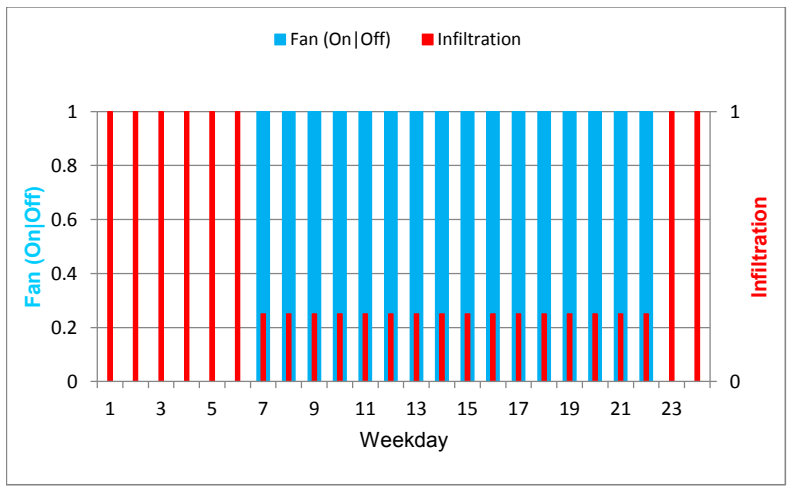
See under **SimulationResults**

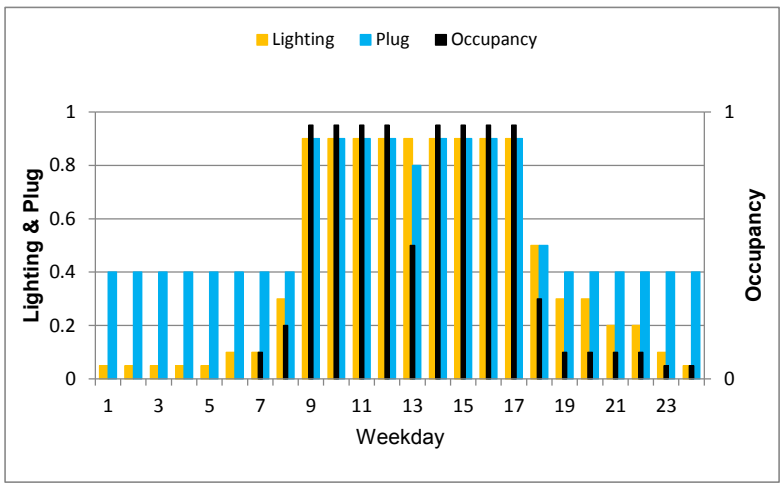
Minimum Outdoor Ventilation Air Requirements

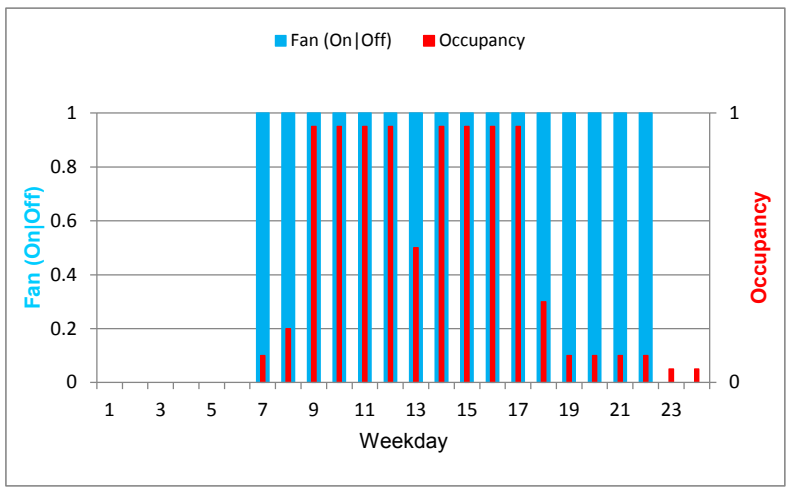
Based on Std 62 prescriptive requirement (Table 6.1)s; actual rates vary due to impact of multiple-spaces reset strategy

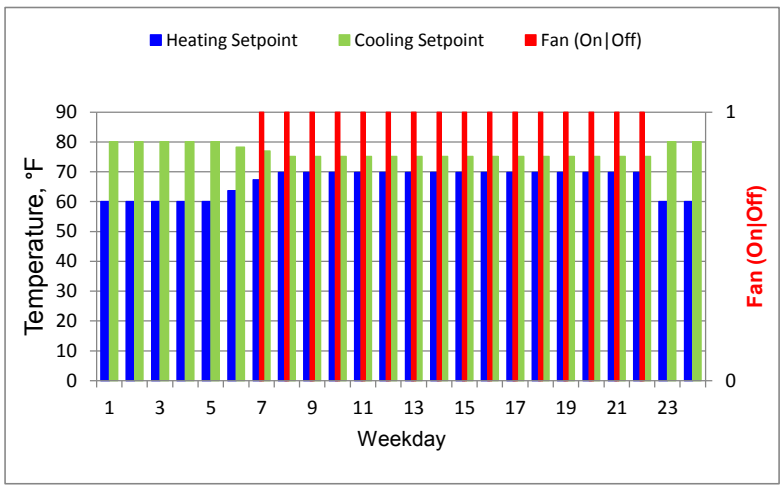
Zone	Area (ft^2)	Multipliers	Assumed Space Type	Total Occupants	Total OSA Ventilation (cfm/zone)			Total OSA Ventilation (cfm/ft2)		
				62.1-2004	90.1-2004 (62-1999)	90.1-2007 (62.1-2004)	90.1-2010 (62.1-2007)	90.1-2004 (62-1999)	90.1-2007 (62.1-2004)	90.1-2010 (62.1-2007)
CORE_BOTTOM	10587.8	1	Office space	53	1,059	900	900	0.100	0.085	0.085
CORE_MID	10587.8	1	Office space	53	1,059	900	900	0.100	0.085	0.085
CORE_TOP	10587.8	1	Office space	53	1,059	900	900	0.100	0.085	0.085
PERIMETER_TOP_ZN_3	2232.0	1	Office space	11	223	190	190	0.100	0.085	0.085
PERIMETER_TOP_ZN_2	1412.9	1	Office space	7	141	120	120	0.100	0.085	0.085
PERIMETER_TOP_ZN_1	2232.0	1	Office space	11	223	190	190	0.100	0.085	0.085
PERIMETER_TOP_ZN_4	1412.9	1	Office space	7	141	120	120	0.100	0.085	0.085
PERIMETER_BOT_ZN_3	2232.0	1	Office space	11	223	190	190	0.100	0.085	0.085
PERIMETER_BOT_ZN_2	1412.9	1	Office space	7	141	120	120	0.100	0.085	0.085
PERIMETER_BOT_ZN_1	2232.0	1	Office space	11	223	190	190	0.100	0.085	0.085
PERIMETER_BOT_ZN_4	1412.9	1	Office space	7	141	120	120	0.100	0.085	0.085
PERIMETER_MID_ZN_3	2232.0	1	Office space	11	223	190	190	0.100	0.085	0.085
PERIMETER_MID_ZN_2	1412.9	1	Office space	7	141	120	120	0.100	0.085	0.085
PERIMETER_MID_ZN_1	2232.0	1	Office space	11	223	190	190	0.100	0.085	0.085
PERIMETER_MID_ZN_4	1412.9	1	Office space	7	141	120	120	0.100	0.085	0.085
TOTAL	**53,633**			**268**	**5,363**	**4,559**	**4,559**	**0.100**	**0.085**	**0.085**

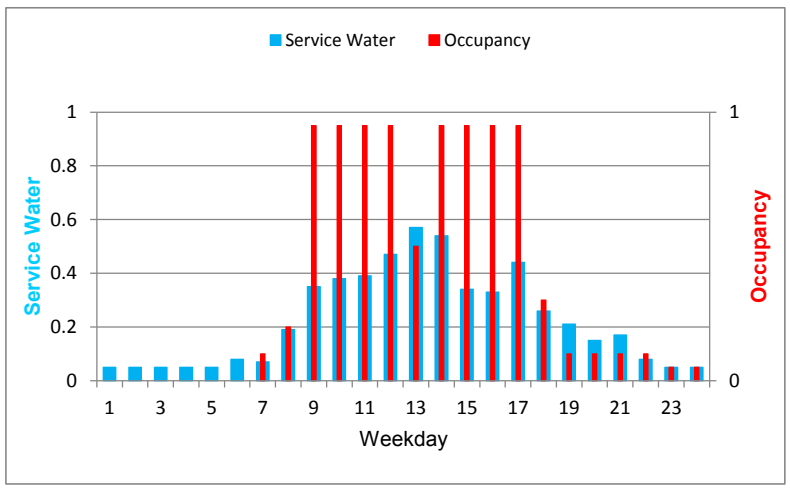
Schedule	Type	Through	Day of Week	1 am	2 am	3 am	4 am	5 am	6 am	7 am	8 am	9 am	10 am	11 am	Noon	1 pm	2 pm	3 pm	4 pm	5 pm	6 pm	7 pm	8 pm	9 pm	10 pm	11 pm	12 pm
Internal Loads Schedules																											
BLDG_LIGHT_SCH	Fraction	Through 12/31	Weekday	0.05	0.05	0.05	0.05	0.05	0.1	0.1	0.3	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.5	0.3	0.3	0.2	0.2	0.1	0.05
			Saturday	0.05	0.05	0.05	0.05	0.05	0.05	0.1	0.1	0.3	0.3	0.3	0.3	0.15	0.15	0.15	0.15	0.15	0.05	0.05	0.05	0.05	0.05	0.05	0.05
			Sun, Hol, Other	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
			SummerDesign	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
			WinterDesign	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
BLDG_EQUIP_SCH	Fraction	Through 12/31	Weekday	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.9	0.9	0.9	0.9	0.8	0.9	0.9	0.9	0.9	0.5	0.4	0.4	0.4	0.4	0.4	0.4
			Saturday	0.3	0.3	0.3	0.3	0.3	0.3	0.4	0.4	0.5	0.5	0.5	0.5	0.35	0.35	0.35	0.35	0.35	0.3	0.3	0.3	0.3	0.3	0.3	0.3
			Sun, Hol, Other	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
			SummerDesign	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
			WinterDesign	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
BLDG_OCC_SCH	Fraction	Through 12/31	Weekday	0	0	0	0	0	0	0.1	0.2	0.95	0.95	0.95	0.95	0.5	0.95	0.95	0.95	0.95	0.3	0.1	0.1	0.1	0.1	0.05	0.05
			Saturday	0	0	0	0	0	0	0.1	0.1	0.3	0.3	0.3	0.3	0.1	0.1	0.1	0.1	0.1	0.05	0.05	0	0	0	0	0
			Sun, Hol, Other	0	0	0	0	0	0	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0	0	0	0	0	0
			SummerDesign	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
			WinterDesign	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
BLDG_ELEVATORS	Fraction	Through 12/31	Weekday	0	0	0	0	0	0	0	0.35	0.69	0.43	0.37	0.43	0.58	0.48	0.37	0.37	0.46	0.62	0.12	0.04	0.04	0	0	0
			Saturday	0	0	0	0	0	0	0	0.16	0.14	0.21	0.18	0.25	0.21	0.13	0.08	0.04	0.05	0.06	0	0	0	0	0	0
			Sun, Hol, Other	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Exterior Lighting Schedule																											
BLDG_EXTERIOR_LIGHT	Fraction	Through 12/31	All Days	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
(AstronomicalClock control)																											
Service Water Heater Load Schedule																											
BLDG_SWH_SCH	Fraction	Through 12/31	Weekday	0.05	0.05	0.05	0.05	0.05	0.08	0.07	0.19	0.35	0.38	0.39	0.47	0.57	0.54	0.34	0.33	0.44	0.26	0.21	0.15	0.17	0.08	0.05	0.05
			Saturday	0.05	0.05	0.05	0.05	0.05	0.08	0.07	0.11	0.15	0.21	0.19	0.23	0.2	0.19	0.15	0.13	0.14	0.07	0.07	0.07	0.07	0.09	0.05	0.05
			Sun, Hol, Other	0.04	0.04	0.04	0.04	0.04	0.07	0.04	0.04	0.04	0.04	0.04	0.06	0.06	0.09	0.06	0.04	0.04	0.04	0.04	0.04	0.04	0.07	0.04	0.04
Infiltration Schedule																											
INFIL_SCH_PNNL	Fraction	Through 12/31	Weekday, SummerDesign	1	1	1	1	1	1	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	1	1
			Saturday, WinterDesign	1	1	1	1	1	1	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	1	1	1	1	1	1
			Sun, Hol, Other	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
HVAC Schedules																											
HVACOperationSchd	On/off	Through 12/31	Weekday, SummerDesign	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0
(Fan Schedule)			Saturday, WinterDesign	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0
			Sun, Hol, Other	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
HTGSETP_SCH	Temperature	Through 12/31	Weekday	15.6	15.6	15.6	15.6	15.6	17.6	19.6	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	15.6	15.6
			Saturday	15.6	15.6	15.6	15.6	15.6	17.6	19.6	21	21	21	21	21	21	21	21	21	21	15.6	15.6	15.6	15.6	15.6	15.6	15.6
			Sun, Hol, Other	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6
			WinterDesign	15.6	15.6	15.6	15.6	15.6	17.6	19.6	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	15.6	15.6
CLGSETP_SCH	Temperature	Through 12/31	Weekday	26.7	26.7	26.7	26.7	26.7	25.7	25	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	26.7	26.7
			Saturday	26.7	26.7	26.7	26.7	26.7	25.7	25	24	24	24	24	24	24	24	24	24	24	26.7	26.7	26.7	26.7	26.7	26.7	26.7
			Sun, Hol, Other	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7	26.7
	(°C)		SummerDesign	26.7	26.7	26.7	26.7	26.7	25.7	25	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	26.7	26.7
HTGSETP_SCH	Temperature	Through 12/31	Weekday	60	60	60	60	60	64	67	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	60	60
	(°F)		Saturday	60	60	60	60	60	64	67	70	70	70	70	70	70	70	70	70	70	60	60	60	60	60	60	60
			Sun, Hol, Other	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
			WinterDesign	60	60	60	60	60	64	67	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	60	60
CLGSETP_SCH	Temperature	Through 12/31	Weekday	80	80	80	80	80	78	77	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	80	80
	(°F)		Saturday	80	80	80	80	80	78	77	75	75	75	75	75	75	75	75	75	75	80	80	80	80	80	80	80
			Sun, Hol, Other	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80
			SummerDesign	80	80	80	80	80	78	77	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	80	80
MinOA_MotorizedDamper_Sched	Fraction	Through 12/31	WD, SummerDesign	0	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0
			Sat, WinterDesign	0	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0
			Sun, Hol, Other	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
MinOA_Sched	Fraction	Through 12/31	All	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1

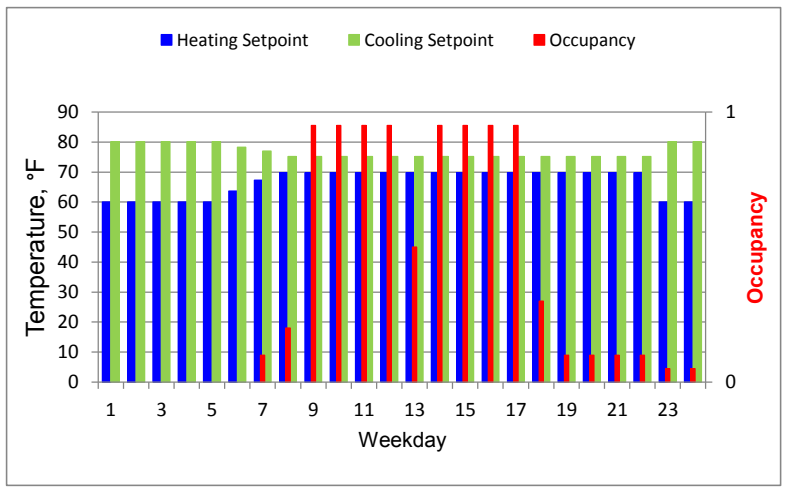

Fan (On|Off)
Infiltration
Fan (On|Off)
Infiltration
Weekday

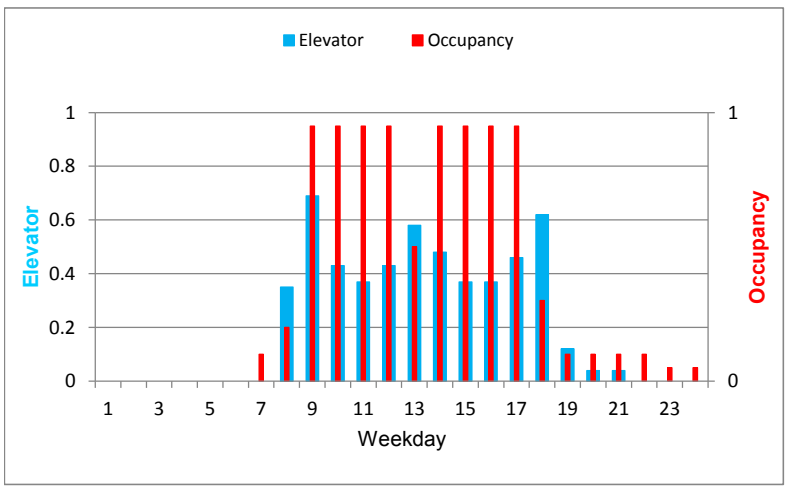

Lighting
Plug
Occupancy
Lighting & Plug
Occupancy
Weekday

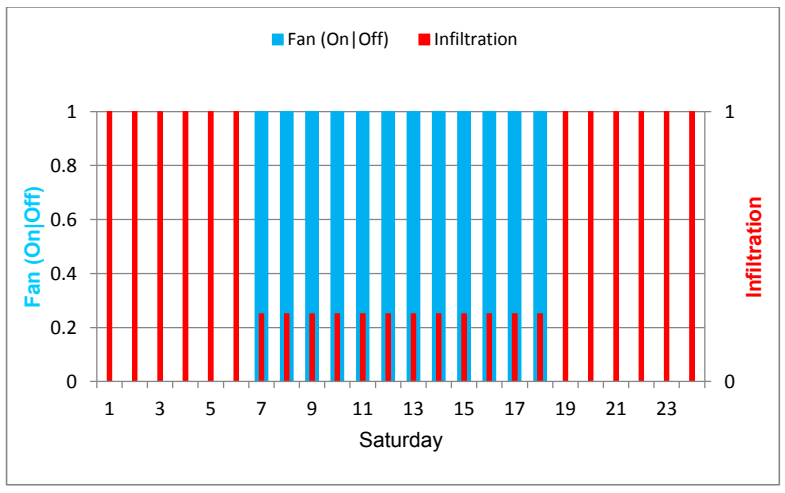

Fan (On|Off)
Occupancy
Fan (On|Off)
Occupancy
Weekday

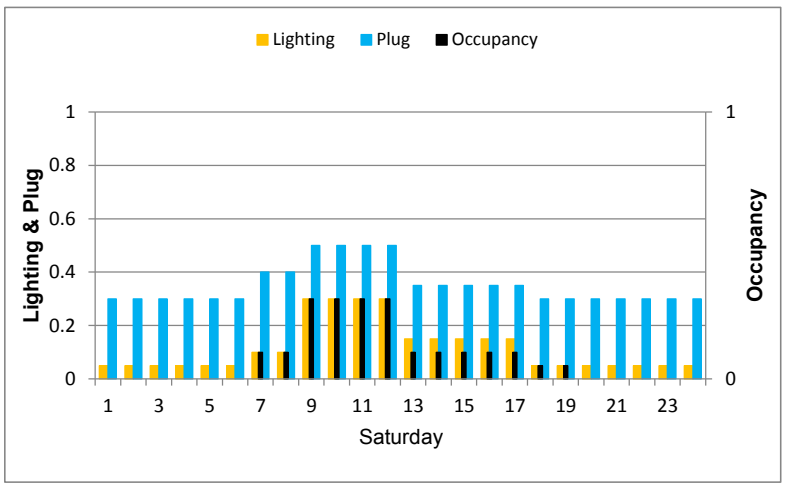

Heating Setpoint
Cooling Setpoint
Fan (On|Off)
Temperature, °F
Fan (On|Off)
Weekday

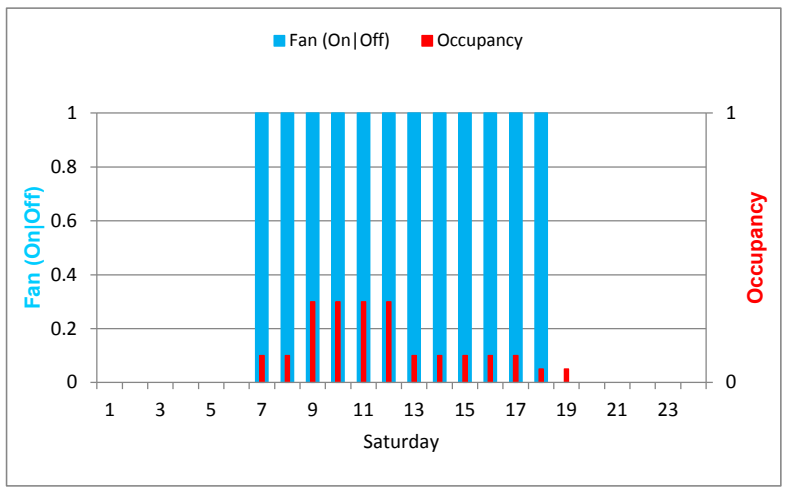

Service Water
Occupancy
Service Water
Occupancy
Weekday

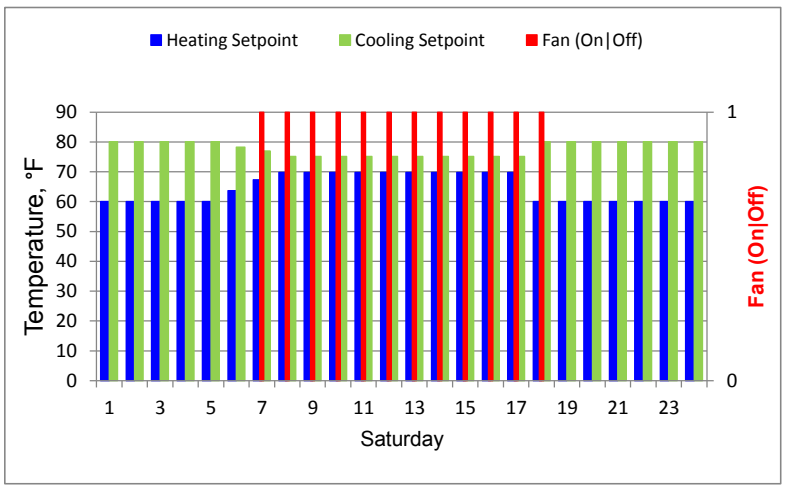

Heating Setpoint
Cooling Setpoint
Occupancy
Temperature, °F
Occupancy
Weekday

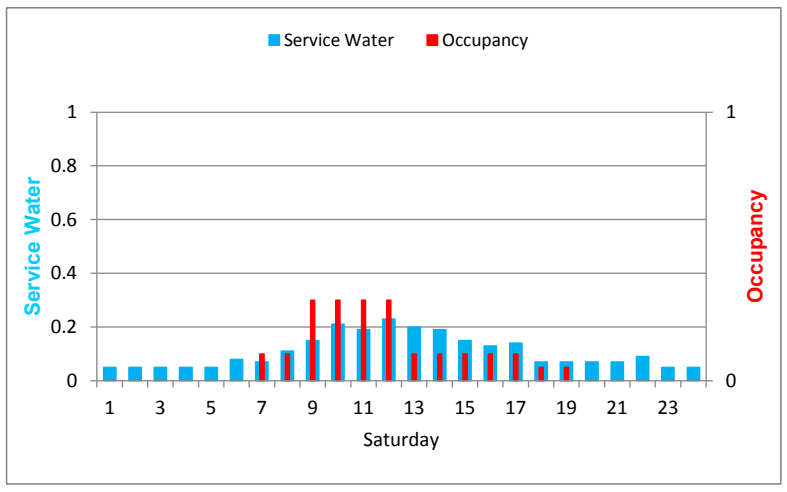

Elevator
Occupancy
Elevator
Occupancy
Weekday

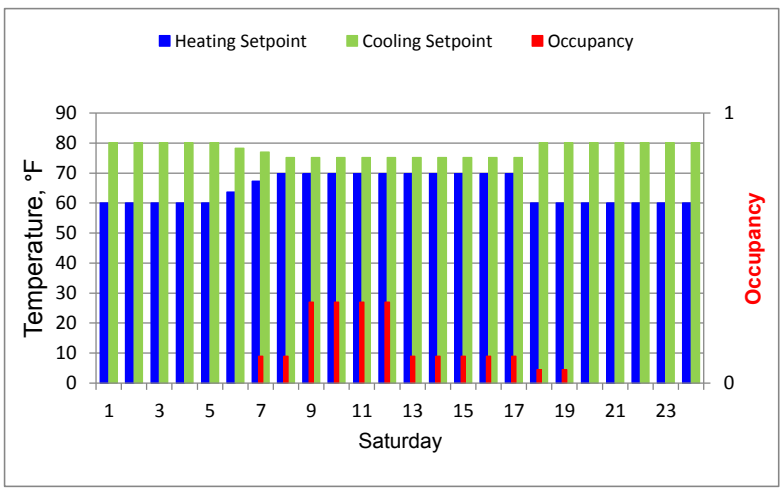

Fan (On|Off)
Infiltration
Fan (On|Off)
Infiltration
Saturday

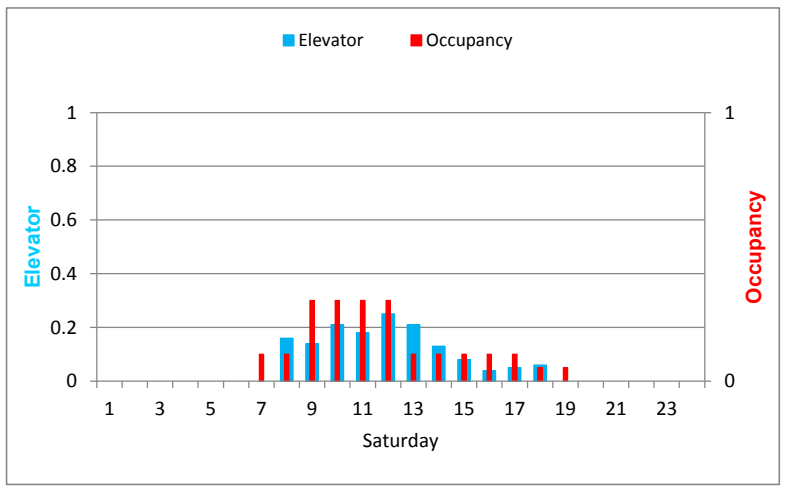

Lighting
Plug
Occupancy
Lighting & Plug
Occupancy
Saturday


Fan (On|Off)
Occupancy
Fan (On|Off)
Occupancy
Saturday


Heating Setpoint
Cooling Setpoint
Fan (On|Off)
Temperature, °F
Fan (On|Off)
Saturday


Service Water
Occupancy
Service Water
Occupancy
Saturday


Heating Setpoint
Cooling Setpoint
Occupancy
Temperature, °F
Occupancy
Saturday


Elevator
Occupancy
Elevator
Occupancy
Saturday

Climate zone	Representa Items		ASHRAE 90.1-2010	
			U value (Btu/h-ft2-F)	0.124
		Exterior wall (steel framed)	Specifications	Steel-Frame Walls (2X4 16IN OC) 0.4 in. Stucco+5/8 in. gypsum board +(R-13.0)+5/8 in.
			U value (Btu/h-ft2-F)	0.055
		Roof	Specifications	Roof membrane+Roof insulation+metal decking
			Uvalue (Btu/h-ft2-F)_SHGC	0.75_0.25
2A	Houston	Windows	Specifications (Window6)	6032+Al w/break_U=0.95 (0.758_0.256)
			U value (Btu/h-ft2-F)	0.124
		Exterior wall	Specifications	Steel-Frame Walls (2X4 16IN OC) 0.4 in. Stucco+5/8 in. gypsum board +(R-13.0)+5/8 in.
			U value (Btu/h-ft2-F)	0.055
		Roof	Specifications	Roof membrane+Roof insulation+metal decking
			Uvalue (Btu/h-ft2-F)_SHGC	0.75_0.25
2B	Phoenix	Windows	Specifications	6032+Al w/break_U=0.95 (0.758_0.256)
			U value (Btu/h-ft2-F)	0.084
		Exterior wall	Specifications	Steel-Frame Walls (2X4 16IN OC) 0.4 in. Stucco+5/8 in. gypsum board + (R-13.0 + R-3.8 c.i.)+5/8 in.
			U value (Btu/h-ft2-F)	0.055
		Roof	Specifications	Roof membrane+Roof insulation+metal decking
			Uvalue (Btu/h-ft2-F)_SHGC	0.65_0.25
3B	Sacramemto	Windows	Specifications	11034+air25in+103+Al w/break_U=1.2(0.624_0.243) (from Andy McNell)
			U value (Btu/h-ft2-F)	0.084
		Exterior wall	Specifications	Steel-Frame Walls (2X4 16IN OC) 0.4 in. Stucco+5/8 in. gypsum board + (R-13.0 + R-3.8 c.i.)+5/8 in.
			U value (Btu/h-ft2-F)	0.055
		Roof	Specifications	Roof membrane+Roof insulation+metal decking
			Uvalue (Btu/h-ft2-F)_SHGC	0.65_0.25
3C	San Francisco	Windows	Specifications	11034+air25in+103+Al w/break_U=1.2(0.624_0.243) (from Andy McNell)
			U value (Btu/h-ft2-F)	0.064
		Exterior wall	Specifications	Steel-Frame Walls (2X4 16IN OC) 0.4 in. Stucco+5/8 in. gypsum board + (R-13.0 + R-7.5 c.i.) +5/8 in.
			U value (Btu/h-ft2-F)	0.055
		Roof	Specifications	Roof membrane+Roof insulation+metal decking
			Uvalue (Btu/h-ft2-F)_SHGC	0.55_0.4
4A	Baltimore	Windows	Specifications	433+air25in+5248+Al w/break_U=1 (0.55_0.4) (from Andy McNell)
				0.064
		Exterior wall	Specifications	Steel-Frame Walls (2X4 16IN OC) 0.4 in. Stucco+5/8 in. gypsum board + (R-13.0 + R-7.5 c.i.) +5/8 in.
			U value (Btu/h-ft2-F)	0.055
		Roof	Specifications	
			Uvalue (Btu/h-ft2-F)_SHGC	0.55_0.4
5A	Chicago	Windows	Specifications	433+air25in+5248+Al w/break_U=1 (0.55_0.4) (from Andy McNell)
			U value (Btu/h-ft2-F)	0.064
		Exterior wall	Specifications	Steel-Frame Walls (2X4 16IN OC) 0.4 in. Stucco+5/8 in. gypsum board + (R-13.0 + R-7.5 c.i.) +5/8 in.
			U value (Btu/h-ft2-F)	0.049
		Roof	Specifications	Roof membrane+Roof insulation+metal decking
			Uvalue (Btu/h-ft2-F)_SHGC	0.55_0.4
6A	Minneapolis	Windows	Specifications	433+air25in+5248+Al w/break_U=1 (0.55_0.4) (from Andy McNell)

Filtrasol 6mm grey (IGDB id# 433)

0.25" Air

PPG Sunqate 500 on clear (IGDB id# 5249)

Thermally broken aluminum frame

Window Configuration			
	U	**SHGC**	**Configuration**
90.1 2010 requirement	*0.75*	*0.25*	
Window6 (Dove)	*0.758*	*0.256*	*6032(Blue on clear)+Al w/break_U=0.95*
Prototype model	*0.81*	*0.293*	*Window_U_0.72_SHGC_0.25: REF B TINT MID 6MM*
90.1 2010 requirement	*0.75*	*0.25*	
Window6 (Dove)	*0.758*	*0.256*	*6032+Al w/break_U=0.95*
Prototype model	*0.81*	*0.293*	*Window_U_0.72_SHGC_0.25: REF B TINT MID 6MM*
90.1 2010 requirement	*0.65*	*0.25*	
Window6 (Andy McNell)	*0.624*	*0.243*	11034+air25in+103+Al w/break_U=1.2(0.624_0.243)
Prototype model	*0.502*	*0.287*	*Window_U_0.62_SHGC_0.25: REF C CLEAR HI 6MM+AIR 6MM+ CLEAR 6MM;*
90.1 2010 requirement	*0.65*	*0.25*	
Window6	*0.624*	*0.243*	11034+air25in+103+Al w/break_U=1.2(0.624_0.243)
Prototype model	*0.502*	*0.287*	*Window_U_0.62_SHGC_0.25: REF C CLEAR HI 6MM+AIR 6MM+ CLEAR 6MM;*
90.1 2010 requirement	*0.55*	*0.4*	
Window6 (Andy McNell)	*0.55*	*0.399*	433+air25in+5248+Al w/break_U=1
Prototype model	*0.467*	*0.426*	*Window_U_0.52_SHGC_0.4: REF D CLEAR 6MM+AIR 13MM+ CLEAR 6MM;*
90.1 2010 requirement	*0.55*	*0.4*	
Window6 (Andy McNell)	*0.55*	*0.399*	433+air25in+5248+Al w/break_U=1
Prototype model	*0.467*	*0.426*	*Window_U_0.52_SHGC_0.4: REF D CLEAR 6MM+AIR 13MM+ CLEAR 6MM;*
90.1 2010 requirement	*0.55*	*0.4*	
Window6 (Andy McNell)	*0.55*	*0.399*	433+air25in+5248+Al w/break_U=1
Prototype model	*0.467*	*0.426*	*Window_U_0.52_SHGC_0.4: REF D CLEAR 6MM+AIR 13MM+ CLEAR 6MM;*

	Title 24		Window5			
Climate 3/4/5 (SF)	U	SHGC	U	SHGC	Glazing	Frame
	Non-North			Non-North		
11-20% WWR	0.77	0.55	0.612	0.396	5284+Air+103	Al No Break U = 1.9
31-40% WWR	0.77	0.41	0.485	0.390	5284+Air+103	Al No Break U = 1.9
41-60% WWR	not specified		0.441	0.387	5284+Air+103	Al No Break U = 1.9
Climate 3/4/5 (SF)	U	SHGC	U	SHGC	Glazing	Frame
		North		North		
11-20% WWR	0.77	0.61	0.626	0.547	159+Air+10	Al No Break U = 1.9
31-40% WWR	0.77	0.61	0.500	0.553	159+Air+10	Al No Break U = 1.9
41-60% WWR	not specified		0.457	0.555	159+Air+10	Al No Break U = 1.9
Climate 6/7/8/9 (LA)	U	SHGC	U	SHGC	Glazing	Frame
	Non-North			Non-North		
11-20% WWR	0.77	0.61	0.613	0.326	5286+Air+103	Al No Break U = 1.9
31-40% WWR	0.77	0.34	0.486	0.314	5286+Air+103	Al No Break U = 1.9
41-60% WWR	not specified		0.442	0.310	5286+Air+103	Al No Break U = 1.9
Climate 6/7/8/9 (LA)	U	SHGC	U	SHGC	Glazing	Frame
		North		North		
11-20% WWR	0.77	0.61	0.626	0.547	159+Air+10	Al No Break U = 1.9
31-40% WWR	0.77	0.61	**0.500**	**0.553**	**159+Air+10**	**Al No Break U = 1.9**
41-60% WWR	not specified		0.457	0.555	159+Air+10	Al No Break U = 1.9
Climate 2/10/11/12/13 (SAC)	U	SHGC	U	SHGC	Glazing	Frame
	Non-North			Non-North		
11-20% WWR	0.47	0.36	0.456	0.288	5286+Air+103	Al Break U = 1
31-40% WWR	0.47	0.31	**0.391**	**0.291**	**5286+Air+103**	**Al Break U = 1**
41-60% WWR	not specified		0.368	0.292	5286+Air+103	Al Break U = 1
Climate 2/10/11/12/13 (SAC)	U	SHGC	U	SHGC	Glazing	Frame
		North		North		
11-20% WWR	0.47	0.51	0.456	0.360	196+Air+10	Al Break U = 1
31-40% WWR	0.47	0.47	**0.391**	**0.369**	**196+Air+10**	**Al Break U = 1**
41-60% WWR	not specified		0.369	0.372	196+Air+10	Al Break U = 1

20% WWR, H=31.2", W=62.4"
40% WWR, H=62.4", W=62.4"
60% WWR, H=93.6", W=62.4"

Schedules

HVAC schedules

A. 12-hour Office														
		Day	Hour of Day											
		type*	1	2	3	4	5	6	7	8	9	10	11	12
	Fan	WD	off	off	off	off	off	off	on	on	on	on	on	on
	"	WEH	off	off	off	off	off	off	on	on	on	on	on	on
	Cooling	WD	90	90	90	90	90	90	75	75	75	75	75	75
	"	WEH	90	90	90	90	90	90	75	75	75	75	75	75
	Heating	WD	55	55	55	55	55	55	70	70	70	70	70	70
	"	WEH	55	55	55	55	55	55	70	70	70	70	70	70
	Vent	WD	off	off	off	off	off	off	off	vent	vent	vent	vent	vent
	"	WEH	off	off	off	off	off	off	off	vent	vent	vent	vent	vent
		Day	Hour of Day											
		type*	13	14	15	16	17	18	19	20	21	22	23	24
	Fan	WD	on	on	on	on	on	on	off	off	off	off	off	off
	"	WEH	on	off	off	off	off	off	off	off	off	off	off	off
	Cooling	WD	75	75	75	75	75	75	90	90	90	90	90	90
	"	WEH	75	90	90	90	90	90	90	90	90	90	90	90
	Heating	WD	70	70	70	70	70	70	55	55	55	55	55	55
	"	WEH	70	55	55	55	55	55	55	55	55	55	55	55
	Vent	WD	vent	vent	vent	vent	vent	off	off	off	off	off	off	off
	"	WEH	off	off	off	off	off	off	off	off	off	off	off	off

Loads schedules

A. 12-hour Office										
		Day	Hour of Day							
Zone	Parameter	type*	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00
Bldg	People, Hot V	WD	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.33
	* *	WEH	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.07
Bldg	Lights	WD	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.90
	*	WEH	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.40
Bldg	Equipment,	WD	0.17	0.17	0.17	0.17	0.17	0.17	0.17	1.00
	Elevators	WEH	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17
		Day	Hour of Day							
Zone	Parameter	type*	9.00	10.00	11.00	12.00	13.00	14.00	15.00	16.00
Bldg	People, Hot V	WD	0.67	1.00	1.00	1.00	1.00	1.00	1.00	1.00
	* *	WEH	0.13	0.20	0.20	0.20	0.10	0.00	0.00	0.00
Bldg	Lights	WD	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90
	*	WEH	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30
Bldg	Equipment,	WD	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
	Elevators	WEH	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17
		Day	Hour of Day							
Zone	Parameter	type*	17.00	18.00	19.00	20.00	21.00	22.00	23.00	24.00
Bldg	People, Hot V	WD	1.00	0.50	0.00	0.00	0.00	0.00	0.00	0.00
	* *	WEH	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Bldg	Lights	WD	0.90	0.30	0.30	0.30	0.30	0.30	0.30	0.30
	*	WEH	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30
Bldg	Equipment,	WD	1.00	0.17	0.17	0.17	0.17	0.17	0.17	0.17
	Elevators	WEH	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17
B. 24-hour Office										
		Day	Hour of Day							
Zone	Parameter	type*	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00
Bldg	People, Hot V	WD	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.67
	* *	WEH	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33
Bldg	Lights	WD	0.33	0.33	0.33	0.33	0.33	0.33	0.90	0.90
	*	WEH	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33
Bldg	Equipment,	WD	0.17	0.17	0.17	0.17	0.17	0.17	0.17	1.00
	Elevators	WEH	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17
		Day	Hour of Day							
Zone	Parameter	type*	9.00	10.00	11.00	12.00	13.00	14.00	15.00	16.00
Bldg	People, Hot V	WD	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
	* *	WEH	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33
Bldg	Lights	WD	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90
	*	WEH	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33
Bldg	Equipment,	WD	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
	Elevators	WEH	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17
		Day	Hour of Day							
Zone	Parameter	type*	17.00	18.00	19.00	20.00	21.00	22.00	23.00	24.00
Bldg	People, Hot V	WD	1.00	0.50	0.33	0.33	0.33	0.33	0.33	0.33
	* *	WEH	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33
Bldg	Lights	WD	0.90	0.33	0.33	0.33	0.33	0.33	0.33	0.33
	*	WEH	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33
Bldg	Equipment,	WD	1.00	0.17	0.17	0.17	0.17	0.17	0.17	0.17
	Elevators	WEH	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17

Prototypical large office developed for ASHRAE (PNL) cited in Huang et al. 1991

Benchmark

Schedule	Type	Through	Day of Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
BLDG_LIGHT_SCH	Fraction	Through 12/31	WD	0.05	0.05	0.05	0.05	0.05	0.1	0.1	0.3	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.5	0.3	0.3	0.2	0.2	0.1	0.05
			SummerDesign	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
			Sat	0.05	0.05	0.05	0.05	0.05	0.05	0.1	0.1	0.3	0.3	0.3	0.3	0.15	0.15	0.15	0.15	0.15	0.05	0.05	0.05	0.05	0.05	0.05	0.05
			WinterDesign	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
			Sun, Hol, Other	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
BLDG_EQUIP_SCH	Fraction	Through 12/31	WD	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.9	0.9	0.9	0.9	0.8	0.9	0.9	0.9	0.9	0.5	0.4	0.4	0.4	0.4	0.4	0.4
			SummerDesign	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
			Sat	0.3	0.3	0.3	0.3	0.3	0.3	0.4	0.4	0.5	0.5	0.5	0.5	0.35	0.35	0.35	0.35	0.35	0.3	0.3	0.3	0.3	0.3	0.3	0.3
			WinterDesign	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
			Sun, Hol, Other	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
BLDG_OCC_SCH	Fraction	Through 12/31	WD	0	0	0	0	0	0	0.1	0.2	0.95	0.95	0.95	0.95	0.5	0.95	0.95	0.95	0.95	0.3	0.1	0.1	0.05	0.05	0.05	0.05
			SummerDesign	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0.05	0.05
			Sat	0	0	0	0	0	0	0.1	0.1	0.3	0.3	0.3	0.3	0.1	0.1	0.1	0.1	0.1	0	0	0	0	0	0	0
			WinterDesign	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
			Sun, Hol, Other	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
BLDG_ELEVATORS	Fraction	Through 12/31	WD, SummerDesi	0	0	0	0	0	0	0	0.35	0.69	0.43	0.37	0.43	0.58	0.48	0.37	0.37	0.46	0.62	0.12	0.04	0.04	0	0	0
			Sat, WinterDesign	0	0	0	0	0	0	0	0.16	0.14	0.21	0.18	0.25	0.21	0.13	0.08	0.04	0.05	0.06	0	0	0	0	0	0
			Sun, Hol, Other	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
INFIL_SCH	Fraction	Through 12/31	WD, SummerDesi	1	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1
			Sat, WinterDesign	1	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	1	1	1
			Sun, Hol, Other	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
INFIL_HALF_ON_SCH	Fraction	Through 12/31	WD, SummerDesi	1	1	1	1	1	1	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	1	1
			Sat, WinterDesign	1	1	1	1	1	1	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	1	1	1	1	1	1
			Sun, Hol, Other	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
INFIL_QUARTER_ON_SCH	Fraction	Through 12/31	WD, SummerDesi	1	1	1	1	1	1	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	1	1
			Sat, WinterDesign	1	1	1	1	1	1	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	1	1	1	1	1	1
			Sun, Hol, Other	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
BLDG_SWH_SCH	Fraction	Through 12/31	WD, SummerDesi	0.05	0.05	0.05	0.05	0.05	0.08	0.07	0.19	0.35	0.38	0.39	0.47	0.57	0.54	0.34	0.33	0.44	0.26	0.21	0.15	0.17	0.08	0.05	0.05
			Sat, WinterDesign	0.05	0.05	0.05	0.05	0.05	0.08	0.07	0.11	0.15	0.21	0.19	0.23	0.2	0.19	0.15	0.13	0.14	0.07	0.07	0.07	0.07	0.09	0.05	0.05
			Sun, Hol, Other	0.04	0.04	0.04	0.04	0.04	0.07	0.04	0.04	0.04	0.04	0.04	0.06	0.06	0.09	0.06	0.04	0.04	0.04	0.04	0.04	0.04	0.07	0.04	0.04
Hours_of_operation	On/Off	Through 12/31	WD, SummerDesi	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0
			Sat, WinterDesign	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0
			Sun, Hol, Other	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
ALWAYS_ON	Fraction	Through 12/31	All	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
ALWAYS_OFF	Fraction	Through 12/31	All	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
HVACOperationSchd	On/Off	Through 12/31	WD, SummerDesi	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0
			Sat, WinterDesign	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0
			Sun, Hol, Other	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
PlantOnSched	On/Off	Through 12/31	All	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
FAN_SCH	Fraction	Through 12/31	All	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
ReheatCoilAvailSched	Fraction	Through 12/31	All	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
CoolingCoilAvailSched	Fraction	Through 12/31	All	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
HTGSETP_SCH	Temperat	Through 12/31	WD	15.6	15.6	15.6	15.6	15.6	21	21	21	21	21	21	21	21	21	21	21	21	21	21	15.6	15.6	15.6	15.6	15.6
			SummerDesign	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6
			Sat	15.6	15.6	15.6	15.6	15.6	15.6	21	21	21	21	21	21	21	21	21	21	21	15.6	15.6	15.6	15.6	15.6	15.6	15.6
			WinterDesign	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21
			Sun, Hol, Other	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6
CLGSETP_SCH	Temperat	Through 12/31	WD, SummerDesi	30	30	30	30	30	30	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	30	30
			Sat	30	30	30	30	30	30	24	24	24	24	24	24	24	24	24	24	24	24	30	30	30	30	30	30
			WinterDesign	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30
			Sun, Hol, Other	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30
Humidity Setpoint Schedule	Humidity	Through 12/31	WD, SummerDesi	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50
			Sat, WinterDesign	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50
			Sun, Hol, Other	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50
MinRelHumSetSch	Humidity	Through 12/31	All	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30
MaxRelHumSetSch	Humidity	Through 12/31	All	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
MinOA_MotorizedDamper_S	Fraction	Through 12/31	WD, SummerDesi	0	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0

Largeoffice.IDF

Schedule	Type	Through	Day of Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Sch HVAC System Availability	On/Off	Through: 12/31	WD	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0
			SummerDesignDay	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0
			Sat	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0
			WinterDesign	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0
			Sun, Hol, Othe	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Sch LBL Equipment.	Fraction	Through: 12/31	WD	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.9	0.9	0.9	0.9	0.8	0.9	0.9	0.9	0.9	0.5	0.4	0.4	0.4	0.4	0.4	0.4
			SummerDesignDay	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
			Sat	0.3	0.3	0.3	0.3	0.3	0.3	0.4	0.4	0.5	0.5	0.5	0.5	0.35	0.35	0.35	0.35	0.35	0.3	0.3	0.3	0.3	0.3	0.3	0.3
			WinterDesign	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
			Sun, Hol, Othe	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
BLDG LIGHT SCH	Fraction	Through: 12/31	WD	0.05	0.05	0.05	0.05	0.05	0.1	0.1	0.3	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.5	0.3	0.3	0.2	0.2	0.1	0.05
			SummerDesignDay	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
			Sat	0.05	0.05	0.05	0.05	0.05	0.05	0.1	0.1	0.3	0.3	0.3	0.3	0.15	0.15	0.15	0.15	0.15	0.05	0.05	0.05	0.05	0.05	0.05	0.05
			WinterDesign	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
			Sun, Hol, Othe	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
BLDG OCC SCH	Fraction	Through 12/31	WD	0	0	0	0	0	0	0.1	0.2	0.95	0.95	0.95	0.95	0.5	0.95	0.95	0.95	0.95	0.3	0.1	0.1	0.05	0.05	0.05	0.05
			SummerDesign	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0.05	0.05
			Sat	0	0	0	0	0	0	0.1	0.1	0.3	0.3	0.3	0.3	0.1	0.1	0.1	0.1	0.1	0	0	0	0	0	0	0
			WinterDesign	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
			Sun, Hol, Othe	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
CLGSETP SCH	Temperatur	Through 12/31	WD, SummerDesign	30	30	30	30	30	30	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	30	30
			Sat	30	30	30	30	30	30	24	24	24	24	24	24	24	24	24	24	24	24	30	30	30	30	30	30
			WinterDesign	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30
			Sun, Hol, Othe	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30
HTGSETP SCH	Temperatur	Through 12/31	WD	15.6	15.6	15.6	15.6	15.6	15.6	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	15.6	15.6
			SummerDesign	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6
			Sat	15.6	15.6	15.6	15.6	15.6	15.6	21	21	21	21	21	21	21	21	21	21	21	21	15.6	15.6	15.6	15.6	15.6	15.6
			WinterDesign	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21
			Sun, Hol, Othe	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6

090(ASHRAE 90.1)

Schedule	Type	Through	Day of Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Sch HVAC System Availability	On/Off	Through: 12/31	WD	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0
			SummerDesignDay	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0
			Sat	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0
			WinterDesign	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0
			Sun, Hol, Othe	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Sch LBL Equipment.	Fraction	Through: 12/31	WD	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.9	0.9	0.9	0.9	0.8	0.9	0.9	0.9	0.9	0.5	0.4	0.4	0.4	0.4	0.4	0.4
			SummerDesignDay	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
			Sat	0.3	0.3	0.3	0.3	0.3	0.3	0.4	0.4	0.5	0.5	0.5	0.5	0.35	0.35	0.35	0.35	0.35	0.3	0.3	0.3	0.3	0.3	0.3	0.3
			WinterDesign	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
			Sun, Hol, Othe	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
Sch LBL Lighting	Fraction	Through: 12/31	WD	0.05	0.05	0.05	0.05	0.05	0.1	0.1	0.3	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.5	0.3	0.3	0.2	0.2	0.1	0.05
			SummerDesignDay	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
			Sat	0.05	0.05	0.05	0.05	0.05	0.05	0.1	0.1	0.3	0.3	0.3	0.3	0.15	0.15	0.15	0.15	0.15	0.05	0.05	0.05	0.05	0.05	0.05	0.05
			WinterDesign	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
			Sun, Hol, Othe	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
Sch LBL Occupancy	Fraction	Through 12/31	WD	0	0	0	0	0	0	0.1	0.2	0.95	0.95	0.95	0.95	0.5	0.95	0.95	0.95	0.95	0.3	0.1	0.1	0.05	0.05	0.05	0.05
			SummerDesign	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0.05	0.05
			Sat	0	0	0	0	0	0	0.1	0.1	0.3	0.3	0.3	0.3	0.1	0.1	0.1	0.1	0.1	0	0	0	0	0	0	0
			WinterDesign	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
			Sun, Hol, Othe	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0

090 (LBL)

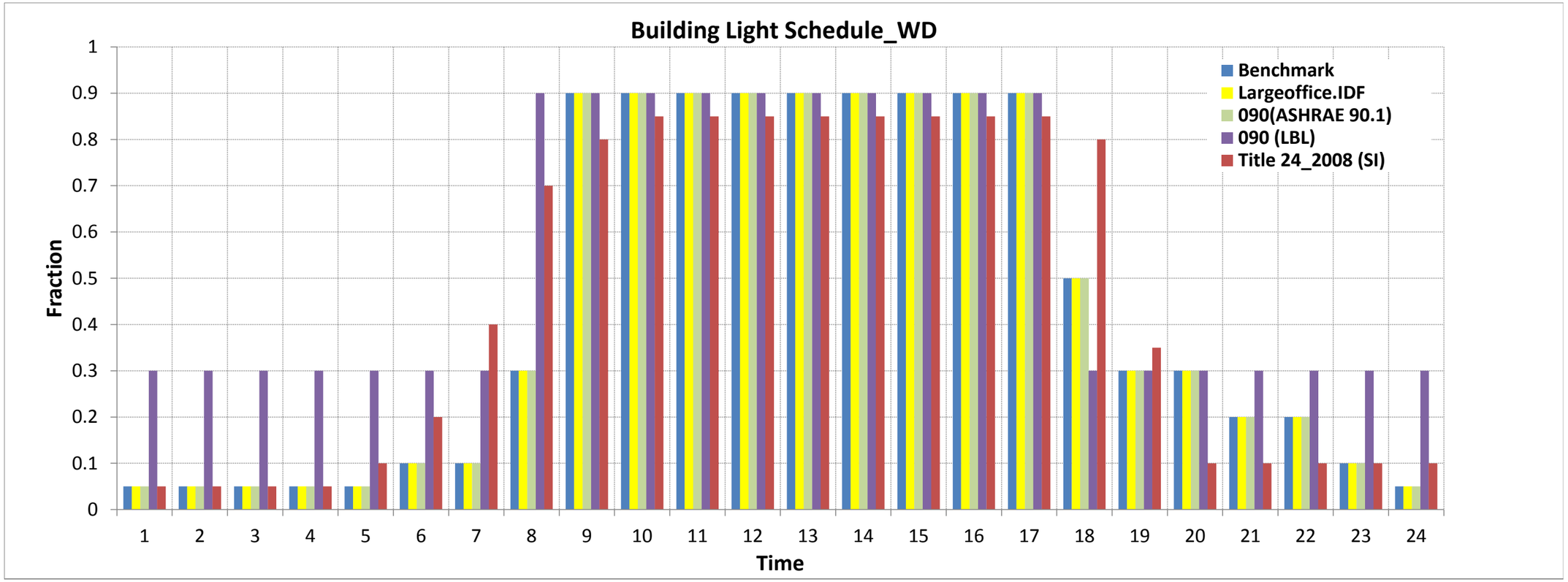
Schedule	Type	Through	Day of Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Sch HVAC System Availability	On/Off	Through: 12/31	WD	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0
			SummerDesignDay	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0
			Sat	0	0	0	0	0	0	1	1	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	0
			WinterDesign	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0
			Sun, Hol, Othe	0	0	0	0	0	0	1	1	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	0
Sch AHU Coil Availability	On/Off	Through: 12/31	WD	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0
			SummerDesignDay	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0
			Sat	0	0	0	0	0	0	1	1	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	0
			WinterDesign	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0
			Sun, Hol, Othe	0	0	0	0	0	0	1	1	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	0
Sch LBL Occupancy.	Fraction	Through: 12/31	WD	0	0	0	0	0	0	0	0.33	0.67	1	1	1	1	1	1	1	1	0.5	0	0	0	0	0	0
			SummerDesignDay	0	0	0	0	0	0	0	0.33	0.67	1	1	1	1	1	1	1	1	0.5	0	0	0	0	0	0
			Sat	0	0	0	0	0	0	0	0.07	0.13	0.2	0.2	0.2	0.1	0	0	0	0	0	0	0	0	0	0	0
			WinterDesign	0	0	0	0	0	0	0	0.33	0.67	1	1	1	1	1	1	1	1	0.5	0	0	0	0	0	0
			Sun, Hol, Othe	0	0	0	0	0	0	0	0.07	0.13	0.2	0.2	0.2	0.1	0	0	0	0	0	0	0	0	0	0	0
Sch LBL Lighting.	Fraction	Through: 12/31	WD	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.3	0.3	0.3	0.3	0.3	0.3	0.3
			SummerDesignDay	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.3	0.3	0.3	0.3	0.3	0.3	0.3
			Sat	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
			WinterDesign	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.3	0.3	0.3	0.3	0.3	0.3	0.3
			Sun, Hol, Othe	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
Sch LBL Equipment.	Fraction	Through: 12/31	WD	0.17	0.17	0.17	0.17	0.17	0.17	0.17	1	1	1	1	1	1	1	1	1	1	0.17	0.17	0.17	0.17	0.17	0.17	0.17
			SummerDesignDay	0.17	0.17	0.17	0.17	0.17	0.17	0.17	1	1	1	1	1	1	1	1	1	1	0.17	0.17	0.17	0.17	0.17	0.17	0.17
			Sat	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17
			WinterDesign	0.17	0.17	0.17	0.17	0.17	0.17	0.17	1	1	1	1	1	1	1	1	1	1	0.17	0.17	0.17	0.17	0.17	0.17	0.17
			Sun, Hol, Othe	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17

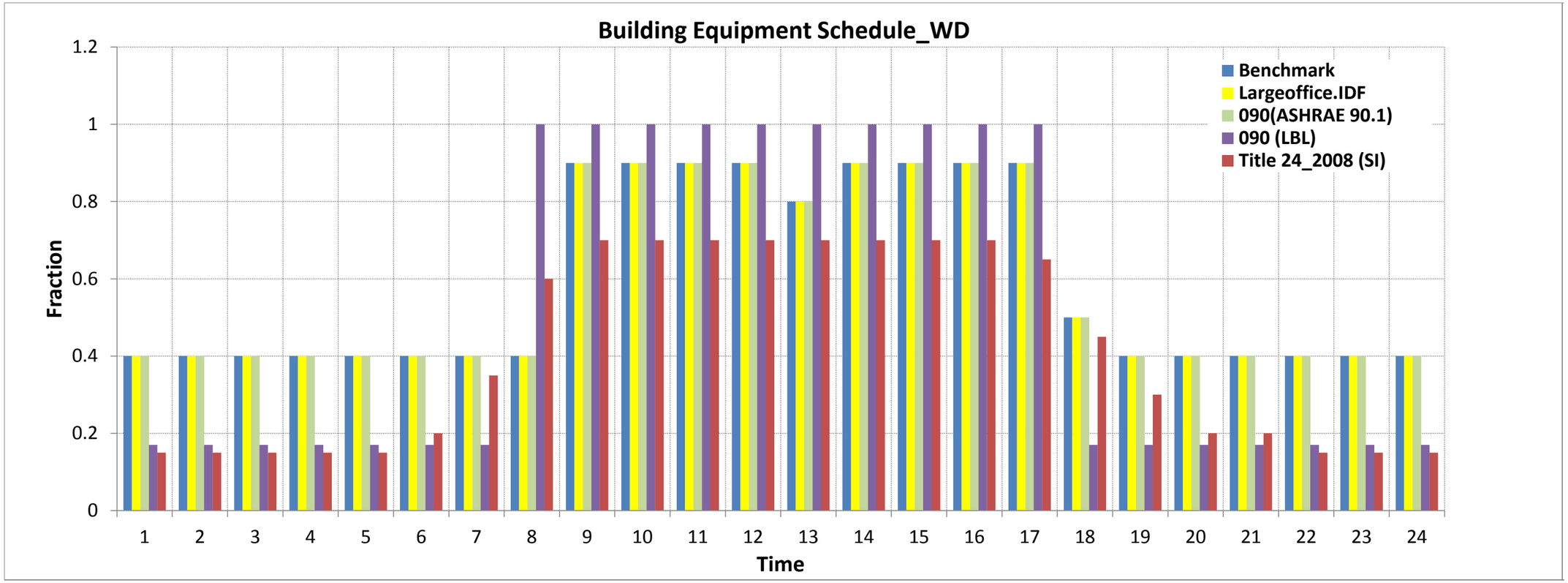
Title 24_2008 (SI)

Time	Type	Through	Day of Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
	Temperatur	Through: 12/31	WD	15.6	15.6	15.6	15.6	15.6	18.3	18.3	21.1	21.1	21.1	21.1	21.1	21.1	21.1	21.1	21.1	21.1	21.1	18.3	15.6	15.6	15.6	15.6	15.6
			Sat	15.6	15.6	15.6	15.6	15.6	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6
Heating			Sun	15.6	15.6	15.6	15.6	15.6	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	18.3	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6
	Temperatur	Through: 12/31	WD	25.0	25.0	25.0	25.0	25.0	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	25.0	25.0	25.0	25.0	25.0	25.0
			Sat	25.0	25.0	25.0	25.0	25.0	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	25.0	25.0	25.0	25.0	25.0	25.0
Cooling			Sun	25.0	25.0	25.0	25.0	25.0	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	22.8	25.0	25.0	25.0	25.0	25.0	25.0
	Fraction	Through: 12/31	WD	0.05	0.05	0.05	0.05	0.1	0.2	0.4	0.7	0.8	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.8	0.35	0.1	0.1	0.1	0.1	0.1
			Sat	0.05	0.05	0.05	0.05	0.05	0.1	0.15	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.2	0.2	0.2	0.15	0.1	0.1	0.1	0.1	0.1	0.1
Lights			Sun	0.05	0.05	0.05	0.05	0.05	0.1	0.1	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.1	0.1	0.1	0.05	0.05	0.05	0.05
	Fraction	Through 12/31	WD	0.15	0.15	0.15	0.15	0.15	0.2	0.35	0.6	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.65	0.45	0.3	0.2	0.2	0.15	0.15	0.15
			Sat	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.2	0.25	0.25	0.25	0.25	0.25	0.25	0.2	0.2	0.2	0.15	0.15	0.15	0.15	0.15	0.15	0.15
Equipment			Sun	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.15	0.15	0.15	0.15	0.15	0.15	0.15
	On/off	Through 12/31	WD	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0
			Sat	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0	0	0	0
Fan(%)			Sun	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Fraction	Through 12/31	WD	100	100	100	100	100	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	100	100	100	100
			Sat	100	100	100	100	100	0	0	0	0	0	0	0	0	0	0	100	100	100	100	100	100	100	100	100
Infiltration			Sun	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
	Fraction	Through 12/31	WD	0	0	0	0	0.05	0.1	0.25	0.65	0.65	0.65	0.65	0.6	0.6	0.65	0.65	0.65	0.65	0.4	0.25	0.1	0.05	0.05	0.05	0
			Sat	0		0	0	0	0	0.05	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.05	0.05	0.05	0	0	0	0
People			Sun	0	0	0	0	0	0	0	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0	0	0	0

Title 24_2008 (IP)

Time		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
	WD	60	60	60	60	60	65	65	70	70	70	70	70	70	70	70	70	70	70	65	60	60	60	60	60
	Sat	60	60	60	60	60	65	65	65	65	65	65	65	65	65	65	65	60	60	60	60	60	60	60	60
Heating	Sun	60	60	60	60	60	65	65	65	65	65	65	65	65	65	65	65	60	60	60	60	60	60	60	60
	WD	77	77	77	77	77	73	73	73	73	73	73	73	73	73	73	73	73	73	77	77	77	77	77	77
	Sat	77	77	77	77	77	73	73	73	73	73	73	73	73	73	73	73	73	73	77	77	77	77	77	77
Cooling	Sun	77	77	77	77	77	73	73	73	73	73	73	73	73	73	73	73	73	73	77	77	77	77	77	77
	WD	5	5	5	5	10	20	40	70	80	85	85	85	85	85	85	85	85	80	35	10	10	10	10	10
	Sat	5	5	5	5	5	10	15	25	25	25	25	25	25	25	20	20	20	15	10	10	10	10	10	10
Lights	Sun	5	5	5	5	5	10	10	15	15	15	15	15	15	15	15	15	15	10	10	10	5	5	5	5
	WD	15	15	15	15	15	20	35	60	70	70	70	70	70	70	70	70	65	45	30	20	20	15	15	15
	Sat	15	15	15	15	15	15	15	20	25	25	25	25	25	25	20	20	20	15	15	15	15	15	15	15
Equipment	Sun	15	15	15	15	15	15	15	20	20	20	20	20	20	20	20	20	20	15	15	15	15	15	15	15
	WD	off	off	off	off	off	on	on	on	on	on	on	on	on	on	on	on	on	on	on	on	off	off	off	off
	Sat	off	off	off	off	off	on	on	on	on	on	on	on	on	on	on	off	off	off	off	off	off	off	off	off
Fan(%)	Sun	off	off	off	off	off	off	off	off	off	off	off	off	off	off	off	off	off	off	off	off	off	off	off	off
	WD	100	100	100	100	100	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	100	100	100	100
	Sat	100	100	100	100	100	0	0	0	0	0	0	0	0	0	0	100	100	100	100	100	100	100	100	100
Infiltration	Sun	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
	WD	0	0	0	0	5	10	25	65	65	65	65	60	60	65	65	65	65	40	25	10	5	5	5	0
	Sat	0	0	0	0	0	0	5	15	15	15	15	15	15	15	15	15	15	5	5	5	0	0	0	0
People	Sun	0	0	0	0	0	0	0	5	5	5	5	5	5	5	5	5	5	5	5	5	0	0	0	0


Building Light Schedule_WD
Fraction
Time
Benchmark
Largeoffice.IDF
090(ASHRAE 90.1)
090 (LBL)
Title 24_2008 (SI)
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24


Building Equipment Schedule_WD
Benchmark
Largeoffice.IDF
090(ASHRAE 90.1)
090 (LBL)
Title 24_2008 (SI)
Fraction
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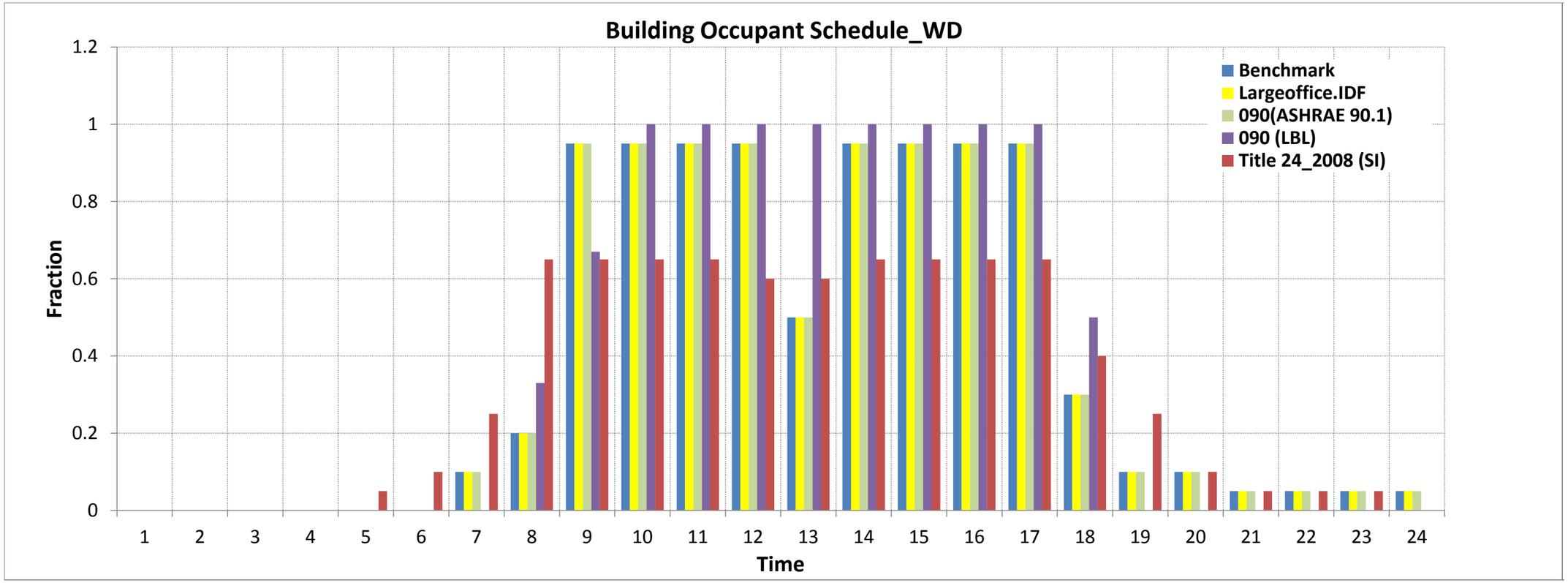
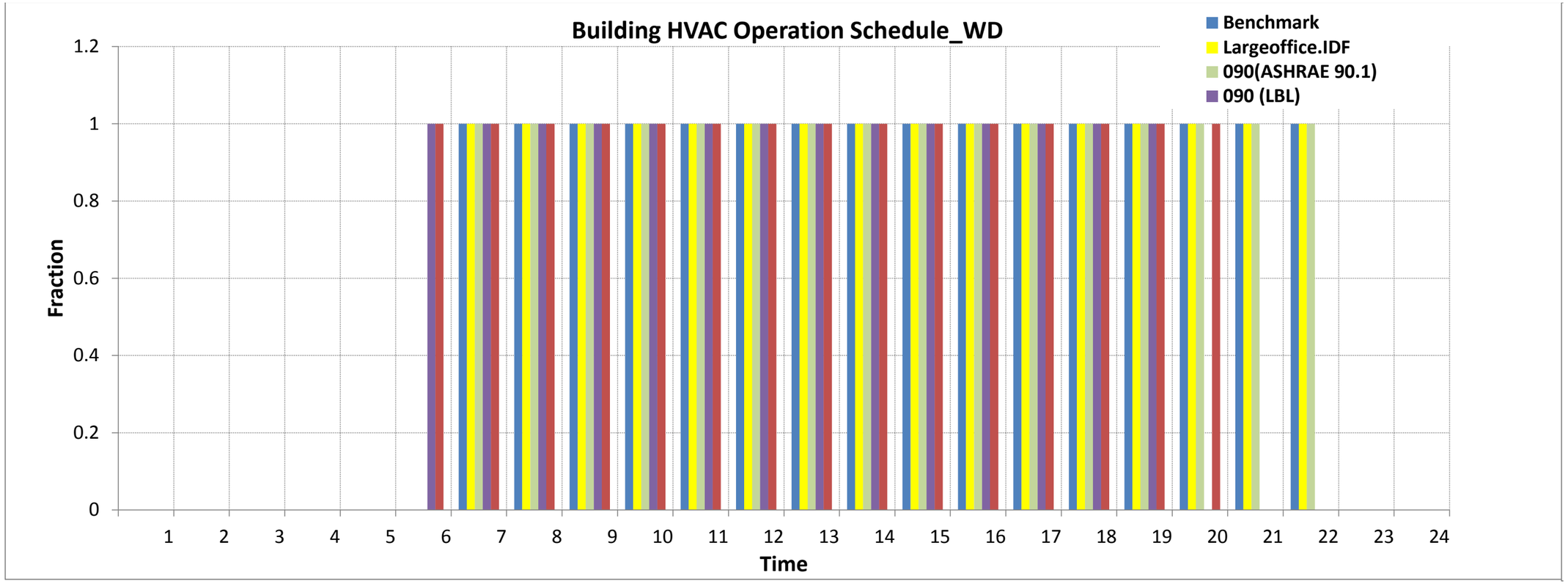
0.8
0.6
0.4
0.2
0
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24
Time


Building Occupant Schedule_WD
Fraction
Time
Benchmark
Largeoffice.IDF
090(ASHRAE 90.1)
090 (LBL)
Title 24_2008 (SI)
1.2
1
0.8
0.6
0.4
0.2
0
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24


Building HVAC Operation Schedule_WD
Benchmark
Largeoffice.IDF
090(ASHRAE 90.1)
090 (LBL)
Fraction
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1
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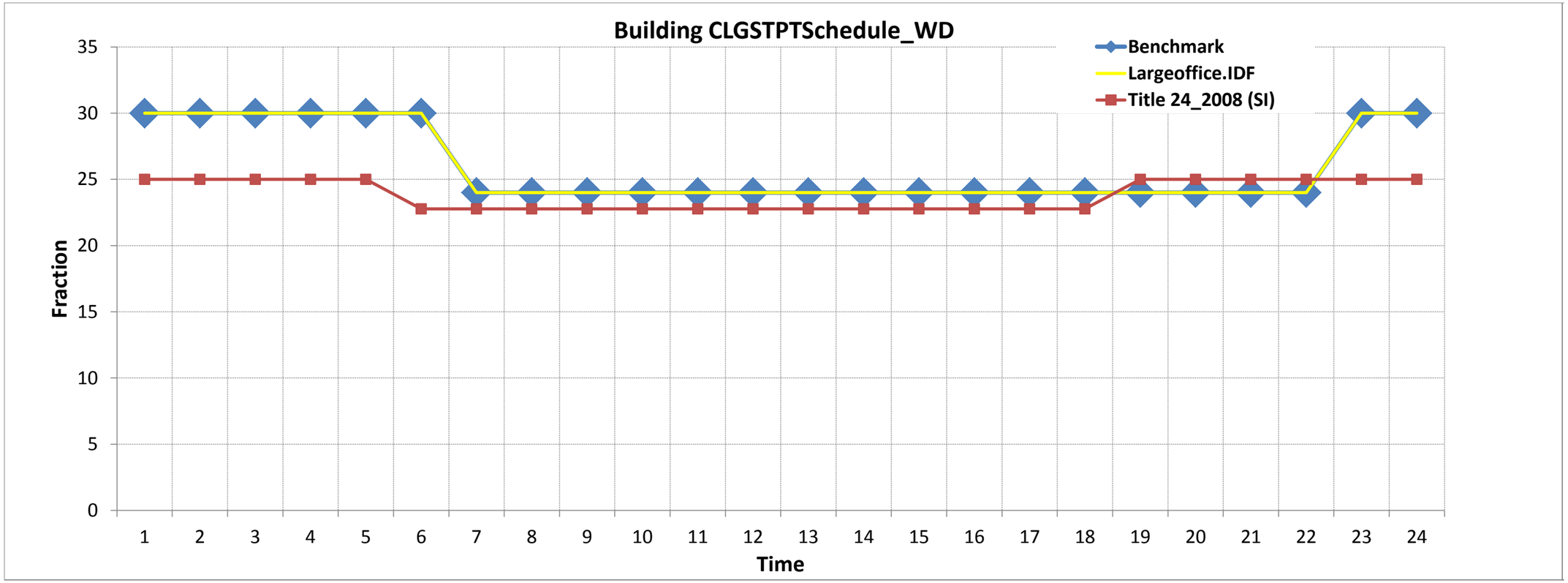
0.6
0.4
0.2
0
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24
Time


Building CLGSTPTSchedule_WD
Benchmark
Largeoffice.IDF
Title 24_2008 (SI)
Fraction
35
30
25
20
15
10
5
0
1
2
3
4
5
6
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10
11
12
13
14
15
16
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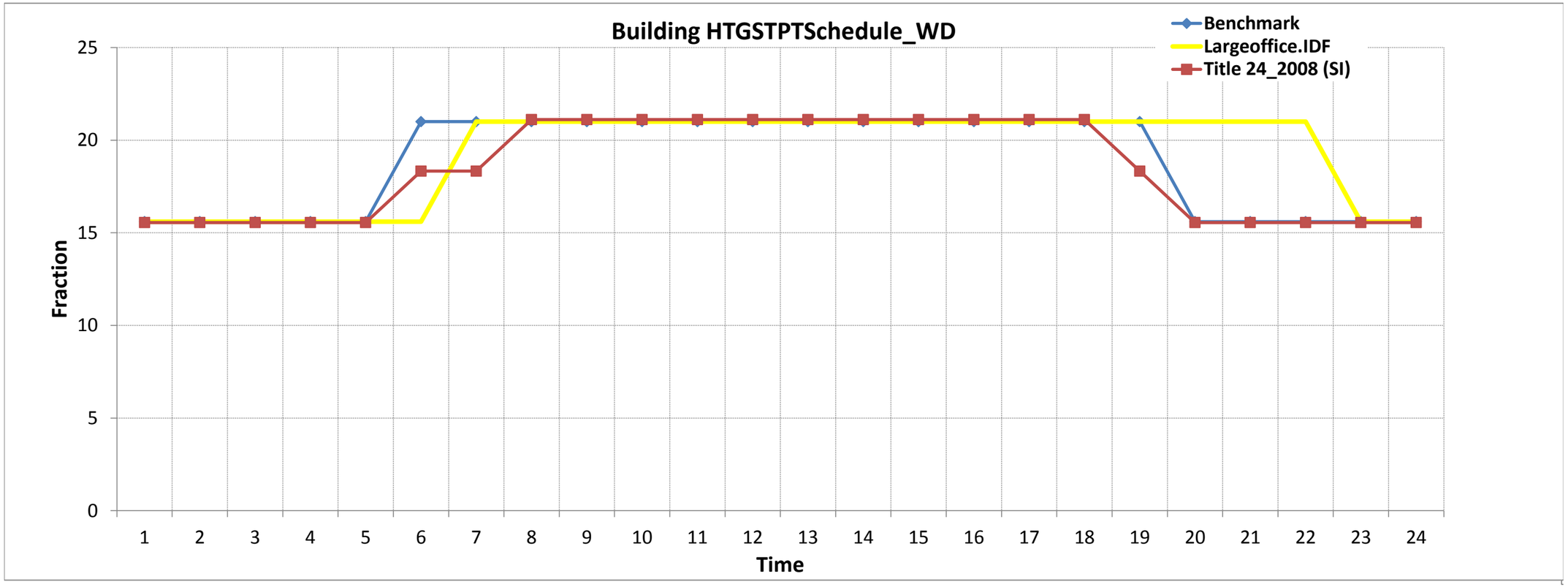
18
19
20
21
22
23
24
Time


Building HTGSTPTSchedule_WD
Benchmark
Largeoffice.IDF
Title 24_2008 (SI)
Fraction
0
5
10
15
20
25
1
2
3
4
5
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10
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14
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16
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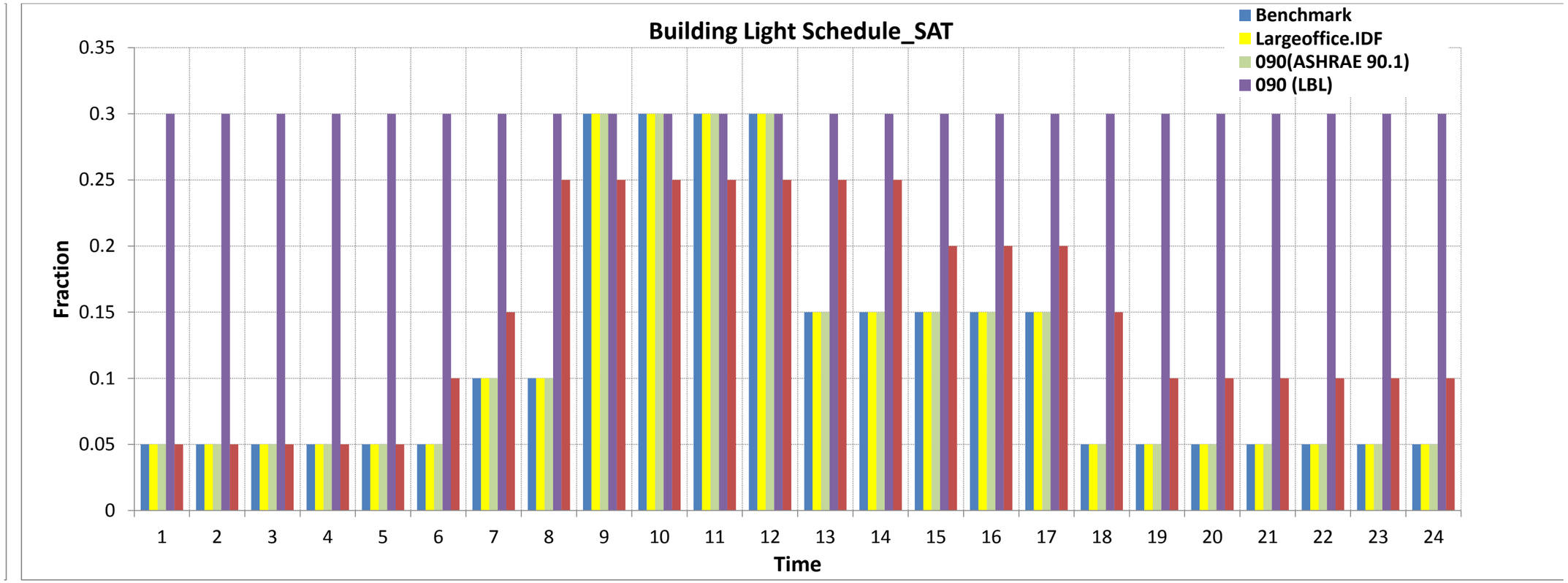
20
21
22
23
24
Time


Building Light Schedule_SAT
Benchmark
Largeoffice.IDF
090(ASHRAE 90.1)
090 (LBL)
Fraction
0.35
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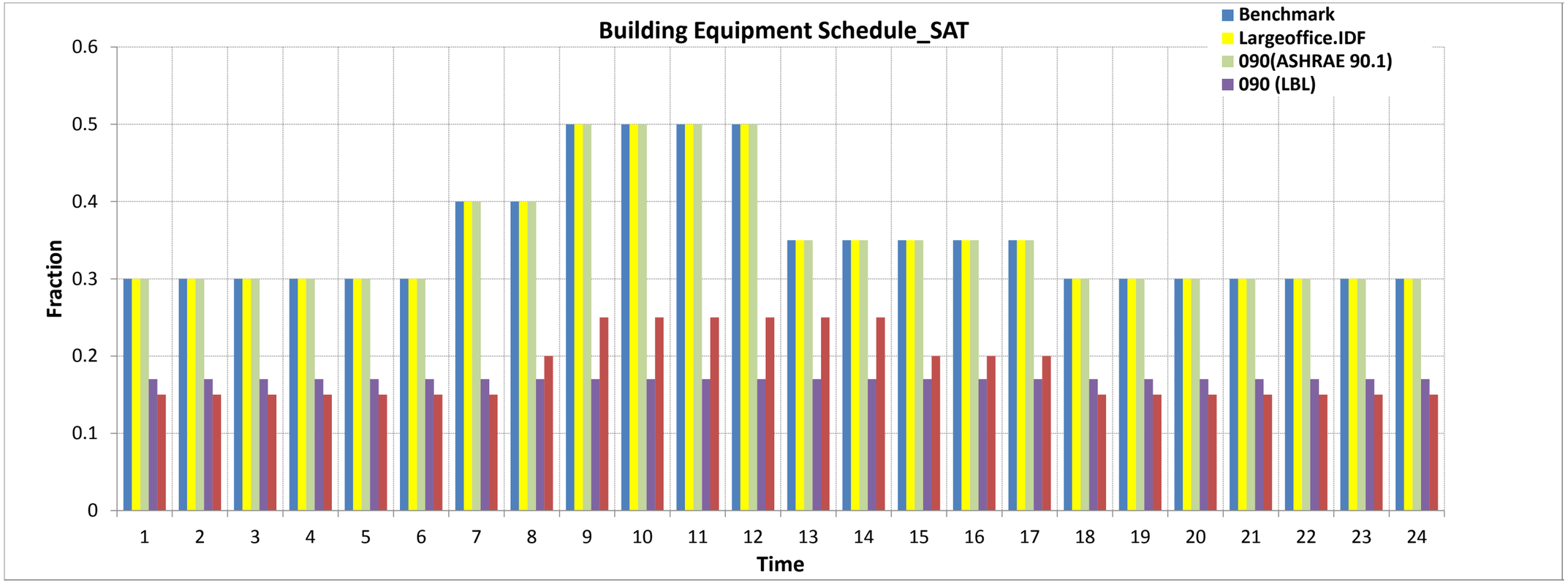
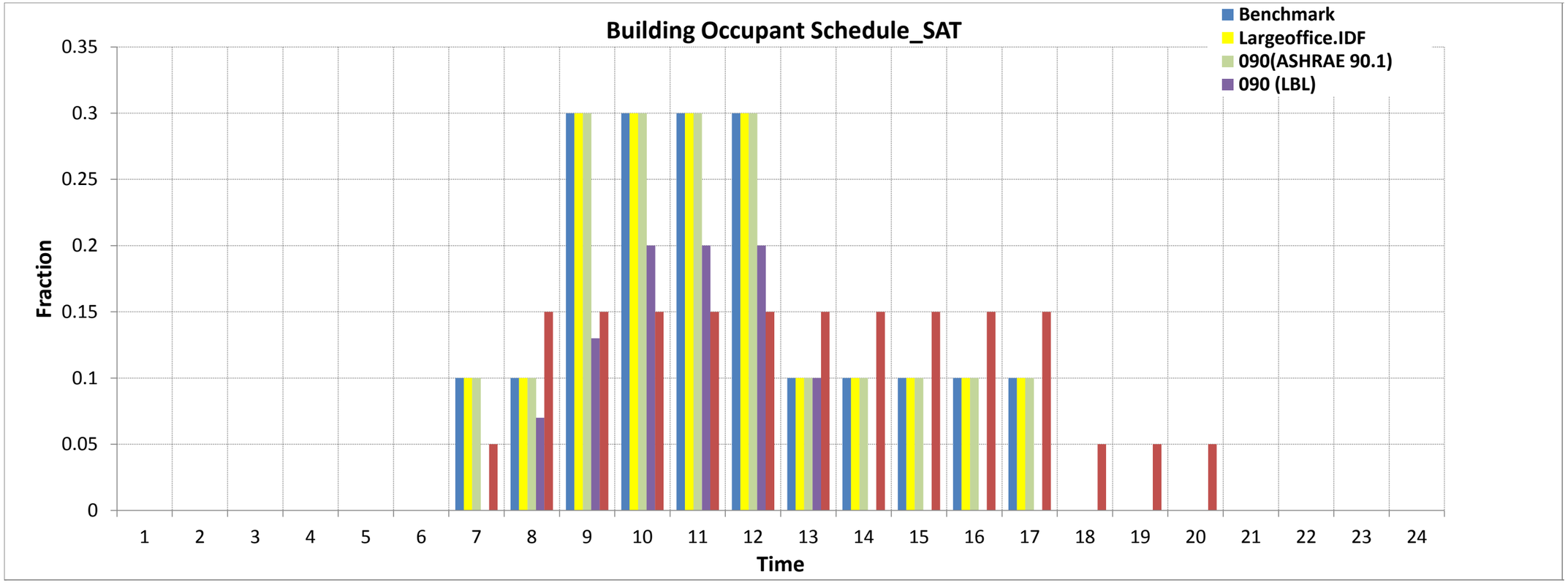
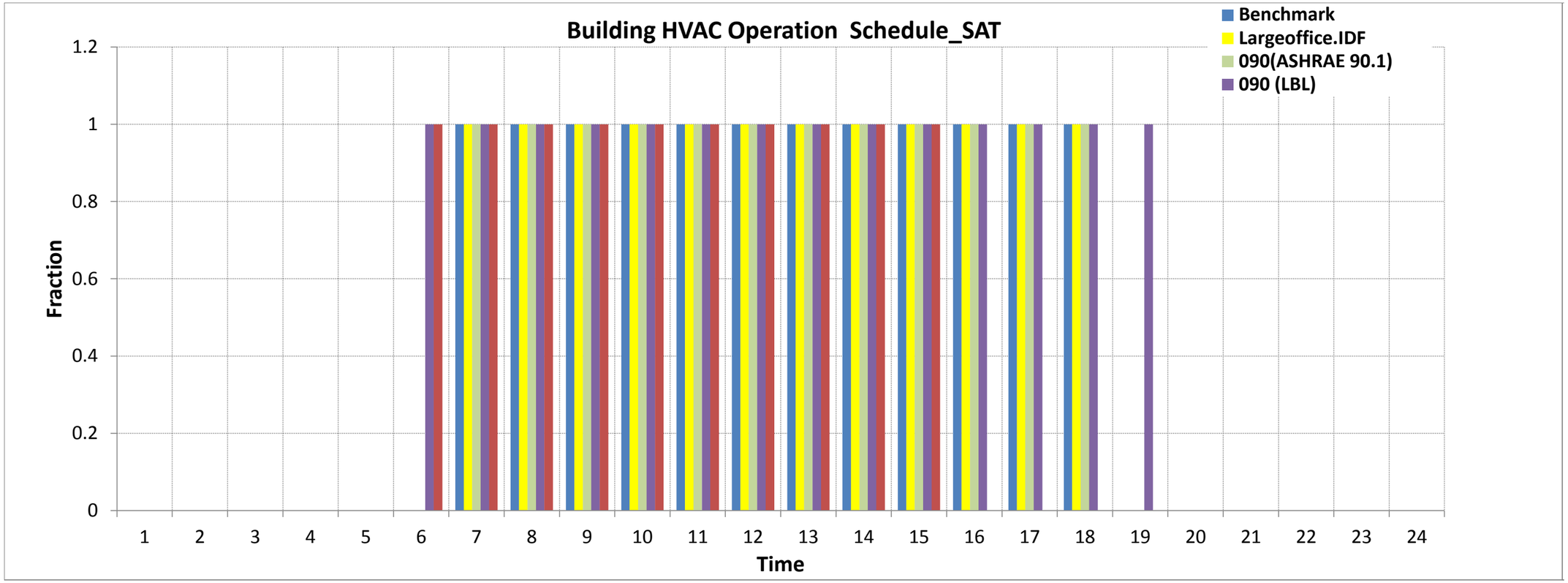
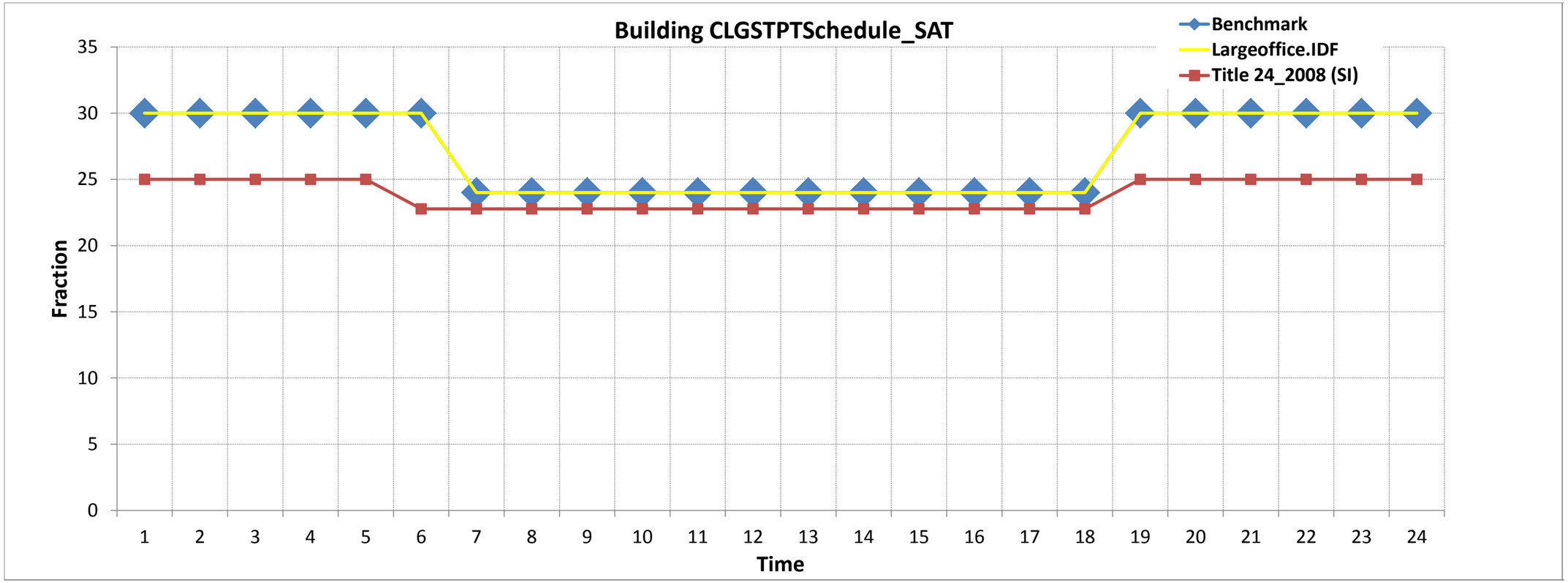
0.25
0.2
0.15
0.1
0.05
0
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24
Time


Building Equipment Schedule_SAT
Benchmark
Largeoffice.IDF
090(ASHRAE 90.1)
090 (LBL)
Fraction
0.6
0.5
0.4
0.3
0.2
0.1
0
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24
Time


Building Occupant Schedule_SAT
Benchmark
Largeoffice.IDF
090(ASHRAE 90.1)
090 (LBL)
Fraction
0.35
0.3
0.25
0.2
0.15
0.1
0.05
0
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24
Time


Building HVAC Operation Schedule_SAT
Benchmark
Largeoffice.IDF
090(ASHRAE 90.1)
090 (LBL)
Fraction
1.2
1
0.8
0.6
0.4
0.2
0
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24
Time


Building CLGSTPTSchedule_SAT
Benchmark
Largeoffice.IDF
Title 24_2008 (SI)
Fraction
35
30
25
20
15
10
5
0
1
2
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6
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11
12
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14
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18
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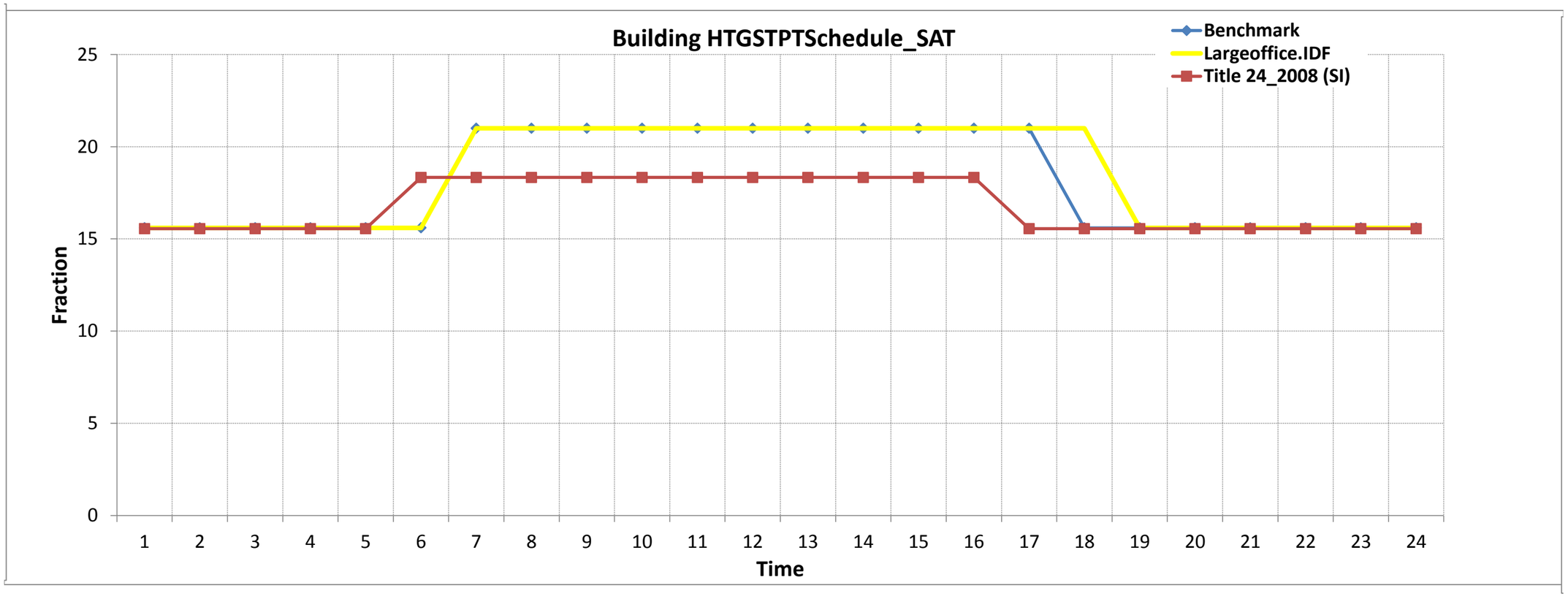
20
21
22
23
24
Time


Building HTGSTPTSchedule_SAT
Benchmark
Largeoffice.IDF
Title 24_2008 (SI)
Fraction
0
5
10
15
20
25
1
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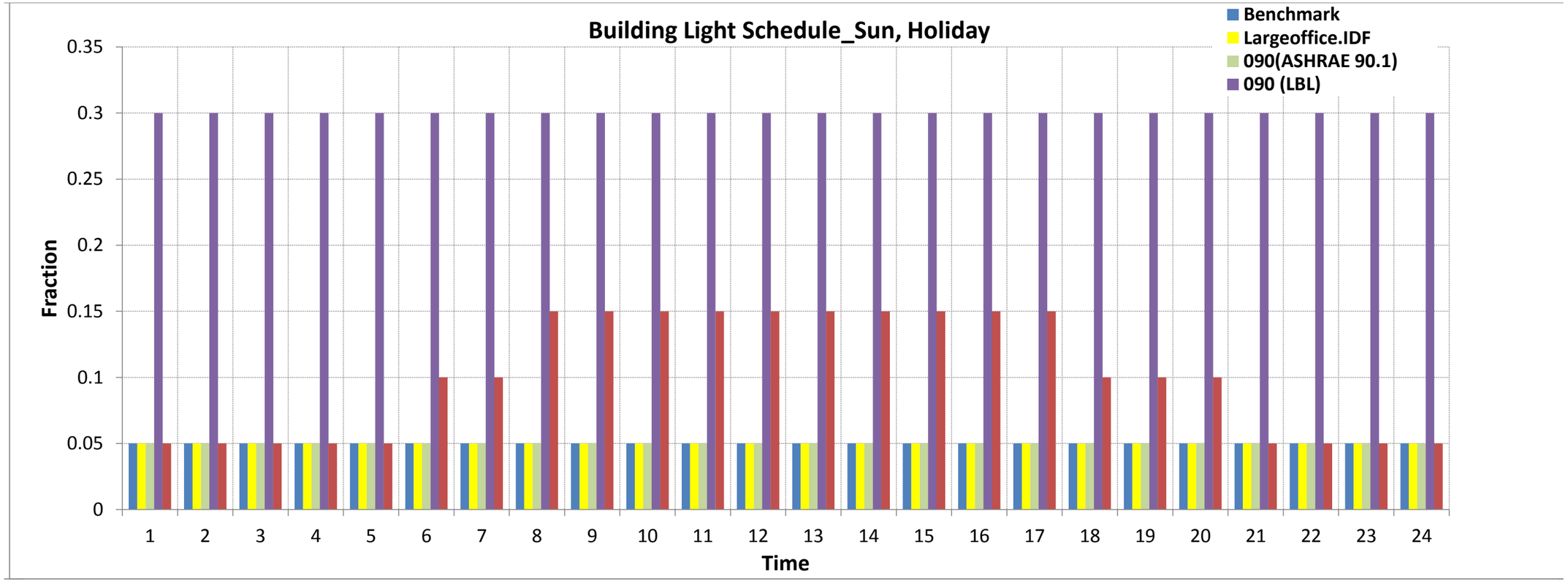
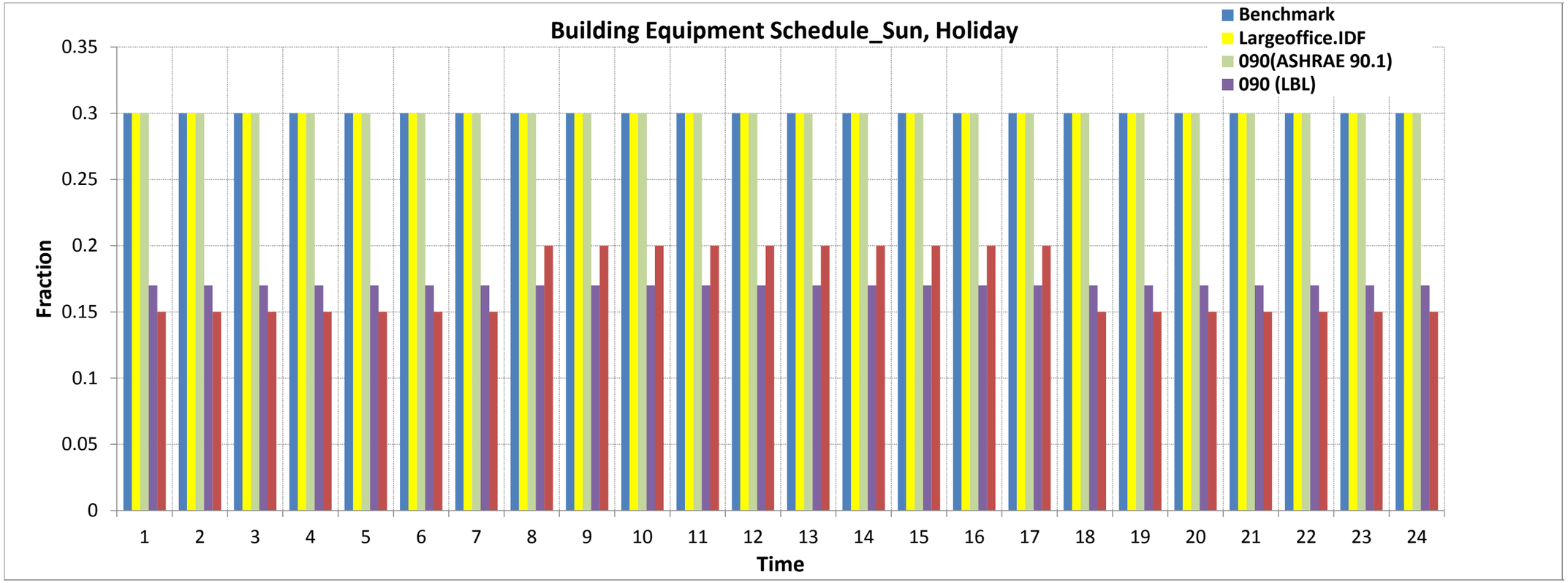
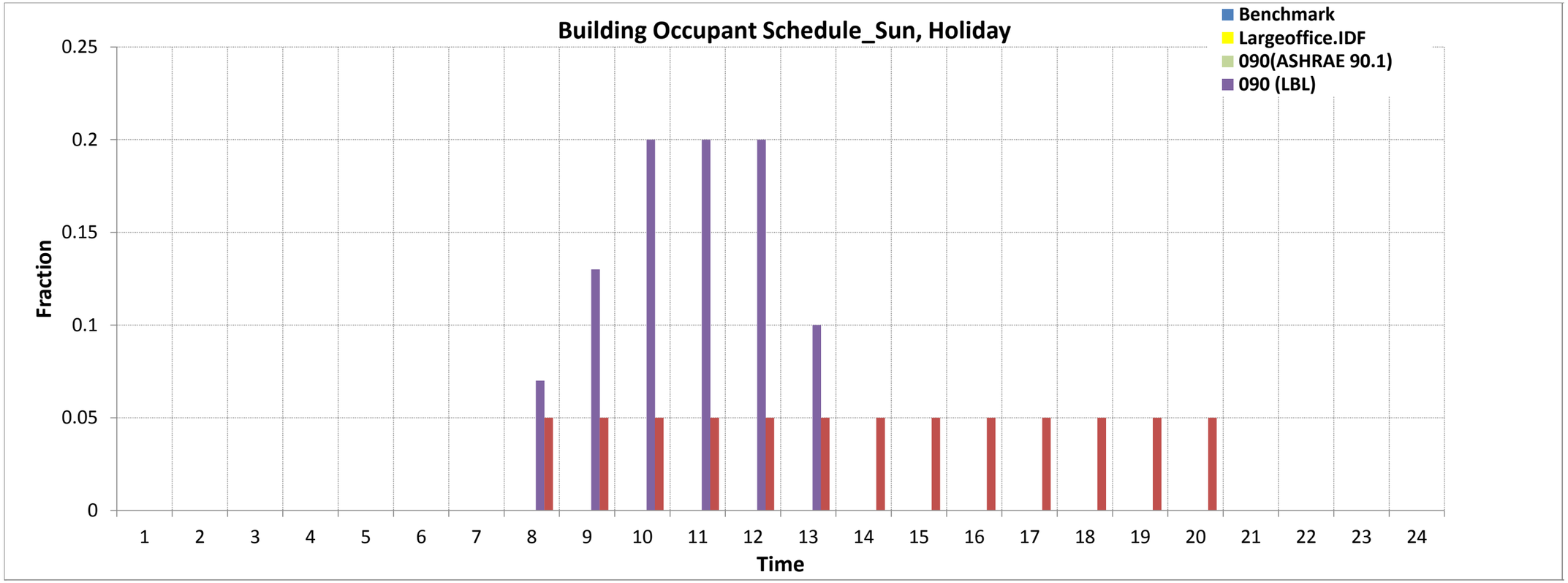
20
21
22
23
24
Time


Building Light Schedule_Sun, Holiday
Benchmark
Largeoffice.IDF
090(ASHRAE 90.1)
090 (LBL)
Fraction
0.35
0.3
0.25
0.2
0.15
0.1
0.05
0
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24
Time


Building Equipment Schedule_Sun, Holiday
Benchmark
Largeoffice.IDF
090(ASHRAE 90.1)
090 (LBL)
Fraction
0.35
0.3
0.25
0.2
0.15
0.1
0.05
0
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24
Time


Building Occupant Schedule_Sun, Holiday
Benchmark
Largeoffice.IDF
090(ASHRAE 90.1)
090 (LBL)
Fraction
0.25
0.2
0.15
0.1
0.05
0
1
2
3
4
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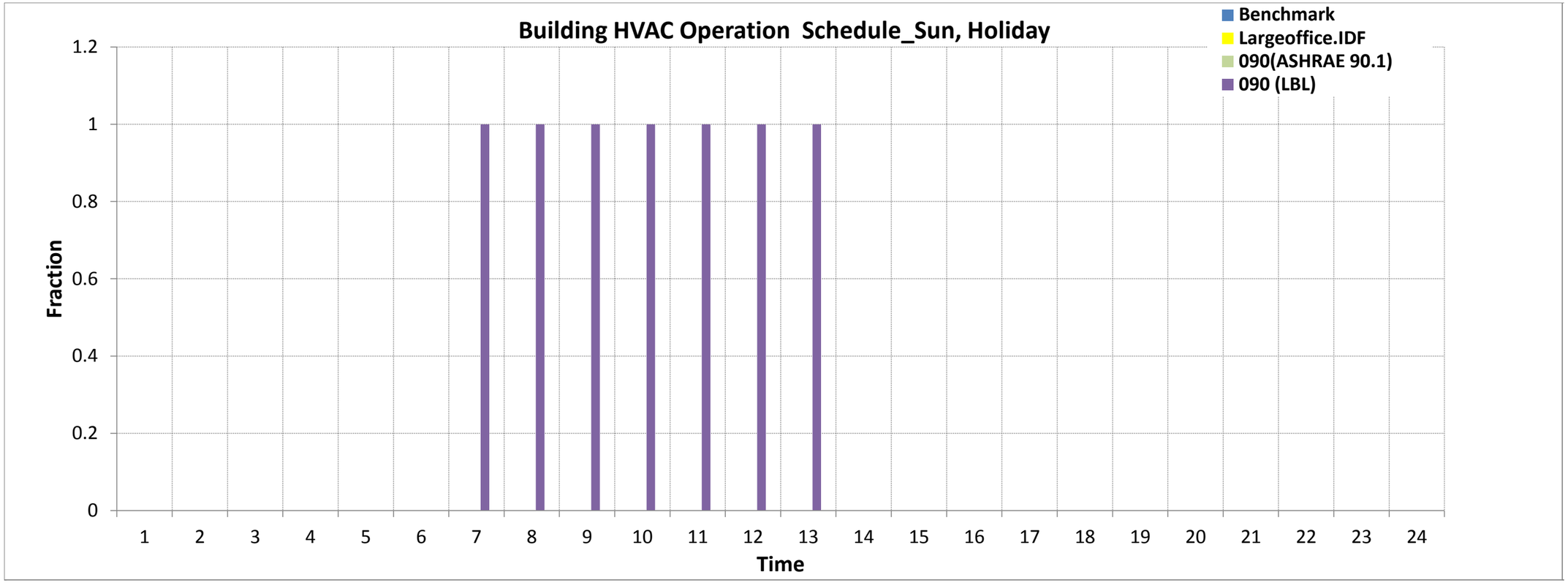
21
22
23
24
Time


Building HVAC Operation Schedule_Sun, Holiday
Benchmark
Largeoffice.IDF
090(ASHRAE 90.1)
090 (LBL)
Fraction
1.2
1
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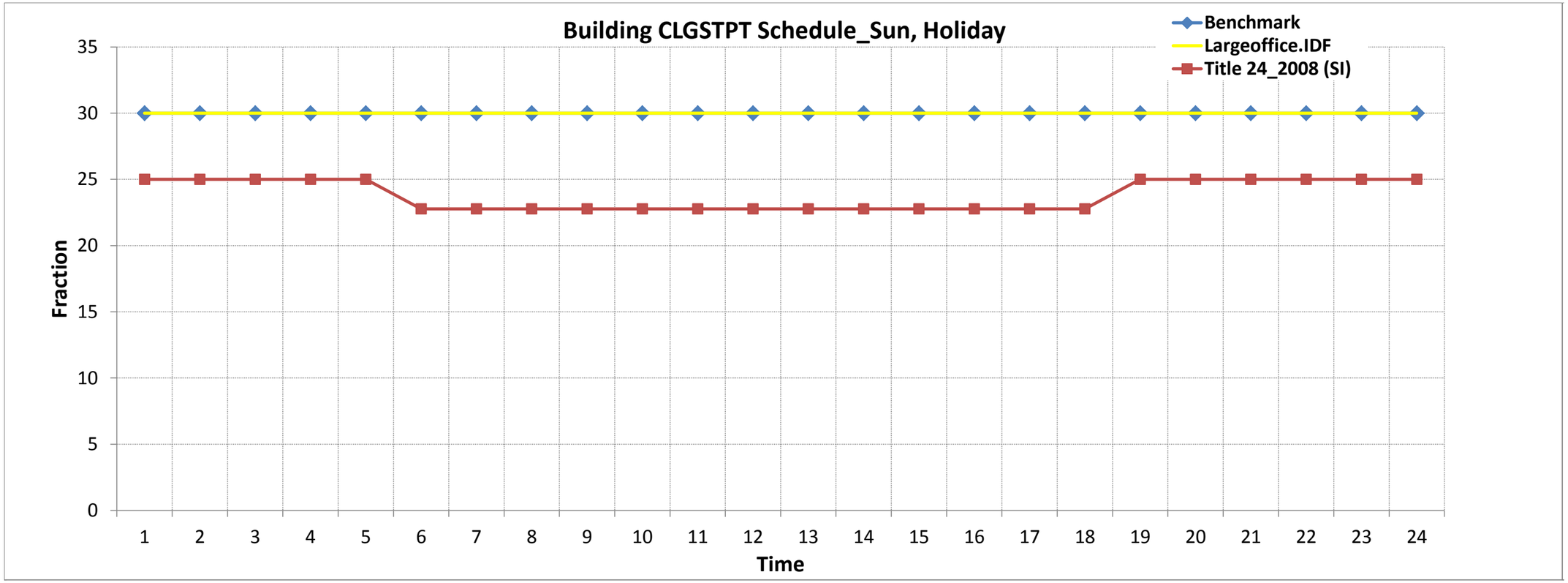
0.6
0.4
0.2
0
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24
Time


Building CLGSTPT Schedule_Sun, Holiday
Benchmark
Largeoffice.IDF
Title 24_2008 (SI)
Fraction
35
30
25
20
15
10
5
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
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18
19
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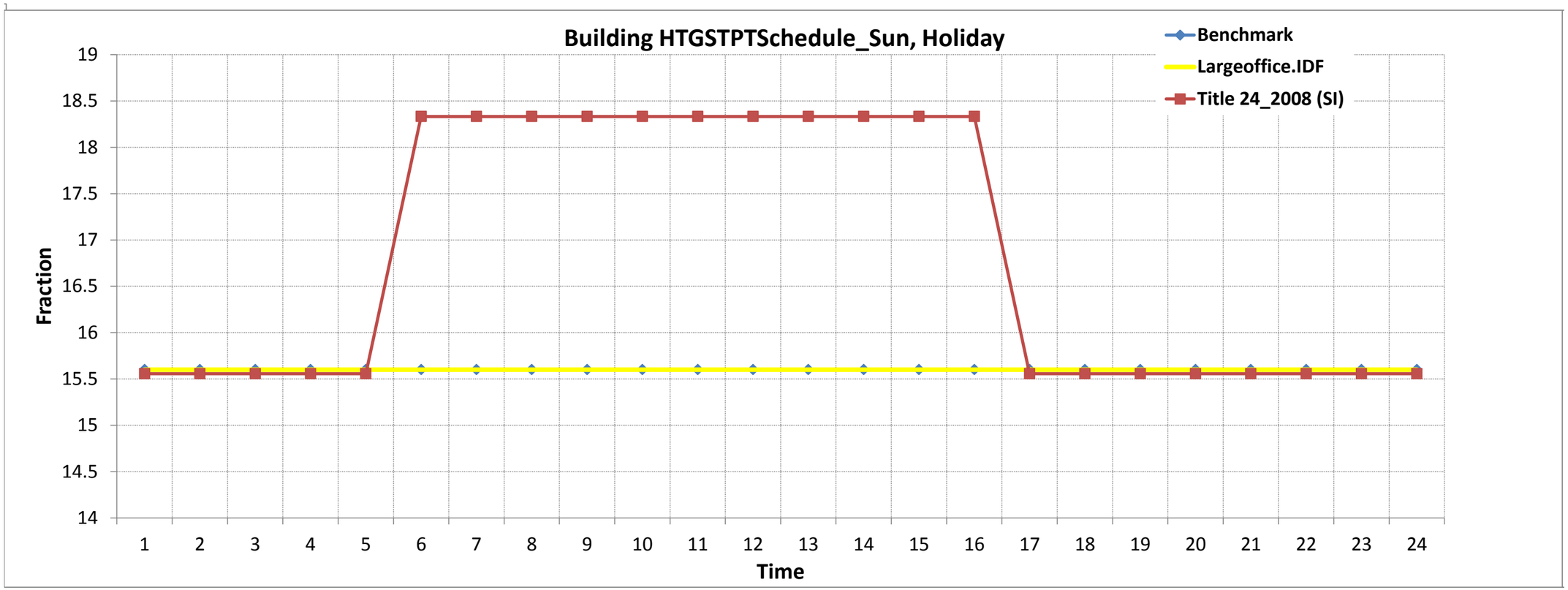
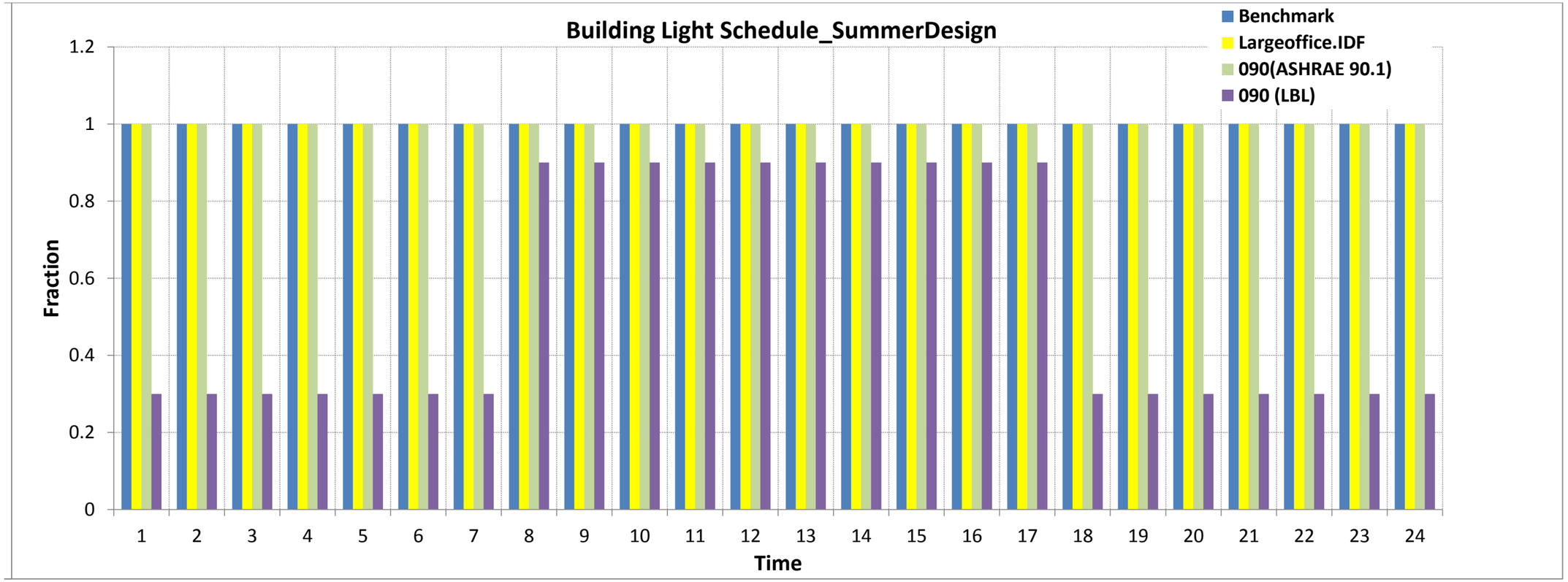
21
22
23
24
Time


Building HTGSTPTSchedule_Sun, Holiday
Benchmark
Largeoffice.IDF
Title 24_2008 (SI)
Fraction
19
18.5
18
17.5
17
16.5
16
15.5
15
14.5
14
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24
Time


Building Light Schedule_SummerDesign
Benchmark
Largeoffice.IDF
090(ASHRAE 90.1)
090 (LBL)
Fraction
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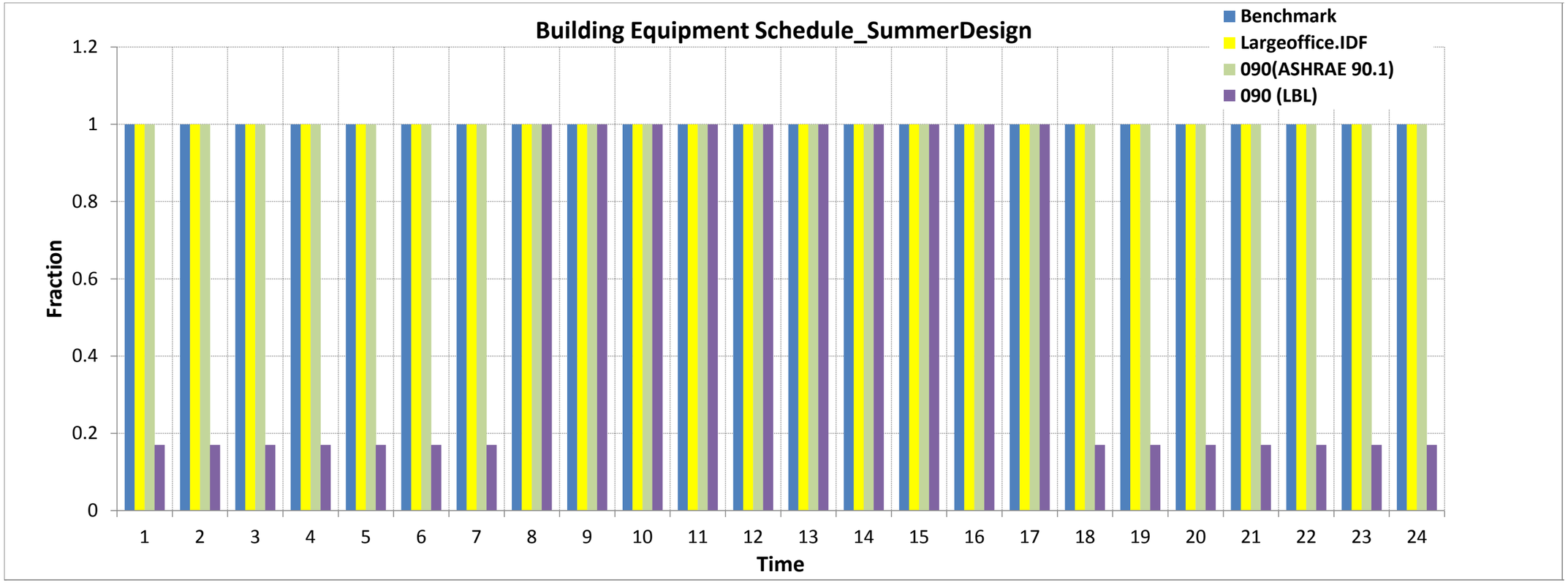
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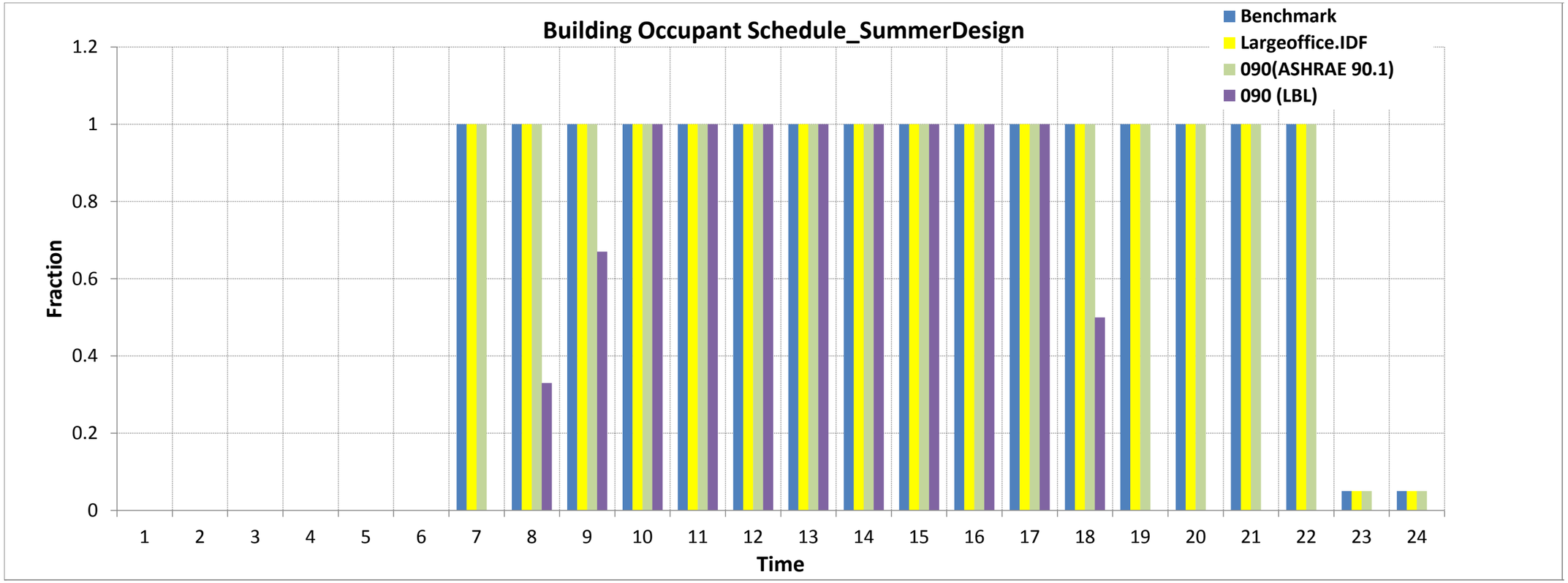
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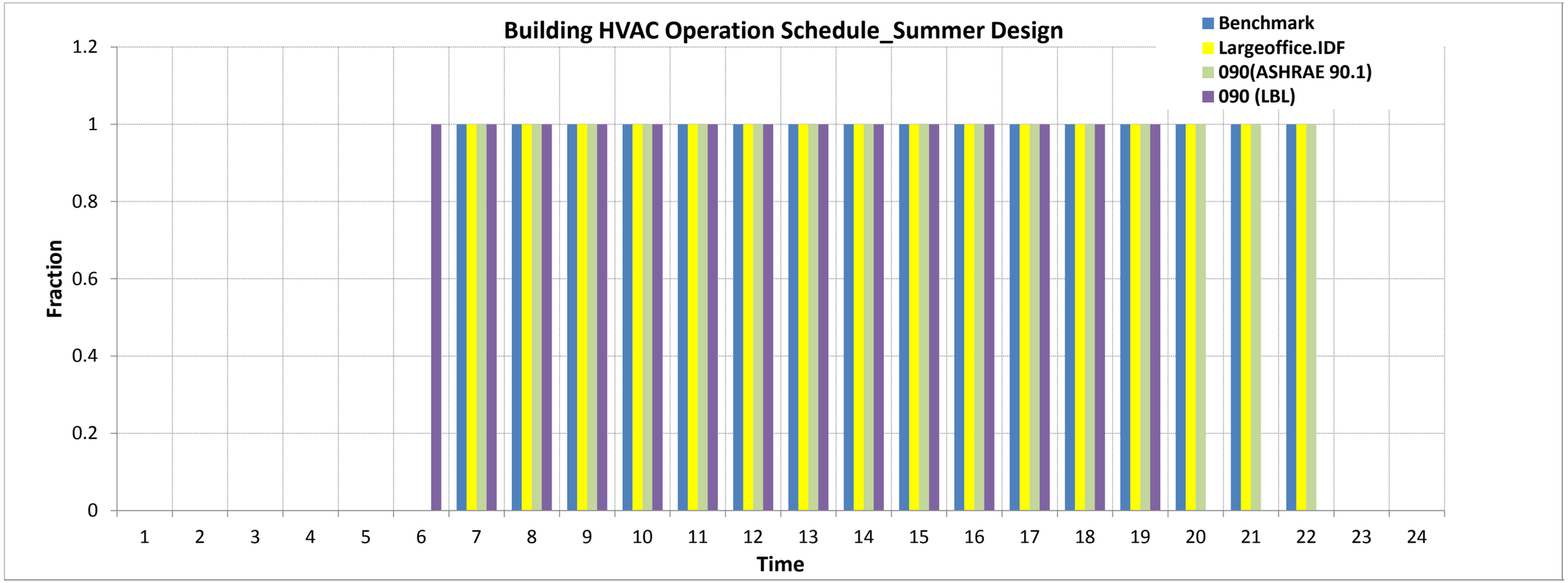
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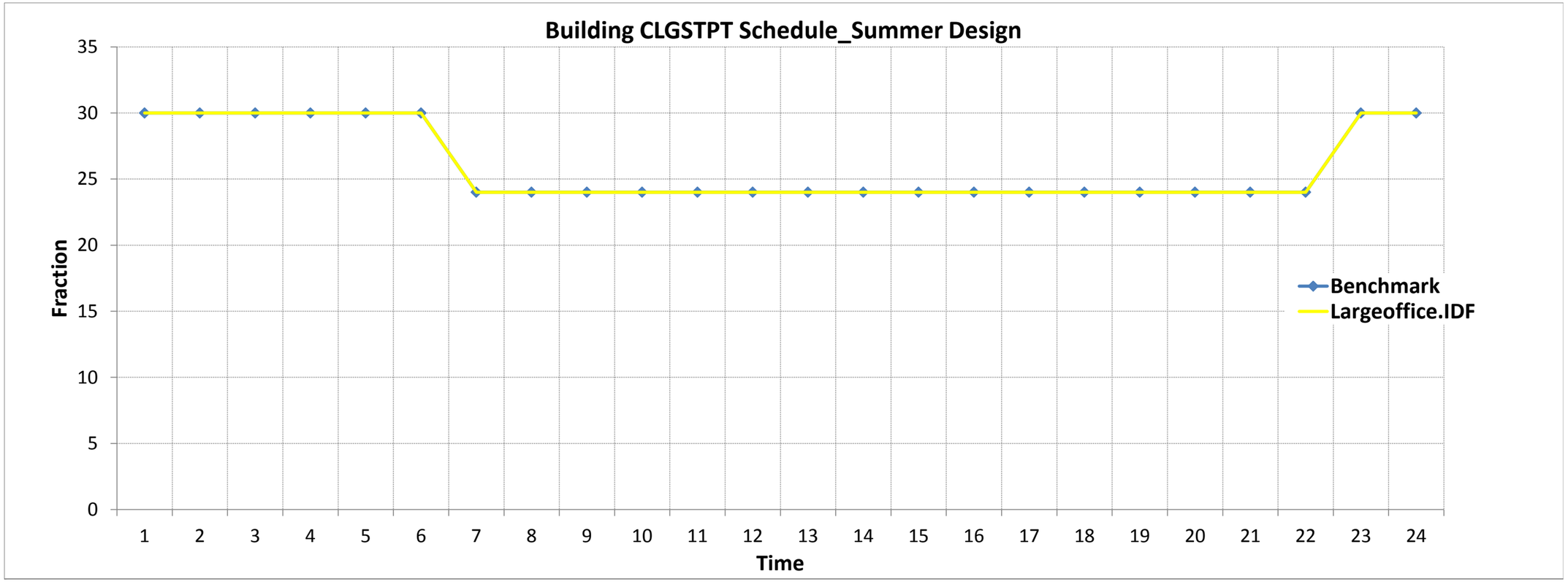
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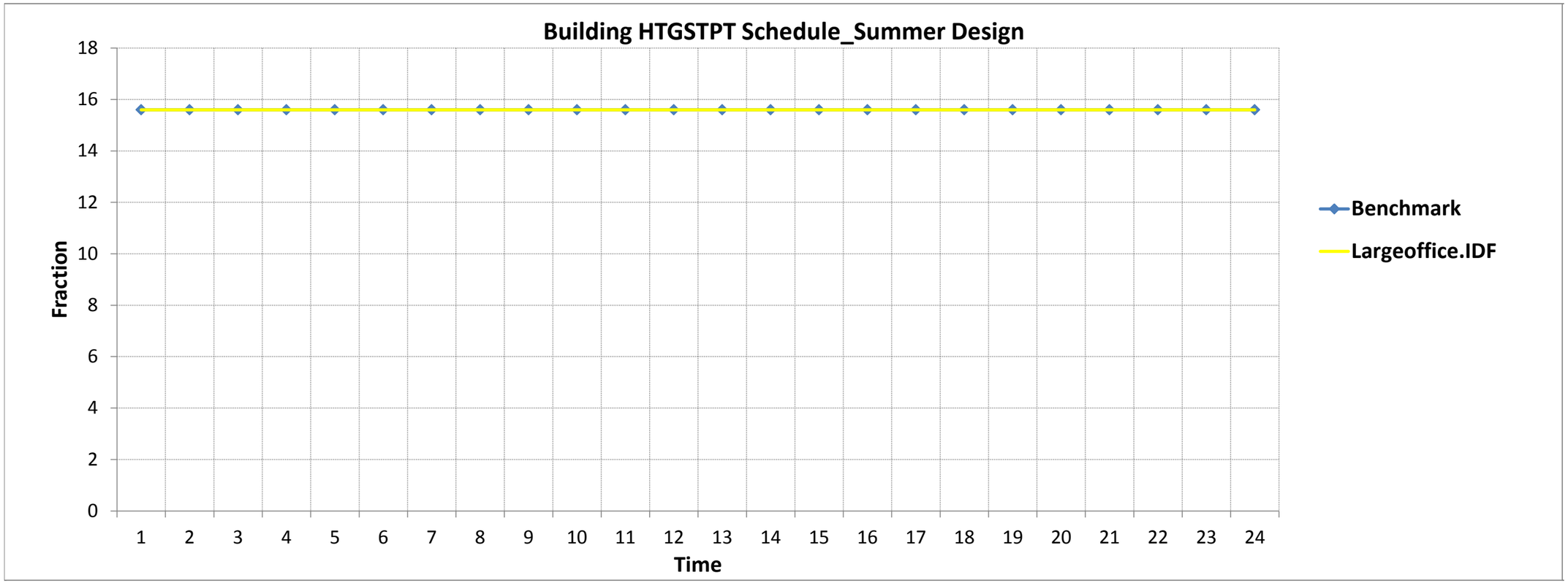
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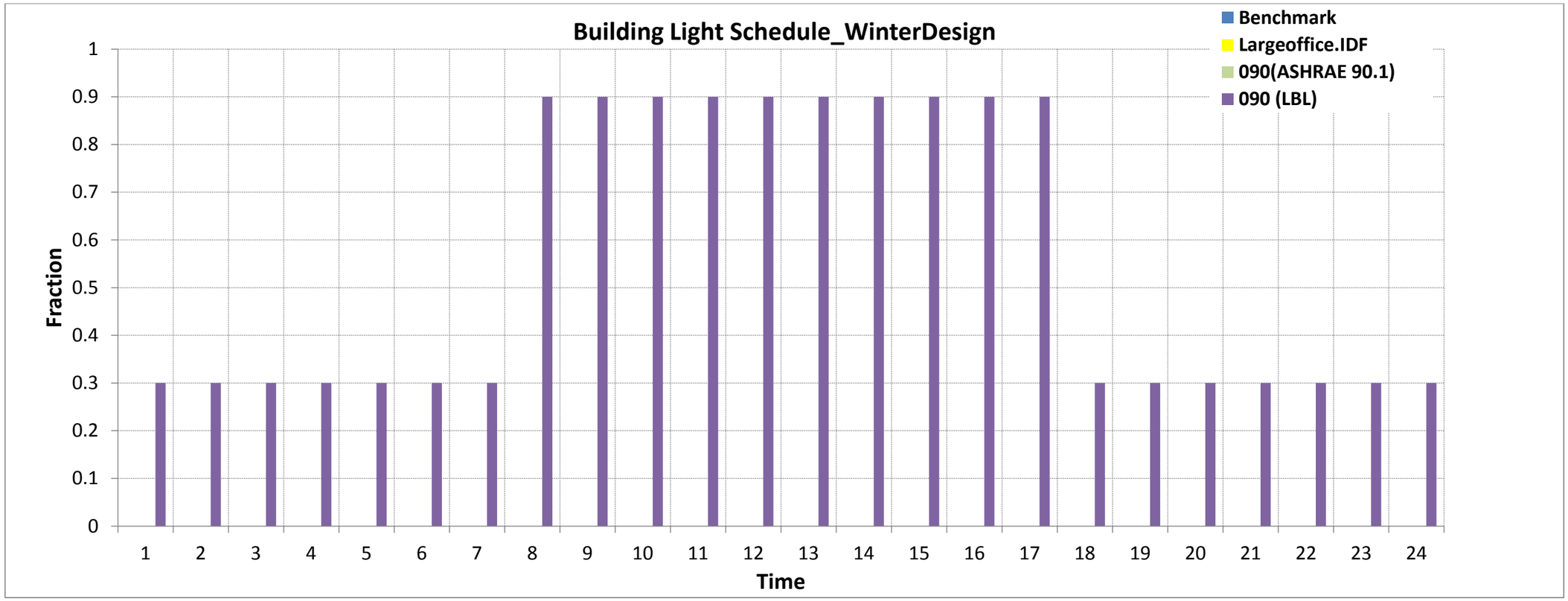
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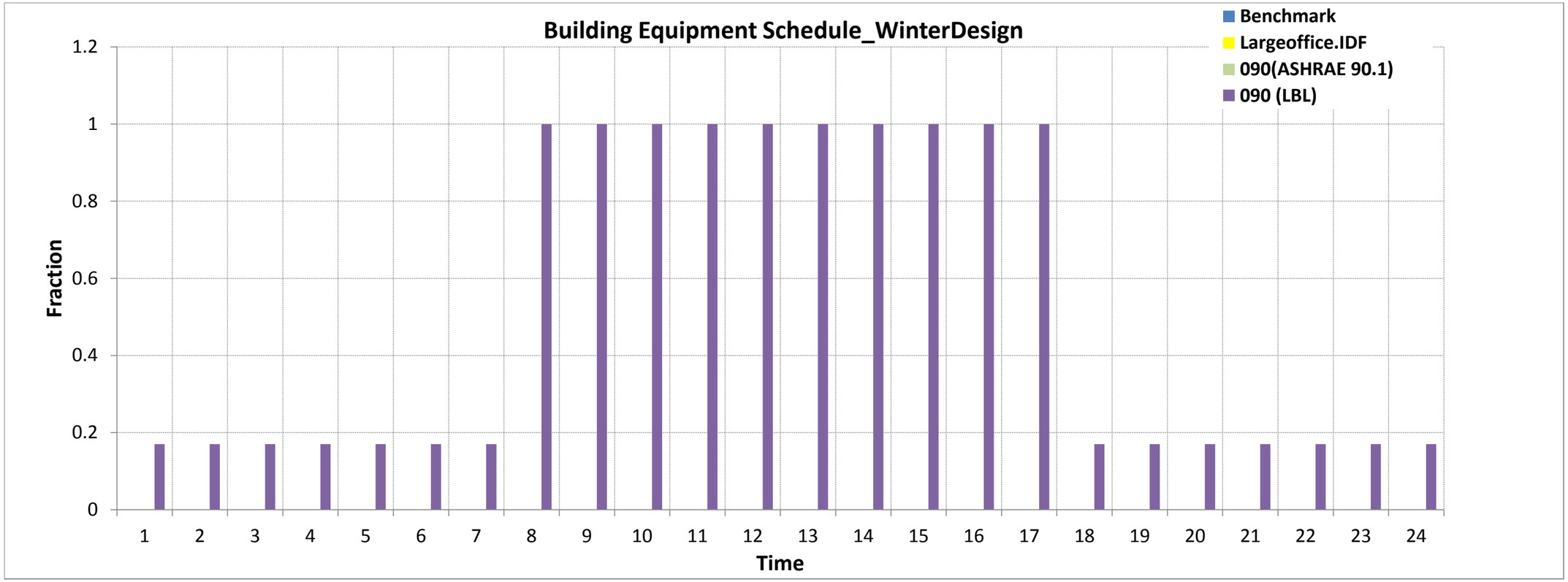
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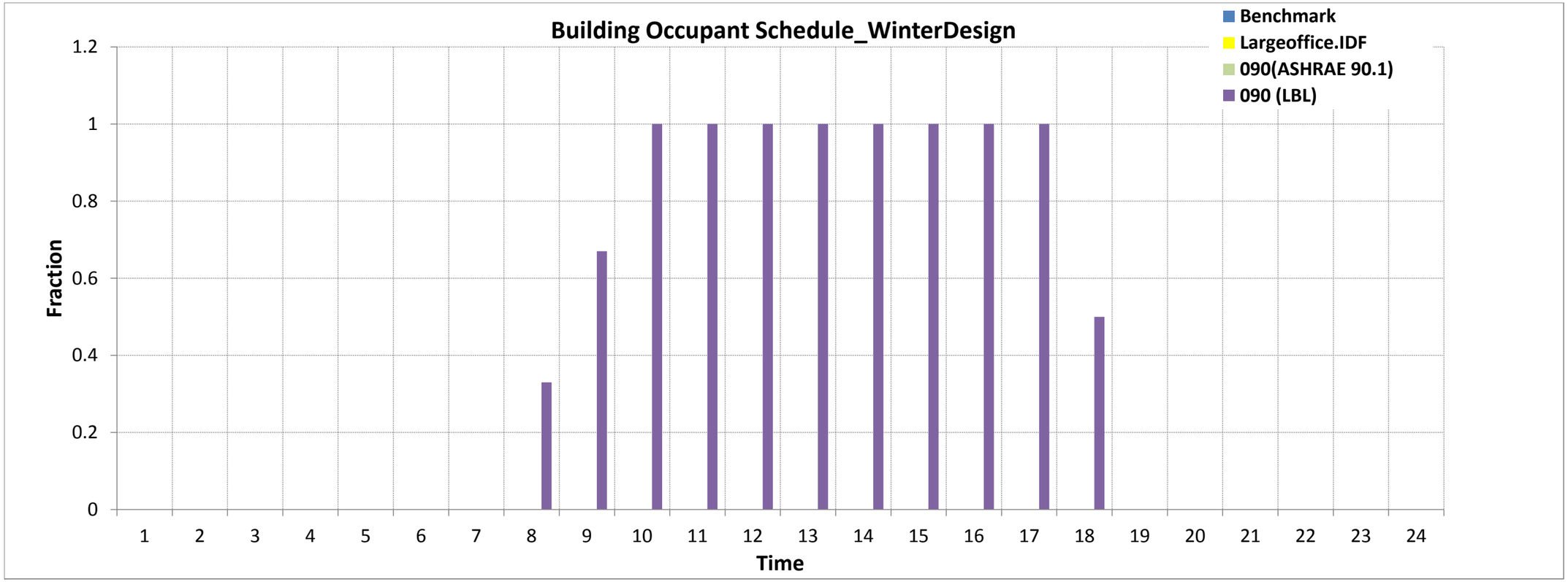
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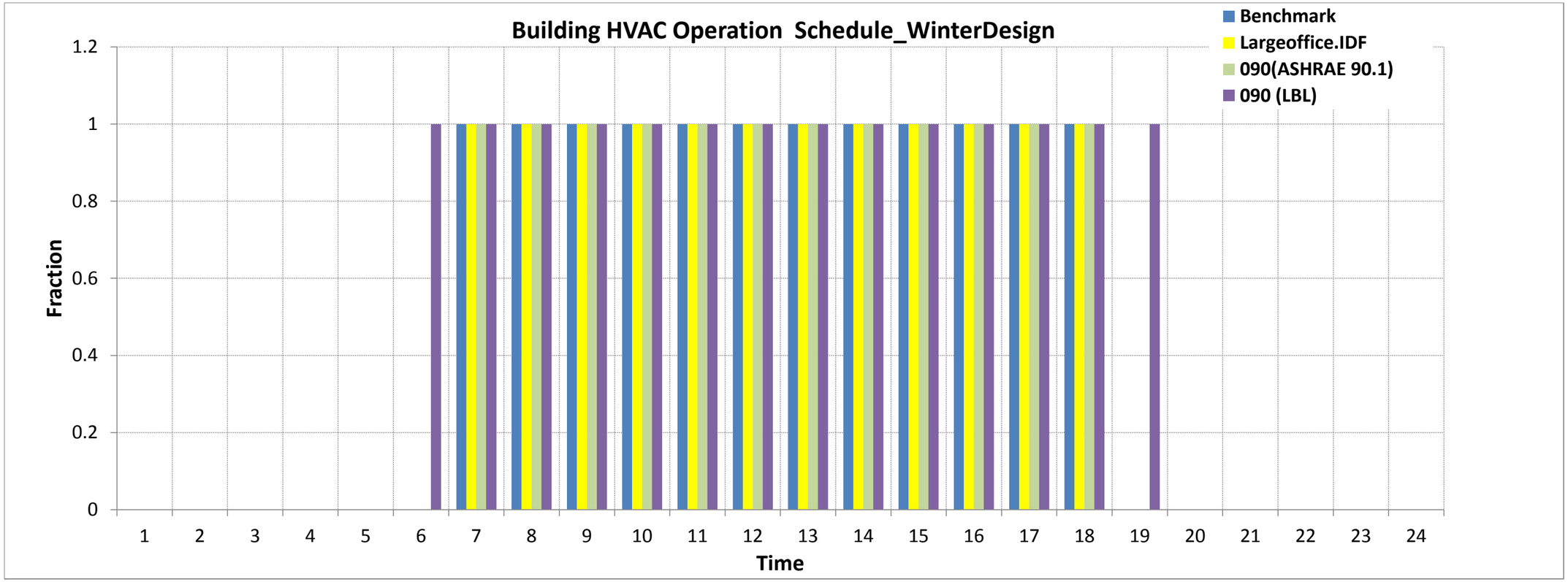
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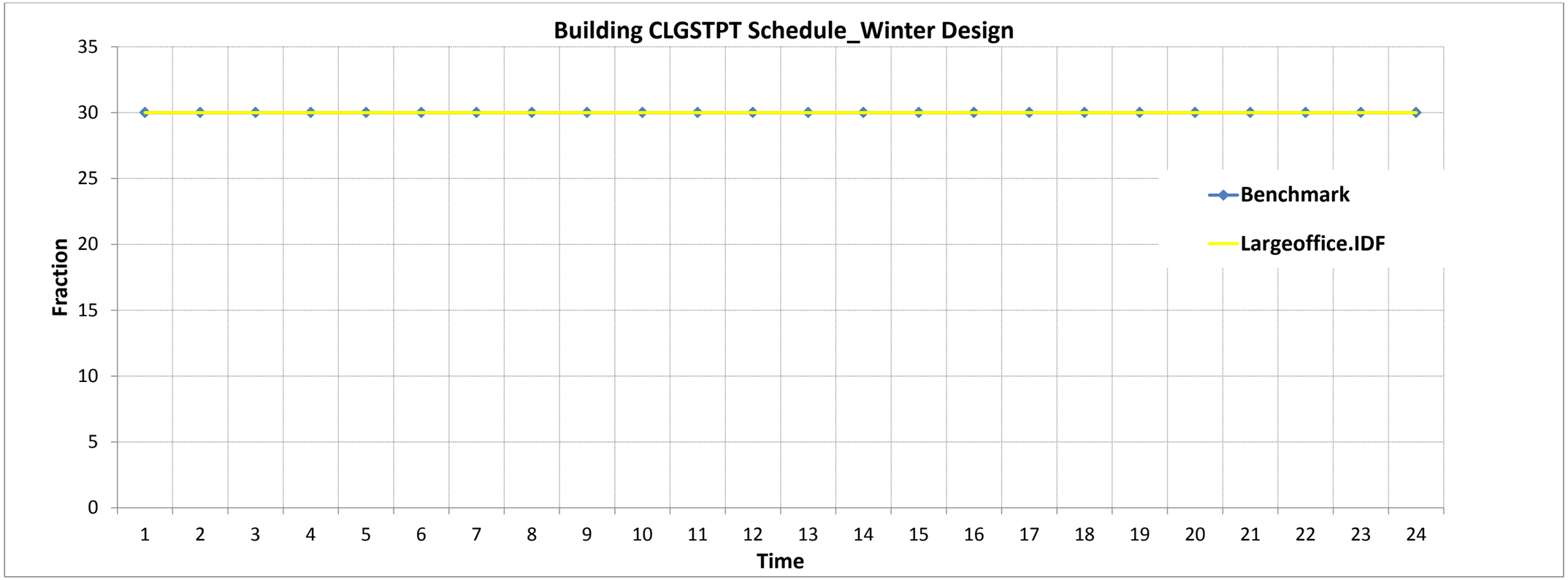
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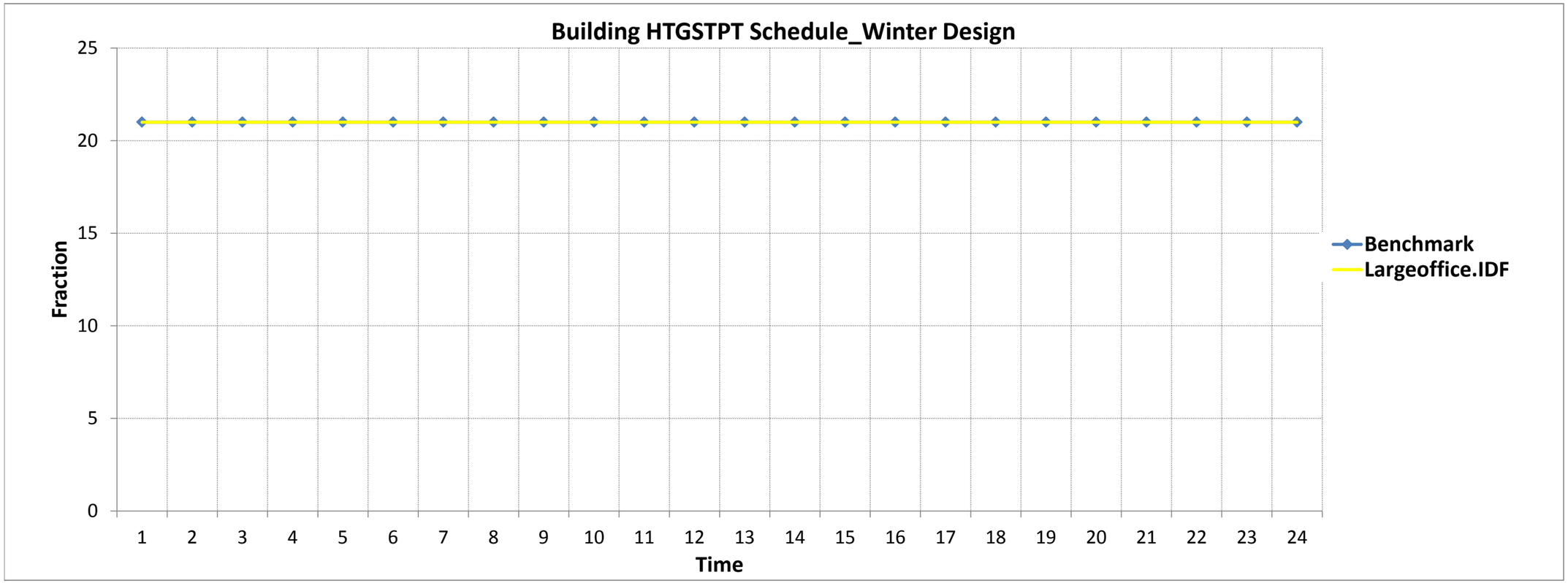
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Interim report March 2012

CBE EnergyPlus Modeling Methods for UFAD Systems

T. Webster , F. Bauman, K. Lee, S. Schiavon, A. Daly, T. Hoyt

Revised 12-2012

STRATIFICATION MODELING: GAMMA-PHI FORMULATIONS

This report documents the UFAD module specifications for EnergyPlus v3.1 and higher. We have conducted extensive validation of the interior zone model for Price swirl and York MIT1 variable area (VA) diffusers. These correlations should not be considered to be applicable to other diffuser types. (See below for user procedures to support other diffuser brands and types.)

INTERIOR ZONE

We have shown [Bauman et. al. 2006; Webster et. al. 2008] that diffusion plots for interior zones show a good correlation between the parameters PHIoz-avg and Gamma as described in the following:

$$PHI = \frac{T_{oz-avg} - T_{SAT}}{T_R - T_{SAT}} \qquad \text{Equation 1}$$

Where,

T_R	return air temperature [°F]
T_{oz-avg}	occupied zone average temperature [°F], calculated as average temperature between heights of 4 inches and 67 inches in room
T_{SAT}	diffuser supply air temperature [°F].

PHI represents the stratification in the room; lower values mean more stratification. [1] **Error! Reference source not found.** shows the definition of Gamma for interior zones derived from theory for point source

[1] The temperature difference between the upper and lower layers in the room is only an approximation to the actual stratification that we would expect. Stratification normally occurs primarily in the lower zone and is characterized by the difference between the temperature near the floor and one at head height. This defines the occupied zone (OZ) which is the region between 0.1m (4 inches) and 1.7 m (67 inches) from the floor. For transition heights fixed at 1.7 m (67 inches) we can obtain a rough estimate of the "real" stratification by assuming a linear profile where the average is the lower layer temperature; in this case the stratification will be two times the upper to lower temperature difference.

thermal plumes. PHI is equal to the term Kc used in the Energyplus documentation. It is the ratio of the heat gain in the lower layer to that of the entire room.

$$\Gamma = \frac{(Q \cdot \cos\varphi)^{\frac{3}{2}}}{m \cdot \left(\frac{n}{m} \cdot A_{eff}\right)^{\frac{5}{4}} \cdot (0.0281 \cdot W)^{\frac{1}{2}}} \qquad \text{Equation 2}$$

Q	total room airflow [m³/s]
$\cos(\varphi)$	discharge angle from vertical [°]
n	number of diffusers [1]
m	number of plumes (i.e., workstations) [1]
A_{eff}	effective diffuser area [m²]
W	room convective heat extraction (i.e., cooling load) [kW]
Γ	Gamma, a dimensionless number representing the ratio of momentum to buoyancy forces

Interior correlations

Figure 3 shows regressions from full scale laboratory testing of typical office workstation configurations for a wide range of operating conditions.

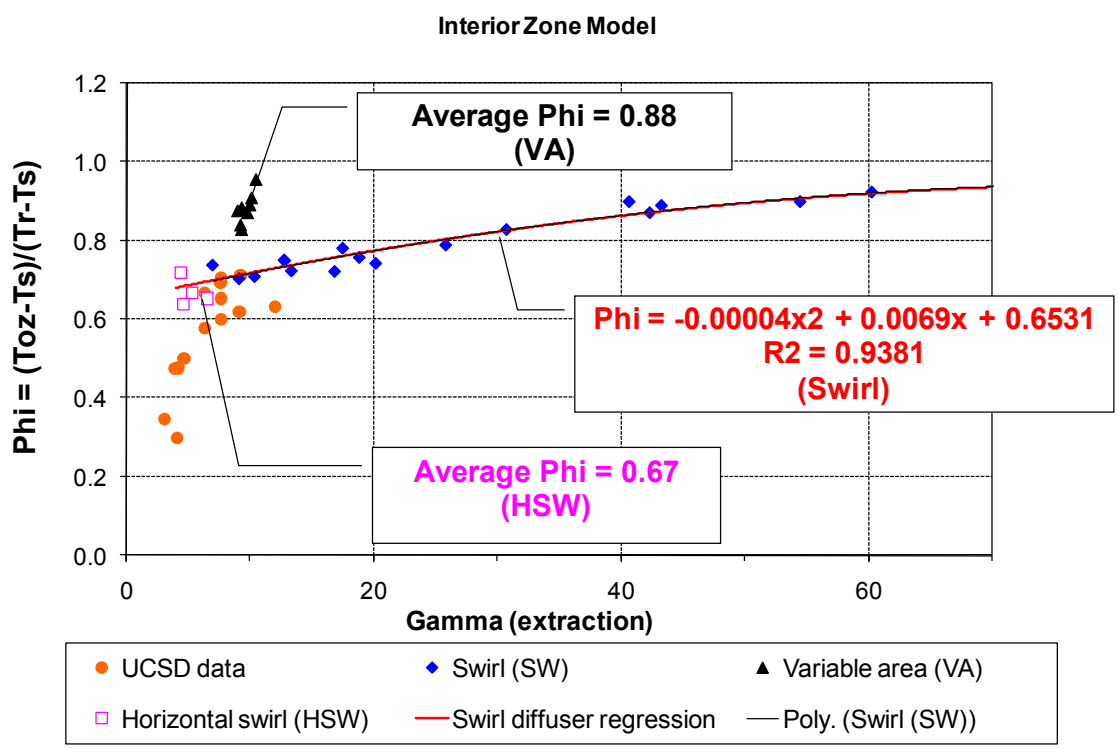

Figure 3. Interior zone correlations

PERIMETER ZONE

For perimeter zones, Gamma is developed based on theory of line plumes generated by heat gain from exterior windows and walls. In this case a linearized formulation for Gamma is used:

$$\Gamma_L = \frac{(Q \cdot \cos\varphi)}{(n \cdot A_{eff}) \cdot (B_L)^{\frac{1}{3}}}$$

Equation 1

Where,

Q	total perimeter zone airflow [m³/s]
(φ)	discharge angle from vertical [°]
n	number of diffusers [1]
A_{eff}	effective diffuser area of single diffuser [m²]
B_L	buoyancy per unit length of zone [m^3/s^3]

Converting buoyancy to heat gain and following the strategy we used for interior zones of using the room extraction rate for the buoyancy term we have:

$$\Gamma_L = \frac{(Q \cdot \cos\varphi)}{(n \cdot A_{eff}) \cdot (0.0281 \cdot W_L)^{\frac{1}{3}}}$$

Equation 4

Where,

W_L zone extraction rate per unit length of zone [kW/m]

Perimeter correlations

Figure 4 shows (revised based on testing conducted in 2010) correlations for two tested diffuser types: York/JCI MIT2), and linear bar grilles (Titus and Walnut 6" wide linear). Note that based on previous testing EnergyPlus applies a consistent reduction of 0.2 Phi for all diffusers when blinds are down. Linear bar grille tests were conducted without interior zone diffusers, i.e., diffusers only in perimeter zone that included internal workstation loads as well as solar gains derived from a solar simulator consisting of an array of HID lamps. For these reasons, only the variable area and linear bar grilles are valid types to use in perimeter zones at the present time. **Note that the older correlations for York lab testing are no longer recommend to be used, see discussion under diffusers below.**

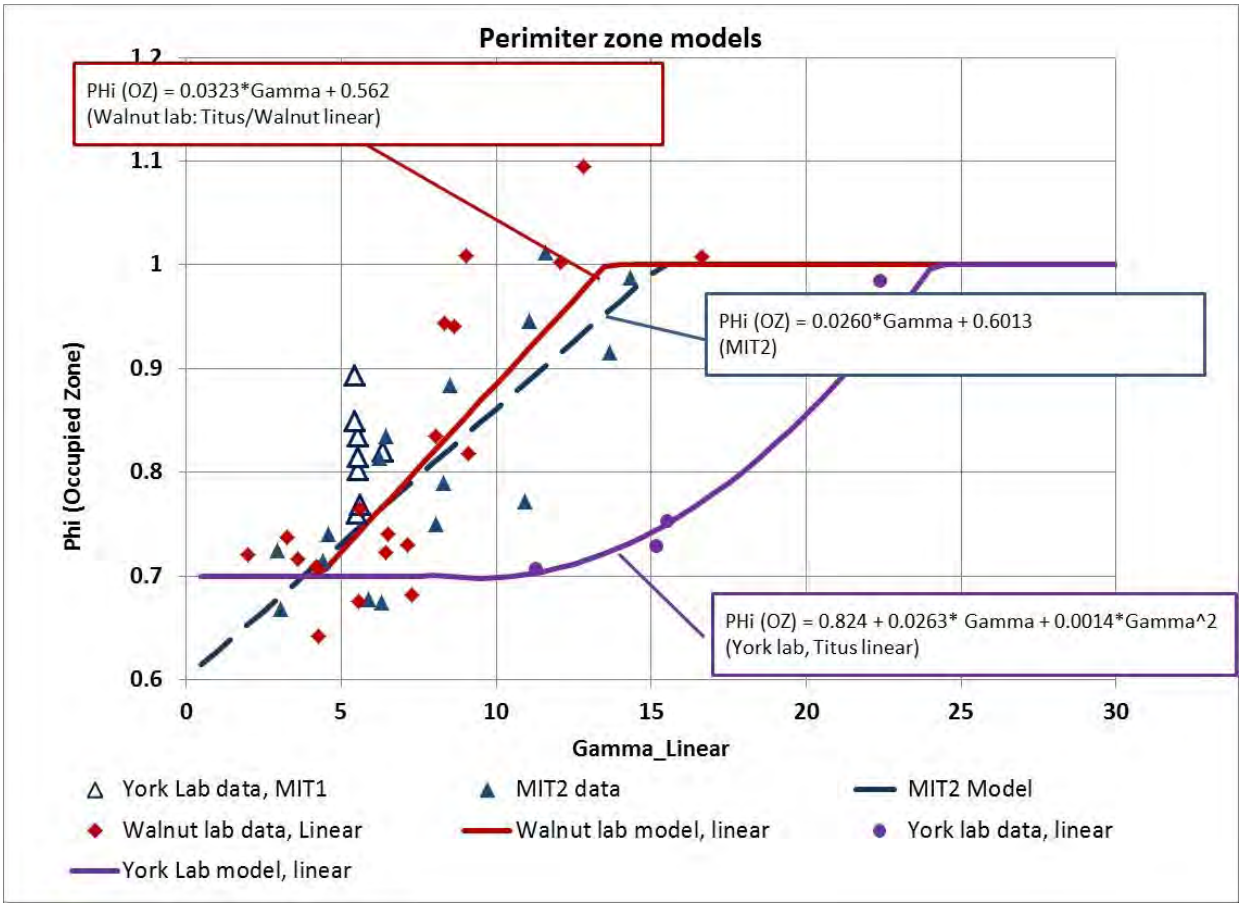

*Figure 4. Perimeter zone stratification correlations (**NOTE: DO NOT USE York lab results**)*

ENERGYPLUS UFAD INPUT REQUIREMENTS

Complete descriptions of required inputs and their defaults can be found in the EnergyPlus Input/Output Reference document. Most defaults are based on tested diffusers discussed above. The following provides more detail on selected input parameters to the help the user better understand EnergyPlus methods and the connection to plume theory.

Default values for all the required inputs are embedded in EnergyPlus and are activated when the *Auto-calculate* command is invoked. This is intended to provide reasonable default values where information is left out or lacking. It is not the same as "*Autosize;"* there are no sizing calculations done for these inputs (other than those performed in the CBE EnergyPlus/UFAD User Interface; see below).

DIFFUSER OBJECT INPUT FIELDS

Detailed input fields for diffuser characteristics are describe here and in the EnergyPlus Input/Output Reference. These inputs are used to calculate Gamma which is in turn is used to calculate Phi from the regression equations. As stated above, Phi is then used to apportion the zone convective gains between the upper and lower layers of the EnergyPlus/UFAD two layer room model. This allows a dynamic hour by hour simulation of stratification.

The primary inputs for diffusers are:

- Effective area (Ae) – the effective discharge area for one diffuser; determined from manufacturers data or by testing (Methods for determining effective area are discussed below)
- Number of diffusers (n) – Total number of diffusers per zone; user assigned (i.e., derived from peak flow rates and diffuser design airflow or other criteria) or defaulted to design number of occupants or based on assumption of a nominal 9.3 m^2 (100 $ft^{2)}$ per occupant.
- Diffuser discharge angle (φ) – Angle from vertical of air discharge from the diffuser. Does not include angles used to induce swirl.
- Diffuser type – Type of diffuser in a zone; only one type is allowed per zone. The options are Swirl, Horizontal Swirl, Variable area, and Linear bar grille any of which can be specified in the Diffuser Type object. If "Custom" is entered for diffuse type, the default types and their corresponding parameters are ignored and the detailed user entered parameters (including regression coefficients) are used instead. (See Other Diffusers below.)

All of these parameters have defaults that are used when a specific diffuser type is selected; the defaults are listed in Table 1.

Alternative diffusers

Other diffusers can be accommodated in EnergyPlus UFAD modeling. All that is required is empirically derived Gamma vs. Phi correlations for the particular diffuser of interest, as well as the other diffuser characteristics, effective area and discharge angle. The regression coefficients can be input into EnergyPlus in either power or polynomial equation formats.

THERMAL PLUMES

These parameters are only relevant to interior zones with internal loads that generate thermal plumes. Although perimeter zones have internal plume generating loads also, we assume that all loads (internal and solar) behave like an equivalent wall plume which is represented by the Gamma linear vs. Phi formulation. Likewise, for interior zones we do not distinguish between individual load components (people, computers, etc.) rather we represent these as a single (coalesced) plume; i.e., one plume per workstation. In both of these constructs, the room extraction rate is assumed to be the total convective plume power generated by m number of (interior) zones; one wall plume is assumed for perimeter zones) plumes.

The primary inputs for interior zone plumes are:

- Power-per-plume (PpP) – This is the power (kW) that represents the full load power of an individual plume; i.e., equivalent full load plume power. Users can input a value or use the EnergyPlus default via the auto-calculate command. Defaults are derived from the total design convective heat gains for people and equipment divided by the design number of occupants; i.e., assumes one occupant per workstation or load center. If occupants are not included, one plume per 9.3 m^2 (100 ft^2) is assumed.

Once calculated, power-per-plume is used internally to calculate the number of plumes (m). During simulation m is determined by dividing the total room convective heat gains (~extraction rate) by power-per-plume; m then is the number of equivalent full load plumes for a given hour.

For perimeter zones, plumes are modeled as one continuous wall plume whose strength is proportional to the room cooling load.

TRANSITION HEIGHT

The transition height represents the stratification level, the division between the upper and lower layer. According to plume theory this height varies depending on the balance between buoyancy (via thermal plumes) and momentum (via diffuser airflow) forces. Transition height increases with higher airflow from the diffuser for a given load. However, it also depends on the number and characteristics of the diffusers and is a function of Gamma as is the case for Phi. Gamma is used to calculate the transition height, h. This can result in a complex dynamic relationship between the control point and the occupied zone average temperature.

The transition height can be fixed by user input or auto-calculated from its Gamma correlation. This correlation has not been validated since the height is nearly impossible to measure in real systems. This results from the fact that in most real situations there is not a clear or single valued stratification height; it is most likely a band due to unequal plume strengths and load component vertical locations.

Simulations have been conducted to determine the sensitivity of using fixed Phi and transition height parameters. The transition has been found to have little or no impact in interior zones, and only a minor impact in perimeter zones. These results indicate that using a fixed height as a simplification (e.g., fixed at 67 inches to allow the occupied zone temperature to be the controlled temperature) results in very little error.

DIFFUSER CHARACTERISTICS

The critical parameters required for properly simulating diffusers are effective area, discharge angle, and a corresponding design airflow as noted above. The following describes the basis for the default models installed in EnergyPlus.

SWIRL

The effective area of swirl diffusers is based on the Price model RFTD with an effective diffuser area of 0.0075 m² (0.081 ft^2) and a discharge angle of 28°. These values were derived from bench testing and correspond closely to results from catalogue data. It is also noted that this area was identical for other swirl diffuser brands (i.e., Trox, Nailor, Titus). However, the Gamma-Phi correlations only represent Price diffusers; others need to be tested to provide a corresponding regression. These diffusers are most commonly used in interior zones, but rarely in perimeter zones.

HORIZONTAL SWIRL

The effective area for horizontal discharge diffusers (Price ARFHD) is 0.0060 m² (0.0648 ft^2). These diffusers have horizontal discharge therefore the discharge angle from vertical is much greater than for standard swirl diffusers. We used 73° based on choosing a reasonable comparison to the salt-tank data from UCSD. Only four tests using HSW diffusers were conducted. These are only used in interior zones.

VARIABLE VOLUME

These diffusers (York MIT generation 2 (MIT2) diffusers) operate with constant underfloor plenum pressure (i.e. 0.05 iwc during laboratory tests) and constant air velocities leaving the diffusers. Consequently, in order to vary the amount of airflow entering the room, the variable volume function is derived from pulse-width modulation of the diffuser inlet damper.

By assuming a constant supply air velocity of 2 m/s (400 fpm), the effective area is calculated using **Error! Reference source not found.**. This assumed velocity was verified by staff at York/JCI. For fully open diffusers airflow measurements (shown in **Error! Reference source not found.**) reveal a flow of

about 255 m³/h (150 cfm). If the area of a fully open diffuser is 0.035 m² (0.38 ft²) the outlet velocity at the grille is about 2 m/s (400 fpm).

$$A_{eff} = \frac{Q_{diff}}{v} \qquad \text{Equation 5}$$

with

Q_{diff} airflow per diffuser [m³/s]

v outlet air velocity [m/s].

The estimated discharge angle for MIT diffusers is 45°.

Although the grille area does not change (as it did in previous versions of these products) pulse-width modulation creates a variable volume function, therefore the area and discharge velocity is fixed for the open condition. We therefore consider these "constant Phi" diffusers. These diffusers are commonly used in both interior and perimeter zones.

LINEAR BAR GRILLES

These diffusers are standard products used and are commonly employed in UFAD systems for heating and cooling. The effective area (for Titus diffusers) was derived from bench testing and corroborated by catalogue data. They come in various lengths and widths and usually have a lip on the bars that directs the flow away from the window at about a 15° angle. Sometimes they are equipped with vanes that spread the flow along the window. When this is the case $\cos(\varphi)$ is a compound angle.

Recent testing (2010 at Walnut labs) demonstrated that there is some difference in performance due to the form factor of these diffusers. The original testing was done with small diffusers (Titus CT-481, 3 in x 18 in) with inlet vanes. The results from Walnut lab testing were for a more common size, nominally Titus or Walnut, 6 in wide by 6 ft long without inlet vanes. Other brands with a different form factor (e.g., Price LFG series).have not been tested and we would expect the performance to be somewhat different. To support simulations we have assumed nominal configuration data shown in Table 2 below. For LIN2 this represents a nominal 6 in (0.154 m) wide by 3 ft (1.21m) long diffuser that was extrapolated from the 6 ft long tested diffusers. These represent the default configurations used in EnergyPlus for the respective object label. (See the Appendix A for details of how these values are used for sizing purposes.) The Titus and Walnut data for diffuser LIN2 was combined to create the correlation equations. Note that the data for LIN2 were collected with the diffuser wide open (full flow mode) but due to the pulse width modulated control sequence used, in operation these diffusers operate at a fixed phi (and gamma) of about 0.85.

LIN1 diffusers were tested completely different and no longer need to be used. These were tested along with swirl diffusers that supply the internal loads for the zone in a large 676 sf chamber. (Walnut testing had only linear bar grilles, and a 256 sf chamber).[2] The curves shown in Figure 4 and Table 2 of LIN1 diffusers is inaccurate since the full room flow was assumed to flow through the nominal size of the 3" wide bar grilles (which also have diverting vanes installed). This is incorrect for two reasons; (1) the airflow at the linear diffusers were controlled by a flapper so the affective area (Ae) needs to be corrected

[2] In other words, York data assumes swirl diffusers in the perimeter zone, not an uncommon approach for real systems, but Walnut data assumes only linear bar grilles in the perimeter zone.

for this fact, and (2) the Phi – Gamma relationship is complicated by the fact of using two different types of diffusers. We have attempted several ways to calculate gamma considering these factors, but have yet to determine a scientifically valid way to represent this data. **Therefore, we recommend that the LIN1 option not be used for simulations. This should have little practical effect on results since linear bar grilles produce little stratification in simulation and in practice based on our experience; either model replicates this effect.**

Although other form factors and various lengths could be derived from the basic unit form factor data (i.e., a wider unit would have a larger form factor, so the equivalent length would be shorter) this option requires that the CUSTOM specification be used with user supplied object fields.

For reference purposes, Figures 5a through 5c show catalog performance information for the Titus CT-481 diffusers, and Price swirls. Table 2 below shows simulation specifications used for the default diffuser types.

York/JCI diffusers

Although the phi-gamma curves for the Walnut and Titus diffusers are virtually the same from the testing, there is a fundamental difference in how they are applied. The Walnut diffusers are supplied by York/JCI packaged in a unit that uses their pulse width modulated (PWM) air valves, just as is done with the MIT diffuser units. The effect of this is the same; these diffusers all operate a fixed discharge velocity (equal to that of the design airflow) when the air valve damper is open. This has the effect of making these diffusers "constant phi" diffusers. We make the assumption that these diffusers are selected and controlled to deliver the design airflow when open. This is contrary to how these would work when connected to a fan coil unit (as Titus and other linear bar grilles are normally configured) where the airflow is varied for a fixed area.

Design volumes

Design volumes listed in Table 2 have been scaled to nominal values for simulation purposes and based on our analysis and testing and assumptions about what we consider appropriate default sizes for simulation purposes; they may not conform precisely to manufacturers catalogue data. We have standardized on design pressures as follows: a plenum pressure of 0.050 iwc for swirl and MIT diffusers, and a drop across the diffuser plate of about 0.019 iwc; i.e., pressure just below the diffuser plate, for linear bar grilles. Unfortunately, this pressure was not accurately measured during testing to verify this nominal value that was estimated from Titus diffuser grille data (see Figure 5a below). Walnut specifies plenum pressures to be controlled to 0.05 iwc so we conclude that there is a significant pressure drop through the damper box at full flow to yield these low pressures at the diffuser plate. In real systems it represents the pressure at the inside of the diffuser as supplied by a fan coil unit for Titus diffusers and fan coil unit for the Walnut based systems.

Table 1. Default characteristics of tested diffusers

	EPlus label	Test lab	Make and model	Nominal Size	
				in, ft	m
Interior	Swirl	York/JCI	Price RFTD	8 in dia	0.2
	Horiz Swirl	York/JCI	Price ARFHD	8 in dia.	0.2
	MIT	York/JCI	York/JCI, MIT1	11.4 in x 11.4 in	0.29 x 0.29
	MIT2	Walnut	York/JCI, MIT2	11.4 in x 11.4 in	0.29 x 0.29
Perimeter	LIN1	York/JCI	Titus CT-481	3” x 1.5 ft	0.077 x 0.46
	MIT1	York/JCI	York/JCI, MIT1	11.4 in x 11.4 in	0.29 x 0.29
	LIN2	Walnut	Titus CT-481 & Walnut CLEMIT-3-27-06-015-L6F	6” x 6 ft	0.154 x 1.83
	MIT2	Walnut	York/JCI, MIT2	11.4 in x 11.4 in	0.29 x 0.29

Area, Square Feet*	Nominal Duct Width, nches	Total Pressure		.0125	.027	.050	.079	.111	.151	.199	.252	.310
.082	3"	**CFM per Foot**		**33**	**49**	**66**	**82**	**98**	**115**	**131**	**148**	**164**
		NC		–	12	20	26	31	36	40	43	46
		Throw,	Sill or Floor	2 - 2 - 2	6 - 6 - 6	9 - 9 - 9	12 - 12 - 13	15 - 15 - 16	17 - 18 - 19	20 - 21 - 22	21 - 22 - 23	23 - 23 - 24
		Feet	Side Wall	4 - 6 - 9	7 - 10 - 13	9 - 13 - 17	12 - 16 - 21	15 - 20 - 25	17 - 22 - 28	20 - 26 - 32	21 - 28 - 35	23 - 31 - 39
.186	6"	**CFM per Foot**		**74**	**111**	**149**	**186**	**223**	**260**	**298**	**335**	**372**
		NC		–	14	22	28	34	39	42	46	49
		Throw,	Sill or Floor	4 - 4 - 4	10 - 10 - 10	14 - 14 - 14	18 - 18 - 18	22 - 22 - 22	25 - 25 - 25	28 - 28 - 28	30 - 30 - 30	32 - 32 - 32
		Feet	Side Wall	8 - 10 - 13	11 - 14 - 18	15 - 19 - 23	19 - 23 - 27	23 - 27 - 31	25 - 30 - 35	28 - 33 - 39	31 - 37 - 43	34 - 40 - 46

Figure 5a. Titus Linear bar grille performance data

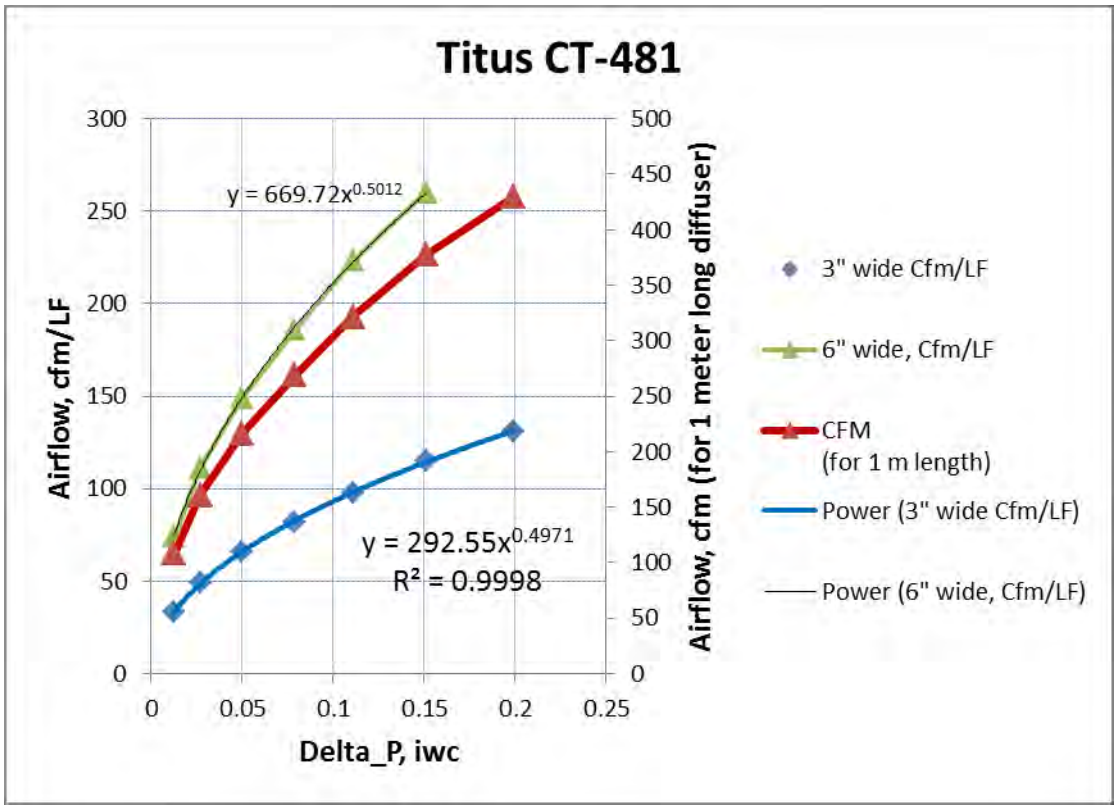

Figure 5b. Titus Linear bar grille performance

DETERMINING DIFFUSER EFFECTIVE AREA

In this section we describe procedures for determining the effective area for diffusers. Given adequate data, this can be accomplished with manufacturer's catalog data.

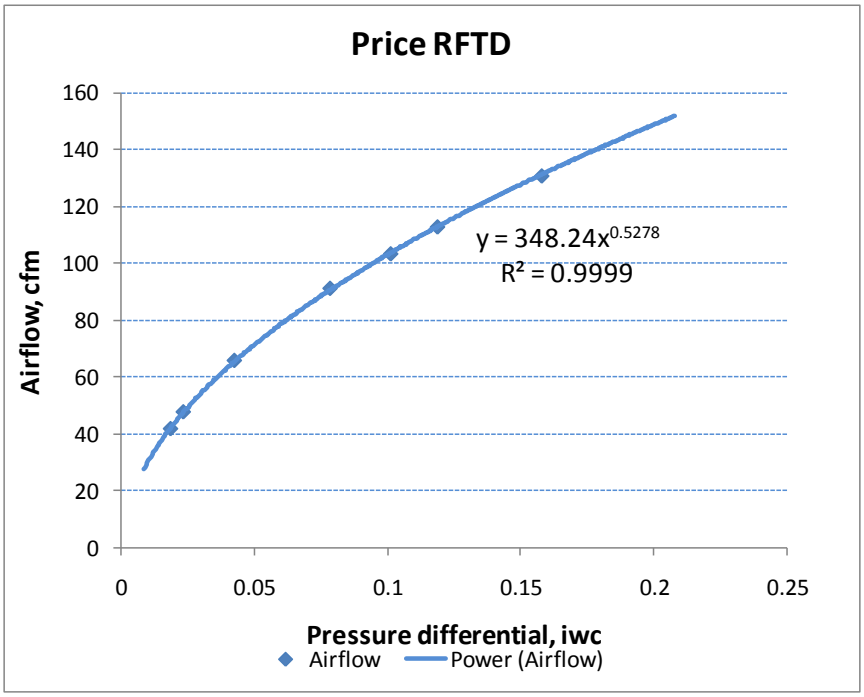

Figure5c. Price swirl diffuser airflow performance

Using results from testing or data from catalogs for a given diffuser the effective area can be determined as follows using the example regression equation from Figure 5c:

To convert results from the linear regression to square root form; select typical operating pressure (e.g., 0.05 iwc) and compute as follows.

$$A_{eff} = \frac{K}{4005}$$ Equation 5

Where,

$$K = \frac{348.2 * 0.05^{.0.50}}{0.05^{0..5278}}$$ Equation 6

DEFAULT DIFFUSER INPUT SPECIFICATIONS SUMMARY

Table 2 summarizes default specifications and correlation equations for all diffusers tested. The design volumes were derived from the test diffuser specifications shown in Table 1.

Table 2. Default specifications for EnergyPlus/UFAD

Zone Type	Diffuser Type (Eplus lable)	Default total effective area per diffuser (unit area)		Discharge Angle from vertical	PHI – Gamma correlations (X = Gamma, X_L = Gamma linear)	Phi limits		Nominal Design volume[3] (flow/Length)	
		ft^2	m^2			min	max	cfm	m^3/s
Interior	Swirl[4]	0.081	0.0075	28°	Phi = $-0.00004x^2 + 0.0069x + 0.6531$	NA	NA	72	0.034
	Horiz Swirl[5]	.0644	0.0060	73°	Phi = 0.67 (constant for all Gamma)	NA	NA	55	0.026
	MIT1	0.38	0.035	45°	Phi = 0.88 (constant for all Gamma)	NA	NA	150	0.071
	MIT2	0.38	0.035	45°	Phi = $0.025x_L + 0.6013$	NA	1.0	150	0.071
Perimeter	LIN1[6]	0.126 (0.084/Lft)	0.0115 (0.025/Lm)	15°	Phi = $0.0014x^2 - 0.0263x + 0.8214$	0.7	1.0	47 (33.4/Lft)	0.022
	LIN2[7]	0.52 (0.18/Lft)	0.048 (0.055/Lm)	15°	Phi = $0.0323x_L + 0.56$	0.7	1.0	225 (75/Lft)	0.106
	MIT2	0.38	0.035	45°	Phi = $0.025x_L + 0.6013$[8]	NA	NA	150	0.071

[3] Flow at 0.05 iwc plenum pressure for swirl and MIT and estimated 0.0127 iwc for linear bar grilles (behind the grille). Design volumes are used in the CBE UFAD interface to facilitate calculation of the number of diffusers.

[4] Airflow based on CBE measurements at .05 iwc

[5] Airflow based on Price catalogue data at 0.05 iwc

[6] Based on 1.5 LF diffuser

[7] Assumes Titus or Walnut 6" wide based on 3ft nominal length diffuser

[8] Although this table shows a gamma-phi curve for MIT2 diffusers, in practice this is used only for design sizing; the associated phi values remains fixed at the design value for subsequent daily and annual runs. We assume MIT2 represents all York/JCI products that use the modulating damper. (See Appendix A)

SUPPLY PLENUM MODELING

Figure 7-1 presents a schematic diagram of how the simplified plenum model can be configured within EnergyPlus. Conditioned air from the air handler enters the underfloor plenum (Plenum 1) at the desired flow rate and plenum inlet temperature (T_{in1}). Similar to other conditioned zones, EnergyPlus performs an energy balance on the plenum, producing a single well-mixed temperature ($T_{plenum1}$). To calculate the energy balance, surface convection coefficients developed by CBE are specified for the slab (h_{s1}) and raised floor (h_{f1}). The well-mixed plenum temperature ($T_{plenum1}$) serves as the average diffuser discharge air temperature (T_{out1}) entering the conditioned space (Zone 1). Note that more than one thermal zone can be served by a single underfloor plenum, and in fact, unless the user has more detailed information about the expected temperature distribution in the underfloor plenum across the entire floor plate, this is the preferred plenum configuration (one single underfloor plenum) for each floor of the building. However, if desired, one or more additional plenum zones may be added in series to the first plenum zone. In this case, as shown in Figure 7-1, T_{out1} is equal to the plenum inlet temperature (T_{in2}) for the second plenum zone (Plenum 2), i.e., series plenums. This permits the possibility of simulating an interior plenum zone with a cooler supply air temperature and a perimeter plenum zone with a warmer supply air temperature due to thermal decay in the plenum. Note that in this case, the airflow entering the second plenum will be reduced by the volume of air delivered into Zone 1. Due to the complexity of most underfloor plenum airflow and temperature distributions only the series plenum configuration is assumed as the default for current versions of EnergyPlus. Only knowledgeable users should attempt to configure the model for the configurations listed below. Airflow paths are dictated in the AirLoop objects in EnergyPlus. This includes zones, terminal equipment and splitters and mixers. Currently (2011) only splitters are allowed for supply plenums and mixers for return plenums.

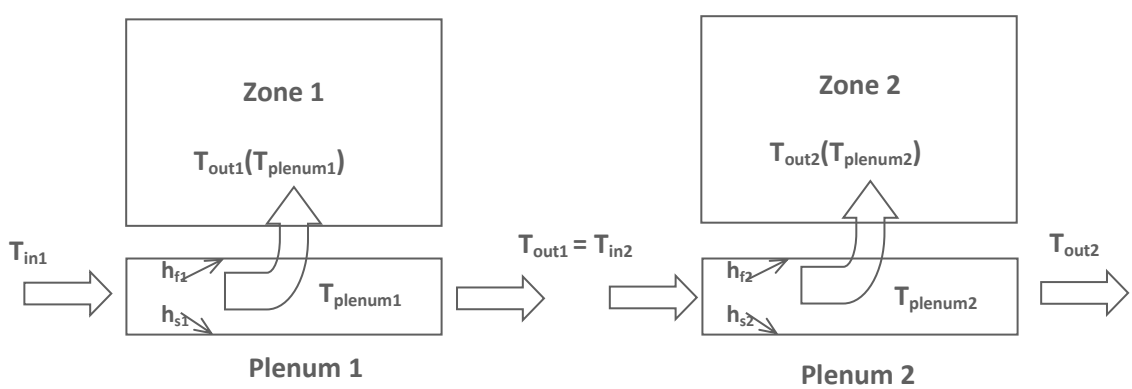

Figure 7-1. Schematic diagram of EnergyPlus plenum model structure for series plenum

Underfloor supply plenums can be installed and zoned in a variety of configurations. Based on current research and practice, it is believed that the choice of plenum configuration can have important impacts on the thermal performance (distribution of heat gains) of the supply plenum. Plenum configurations now available in the CBE development version of EnergyPlus v6.0 are described and illustrated briefly below. The descriptions are somewhat idealized compared to

actual practice, but are intended to capture the expected performance characteristics of different approaches to plenum design and operation.

Figure 7-2 shows plan and section schematic views of a portion of the (default) series plenum configuration available in the public release version of EnergyPlus. This configuration represents an approach that has been used in many UFAD installations in which an open plenum with minimal ductwork serves both an interior and perimeter zone of the building. Supply air from the air handler is delivered from the service core into the interior part of the plenum. Swirl diffusers are used in the interior and linear bar grilles are used at the perimeter. The approximate temperature distribution in the plenum and room (for demonstration purposes) can be read using the color temperature scale. In this case, the series plenum refers to the concept that cool air entering into the plenum (60°F [15.6°C]) will first gain heat as it flows through the interior portion of the plenum (reaching 64°F [17.8°C]), before entering the perimeter portion of the plenum, where it gains additional heat (68°F [20°C]) before entering the room. As will be discussed further, this approach is expected to produce the highest perimeter diffuser discharge temperatures, and should be avoided if possible.

Figure 7-3 shows a reverse series plenum configuration. This is not yet available in EnergyPlus, but should be available later in 2013. This represents an alternative plenum design that uses ductwork (or other means) to deliver the cool supply air from the air handler directly into the perimeter portion of the plenum. Supply air temperature rise proceeds in the reverse direction with the incoming (60°F [15.6°C]) air gaining some heat from the perimeter zone producing (64°F [17.8°C]) supply air at the perimeter bar grilles, before flowing back to the interior where additional supply air temperature rise produces the highest supply temperature. This configuration provides cooler supply air at the perimeter to handle peak cooling loads and warmer temperatures in the interior for comfort purposes.

Figure 7-4 shows the “common” plenum configuration. This refers to a situation where air is delivered into the plenum from a variety of locations (using ductwork or air highways, as shown) and with a variety of inlet velocities. The net result is that while there will be variations in individual diffuser discharge temperatures across the floorplate, the average supply temperature entering the interior zone will be very similar to that entering the perimeter zone. In this case, the term “common” plenum means that the plenum delivers the same average temperature everywhere. This characteristic behavior has been observed in completed UFAD projects.

Figure 7-5 shows a parallel, or zoned, plenum configuration. This configuration requires the installation of a plenum divider separating (in this case) the perimeter and interior portions of the plenum. This is an approach that may be used when the designer wants to ensure improved control of a special zone with unique load requirements. It does result in a more complicated plenum configuration with barriers restricting some of the flexibility. Supply air from the air handler must be delivered directly to each isolated zone.

One other plenum configuration, ducted perimeter, is not illustrated, but represents a combination of the parallel and single plenum schemes. In this case, AHU supply air is ducted directly to the perimeter zone diffusers via an assumed VAV box. The interior portion of the plenum is served in the same manner as parallel plenum configuration.

All of the configurations described above, except reverse series, can be simulated with appropriate changes to the air path objects and careful construction of the zone to plenum boundary surfaces

objects. These are currently (2012) implemented in the CBE prototype building .imf files that can be used as an example for other building models.

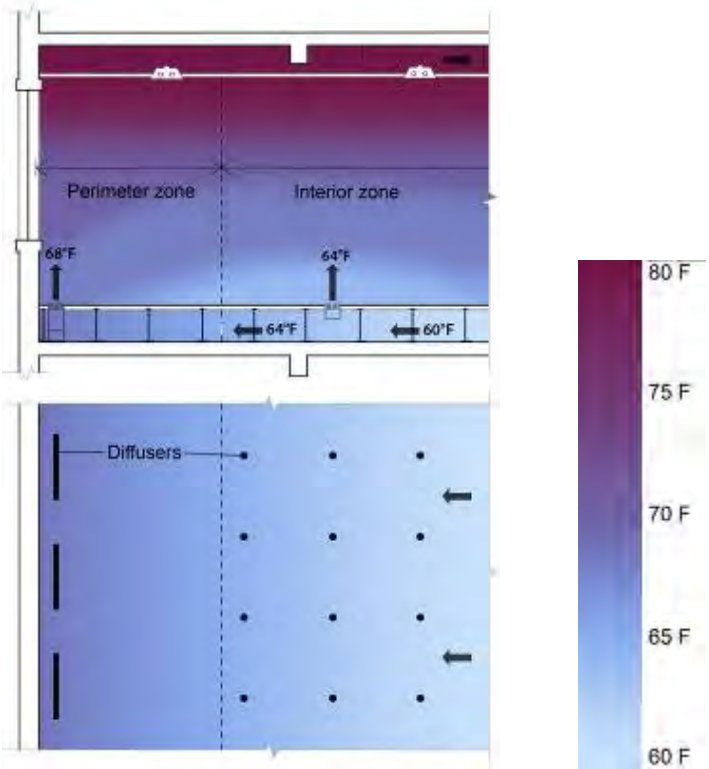

Figure 7-2. Series plenum configuration

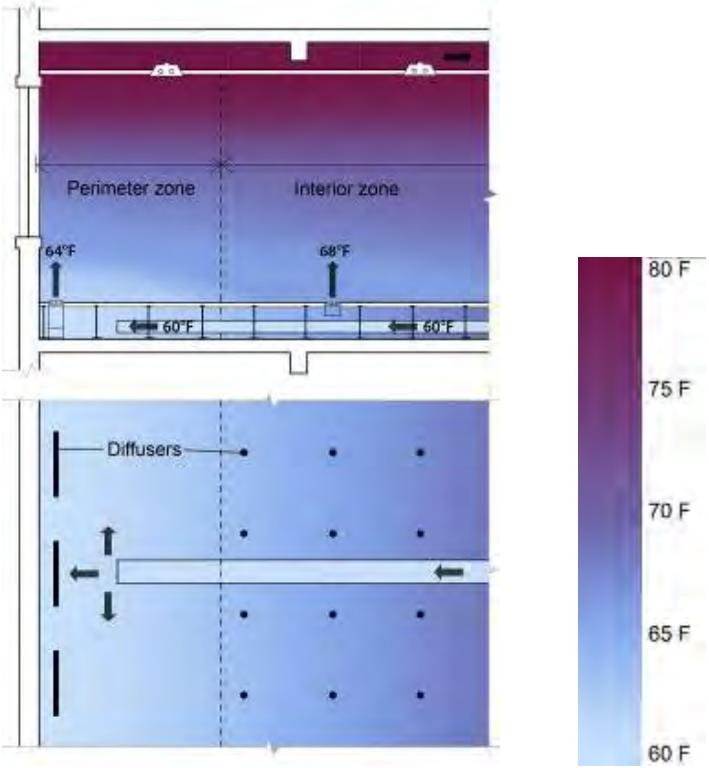

Figure 7-3. Reverse series plenum configuration

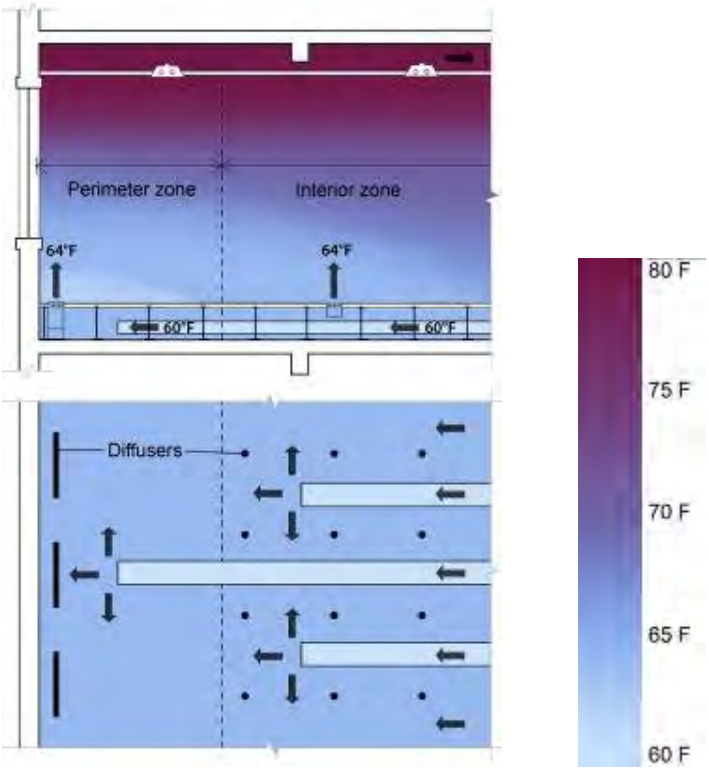

Figure 7-4. Common plenum configuration

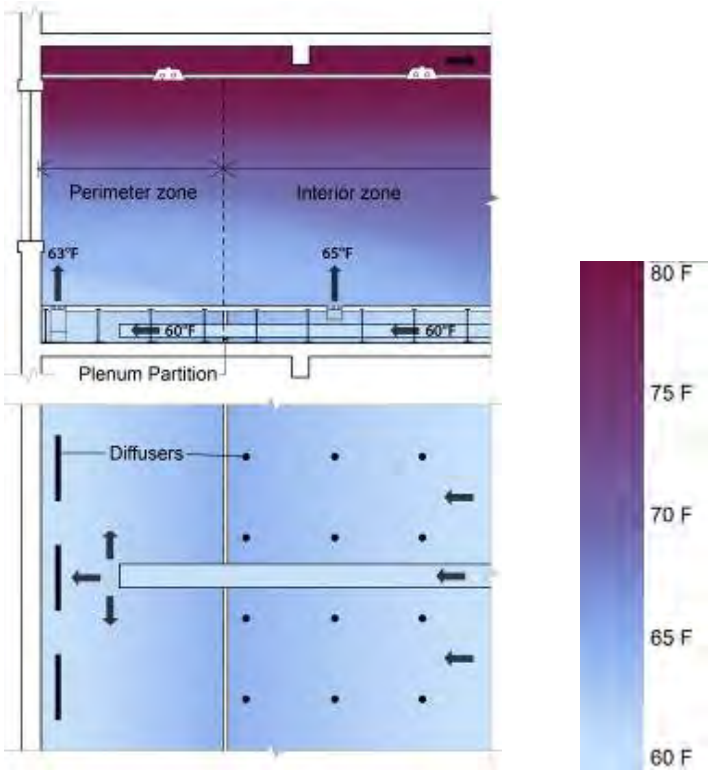

Figure 7-5 Parallel (zoned) plenum configuration

CBE ENERGYPLUS UFAD HVAC MODELING GUIDE

In this section we outline methods used to model various system and plant components typically used for UFAD systems. Each run is assumed to be sized anew for central system and plant components, either using the EnergyPlus auto-sizing capability or some form of manual sizing. This approach assumes that systems are redesigned[9] for the new design conditions implicit in the parametric factors under study. Note also that a number of the features and design work-arounds discussed below are only implemented in CBE's development version of EnergyPlus.

CENTRAL AIR HANDLER MODELING

For these studies the fan design volume (and fan size) will be different for each run. For these we assume that the new operating points all use the same AHU Fan design static pressure (FSP). This implies that the fans are assumed to be resized for each new design point such that the internal losses (filters, coils, etc.) are the same, AND the fan is sized to maintain the same (near peak) design efficiency. Therefore, each UFAD design point is input with the same peak efficiency and UFAD design static pressure, but a new airflow.

Due to problems with auto-sizing the airflow, we use a CBE sizing strategy where the design day output is used for the maximum airflow which is then increased by a global sizing factor of 1.2.

[9] As opposed to maintaining a given size while changing the parameters as would be done if differences in operation were being investigated

Fan design efficiencies

EnergyPlus AHU models require design airflow, static pressure, fan efficiency, and motor efficiency as inputs. For the supply and return fans we assume a variable frequency drive with a high efficiency motor and a belt drive to the fan. Typical values are shown in Table 2.

Table 2. Central air handler efficiencies

	Individual	Combined
Fan static efficiency	75%	75%
Motor efficiency	90%	84%
Belt drive efficiency	95%	
VFD efficiency (100%)	98%	
Total Efficiency	63%	63%

Variable frequency drive (VFD) efficiencies are a function of operating frequency [Wray, Personal communication] (i.e., speed) which are in the range of 60-70% at 30% speed and 98% at 100% speed. We have included only the design point VFD efficiency; a new algorithm will be required to simulate the VFD part load performance accurately. The resulting motor plus overall drive efficiencies are incorporated in the motor efficiency as shown in the Combined column in Table 2.

Fan design static pressure

STATIC PRESSURE BREAKDOWN

To guide the user, Table 3 shows a typical component breakdown of static pressure losses for air system components and how UFAD and overhead systems compare.

Table 3. Fan static pressure guide

	OH		UFAD		
Pressure loss item	**Office supply, iwc**	**Office return/ relief, iwc**	**Office supply, iwc**	**Office return/ relief, iwc**	**comments**
AHU (clean filters, coils, louvers, etc.)	1.1	0.3	1.1	0.3	Include system effect
Dampers	0.1	0.1	0.1	0.1	Economizer dampers
Dirty filter allowance	0.75		0.75		
Ductwork, shafts	0.08	0.03	0.08	0.03	
Ductwork, branches	1.25	0.02	0.75	0.02	Return side = return plenum UFAD: varies depending on plenum ductwork configuration
VAV boxes	0.5		0		
VAV box reheat coil	0.2		0		
Diffusers and low pressure ductwork	0.3	0.1	0.05	0.1	RA grille
Safety factor					
Total	**4.28**	**0.55**	**2.83**	**0.55**	
Fan design static	4.53	0.55	3.08	0.55	Assumes return relief fan. Supply fan static = combined supply and return losses
Fan design eff	0.75	0.54	0.75	0.54	
Motor eff	0.9	0.9	0.9	0.9	
Drive/VFD eff	0.98	0.9	0.98	0.9	
Min fan static (shutoff)	1.5 -0.5	0	0.5	0	Assume 0.5 = static pressure reset

IMPACT OF UFAD PLENUM CONFIGURATIONS

As shown by Table 3 the primary differences between typical OH and UFAD systems is due to branch and low pressure ductwork; Table 3 shows UFAD to be 60% of OH for this component. However, the value for UFAD depends on the configuration of the plenum air distribution system. It can vary widely between extremes of no ductwork to fully ducted (equivalent to an OH system); "air highway" systems are somewhere in between. The user should adjust the values on Table 3 to reflect the design being considered. Examples are shown in Table 3a below.

Table 3a. Design static pressures for various plenum configurations

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

Air handler supply temperature

Two methods can be used to specify air handler supply temperatures (SAT): (1) fixed, where the cooling coil discharge is maintained at a constant temperature, or (2) reset, where the SAT is reset based on outside air temperature or by load demand. Both of these strategies are available in EnergyPlus, but the load based reset has not been shown to be reliable in our experience. Reset is required by ASHRAE standard 90.1-2010.

Part load performance

During simulation, the power at each time step is determined by multiplying the design power by a part load factor from the part load curves shown in Figure 8. Shutoff pressures[10] characterize different control methods and system designs, as indicted in Figure 8. For overhead systems using constant static pressure control strategies, shutoff pressures vary between 0.250 kPa (1.0 iwc) for well-designed systems and 0.375 kPa (1.5 iwc) for typical practice design. [Daly 2009] For UFAD systems, shutoff pressure is assumed to be 0.125 kPa (0.5 iwc) for partly ducted distribution systems, our baseline assumption.

[10] Shutoff pressure represents the design static pressure setpoint for a VAV system without static pressure reset. When static pressure reset is simulated, the shutoff pressure is assumed to be near zero (e.g., 0.5 iwc).

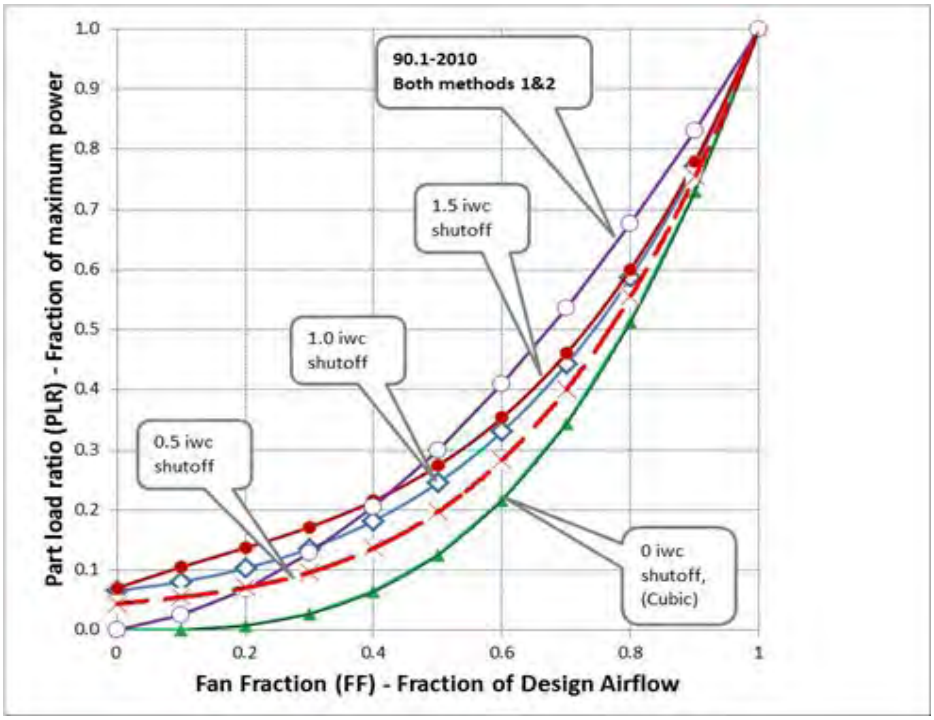

Figure 8. Fan part load curves for central airfoil fans with difference shutoff assumptions and for variable speed VAV terminal units.

Table 4 summarizes the recommended equations used for various fan types.

Table 4: Part load curves

	PLR curves	Remarks
Supply fan, typical overhead system, without static reset	$y = 0.2939x^4 + 0.2227x^3 + 0.3063x^2 + 0.1131x + 0.0662$	Assume airfoil fan with 1.0 min static pressure
Relief fan, and VSFCUs	$y = 1x^3 + 2E\text{-}13x^2 - 6E\text{-}14x - 4E\text{-}15$	Assume propeller relief fan follows cubic curve
UFAD supply with static pressure reset, Return fans	$y = 0.0636x^4 + 0.7993x^3 - 0.0102x^2 + 0.1037x + 0.0441$	Assume airfoil fan with 0.5 min static pressure

ZONE HVAC MODELING

UFAD system models

Although other system types are possible to model with EnergyPlus depending on the skill and knowledge of the user, this guide focuses on the two system types discussed below that we have modeled successfully.

- **System 1, engineered:** UFAD system configured with swirl diffusers in the interior controlled by varying the supply plenum pressure and a perimeter system with a variable speed fan powered terminal unit's (VSFCU) that supplies linear bar grille diffusers.
- **System 2, packaged:** This system is exemplified by the packaged system sold by York/JCI. In this system the plenum pressure is held constant and both interior and perimeter VAV floor diffuser boxes modulate airflow for cooling. For heating, a constant volume fan coil unit provides heat through perimeter diffusers that are switched between heating and cooling as required.

Figure 9 shows illustrations of these two system types.

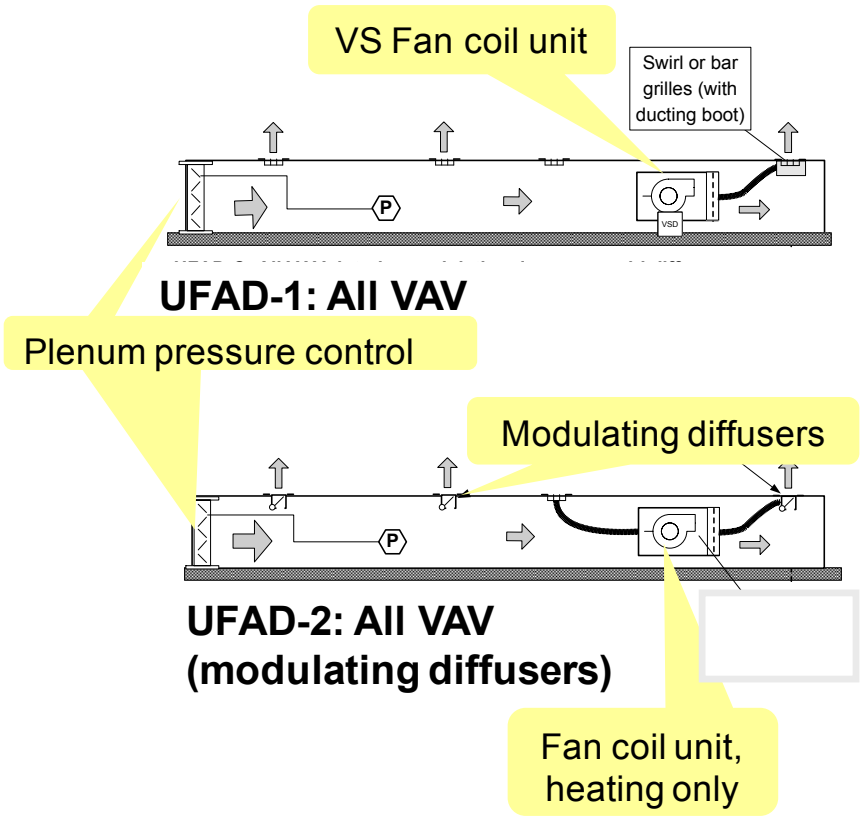

Figure 9: UFAD System configurations modeled by EnergyPlus

Terminal units modeling

FAN POWERED TERMINAL UNIT EFFICIENCIES

Small fan terminal units typically used in VAV and UFAD systems use small direct drive fans, usually with forward curve blades. Direct drive makes it difficult to measure actual fan shaft power thus data sheets typically do not show power performance along with their flow vs. static pressure performance. Also, static pressure performance is usually stated in terms of external static, not total fan static pressure across the fan itself. For these reason, fan efficiency cannot be determined from catalog data. For this project, aid of manufactures' was enlisted to provide data from which the overall unit efficiency (fan plus drive) could be calculated. Data was obtained for throttling performance at constant speed for various ATU models.

From the data provided, the largest units were selected with an assumed design operating point of 0.10 kPa (0.4 iwc) (includes hot water coil and discharge ducting) at maximum speed. Based on interviews of design professionals, this strategy is a reasonable representation of how these units are selected in actual practice; where possible, several large units selected to operate in parallel at their maximum speed to meet the zone design airflow requirements. Design efficiency for this operating point was found to be ~15 percent. The user can modify this parameter based on more recent data. We assume heating fan coils used for System 2 use constant volume fans with a similar overall efficiency.

FAN POWERED TERMINAL UNIT PART LOAD PERFORMANCE

Since variable speed terminal units are not controlled to static pressure set points, they may can be assumed to throttled along a cubic curve to zero pressure and zero flow. For simulation purposes

this is characterized by a cubic curve as shown in Figure 8. This performance was verified by actual measurements provided by an ATU vendor.

Zone control models

INTERIOR ZONES

Since for both types of system, temperature is controlled by varying the volume, these zones are modeled with a non-reheat VAV box. The control model is shown in Figure 10.

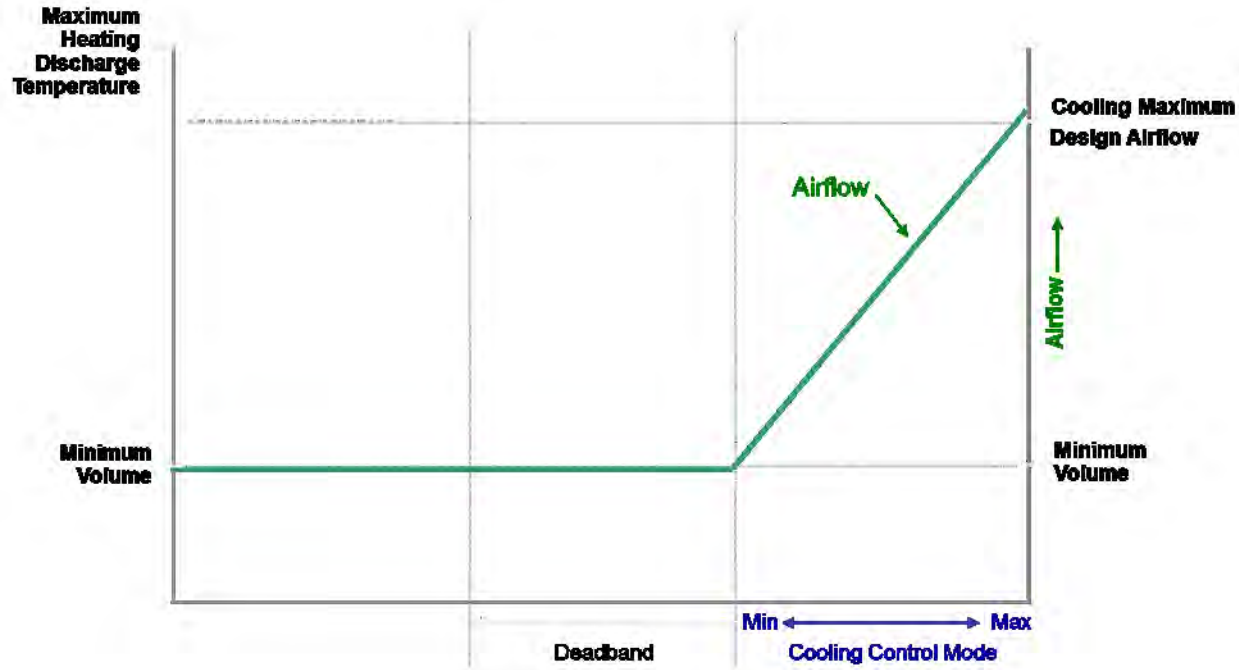

Figure 10. EnergyPlus/UFAD interior zone VAV box control model

Perimeter zones

SYSTEM 1

The EnergyPLus control model for VSFCU's is shown in Figure 11. This control scheme is a "dual max" (i.e., a maximum for each cooling and heating) system with "dual minimums." Dual minimums reflect how these units are typically operated with the unit turned off in the deadband. The lowest minimum reflects leakage flows. The other minimum reflects the minimum speed airflow of ECM powered terminal units when heating starts, typically ~12%. The heating coil controls the output unit the heating maximums discharge temperature is achieved and then the motor speed is increased up to the maximum heating volume. The dual minimum strategy discussed here is only included in CBE's development version.

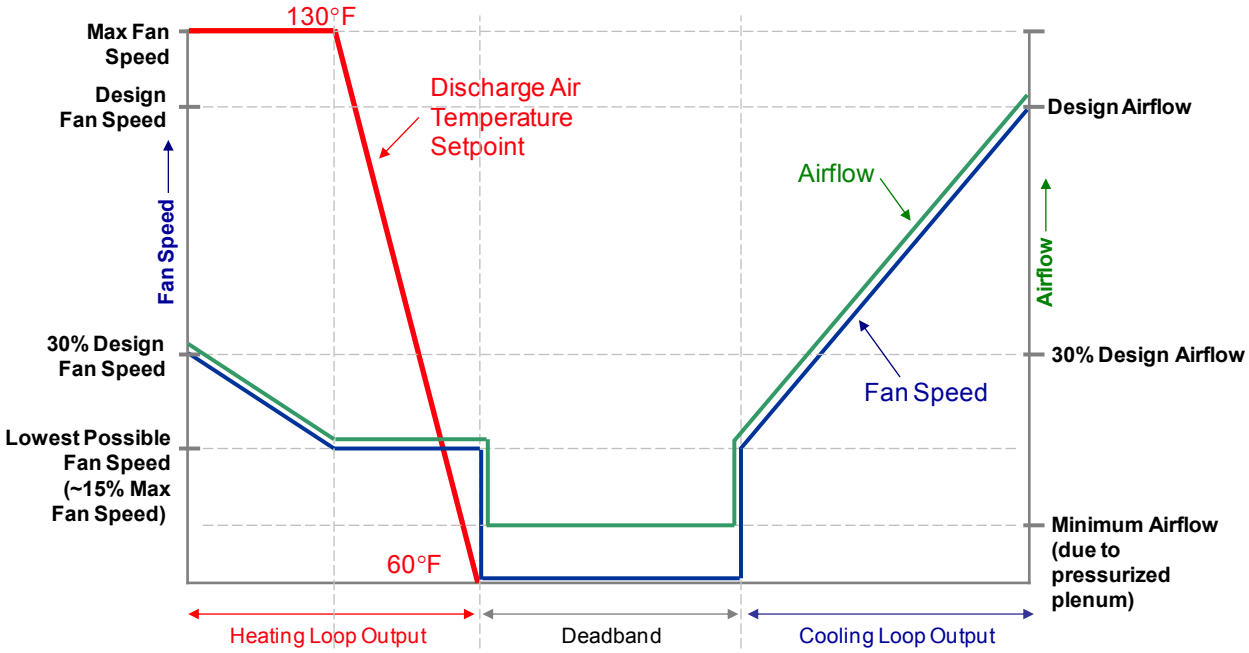

Figure 11. EnergyPlus variable speed terminal unit for UFAD with dual minimums

SYSTEM 2

Although this system type looks similar to a parallel fan powered VAV box, it is not. Fan powered boxes do not allow for zone air (more specifically, lower layer temperature air) to pass through the box during heating. Therefore, modeling this system requires separating the heating and cooling functions but with limitations imposed by EnergyPLus modeling; two terminal units can be used as long as one has its airflow path entirely within the zone. Figure 12 illustrated the EnergyPlus possibilities.

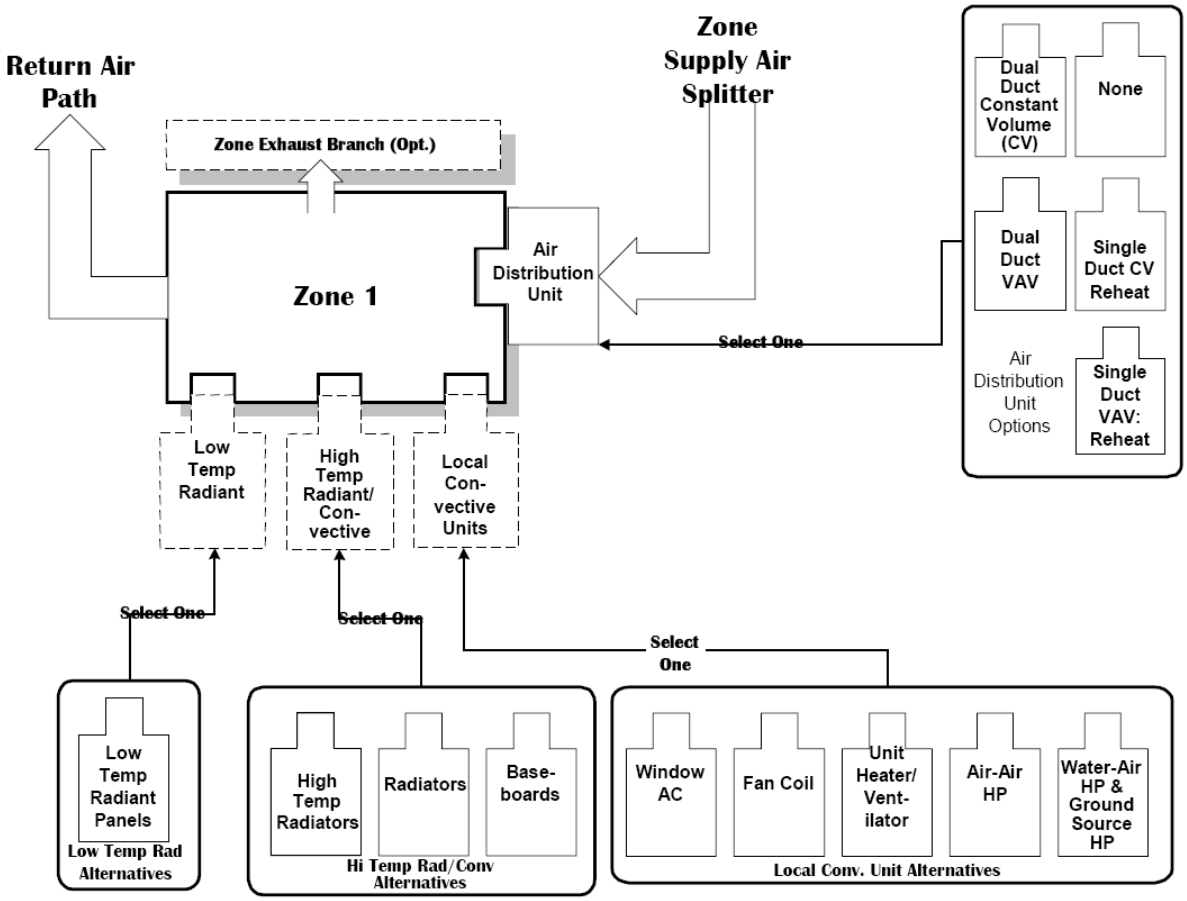

Figure 12. EnergyPlus terminal unit options

In this case we combine a single duct VAV box without reheat with a constant volume unit heater/ventilator local convection unit. The control diagram is shown in Figure 13.

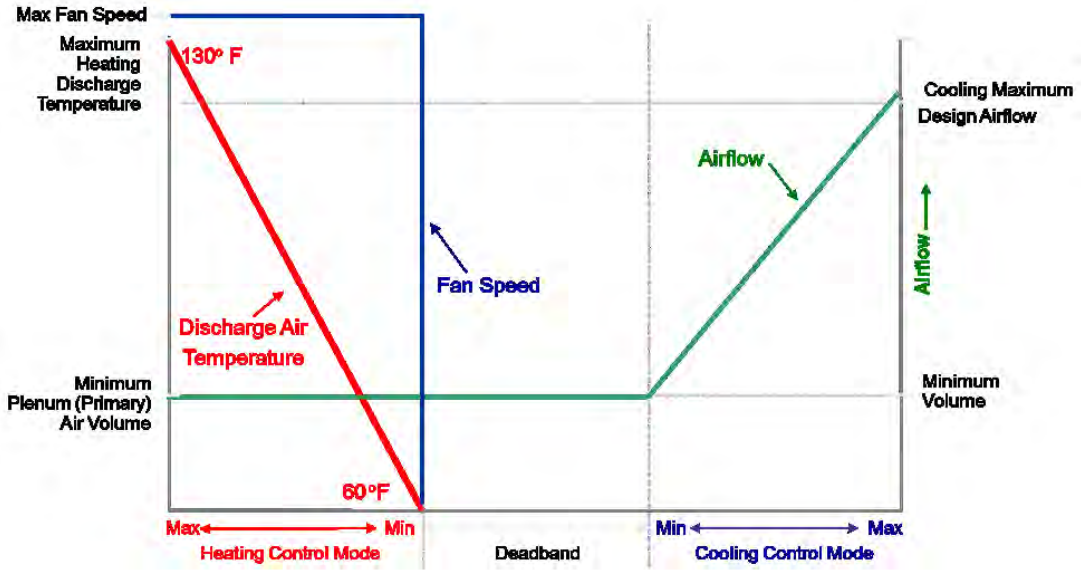

Figure 13. EnergyPlus System 2 control model

PLANT EQUIPMENT

Although ASHRAE 90.1 Appendix G specifies DX packaged rooftop units to be used for medium sizing buildings, for consistency and ease of debugging etc. we use a chilled water system regardless of the size of building we are simulating.

Chiller assumptions

Chillers: As specified in ASHARE 90.1, Appendix G, two screw chillers (using Chiller:EIR object) sized at 50% each (based on autosizing) are served by a single variable speed primary pump. Chiller staging is controlled using an idealized strategy where the first chiller is staged up to its optimum operating point; the second chiller loads to its optimum point, then both chillers increase capacity together to full load. Leaving chilled water setpoint is reset based on outside air temperature using the schedule shown in Figure 14.

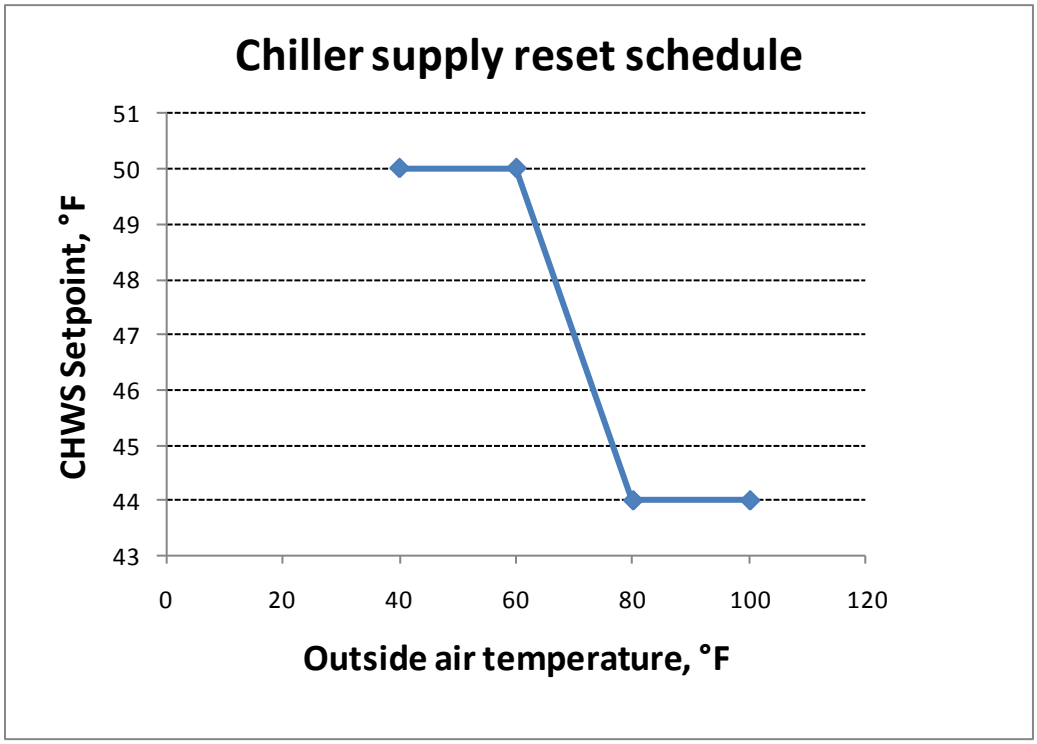

Figure 14. Example chilled water supply reset schedule

Chilled water pump (and boiler) variable speed pumps are modeled with parameters shown in Table 5 and part load curve shown in Figure 15.

Table 5. Chilled water pump performance

Chilled water pump eff	0.75
CHW pump design head	80 ft
Chilled water pump min DP	33%
Motor eff	0.90
Drive/VFD eff	0.98

VS pump PLR	FFP = 0.6307*PLF^3 + 0.0104*PLR^2 + 0.3533*PLR + 0.0114

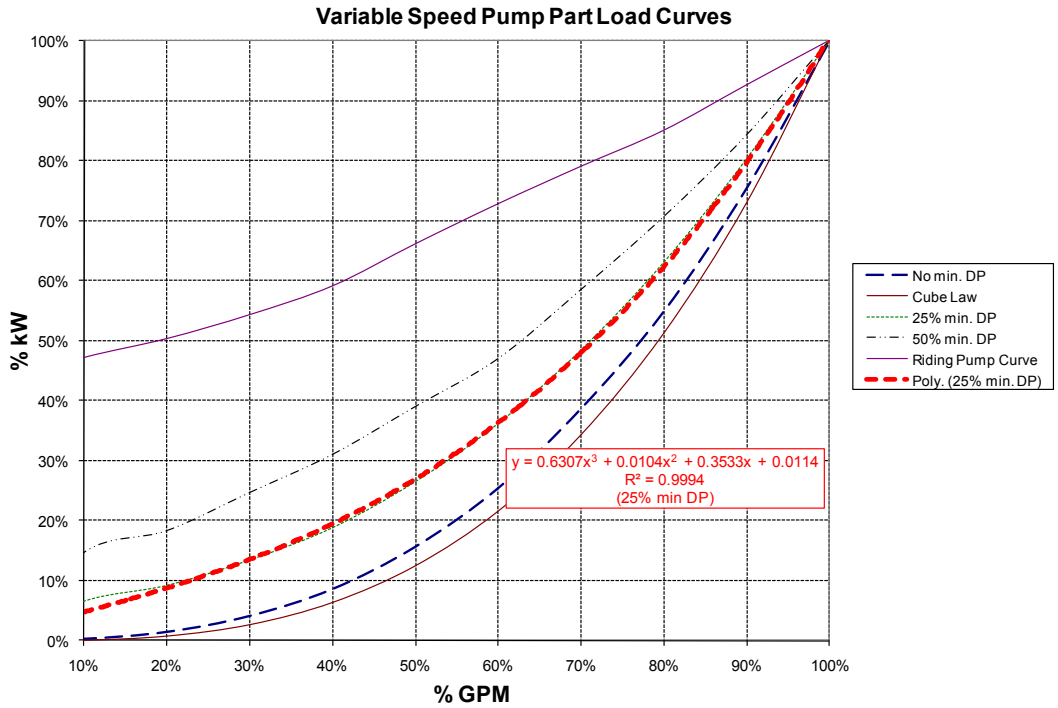

Figure 15. Chilled water and hot water variable speed pump performance curve

Boiler assumptions

Two non-condensing forced draft boilers are assumed using a 50/50% staging strategy. Boiler part load performance is modeled with DOE-2 forced draft curves shown in Figure 16. EnergyPlus uses a normalized part load curve (indicated in red) in its input structure. Hot water supply is reset according to the schedule shown in Figure 17. However, EnergyPlus has limited curve fit correlations none of which represent the type of curve shown. Therefore, we have adopted a two part linear curves shown to more accurately represent the low end performance.

We assume 180°F maximum boiler leaving water temperature reset to 150°F with outside air over the range of 20°F to 50°F, respectively.

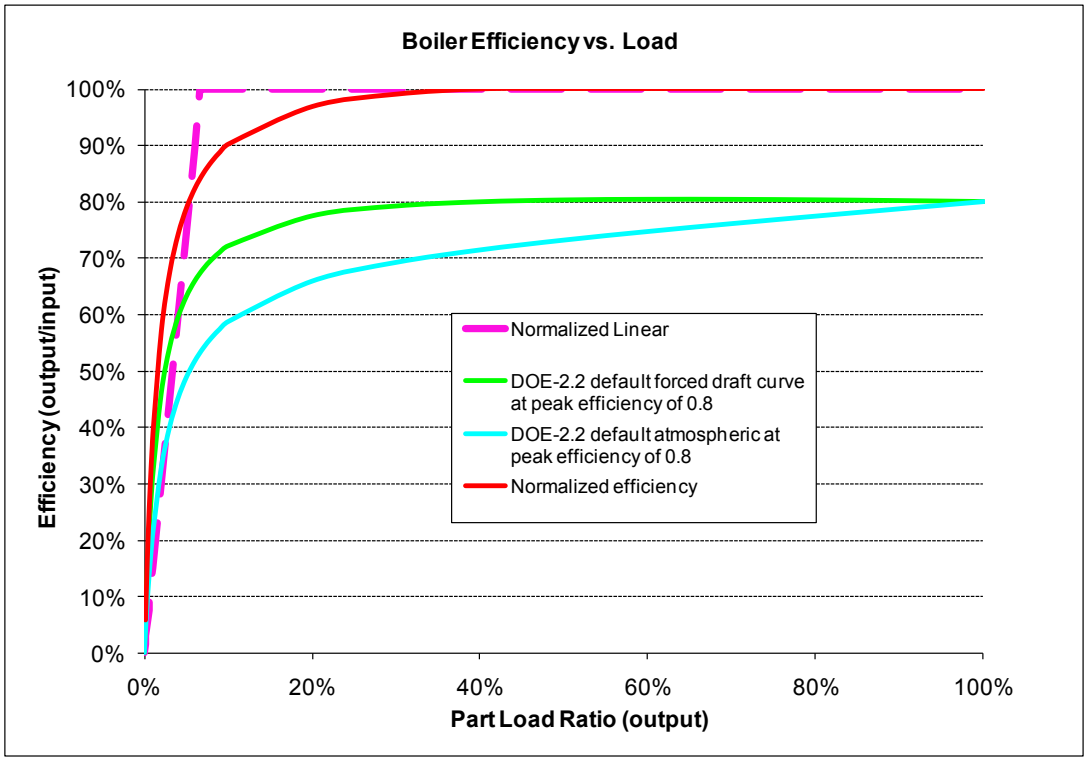

Figure 16. Forced draft boiler curve

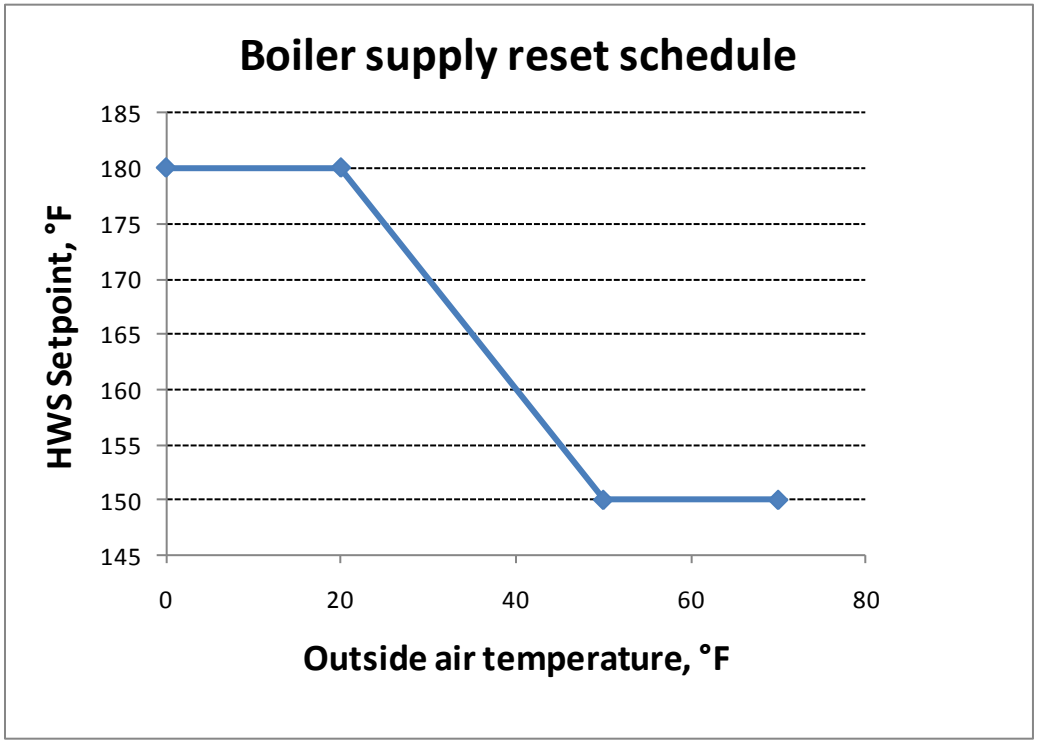

Figure 17. Boiler reset schedule

SIZING STRATEGIES

As mentioned previously in this report, CBE research indicates that EnergyPlus auto-sizing routines result in properly sized AHU fans for OH systems, but usually cause UFAD fans to be oversized. Proper sizing and specification of part load performance for air handlers is critical to accurate results. Erroneous results were encountered when sizing was not correctly specified or EnergyPlus autosizing routines were used for UFAD systems. One lesson learned is that the results from autosizing should always be checked before final simulations are conducted.

Chiller sizing

EnergyPlus chiller auto-sizing is not adequate for UFAD systems, therefore the following work-around was implemented to overcome these deficiencies.. A cooling design day simulation is run where the peak plant loop cooling demand is selected and a sizing factor of 0.6 is applied to each chiller to size the chillers for the annual run. Two chillers having sizing factor of 0.6 results in 20% over-sizing overall. Then the annual simulation is performed based on the chiller size determined from this work around.

Boiler sizing

Since problems similar to chiller sizing were encountered with boiler sizing, an alternative to the EnergyPlus autosizing, is implemented with CBE sizing strategies. A heating design day simulation is run where the peak plant loop heating demand is selected and a sizing factor of 0.6 is applied to each boiler to size the boilers for the annual run. Two boilers having sizing factor of 0.6 results in 20% oversizing overall. Then the annual simulation is performed based on the boiler size determined from this work around.

Terminal unit sizing

OH SYSTEMS

Sizing of VAV boxes is not a straightforward process and requires special care since it is closely related to the sizing of other system components such as chillers, coils, boilers, pumps, cooling towers and fans and has an impact on reheat energy. The user must explicitly specify the VAV box cooling design supply air temperature (SAT) for each VAV box (EnergyPlus object name: Zone Sizing). Using this SAT input, EnergyPlus sizes each VAV box based on the peak cooling load determined from a design day calculation. The higher the design SATs, the larger the VAV box size. For OH systems, we set the VAV box design SAT 0.5°C (0.9°F) higher than the air handling unit (AHU) SAT to provide a small safety factor to meet the high cooling loads during the annual simulation. The same design SATs are used for OH regardless of the different zones since the actual discharge air temperature from the diffusers are assumed to be equal to the AHU SAT without duct heat gains.[11] As of V6.0 release, EnergyPlus now allows minimum flowrates for VAV boxes to be set at a fixed fraction (of design cooling airflow) or a fixed amount (typically equal to the minimum ventilation requirement). Also, implemented is a "dual max" strategy where both cooling and heating maximums can be specified. These minimum settings are determined by a

[11] This represents a serious deficiency when comparing UFAD and OH, since it is known that OH systems also have duct heat gain; however, EnergyPlus has yet to implement a method for simulating this.

variety of factors but are primarily based on standards T-24 and ASHRAE 90.1, Appendix G. (See Appendix B.)

UFAD SYSTEMS

On the other hand for UFAD systems, different design SATs are used for each zone of the UFAD model to account for the impact of thermal decay in the supply plenum. These procedures are covered in more detail in Appendix A.

Determining the number of diffusers is another important element of UFAD sizing and it directly affects UFAD performance because it impacts stratification. Since determining number of diffusers has not been addressed in EnergyPlus autosizing procedures, we have implemented the work-around described in Appendix A.

Appendix A: Terminal Unit Sizing Procedures

OBJECTIVES

This sizing procedure aims to accurately estimate thermal decay with a preliminary design day run. This estimate serves to supply the zone design supply air temperature, which is used for sizing the maximum airflow rate of the ATUs. Thermal decay can be affected by many things, such as floor mass, window configuration, climate, and slab insulation, for example. Accurate sizing of the ATUs is important because it affects stratification heavily, as well as ATU minimum volumes which affect airflow and reheat energy.

BACKGROUND

Original method for ATU Sizing

Thermal Decay was previously accounted for in Air Terminal Unit (ATU) sizing with a coarse correction to the zone design supply air temperature. This input is found in the Sizing:Zone object and is parameterized in the interface as Zone_Cool_SAT_Perim[] and Zone_Cool_SAT_Core[] (at the time of the writing of this note, there is an obsolete variable called Zone_Cool_SAT_Serv[], intended to be the design SAT for the service core zones). For perimeter zones, 4°C was added to the design SAT, and in the core, 1.25 was added. The final design calculations proceed from here, so only one design simulation is performed.

This includes using higher design SATs for perimeter zones than for interior zones. Currently we add 1.3°C (2.3°F) to the AHU SAT setpoint for interior UFAD zones and 3.9°C (7.0°F) for perimeter zones. (This strategy was used circa 6/2011and is being revised to more accurately determine box size, especially for perimeter zones.)

APPROACH

Design Day for thermal decay

The new approach using (Excel macros) has been implemented as follows: A design day run is performed in order to estimate the thermal decay, and appropriately set the design SATs for the ATUs. In this initial run, the design airflow is oversized greatly so all peak loads will be covered; we assume 6 °C of thermal decay in the perimeter, and 2 °.C in the core zones. After the simulation is completed, the thermal decay for each zone can be accurately determined from the design day run. It is defined as the difference between EnergyPlus output “Zone/Sys Air Temperature” and the AHU SAT at the peak airflow condition. This is calculated separately for the middle floor core, and then averaged over the middle floor perimeter zones, for the zone design SAT for the core and perimeter zones respectively.

Design days for each exposure

This sizing strategy can be improved for both the peak airflow calculation and the thermal decay calculation. In order to better estimate the annual peak airflow and thermal decay for each exposure, a special design condition for each exposure can be added to the design day simulation. For example, the south zone peaks during the winter, due to a low sun angle.

ATU and number of diffusers sizing

The following procedure is used to size the ATUs for UFAD systems.

Select default diffuser type – Certain diffuser types for which we have laboratory test data for phi-gamma are included as defaults in EnergyPlus UFAD code. Table 2 summarizes the default types and their characteristics. Other types and/or modification of the defaults can be specified via the Custom command; users should contact the CBE team before attempting this, however. Note that the only factor that is not specified in code is the diffuser design airflow which is applied via the VBA macro procedures in the Excel use interface. [And in python code in the JEPlus parametric generator]

1) Specify the design Phi and "fake" thermal decay (oversizing ATU SAT) – This parameter provides a means for accurately sizing both the ATU and the number of diffusers
2) Run initial "oversizing" design day with autosizing on – Run all of the design days (see above) with the oversizing ATU SAT and design phi. This yields an accurate ATU inlet temperature for all zones since the airflow will be modulated to something less than the large box size.
3) Run second final sizing run with autosizing off? - From previous run, enter the ATU SATs and rerun design day with design phi. This provides an accurate box size.
4) Number of diffusers - Once the peak airflow (ie. Box size) is determined, the number of diffusers for each zone is found by dividing the design airflow by the diffuser nominal design airflow (See Table 2).
    a) York/JCI systems - The design phi is updated to correspond to the phi determined from the gamma-phi equation at the gamma for the peak design condition. As noted in Table 2, the MIT2 gamma-phi correlation is assumed represent all diffusers for York/JCI products. This phi is then fixed for all runs; i.e., the gamma-phi calculation is overridden.

        The design number of diffusers can also be entered by the user to override the design calculation above (e.g., to study impact of increasing number of diffusers). When this is done for the Walnut/JCI products it assumes that the plenum pressure has been lowered to provide a lower design volume; the constant phi is then entered corresponding to the new design gamma based on the user entered number of diffusers.
    b) For other diffusers, the phi is calculated hourly from the gamma-phi equation after the design number of diffusers is entered.
    c) For linear bar grilles the total length of n diffusers should not exceed the length of the zone; there is no check for this.
5) Conduct annual or other runs – The interface (and python code) allows for making "operating runs" or "design runs". In the case of operating runs, the sizing procedure is run once and then all subsequent runs are made with the same sizing results.

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Cooling load differences between radiant and air systems

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Abstract

Unlike the case of air systems where the cooling load is purely convective, the cooling load for radiant systems consists of both convective and radiant components. The objectives of this simulation study were to investigate whether the same design cooling load calculation methods can be used for radiant and air systems by studying the magnitude of the cooling load differences between the two systems over a range of design configurations. Simulation results show that 1) zone level 24-hour total cooling energy of radiant systems can be 5-15% higher than air systems due to higher conduction load through the envelope; 2) peak cooling rate at the radiant cooled surface can be 7-35% higher than air system for zones without solar load. This difference can increase up to 85% for floor system in zones with solar load; 3) the peak cooling rate differences originate from: a) radiant cooling surface(s) reduce radiant heat gain accumulation in the building mass; b) only part of the convective heat gain becomes instantaneous cooling load. As a result, tools using response factor methods such as Radiant Time Series Method for cooling load calculations are not appropriate for radiant system design.

1. Introduction

Water-based radiant cooling systems are gaining popularity as an energy efficient approach for conditioning buildings [1-3]. The design of radiant systems is complicated because of the coupling between thermal load, building structure and the hydronic system and because of the important impact of both radiation and convection on thermal comfort. Dedicated radiant system design and testing standards have been developed to address issues like system sizing, installation, operation and control [4-9]. However, radiant cooling systems are still considered as an innovative approach, and their application in North America is still limited [10, 11]. In this study, we investigated the impacts of the presence of activated cooled surface on zone cooling loads.

Cooling load calculations are a crucial step in designing any HVAC system. Compared to air systems, the presence of an actively cooled surface changes the heat transfer dynamics in the room, and two potential impacts on zone cooling loads studied here are: 1) cooled surfaces may create different inside surface temperatures of the non-active exterior building walls, causing different heat gain through the building envelope, and in turn different zone level total energy, and 2) changes the effect of thermal mass on cooling loads, and therefore creating different peak cooling load.

Two research studies were identified that looked at heating load calculations in terms of the impact of the radiant system on wall surface temperatures and the resultant room load [12, 13]. However, both studies focused on heating load calculation under steady-state conditions. In another study, Chen [14] suggested that the total heating load of a ceiling radiant heating system was 17% higher than that of the air heating system because of the role of thermal mass and higher heat loss through the building envelope due to slightly higher inside surface temperatures. For cooling applications, no studies were found on this topic, and in current radiant system design guidelines [4, 8], such impacts are not considered or evaluated.

Secondly, the interaction of building mass with heat source is influenced by the presence of activated radiant cooling surface(s). One phenomenon mentioned in the literature was radiant surface(s) as part of the building mass, instead of storing themal energy as in the case of air systems, removes radiant heat gain (e.g. solar, radiative internal load and radiative envelope load) that is directly impinging on it. This phonomenon fundamentally changes the cooling load dynamics in a room. Niu [15] pointed out that this direct radiation may create high peak cooling loads. He modified the thermal analysis program ACCURACY [16] to account for the direct radiant heat gain as instantaneous cooling load for radiant systems. However, no information can be found on how he implemented the modification and the software is not accessible for the public. In an effort to develop a new cooling load calculation approach for radiant systems, Corgnati [17] also tackled the direct radiant heat gain effect using a similar strategy to Niu. Based on Corgnati's work, Causone et al. [18] focused on the cases with the presence of direct solar gain. However, the methods proposed in these research studies only looked at the effect of direct radiant heat gain on cooling load, and the rest of the radiant heat gain and the convective heat gain are still considered to interact with building mass as if the radiant system does not exist. In addition, no research can be found that fundamentally studies the differences of the heat transfer process in zones conditioned by an air and a radiant system, and how these differences are going to impact the cooling load calculation and what could be the magnitude of the differences.

Although research has demonstrated that cooling loads for radiant systems need to be considered differently than for air systems, current radiant design standards do not explicitly acknowledge these differences. Several standards and handbooks were reviewed, including: chapter 6 of *ASHRAE Equipment and HVAC systems* [19], radiant heating and cooling handbook (2002) [9], chapter 18 of *ASHRAE Fundamental* (2012), ISO 11855 (2012) [4], and European standard EN 15377 (2008) [8]. The first three do not offer any guidance on the selection of the calculation methods when radiant systems are involved. In chapter 18 of *ASHRAE Fundamental* (2012) handbook, the description of the cooling load calculation process is based on the implicit assumption that an air system is used for conditioning the space. Some simplified cooling load calculation methods, such as Transfer Function method (TF)[20] and Radiant Time Series method (RTS)[21], have also been developed for air system. These algorithms are widely implemented in building thermal simulation or load calculation tools, including HAP (TF), TRANE TRACE (RTS), BLAST, and DOE-2 (TF) based tools such as eQuest, Energy-pro, Green Building Studio and VisualDOE. These tools are often used for cooling load estimates during initial design stage and for detailed energy and comfort analysis even when radiant systems are involved [22]. The European standards reviewed indirecly reference EN 15255 [23] for cooling load calculation procedure. EN 15255 classified all cooling load calculation methods into different catogories according to their capability to model different types of cooling system and control method. Methods that are able to simulate radiant systems controlled by operative temperature are in Class 4b. This implies that cooling load calculation method for radiant systems should be properly distinguished from air systems. However, this standard does not explicitly provide cooling load calculation mehtods for radiant system.

A recent survey conducted by the authors of radiant cooling design practitioners revealed that the differences in cooling load between radiant and air systems are not fully understood. Some of the most experienced professionals acknowledge the complications and lack of guidance in the standards and developed rule-of-thumb methods for initial system design calculation. Among those methods, either heat gain is directly used as cooling load for system sizing [24, 25], or a portion of the heat gain is considered as direct heat removal by the active radiant surface. The percentages of the direct removal depend on load type (lighting/people/equipment), and are obtained based on experience [26]. In design practice, it is not often that dynamic simulation tools that can properly model radiation heat transfer are used at the cooling load estimation stage. Radiant system manufacturers have developed some tools for system sizing [27], but they are mainly used for heating applications, where steady-state heat transfer is adequate to capture the thermal behavior.

The objectives of this simulation study are to 1) assess the cooling load differences between the two systems by comparing the zone level peak zone cooling load and 24-hour total cooling energy for a

radiant cooling system (with activated chilled surface) vs. an air system; and 2) suggest potential improvements in current design guidelines for radiant cooling system.

2. Background and theory

In this section, we give a brief introduction to the three types of radiant cooling systems investigated in this paper and explain how their thermal characteristics affect the design approach. Since radiant and air systems are different in many ways, the simulation study had to be designed carefully to provide a fair comparison.

2.1. Radiant cooling systems

The REHVA guidebook on radiant systems [7] has roughly categorized these systems into three types: radiant cooling panels (RCP), water-based embedded surface cooling systems (ESCS), and thermally activated building systems (TABS). As shown in Figure 1, RCP are metal panels with integrated pipes usually suspended under the ceiling with heat carrier temperature relatively close to room temperature. ESCS have pipes embedded in plaster or gypsum board or cement screed, and they are thermally decoupled from the main building structure (floor, wall and ceiling) by the use of thermal insulation. They are used in all types of buildings and work with heat carriers at relatively high temperatures for cooling. Finally, "systems with pipes embedded in the building structure (slab, walls), TABS, which are operated at heat carrier temperatures very close to room temperature and take advantage of the thermal storage capacity of the building structure." These systems usually have different applications due to their thermal and control characteristics, and therefore, the design and dimensioning strategies for these systems vary.

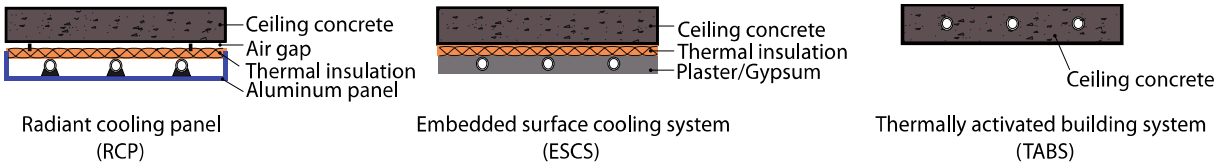

Figure 1: Schematic of the three types of radiant surface ceiling systems (not to scale)

2.2. Radiant vs. air systems

A comparison between radiant and air systems is challenging. In this section, we discuss the differences between the two systems that dictate the modeling approach used in this study. Besides those mentioned in the literature [28], the main difficulties include:

- Types of load (sensible/latent) and the expected amount of load to be handled by the two systems are different. Air systems are usually designed to be the only system to handle both latent and sensible loads, while radiant systems must operate in hybrid mode with a reduced-sized air system (for ventilation and latent loads). Radiant cooling systems are always sized to handle a portion (as much as possible) of the sensible-only cooling load. To address this issue, neither the latent load nor ventilation system was simulated. This was to simplify our analysis.
- The design cooling load concept is different for the two systems. According to ASHRAE Handbook [29], the sensible cooling load for an air system is calculated in terms of maintaining a constant zone air temperature, while radiant systems, particularly TABS, are not capable of maintaining a constant zone air temperature due to large thermal inertia of the active surfaces. For this reason, in this comparison study, we sized and controlled the simulated radiant systems to maintain an acceptable thermal comfort range during the simulation period. Operative temperature was used as the control temperature for both systems [28, 30]. To ensure equivalent comfort conditions between the two systems for fair comparison, all simulations of the air system were subsequently controlled to closely track the hourly operative temperature profile derived from the radiant system simulation for the identical input conditions.
- For an air system the zone cooling load is equal to the heat extraction rate by the mechanical system when the room air temperature and humidity are constant. But this is not always the

case in a radiant system. Other than panel systems, radiant cooling systems (ESCS and TABS) are integrated with the building structure with hydronic pipes embedded in the mass. As a result, heat removed from the zone at the chilled surface can be quite different from the heat removed by the hydronic loop. Sizing of the radiant system cooling equipment is highly dependent on specifications of the cooling surface (slab material/thickness, tube spacing, and surface finishing). This indicated that we needed to investigate heat transfer of the radiant system at both the surface and hydronic levels, which is discussed in detail below.

2.3. Heat transfer at radiant surface and hydronic level

Radiant systems remove the sensible heat in a room at the cooling surface. We define this cooling rate as surface cooling rate. Define the control volume as the inside face of the cooling slab, with positive sign means heat being transferred into the control volume and negative indicates heat leaving the control volume, the heat balance for the cooling surface can be written as follows (1) [31]:

$$q''_{surf} = q''_{surf,conv} + q''_{surf,rad} = -q''_{surf,cond} \quad (1)$$

Surface cooling rate serves as one key design parameter for determining required radiant system area and selection of system type.

Hydronic cooling rate is the heat extraction rate based on an energy balance on the hydronic circuit. The hydronic cooling rate is important for sizing of waterside equipment, such as pumps, chillers and cooling tower. Hydronic cooling rate can be calculated by equation (2) [31]:

$$q''_{hyd} = \left(\dot{m}c_p\right)_{water}(T_{wi} - T_{wo}) \quad (2)$$

Both RCP and most ESCS operate during occupied hours to maintain a relatively constant comfort condition in the space, so the difference between the surface and hydronic rate is only a function of thermal properties of the panel/slab. For RCP systems, if insulation is installed on the backside of the panel, hydronic cooling rate can be assumed to be the same as surface cooling output due to high conductivity of the surface material [6], which is usually desired. TABS are usually designed and operated to take advantage of the thermal storage effect of the slab, so the difference between the surface and hydronic rate is also a function of the operational strategies, which will be discussed later.

3. Methodology and modelling approach

To investigate the impacts of the presence of activated cooled surface on zone cooling load, we adopted the following methodology:

- Two single zone models, one conditioned by an air system and one by radiant system were developed in EnergyPlus v7.1 for comparison. All three radiant systems (RCP/ESCS/TABS) were studied. Because the construction of each radiant system type is different and is highly influential on overall building response, the comparison air models were configured to match the construction of the radiant systems.
- The models were parameterized for studying the influences of envelope thermal insulation, thermal mass, type of internal gain, solar heat gain with different shading options, and radiant surface orientation (ceiling, floor).

EnergyPlus v7.1 was used for the simulation study because it performs a fundamental heat balance on all surfaces in the zone. The heat balance model ensures that all energy flows in each zone are balanced and involve the solution of a set of energy balance equations for zone air and the interior and exterior surfaces of each wall, roof, and floor. It captures both longwave and shortwave radiation heat transfer and has been extensively validated [32, 33]. In addition, EnergyPlus is able to integrate the heat transfer calculation in the radiant cooling systems with changing zone conditions; therefore it is able to capture the transient behavior of the systems [31].

3.1. Simulation Runs

In total, seventy-four simulation cases were configured, including 13 (11 for RCP) variations for the three types of radiant systems and their equivalent air systems. The different combinations and ranges of parameters are listed in Table 1.

Cases hw_r2 and hw_r1 in Group 1 are designed for studies of the impact of thermal insulation, and hw_r2 and lw_r1 in Group 2 are for studies of thermal mass. These represent perimeter zones without windows, only subjected to building envelope conductive heat gains. Cases in G3, rad0 to rad1, are to evaluate the impacts of internal load with different radiant fractions, defined as the portion of radiative heat gain to total heat gain given off by a heat source. Radiant fraction of lighting ranges from 0.48 to 1.0 depending on luminaire type [34]; for people the radiant fraction can be from 0.2 to 0.6 depending on surrounding air velocity and people's activity (e.g., walking, running, etc.) [29]; and for office equipment, the range is usually between 0.1 to 0.4 depending on equipment type [35]. For these cases, the building envelope was set to be adiabatic to represent an interior zone and isolate the influences from outside environment. Two windows were modelled on the south wall in the next groups, G4-G6, in order to study the impact of solar gains in perimeter zones. Radiant ceiling and floor systems were both simulated. Case cl_shade_rad0.6 was configured to represent a zone with real internal load and windows with exterior shading that is conditioned by a radiant ceiling system. All three types of radiant systems were modelled for all cases, except that the RCP systems were not simulated for the radiant floor case because it is not a common practice.

Table 1: Simulation runs summary

Group	Case	Building	Int. heat gain[1]	Window	Radiant surface	Boundary conditions[3]
G1: insulation	hw_r2	heavyweight	no	no	ceiling	environment
	hw_r1	hW_smallR[2]	no	no	ceiling	environment
G2: thermal mass	hw_r2	heavyweight	no	no	ceiling	environment
	lw_r2	lightweight	no	no	ceiling	environment
G3: Int. heat gain[1]	rad0	heavyweight	RadFrac[1] = 0	no	ceiling	adiabatic
	rad0.3	heavyweight	RadFrac=0.3	no	ceiling	adiabatic
	rad0.6	heavyweight	RadFrac=0.6	no	ceiling	adiabatic
	rad1	heavyweight	RadFrac=1	no	ceiling	adiabatic
G4: ceiling with solar	cl_ noshade	heavyweight	no	yes	ceiling	environment
	cl_shade	heavyweight	no	yes+shade	ceiling	environment
G5: floor with solar[4]	flr_noshade	heavyweight	no	yes	floor	environment
	flr_shade	heavyweight	no	yes+shade	floor	environment
G6: typical ceiling	cl_shade_rad0.6	heavyweight	RadFrac = 0.6	yes+shade	ceiling	environment

Note:

1. Int. heat gain= Internal heat gain; RadFrac = Radiative fraction of internal heat gain;
2. HW_smallR=Heavy weight construction with half thermal insulation at exterior walls;
3. Both roof and floor have boundary conditions set to adiabatic for simplicity, and the boundary conditions specified in this column are for exterior walls;
4. These cases are not simulated for radiant panel systems.

3.2. Model Specifications

Since the objective of the study was to understand the heat transfer and the resultant cooling load differences between a radiant and an air system, a representative single zone model is adequate. The model was developed primarily based on ASHRAE Standard 140 [36]. Weather file provided in the standard was used. System and design parameters for the radiant system were adopted from RADTEST [37]. Additional details are summarized below.

The test case (Figure 2) was a rectangular, heavy weight construction single zone building (8 m wide × 6 m long × 2.7 m high) with no interior partitions. Both the floor and roof boundary conditions were set to be adiabatic to simplify the analysis. Only cases in G4-G6 have 12 m^2 of south-facing windows. The overall U-Factor was 2.721 $W/(m^2 \cdot K)$ with Glass SHGC at 0.788. The baseline construction was based on case 900 (ASHRAE 140 2007 Table 11), except that the ceiling/floor

constructions were modified so that radiant ceiling/floor systems can be simulated. Exterior walls for Case hw_r2 had U-value of 0.454 W/(m^2·K). Case hw_r1 was modified to have U-value of 0.83 W/(m^2·K), and Case lw_r2 was modified with lightweight construction based on case 600 (ASHRAE 140 2007 Table 1). Floor and ceilings were configured separately for each case depending on location of the activated cooling surface and radiant system types. Table 2 is a summary of the radiant ceiling/floor construction specifications. For cases in G3, the internal gain was 720 W from 6:00 to 18:00. The radiant fraction was different for each run as specified in Table 1. There was zero air infiltration for all runs because we did not want to have an additional confounding factor. Table 3 lists the radiant system design specifications that are developed based on RADTEST case 2800. When windows were simulated, tube spacing changed from 0.3 m to 0.15 m in order to maintain similar thermal comfort level. Design flow rates for RCP were reduced for cases in Group 1 and 2, since these systems have higher cooling capacity as compared to the other two radiant systems. As for control, the goal was to maintain operative temperature setpoint at 23°C for 24 hours with a 2°C deadband [31]. For the air system models, the EnergyPlus object "IdealLoadsAirSystem" was used for simplicity to ensure the same operative temperature as the corresponding radiant systems.

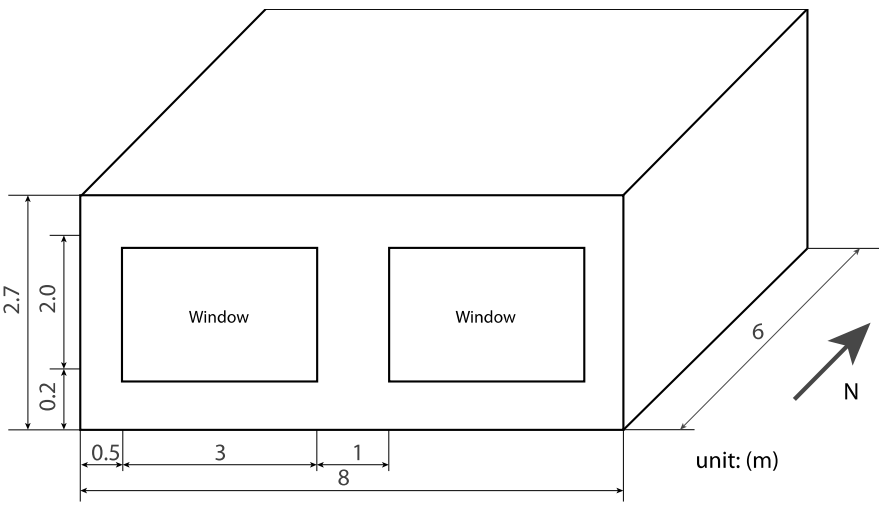

Figure 2: Isometric Base Case (Only G4-G6 have windows)

Table 2: Radiant surface constructions specifications (inside to outside)

	Thickness (m)	Specific Heat (J/kg·K)	Density (kg/m^3)	Conductivity (W/m·K)
RCP ceiling				
Aluminum panel	0.001	910	2800	273.0
Water Tube				
Insulation	0.05	1210	56	0.02
Concrete slab	0.08	1000	1400	1.13
Insulations	0.1118	840	12	0.04
Roof deck	0.019	900	530	0.14
ECS ceiling				
Lime plaster	0.012	840	1050	0.7
Water Tube				
Lime plaster	0.014	840	1050	0.7
Insulation	0.05	1210	56	0.02
Concrete	0.08	1000	1400	1.13

Insulations	0.1118	840	12	0.04
Roof deck	0.019	900	530	0.14
ECS floor				
Floor finish	0.0016	1250	1922	0.17
Cement Screed	0.04	988	1842	1.2
Water Tube				
Cement Screed	0.01	988	1842	1.2
Insulation	0.05	1210	56	0.02
Concrete	0.08	1000	1400	1.13
Insulation	1.007	n/a	n/a	0.04
TABS ceiling				
Concrete	0.04	1000	1400	1.13
Water Tube				
Concrete	0.04	1000	1400	1.13
Insulations	0.1118	840	12	0.04
Roof deck	0.019	900	530	0.14
TABS floor				
Concrete	0.04	1000	1400	1.13
Water Tube				
Concrete	0.04	1000	1400	1.13
Insulations	1.007	n/a	n/a	0.04

Table 3: Hydronic loop specifications

Inner diameter (m)	0.015
Total pipe length (m)	139.2
Inlet water temp (°C)	15
Tube spacing (m)	0.3 (0.15 for cases with windows)
Design mass flow rate (kg/s)	0.167 (0.06 for RCP system in cases without window)

4. Parameters investigated

Table 4 lists the parameters that were evaluated during the simulations. Peak cooling rate is commonly used for equipment sizing in the case of air system and the fast responsive RCP and lightweight ESCS. 24-hour total cooling energy is studied for all radiant systems because it reflects the consequence of the impact of radiant cooling system on exterior wall surface temperature. Comparisons were made at both the surface and hydronic levels for the radiant systems. Percentage differences between the radiant and air systems were reported, and are defined in the last two rows in Table 4.

Table 4: Parameters analyzed

	24 hour-total cooling energy	Peak cooling rate
Air system	24-hour total sensible cooling energy, *kJ/m²* ($\dot{q}_{air,tot}$)	Specific peak sensible cooling rate, *W/m²* ($q''_{air,pk}$)
Radiant system	24-hour total surface cooling energy, *kJ/m²* ($\dot{q}_{surf,tot}$)	Specific peak surface cooling rate, *W/m²* ($q''_{surf,pk}$)
	24-hour total hydronic cooling energy, *kJ/m²* ($\dot{q}_{hyd,tot}$)	Specific peak hydronic cooling rate, *W/m²* ($q''_{hyd,pk}$)
Percentage difference	$P_{surf,tot} = \frac{(\dot{q}_{surf,tot} - \dot{q}_{air,tot})}{\dot{q}_{air,tot}} \times 100\ \%$	$P_{surf,pk} = \frac{(q''_{surf,pk} - q''_{air,pk})}{q''_{air,pk}} \times 100\ \%$

	$P_{hyd,tot} = \frac{(\dot{q}_{hyd,tot} - \dot{q}_{air,tot})}{\dot{q}_{air,tot}} (\%) \times 100\ \%$	$P_{hyd,pk} = \frac{(q''_{hyd,pk} - q''_{air,pk})}{q''_{air,pk}} \times 100\ \%$

5. Results

Results from the 99.6% cooling design day simulations are reported and compared for surface cooling rate, hydronic cooling rate and air system cooling rate in this section. To evaluate the influence of each investigated parameter, the ranges of the $P_{surf,pk}$, $P_{hyd,pk}$, $P_{surf,tot}$, and $P_{hyd,tot}$ are reported graphically.

5.1. 24-hour total cooling energy

The expected impact of the radiant cooling system is to cause lower surface temperatures at the inside of building envelope, resulting in higher envelope heat gain and total cooling energy. This hypothesis was tested by a comparison of the 24-hour total envelope heat gain for a zone conditioned by a radiant vs. air system, as shown in Table 5. For cases in G1 and G2, the heat gains were merely heat conduction through exterior walls, and for the other cases, the heat gains also included solar radiation through windows. G3 cases were not reported because they were modeled to have adiabatic boundary conditions for all exterior surfaces that resulted in near zero heat gain through the building envelope. Table 5 shows higher conductive heat transfer through the building envelope for the radiant system. The reason for this finding was the lower surface temperature (at an average of 0.5°C) at the inside face of the exterior walls caused by the radiant system, as is proved by Figure 3. Table 6 presents the summer design day 24-hour total cooling energy for both radiant and air systems. Comparing heat gain differences between the two systems reported in Table 5 and the 24-hour total energy differences reported in

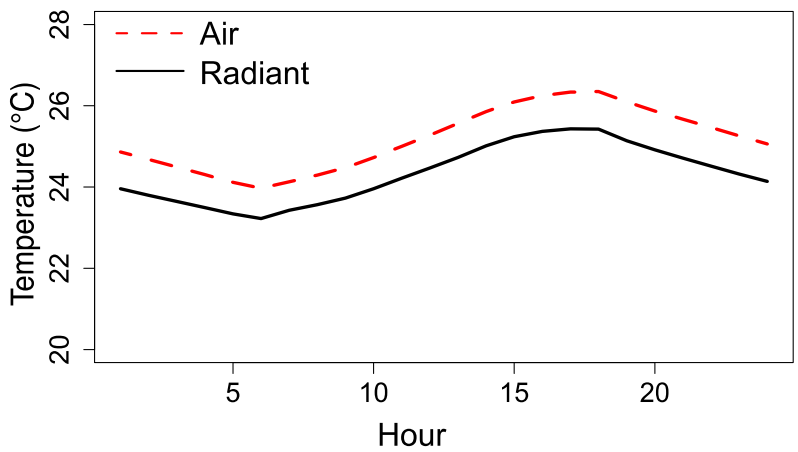

Figure 3: Comparison of temperatures at the inside surface of exterior wall between radiant and air systems. (G6 typical ceiling: cl_shade_rad0.6)

Table 6, we can confirm that heat gain through the building envelope caused higher 24-hour total cooling energy for the radiant systems.

Table 5: Comparison of 24-hour total heat gain through building envelope

Group	Cases	RCP (kJ/m^2)	Air (kJ/m^2)	% diff	ESCS (kJ/m^2)	Air (kJ/m^2)	% diff	TABS (kJ/m^2)	Air (kJ/m^2)	% diff
G1	hw_r2	391	368	6.2%	401	376	6.6%	403	377	6.9%
	hw_r1	630	582	8.2%	651	600	8.5%	652	600	8.8%
G2	hw_r2	391	368	6.2%	401	376	6.6%	403	377	6.9%
	lw_r2	440	424	3.9%	443	425	4.3%	445	422	5.5%
G4	cl_ noshade	1,956	1,735	12.7%	1,898	1,678	13.1%	1,902	1,679	13.3%
	cl_shade	1,245	1,155	7.8%	1,226	1,137	7.8%	1,230	1,139	8.0%

G5	flr_noshade	NA	NA	NA	1,946	1,710	13.8%	1,909	1,674	14.0%
	flr_shade	NA	NA	NA	1,249	1,147	8.9%	1,239	1,137	9.0%
G6	cl_shade_rad0.6	1,244	1,132	9.9%	1,195	1,086	10.1%	1,200	1,088	10.3%

Note: Group 2 cases have adiabatic boundary conditions, therefore, no heat transmission through building envelope

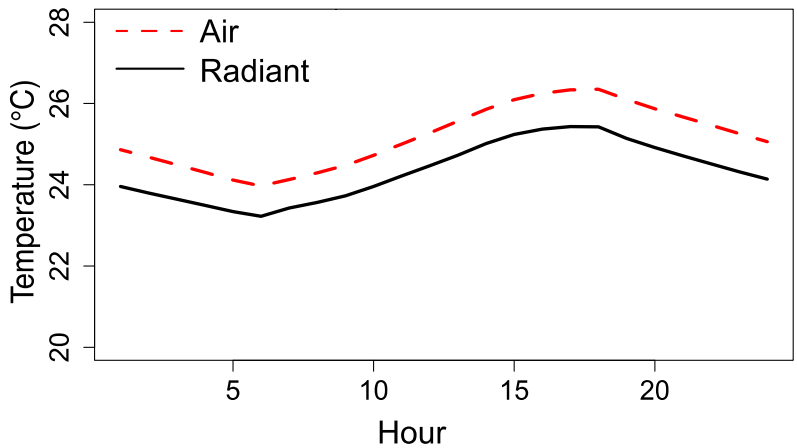

Figure 3: Comparison of temperatures at the inside surface of exterior wall between radiant and air systems. (G6 typical ceiling: cl_shade_rad0.6)

Table 6: 24-hour total cooling energy comparison for summer design day

Group	Cases	RCP vs. Air (kJ/m^2)			ESCS vs. Air (kJ/m^2)			TABS vs. Air (kJ/m^2)		
		$q''_{surf,tot}$	$q''_{hyd,tot}$	$\dot{q}_{air,tot}$	$q''_{surf,tot}$	$q''_{hyd,tot}$	$q''_{air,tot}$	$q''_{surf,tot}$	$q''_{hyd,tot}$	$q''_{air,tot}$
G1	hw_r2	391	391	368	401	403	376	403	406	377
	hw_r1	630	630	582	651	654	600	654	659	600
G2	hw_r2	391	391	368	401	403	376	403	406	377
	lw_r2	441	441	421	444	445	419	446	445	420
G3	rad0	647	646	647	644	636	647	647	642	649
	rad0.3	648	647	647	650	647	647	648	647	646
	rad0.6	648	649	648	648	651	648	646	647	648
	rad1	648	648	648	648	652	649	648	656	648
G4	cl_ noshade	1,949	1,948	1,730	1,892	1,903	1,676	1,897	1,920	1,679
	cl_shade	1,236	1,234	1,153	1,221	1,229	1,136	1,226	1,244	1,143
G5	flr_noshade	NA	NA	NA	1,936	1,954	1,699	1,899	1,906	1,674
	flr_shade	NA	NA	NA	1,244	1,259	1,140	1,234	1,241	1,141
G6	cl_shade_rad0.6	1,861	1,858	1,754	1,816	1,827	1,717	1,823	1,848	1,722

Figure 4 plots the range of $P_{surf,tot}$ (left) and $P_{hyd,tot}$ (right) for each group investigated for RCP, ESCS, and TABS. For example, in the left plot, the first black bar in "G1: insulation" represents the range of $P_{surf,tot}$ for cases in the first group, with the lower end representing $P_{surf,tot}$ for case hw_r2, and the high end representing $P_{surf,tot}$ for case hw_r1. $P_{surf,tot}$ and $P_{hyd,tot}$ are defined in Table 4, and can be calculated using data from
Table 6. Note that since there is only one case in G6 for each type of radiant system, the single lines represent $P_{surf,tot}$ for the cases cl_shade_rad0.6.

From Figure 4, we can see that the differences in surface/hydronic level 24-hour total energy between the two conditioning systems were influenced by the thermal insulation in exterior walls but only slightly influenced by thermal mass of the building. Compared to Case hw_r2, Case hw_r1 had half the thermal insulation in exterior walls and the percentage difference in hydronic total cooling energy increased from 6% to 8% for the RCPs, 7-9% for ESCS, and 8-10% for the TABS; similar ranges were seen at the surface level. G3 cases have adiabatic boundary conditions, and therefore, have negligible differences in total cooling energy. For G4 and G5, the total surface energy was 6-14% higher, and hydronic energy was 6-15% higher. The difference in total energy was not sensitive to the type of radiant surface (ceiling or floor), but was sensitive to the amount of direct solar radiation. When exterior shadings were modeled, $P_{surf,tot}$ and $P_{hyd,tot}$ decreased about 5%. This means higher window surface temperature (caused by direct solar) enhanced the radiation heat transfer between the window surfaces and radiant cooling surface, and resulted in larger heat gain through the window for radiant system.

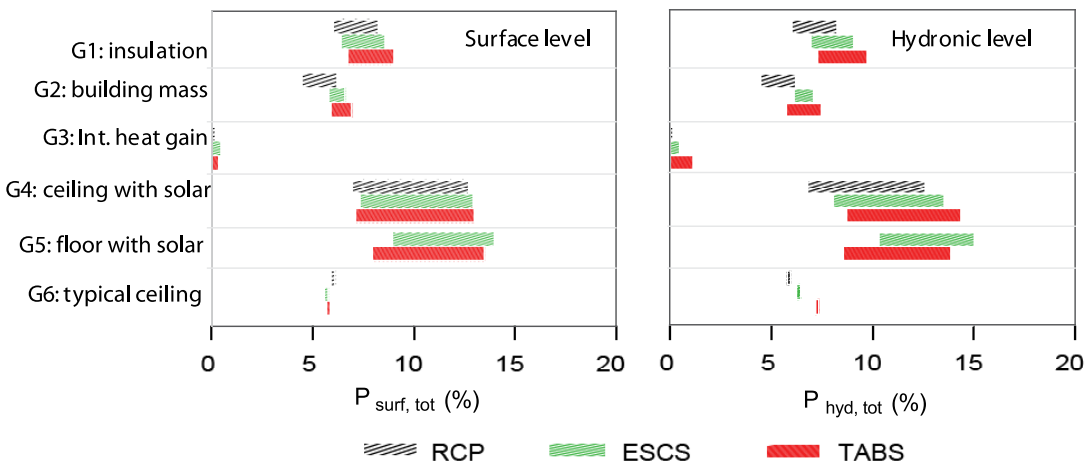

Figure 4: Range of 24-hour total energy percentage difference between air system and radiant system at surface level (left) and hydronic level (right)

The three types of radiant systems displayed similar trends. For RCP systems, zone hydronic level cooling energy was almost the same as surface level, while for the ESCS and TABS, hydronic level energy was always slightly higher than surface level total energy. The difference was the energy used to cool the mass of the slab itself.

In general, even if the total zone level cooling energy may be 5-15% higher for radiant systems compared to air systems, there are many potential advantages of using hydronic-based radiant systems such as, improved plant-side equipment efficiency with warmer chilled water temperatures [38], possibility of nighttime pre-cooling to reduce peak demand [39], utilization of natural cooling resources, and energy efficiency in transporting energy with water compared to air [3]. The combination of all these factors has the potential to produce lower energy consumption for radiant cooling vs. air systems.

5.2. Peak cooling rate

Figure 5 gives an example (G6: typical ceiling) of the cooling rate profiles for the radiant systems and their equivalent air systems. It can be seen that radiant system cooling rate profiles were different from the case of an air system. In general, a large portion of the heat was removed during the occupied period for the radiant case, and the radiant systems peak cooling rates were higher than the air system. Table 7 reports the values of the specific peak cooling rate for the radiant (both hydronic and surface) and the air systems.

Figure 6 plots the ranges of $P_{surf,pk}$ and $P_{hyd,pk}$ for RCP, ESCS, and TABS. Results show that the radiant system peak surface/hydronic cooling rates exceed that of the air system by a wide range depending on radiant system type and zone load conditions.

- For cases in G1 and G2, representing perimeter zones that are only subjected to building envelope load, $P_{surf,pk}$ ranged from 12-25% for the RCPs, and 16 - 27% for the ESCS. For RCP and ESCS, $P_{hyd,pk}$ was in a similar range as $P_{surf,pk}$, While little variation in both $P_{surf,pk}$ and $P_{hyd,pk}$ can be noted for changes in thermal insulation conditions, reduction of thermal mass resulted in much less peak load differences between the radiant and air systems.
- For G3, the total internal load was the same for all cases but with different radiant and convective splits for each case. The peak cooling rate differences ranged from 7- 27% at the surface level and from 7- 33% at the hydronic level. Higher radiant fraction in heat gain produces larger differences in peak loads between the two systems at the surface level. This was further demonstrated in G4-G6.
- For G4, solar gain contributed to a pronounced increase in the radiation heat transfer at the radiant surface(s). When exterior shading was not modeled, RCP ceiling surface peak cooling rate is 36% higher than the air system, and for ESCS ceiling system it is 35%. When exterior shading was modeled, the transmitted solar gain was mostly diffuse allowing it to be evenly distributed among all surfaces. Exterior shading reduced the direct solar impact, but the surface peak cooling rates were still 24-33% higher for the ceiling system.
- When the floor was used as the radiant cooling surface and when it was illuminated by direct solar, both $P_{surf,pk}$ and $P_{hyd,pk}$ increased dramatically compared to the ceiling cases. The ESCS surface peak cooling rate was 69% higher and for TABS it was 85% higher. Exterior shadings greatly reduced the absolute values of the peak load in all systems and the difference between radiant and air systems at the surface level for both radiant systems.

While the high peak-cooling rate shown maybe regarded as an enhancement of cooling capacity of the radiant cooling system [40, 41], the sizing of the associated waterside equipment must take this increase into account.

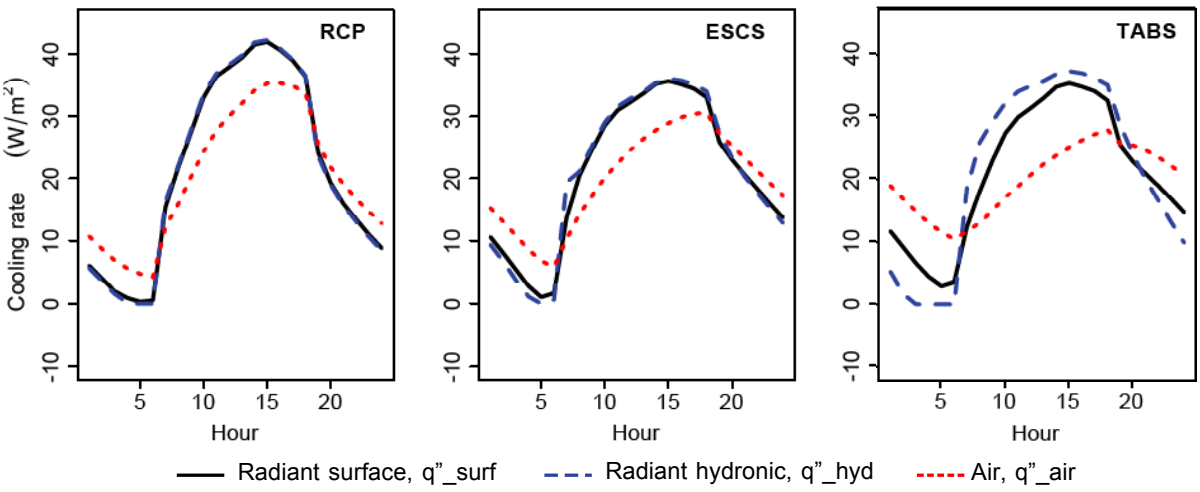

Figure 5: Comparison of design day cooling rate profiles between radiant and air systems. (G6 typical ceiling: cl_shade_rad0.6)

Table 7: Peak cooling rate comparison for summer design day

Group	Cases	RCP vs. Air (W/m^2)			ESCS vs. Air (W/m^2)			TABS vs. Air (W/m^2)		
		$q''_{surf,pk}$	$q''_{hyd,pk}$	$q''_{air,pk}$	$q''_{surf,pk}$	$q''_{hyd,pk}$	$q''_{air,pk}$	$q''_{surf,pk}$	$q''_{hyd,pk}$	$q''_{air,pk}$
G1	hw_r2	7.7	7.8	6.2	8.5	8.7	6.7	8.5	9.7	6.3
	hw_r1	12.9	13.0	10.4	13.9	14.2	11.0	13.6	15.1	10.1
G2	hw_r2	7.7	7.8	6.2	8.5	8.7	6.7	8.5	9.7	6.3
	lw_r2	14.1	14.0	12.6	14.4	14.6	12.4	14.4	16.6	11.0

G3	rad0	14.5	14.5	13.6	14.6	14.7	13.6	14.0	15.0	12.8
	rad0.3	13.9	13.9	12.6	14.5	14.6	12.7	14.0	15.1	11.4
	rad0.6	13.2	13.2	11.7	13.8	13.9	12.0	13.8	14.9	11.2
	rad1	12.5	12.6	10.9	13.1	13.3	11.3	13.0	13.7	10.3
G4	cl_ noshade	51.7	52.2	37.9	39.8	39.5	29.4	39.9	40.6	26.8
	cl_shade	29.4	29.7	23.5	26.0	26.7	21.0	25.6	29.0	19.3
G5	flr_noshade	NA	NA	NA	54.6	62.1	32.2	48.4	44.7	26.2
	flr_shade	NA	NA	NA	28.8	33.0	20.6	25.1	30.7	18.3
G6	cl_shade_rad0.6	41.9	42.2	35.2	35.6	36.0	30.6	35.5	37.3	28.0

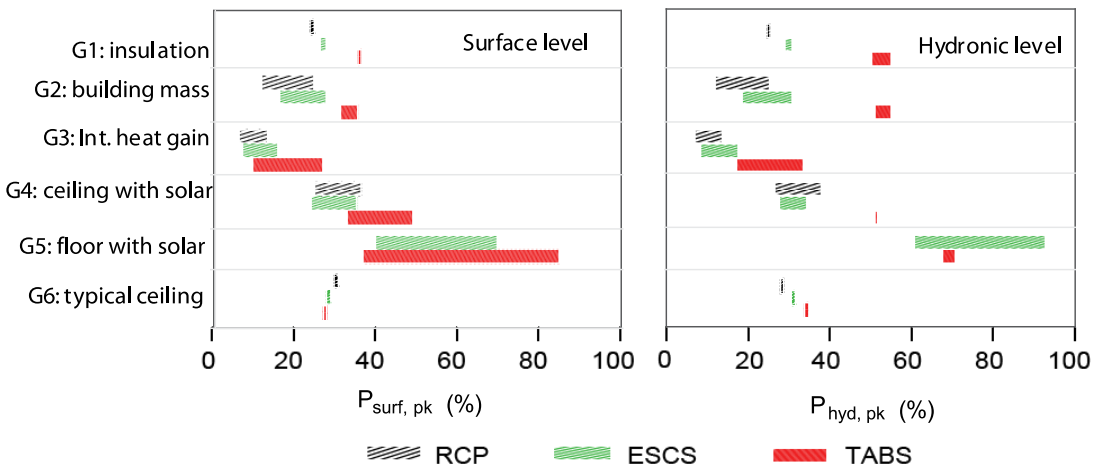

Figure 6: Peak cooling rate percentage difference between radiant and air systems

6. Discussion

6.1. Cooling load dynamic for radiant system

In order to explain why the radiant system peak cooling rate is higher than the equivalent air systems, Figure 7 investigates zone cooling load dynamics for the two systems. Using case rad0.6 (RCP) as an example, the figure compares the processes of how radiative and convective heat gains are converted into zone cooling load for the two systems. To assist the explanation, Figure 8 plots the operative temperature, air temperatures, and active and non-active wall surface temperatures for the two systems. Radiant cooling surface temperature is also plotted. For Case rad0.6, the total internal heat gain (15 W/m^2 during occupied hours) was divided into convective heat gain (6 W/m^2) and radiative heat gain (9 W/m^2). As shown, the cooling load for both systems was composed of two components, one that originated as convective heat gain from internal loads, and one that originated as radiative heat gain from internal loads. The instantaneous cooling load depends both on the magnitude and on the nature of the heat gains acting at the same instant. In a zone conditioned by an air system, the cooling load is 100% convective, while for the radiant systems the cooling load represents the total heat removed at the activated ceiling surface, which includes incident radiant loads, longwave radiation with non-activated zone surfaces and convective heat exchange with the warmer room air. In the case of air system (left plots), convective heat gain becomes cooling load instantaneously, and radiative gains are absorbed by zone thermal mass and re-released as convective load. The fact that building mass delays and dampens the instantaneous heat gain is well recognized by cooling load calculation methods. For the radiant cooling system (right plots), a large portion of the radiative heat gain converts to cooling load directly during the occupied period due to the presence of the cooling surface(s). Not all convective gains instantaneously contribute to cooling load, a smaller amount compared to the air system, during the occupied hours because a higher zone air temperature is reached to balance the cooler ceiling surface temperature, thereby maintaining an equivalent operative temperature, as is shown in Figure 8. And because of the higher zone air temperature, a small part of the convective heat gain is absorbed by non-activated building mass and removed by the radiant

surface via longwave radiation. The bottom plots stack up the two cooling load components, and the solid black lines in the bottom plots are hourly cooling loads, which reach their peak value at the end of the occupied period for both systems. These predicted cooling loads represent the total amount of heat being removed by each system to maintain the same operative temperature profile. Note that the peak cooling rate for the radiant system is predicted to be 13.0% greater than that for the air system.

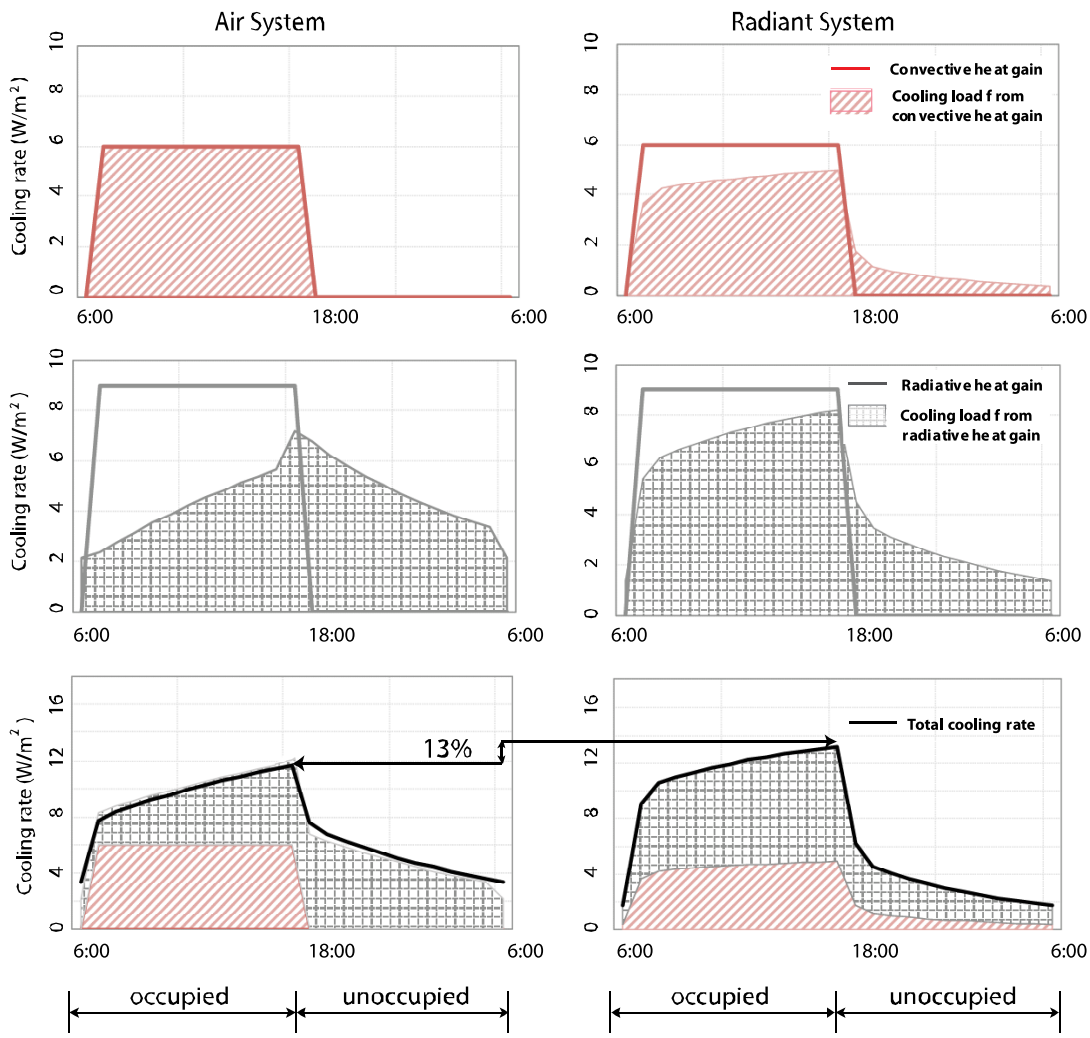

Figure 7: Comparison of surface cooling breakdown (convective and radiative part) for G3 Case rad0.6: air system (left) and radiant cooling panel (RCP) system (right)

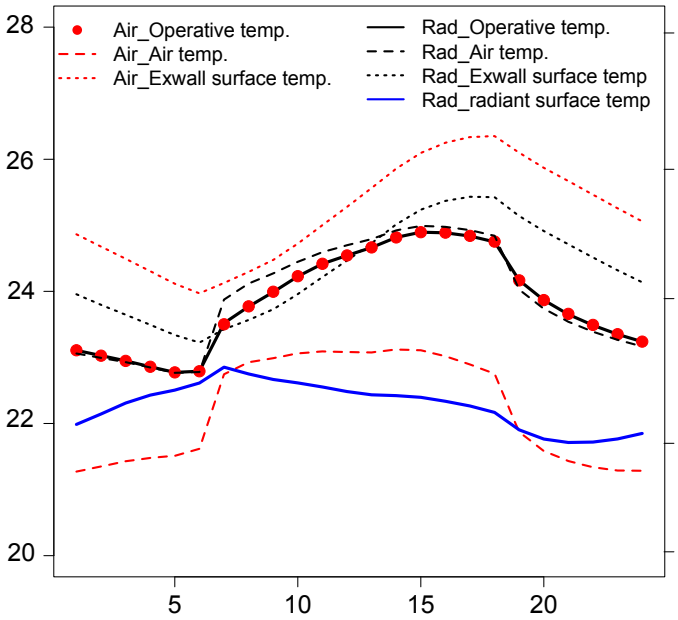

Figure 8: Comparison of zone air temperatures, operative temperatures, active and non-active surface temperatures between radiant and air systems (G6 typical ceiling: cl_shade_rad0.6)

Based on the discussion above, the author modified the cooling load generation diagram presented in chapter 18 of ASHRAE Fundamentals (2009) to represent the cooling load generation process when the zone is cooled by a radiant system (Figure 9). The original diagram was used to explain the cooling load generation process for an air system, and based on which most of the simplified cooling load calculation methods have been developed. The modifications are highlighted in red lines.

This modified diagram illustrates that the cooling load differences between the two systems originate from two aspects: 1) radiant cooling surface(s) directly remove part of the radiant heat gain and reduce heat accumulation in the building mass; 2) only part of the convective heat gain becomes instantaneous cooling load, and the remainder partly contributes to increased air temperature and partly is stored in building mass and removed by the radiant surface as surface cooling load.

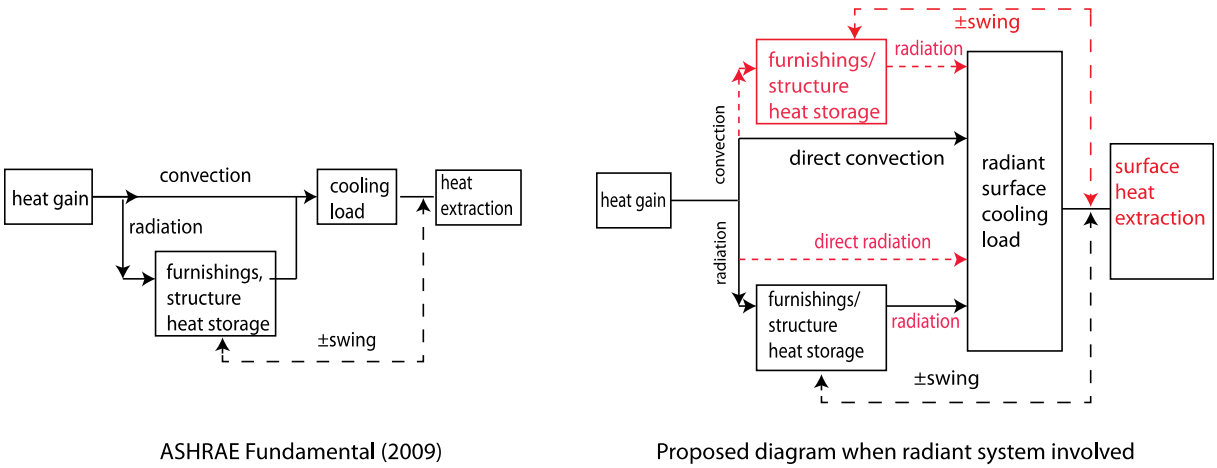

Figure 9: The cooling load generation scheme for air system adapted from ASHRAE Fundamentals (2009) and proposed modifications for radiant system

6.2. Definition of cooling load for different radiant system types

Throughout this study, we felt there is a need to clarify the definitions of design cooling load for sizing radiant systems and to distinguish between the three types of systems for the following reasons:

1. There is no clear definition of design cooling load for sizing radiant systems. According to ASHRAE Handbook [29], cooling load is defined as: "*the rate at which sensible and latent heat must be removed from the zone to maintain a constant zone air temperature and humidity*". However, zone air temperature is not recommended as the control temperature when radiant systems are involved [4]. In addition, in ISO 11855 (2012), design sensible cooling load is defined as: "*required sensible thermal output necessary to achieve the specified design conditions at the outside summer conditions*." It is not clear from this definition what the "specified design conditions" is.
2. Differences in thermal and control characteristics of the three radiant system types are usually not accounted for when determining design cooling loads. Peak instantaneous cooling load is normally used for sizing air system equipment, but it is not the most relevant for sizing all types of radiant systems. One example is the TABS. The reasons are: 1) intermittent or night-time operation is often implemented in order to take advantage of the storage capability of the active surface for load shifting, 2) time constants for these systems are large so it is not feasible to control the hydronic system in response to short-term environmental changes (load, setpoint changes) [30, 42].
3. As mentioned before, radiant cooling systems (ESCS and TABS) are integrated with the building mass. As a result, cooling rates at the surface and at the hydronic level are different due to the mass (thermal storage and delay). In cases of air systems, zone cooling load is directly used for sizing the HVAC systems, while in the case of a radiant system, the cooling load imposed on the hydronic loop is a better reference for sizing of cooling plant equipment.

Based on the discussion above, we propose to: 1) distinguish the design cooling load definition for sizing the quicker-response RCP/ESCS from the slower-response TABS; 2) define surface cooling load for the determination of required cooling surface area, and define hydronic cooling load for sizing hydronic equipment (pumps, cooling plant, etc.).

For RCP and lightweight ESCS, the cooling load definitions are:
<u>Surface cooling load</u>: the rate at which sensible heat must be removed by the actively cooled surface(s) from the zone to maintain a constant zone operative temperature during cooling design day. Peak surface cooling load should be used for determining total required cooling surface area.
<u>Hydronic cooling load</u>: the rate at which heat must be removed by the hydronic loop to maintain a constant zone operative temperature during cooling design day. Peak hydronic cooling load should be used for sizing cooling plant equipment.

The specific surface cooling load can be theoretically calculated by Eq. (1) at design conditions. If we further breakdown the surface radiation term into different radiation components, Eq. (1) can be expanded as following,

$$q''_{surf} = q''_{conv} + q''_{lw_surf} + q''_{lw_int} + q''_{sw_sol} + q''_{sw_int} \quad (3)$$

The last three terms, longwave radiant exchange flux from internal loads, transmitted solar radiation flux absorbed at surface and net shortwave radiation flux to surface from internal loads (lights), are the incident radiation that we discussed in the previous sections. During the sizing process, these three terms can be considered as an enhancement of cooling capacity [41], therefore, even if the peak cooling load of a radiant system may be higher than the cooling load calculated using traditional tools without capability to capture radiation heat transfer, the total area required may not need to be increased. Future research is needed to quantify how the three incident radiation terms mentioned above will affect sizing of cooling surface area.

For the RCP, hydronic cooling load is the surface cooling load plus heat loss from the backside of the panels, if any. For ESCS, the correlation between surface cooling rate and hydronic cooling rate is complicated by the heat conduction through the slab. Part 2 of ISO 11855 [4] recommends three methods for estimating surface cooling output and correlating the output with hydronic side operating conditions.

Design cooling load calculation for TABS has to take the control and operation strategy into account. For example, Part 4 of ISO 11855 (2012) provides guidance on calculating cooling capacity and cooling power demand on waterside to be used to select the cooling system, and it proposed to size the cooling equipment based on the sum of the heat gain values acting during the whole design day, internal load pattern, hydronic loop operation schedule, as well as radiant system specifications. Therefore, cooling load used for sizing TABS is not a unique value.

6.3. Proposed improvements in the design standards

As mentioned before, current radiant cooling design standards do not explicitly identify the differences in cooling load calculations between radiant and air systems. This results in the misapplication of tools in design practice not just for cooling load calculations but sometimes for detailed energy and comfort analysis. Currently, there are three classes of zone thermal models used in energy simulation tools: Heat Balance (HB), Thermal Network (TN), and Transfer Function (TF) models [43]. The HB and TN methods require relatively extensive computation time and effort from their users, and therefore are not widely adopted in tools used by design practitioners. The tools that use these two zone models, however, have the capability to capture detailed heat transfer processes in the zones and are recommended for use when radiant cooling systems are involved in the design. Modifications to the TF method, in particular the RTS method for radiant system cooling load calculations could be a good solution, but would require future research and is not an easy job due to the coupling of the radiant slab with the building structure.

Radiant systems should be modeled to ensure that the cooled surfaces are participating at the zone level heat transfer during the design calculation. A review of design tool showed that even though dynamic simulation tools are used for energy analysis, the cooling equipment sizing is often based on cooling loads calculated for an ideal air system. For example, the authors observed this in the EnergyPlus "autosizing" algorithm for radiant systems. In EnergyPlus, the HB method is used as the zone heat transfer model so it has the capability to calculate cooling load accurately when radiant systems are involved. However, it also assumes that the cooling load for a radiant system is the same as for an air system. Therefore, if "autosizing" function is used, an "ideal air system" is simulated first for load calculation, and this cooling load is used for sizing all associated cooling equipment, including radiant system design mass flow rate, total tube length, and radiant system plant equipment. The users can manually adjust the design parameters if necessary, but failure to realize the cooling load differences between radiant and air systems can produce significant errors.

In summary, the following text and recommendations could be included in radiant cooling design guidelines to improve understanding of radiant cooling system and to facilitate better design solution:

- The cooling load for zones conditioned by a radiant system is different from the cooling load for zones conditioned by an air system. The differences between the two systems originate from two aspects: 1) active radiant cooling surface(s) directly remove part of the radiant heat gain and reduce heat accumulation in the building mass; 2) only part of the convective heat gains becomes instantaneous zone cooling load (as is the case in an air system), and the other portion partly contributes to increased air temperature and partly is stored in building mass and subsequently removed by the active radiant surface as surface cooling load.
- For RCP and lightweight ESCS, peak surface cooling load shall be used for dimensioning total required cooling surface area, and peak hydronic cooling load shall be used for sizing associated cooling equipment. For TABS, equipment sizing depends on total heat gain, heat pattern, operational strategy, etc.
- Simulation tools that use either heat balance or thermal network methods for zone level thermal modelling are recommended for design cooling load and system sizing calculations for radiant systems. Examples of the recommended tools are: EnergyPlus, IES Virtual Environment, IDA ICE, Esp-r, TRNSYS.
- The following design procedure is recommended for load calculation and system equipment sizing: 1) conduct a basic cooling load calculation as if an ideal air system with unlimited cooling capacity is used for conditioning the space. This basic cooling load value can be used

for comparing different design options. If a radiant system is choosen, the basic cooling load value can be used as a starting point for dimensioning the radiant cooling system; 2) recalculate design surface cooling load and hyrdonic cooling load for the radiant system. During this process, the radiant cooling system should be modeled with a computer program that meets the prescribed requirements mentioned above. Size the radiant system properly to satify prescribed thermal comfort requirements. 3) Size the cooling plant equipment based on design hydronic cooling load calculated from step 2.

7. Conclusions

This simulation study investigated the impacts of the presence of an activated cooled surface on zone cooling loads by comparing the peak cooling rate and the 24-hour total cooling energy for radiant and air systems. Three radiant system types (RCP/ESCS/TABS) and single zone models were developed for comparison between each radiant system type and their equivalent air system. The models were configured to study the impacts of the following parameters: envelope thermal insulation, thermal mass, type of internal gain, solar heat gain with different shading options, and radiant surface orientation (ceiling, floor). The simulation results are summarized below:

- For the simulated cases, when compared to an air system for equivalent comfort conditions (operative temperature), 24-hour total cooling energy removed by the RCP system hydronic loop was 5-13 % higher, 6-15% higher for the ESCS, and 6-15% higher for the TABS. This was caused by lower surface temperatures at the inside surfaces of the building envelope created by the active (cooled) radiant surface.
- For perimeter zones that were only subjected to building envelope heat gain, $P_{surf,pk}$ ranged from 12% to 25% for the RCPs, 16% to 27% for the ESCS, and 31% to 35% for the TABS. For the RCP and the ESCS, $P_{hyd,pk}$ were in the similar range as $P_{surf,,pk}$,but for the TABS, $P_{hyd,pk}$ increased to 50-54%.
- For interior zones with longwave radiant heat gain, the peak cooling rate differences ranged from 7% to 27% at the surface level and from 7% to 33% at the hydronic level. This implies that higher radiant fraction in heat gain produces larger differences in peak cooling rates between the two systems at the surface level. This was further demonstrated in cases with solar load.
- For perimeter zones and atrium where direct solar heat gain constitutes a large portion of the cooling load, the peak cooling load difference is pronounced. When exterior shading was not installed, RCP ceiling surface peak cooling rate is 36% higher than the air system, and for ESCS ceiling system it is 35%, and 49% for TABS ceiling systems. Exterior shading reduced the direct solar impact, but the surface peak cooling rate were still 24-33% higher for the ceiling system.
- When the floor was used as the radiant cooling surface and when it was illuminated by direct solar, both $P_{surf,,pk}$ and $P_{hyd,,pk}$ increased dramatically compared to the ceiling cases. The ESCS surface peak cooling rate was 69% higher and for TABS, 85% higher. Exterior shading greatly reduced the difference between radiant and air systems at the surface level for both radiant systems. However, $P_{hyd,,pk}$ for TABS was not much affected by the installation of shading system.
- Cooling rate differences between the two systems originate from two effects: 1) radiant cooling surface(s) directly remove part of the radiant heat gain and reduce heat accumulation in the building mass; 2) only part of the convective heat gain becomes instantaneous cooling load, the remainder partly contributes to increased air temperature and partly is stored in the building mass and removed by the radiant surface as surface cooling load.

In conclusion, zones conditioned by a radiant system have different peak cooling loads than those conditioned by an air system. While the increase in 24-hour total cooling energy is relatively small and may be offset by other energy savings benefits associated with radiant cooling systems, the differences in peak cooling load both in terms of magnitude and time compared to the air systems require special attention in system and control design. In the radiant design standard, these differences in cooling load should be clearly stated and translated into requirements for tools and methods.

Transfer Function (TF) based methods are not appropriate for cooling load calculation when radiant cooling systems are involved. Radiant systems should be modeled using a dynamic simulation tool that employs either Heat Balance (HB) model or Thermal Network (TN) models during the design process for accurate cooling load calculation.

8. Nomenclature

RCP	Radiant cooling panels
ESCS	Embedded surface cooling systems (lightweight)
TABS	Thermally activated building systems
G1 - G6	Simulation group index
q''	Heat flux, W/m^2
q''_{surf}	Heat flux at the exposed face of the cooling surface(s) , W/m^2
$q''_{surf,cond}$	Conduction heat transfer at the exposed face of the cooling surface(s) , W/m^2
$q''_{surf,conv}$	Convection heat transfer at the exposed face of the cooling surface(s) , W/m^2
$q''_{surf,rad}$	Radiation heat transfer at the exposed face of the cooling surface(s) , W/m^2
q''_{lw_surf}	Net longwave radiation flux to radiant active surface from other surfaces, W/m^2
q''_{lw_int}	Longwave radiant exchange flux from internal load, W/m^2
q''_{sw_sol}	Transmitted solar radiation flux absorbed at surface, W/m^2
q''_{sw_int}	Net shortwave radiation flux to surface from internal load (lights). , W/m^2
$q''_{surf,pk}$	Specific peak radiant system surface cooling load, W/m^2
$q''_{hyd,pk}$	Specific peak radiant system hydronic cooling load, W/m^2
$q''_{air,pk}$	Specific peak sensible cooling load for air system, W/m^2
$\dot{q}_{surf,tot}$	Specific 24-hour total surface cooling energy, kJ/m^2
$\dot{q}_{hyd,tot}$	Specific 24-hour total hydronic cooling energy, kJ/m^2
$\dot{q}_{air,tot}$	Specific 24-hour total sensible cooling energy, kJ/m^2
$P_{surf,pk}$	Percentage difference of surface peak cooling rate between radiant and air system, %
$P_{hyd,pk}$	Percentage difference of hydronic peak cooling rate between radiant and air system, %
$P_{surf,tot}$	Percentage difference of surface level 24-hour total cooling energy between radiant and air system, %
$P_{hyd,tot}$	Percentage difference of hydronic level level 24-hour total cooling between radiant and air system, %
Subscript	
surf	Variable measured at radiant surface level
hyd	Variable measured at radiant cooling water loop
pk	Peak cooling load
tot	24 hour total cooling energy

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PERFORMANCE SPECIFICATION

FAN, LOW POWER
(DESKTOP MOUNTED OR WORKSTATION INTEGRATED. FOR OCCUPANT COOLING)

NOTE: This draft prepared by the U.C. Berkeley Center for the Built Environment has not been approved and is subject to modification.
DO NOT USE FOR ACQUISITION PURPOSES.

1 SCOPE AND CLASSIFICATION

1.1 Scope. This specification describes the performance characteristics of low-power fans for cooling the occupants in office workstations.

1.2 Use. The fan described is for use at an office workstation by individual workers to provide thermal comfort through air movement.

1.3 Classification. The fan shall be of the following types and styles as specified.

1.3.1 Type I: Above-worksurface fans. Fans positioned on, integrated with, affixed on or mounted above the worksurface and primarily directed at the breathing zone and upper chest of the user.

1.3.2 Type II: Below-worksurface fans. Fans integrated with furniture or mounted below the worksurface and primarily directed at the lap and lower trunk region of the user.

1.3.3 Class 1: Discrete Control

1.3.4 Class 2: Continuous Control

2 APPLICABLE DOCUMENTS

2.1 Government Publications. The following documents, of the issues in effect on the date of invitation for bids or request for proposal, form a part of this specification to the extent specified herein:

2.1.1 Federal Specifications (with appropriate sourcing paragraphs!)

2.1.2 Federal Standards (with appropriate sourcing paragraphs!)

2.1.3 Commercial Item Descriptions

2.1.4 Military Specifications

2.1.5 Military Standards (with appropriate sourcing paragraphs!)

2.1.6 Military Handbooks (with appropriate sourcing paragraphs!)

2.1.7 Federal Regulations (with appropriate sourcing paragraphs!)

2.2 Other publications. The following documents form a part of this specification to the extent specified herein. Unless a specific issue is identified, the issue in effect on date of invitation for bids or request for proposal shall apply.

2.2.1 Voluntary Standard. This will be the itemized list of applicable codes and standards from UL, CE, ASTM, FCC etc. (with appropriate sourcing paragraphs!)

3 REQUIREMENTS

3.1 Air Movement. Fans shall conform to the air movement characteristics specified in paragraph 3.1.1 through 3.1.3.

3.1.1 Air Velocity. Fans shall maintain the air velocities specified with a tolerance of ±10% when tested in their installed configuration and directed at the workstation occupant's normal seated position as described in the contract or order and section 3.3.2.

a The maximum air velocity shall be not less than 2.5 m/s.

b The minimum air velocity shall be not greater than 0.3 m/s.

c Air velocities for Type I (Above-worksurface) fans shall be measured at the location of the occupant's head when seated at the workstation with the fan adjusted to direct the airflow directly at the head.

d Air velocities for Type II (Below-worksurface) fans shall be measured at a distance of 0.15 m along the centerline of the occupant's body when seated at the workstation.

3.1.2 Air Volume. The volume of air delivered through the fan outlet shall be not less than ten (10) liters per second ±10% when operated at maximum velocity.

3.1.3 Filtration. (Optional feature.) If included in contract documents or order, provide fibrous, cellular or electrostatic filters as specified.

3.2 Energy and power.

3.2.1 Power. The continuous power consumption of the fan shall be no more than four (4) watts at all operating speed settings.

3.2.2 Standby Power. Continuous power consumption in the standby mode described in paragraph 3.3.4 shall be no greater than 0.2 watts.

3.2.3 Voltage and Frequency. Unless otherwise specified, the fans shall be designed to connect to a rated voltage of 120 volts alternating current ±10 percent, and a rated frequency of 60 hertz ±5 percent.

3.2.4 Power Harmonics. Fans shall return power harmonics that do not exceed acceptable levels to the building electrical system when operated across all speeds.

3.3 User interface.

3.3.1 Speed control. Fan speed shall be adjusted via user controls in order to produce the air velocities as specified in paragraph 3.1.1.

a Class 1 Fans (discrete velocity control) shall include ‘off’ and not fewer than three ‘on’ speed positions. Class 2 Fans (continuous velocity control) shall include a positive ‘off’ position as well as the specified range of continuous control.

b Fan speed controls shall be operable with one hand and require no tight grasping, pinching or twisting of the wrist. Operation of the control shall require no more than 2 newtons of force.

c Fan speed controls shall be within easy reach of the normal workstation position. Forward reach shall be no more than 635 mm and the vertical reach shall be not more than 1120 mm above the floor.

3.3.2 Direction Control. Fans shall permit users to manually direct the air stream by adjustment or repositioning to accommodate differences among users and changes of posture. Once directed, fans shall remain stable until manually repositioned.

a Fan direction controls shall be within easy reach of the normal workstation position. Forward reach shall be no more than 635 mm and the vertical reach shall be not more than 1120 mm above the floor.

b Direction controls shall be operable with one hand and require no tight grasping, pinching or twisting of the wrist. Operation of the control shall require no more than 22.2 newtons of force.

c Type I: (Above-worksurface) Fans shall include a vertical adjustment sufficient to direct the air stream across a range from the user’s hands on the worksurface to the top of the user head. Fans shall be repositionable across a horizontal range encompassing normal seating positions of the workstation.

d Type II: (Below-worksurface) Fans shall provide air movement on the lap and lower trunk region while the occupant is in the normal seating position of the workstation.

3.3.3 Automatic off. When the workstation becomes unoccupied, the fan shall switch automatically to ‘off’ or ‘standby’ after a latency period of no more than ten minutes. When the workstation is reoccupied, the fan shall immediately return to the previous speed setting without additional user input.

a Optional feature. The latency period shall be user adjustable in one-minute increments from a minimum of one minute to a maximum of fifteen minutes. Adjustment shall not require special tools or computer interface.

3.3.4 Safety. The fan shall protect the user and other employees from contact with hazards such as rotating blades, electrical current and pinch points.

3.4 Occupancy sensing

3.4.1 Sensor Field of View. The Field of View (FOV) for occupancy/vacancy detection sensors shall be limited to the user-inhabited portion of the workstation.

a Optional feature: Sensors may be adjustable to limit the FOV. FOV adjustments shall not require special tools.

3.4.2 Passive Sensing. Fan sensors shall emit no electromagnetic, acoustic or other signal in order to detect occupancy.

3.5 Acoustics.

3.5.1 Sound Pressure Level. The fan shall produce a sound pressure level not greater than 25 decibels measured using the A-weighted scale at a distance of one meter over the entire range of available speed settings.

3.5.2 Frequency Distribution. The fan shall operate without unpleasant harmonic tones or physical vibration at all operating speeds.

3.6 Operational Environment.

3.6.1 Continuous Operation. The fan shall be capable of continuous operation without overheating or other failure.

3.6.2 Temperature Range. The fan shall operate in ambient temperatures between 0° C and 40° C.

3.6.3 Placement or Attachment. Fans shall be placed or affixed such that they remain stationary.

a Type I (Above-worksurface fans) designed to be placed on the desk shall be rest securely on the worksurface and be of sufficient weight and strength that they remain stationary at all operating speeds and whilst being adjusted or controlled.

b All Type II (Below-worksurface fans) as well as those Type I (Above-worksurface fans) designed to be integrated with, affixed on or mounted above the worksurface shall incorporate mounting hardware of sufficient strength such that the fan remains stationary at all operating speeds and whilst being adjusted or controlled.

3.7 Design and Construction. Fans covered by this document shall include all components necessary to constitute a complete and functional product.

3.7.1 Materials. The fan and all its component parts shall be constructed of materials sufficiently durable for their purpose and the expected service life of not less than10 years according to manufacturers standard commercial practice.

3.7.2 Recovered materials. The offer or/contractor is encouraged to use recovered materials in accordance with Public Law 94-580 to the maximum practical extent. Recycled or reclaimed materials maybe used in the construction of the equipment described herein. Under no conditions or circumstances shall the contractor

submit to the Government for acceptance reconditioned or rebuilt components as a part of the equipment described herein.

3.7.3 Cleaning. Air filters (if present) should be removable, replaceable, and interchangeable without the use of special tools.

3.8 Codes and Standards. Fans covered by this document shall conform to the applicable requirements of the codes and standards specified in 3.5.1 through 3.5.10.

3.8.1 NFPA. Wiring shall conform to National Fire Protection (NFPA) Standard No. 70. The National Electrical Code.

3.8.2 NEMA. Motors shall conform to National Electrical Manufacturers Association (NEMA) Publication No. MG- 1.

3.8.3 UL. Fans shall conform to the requirements of Underwriters Laboratories Inc. (UL) Standard No. UL 507.

3.8.4 Occupational Safety. Fans shall conform to Occupational Safety and Health Act (OSHA), 29 CFR 1910.

3.8.5 Indoor Air Quality. Fans shall conform to the provisions of California Green Specification section 01350.

4 QUALITY ASSURANCE

4.1 Product conformance. The product offered shall meet the performance requirements of specification, and conform to the producer's own drawings, specifications, standards and quality assurance practices. When specific quality assurance provisions are specified for any characteristic, the contractor shall maintain records resulting from inspection(s) and testing conducted in accordance with the specific quality assurance provisions. The government reserves the right to require proof of such conformance prior to first delivery and thereafter as may be otherwise provided under the provisions of the contract. The government reserves the right to audit the contractor's quality assurance records.

4.2 Place and date of manufacture. Each fan shall bear an alpha-numeric code to indicate the manufacturer's plant where produced and the date that production occurred. This encoded information shall be permanently stamped or affixed to the fan.

4.3 Warranty. Unless otherwise specified in the contract, the manufacturer's standard commercial warranty terms shall apply. The warranty shall become effective from the date of start up of the equipment after installation is complete.

4.4 Quality Control. A representative random sample of fans drawn from each production run shall be tested to ensure they satisfy the provisions of the specification.

4.4.1 Air Velocity. Fans shall be tested to deliver the air velocities as specified in section 3.1.1. Velocity measurements will be taken over a period of not less than two minutes for each velocity setting.

4.4.2 Power Consumption. Fans will be tested that they do not exceed operating or standby power limits specified in section 3.2. Continuous power consumption will

be measured at the plug by averaging the values measured at an interval of no more than one second over a testing period of not less than ten minutes.

4.4.3 Comfort. Human subject test? Mannequin?

5 PACKAGING

5.1 Packaging. Requirements for preservation, packing, packaging, and marking of packages shall be as specified in the contract or order.

6 NOTES AND CONCLUDING REMARKS

6.1 CBE Examples. The Center for the Built Environment (CBE) at the University of California, Berkeley has constructed a number of prototype devices.

6.2 Relevant Scientific Literature. We will site the forthcoming Corrective Power Paper, which will give a nice review of the state of the literiture. In the future we can add documentation about this process.

Thermal comfort and perceived air quality of a PEC system

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SUMMARY
A personal environmental conditioning (PEC) system using air-jet cooling was evaluated for its thermal comfort, perceived air quality (PAQ), and eye comfort. The room surroundings and the air jets were both fixed at 28ºC and 50% RH. Two 4W fans directed room air toward the occupant's breathing zone from opposite sides. The premise was that facial air movement would both cool the occupant and disrupt potential PAQ in the body plume. Eighteen subjects participated in 90 3.5-hour tests. Comfort was assessed both at the workstation and during periodic breaks away from it. Comfort persisted throughout ten-minute standing/conversation breaks. After 15-minute step-climbing breaks had ensured discomfort, comfort resumed immediately upon the occupants' return. The influence of body plume on PAQ was examined using a plume-deflecting collar and a menthol scent applied at the waist. The collar significantly reduced the scent intensity, and the PEC air jets had the same effect.

IMPLICATIONS
PEC systems that do not require a cool air supply can be more easily implemented in buildings, are less expensive, and have very low energy use. By compensating for higher indoor temperatures, they can substantially reduce a building's cooling energy. This paper quantifies several practical issues related to their adoption.

KEYWORDS
Air movement cooling, absence from workstation, body thermal plume

INTRODUCTION
Various studies have found that Personal Environmental Control Systems (PECS) can deliver comfort and the perception of good air quality by providing air movement around the head or other parts of the body with very low energy input (Zhang at al. 2010, Melikov 2003, Amai et al. 2007, Sun et al. 2007). However, there remain practical questions about how PECS might be perceived by occupants over time in a real office environment, and also fundamental questions about how the air movement acts on the body to improve comfort and the perception of air quality (PAQ).

First, how long and to what extent would people feel comfortable when they are not under the direct influence of a PECS (e.g., when away at the copy machine, or exercising moderately as when climbing stairs), and how quickly can PECS restore comfort when people return to their workstations? Answers to these questions might impact the widespread adoption of PECS in practice. Second, is the known improvement in perceived air quality (PAQ) in the presence of air movement caused by a real air quality improvement or a psychological process? Real improvements might come either from the transport of fresh outside air to the breathing zone, or from the air movement stripping the bioeffluent-laden body plume from the breathing zone.

The objectives of the study were to determine: 1) the ability of this low-power PECS to provide comfort in warm environments, 2) occupants' comfort during absence from the PECS, for standing activity and a climbing activity producing a higher metabolic rate, 3) how quickly the PECS restores comfort after the occupants return, 4) air speed and body plume effects on PAQ, and 5) the relationships between PAQ, thermal sensation, and comfort.

METHODS

Tests were conducted in an environmental chamber from March through May 2009. 9 male and 9 female college students participated in one test per week for 5 weeks, for a total of 90 3.5-hour tests. Throughout the entire experiment, the room air temperatures were maintained at 28ºC, RH 50%. Subjects' skin temperatures were measured and they were surveyed for their subjective perceptions of thermal comfort, air quality, scent/body odor intensity, air movement preference, and others.

In each workstation, two 4" 4W muffin fans supply re-circulated room air through two nozzles toward opposite sides of the occupant's face. 4" and 2" diameter nozzles were tested. In some tests the air speed was controlled by the subjects using a slider on a computer screen. Liu (2011) presents the air flow profiles produced by the PEC system in this study.

Workstations were arranged so that two subjects could be tested simultaneously. The subjects wore standardized summer clothing, including cotton pants, short-sleeved cotton shirt, summer socks, and a pair of lightweight shoes (0.5 clo). Figure 1 shows the layout of the workstations and the clothing that subjects wore. The configurations tested were listed in Table 1.

Figure 1. Experimental setup

Table 1. Tested configurations

air speed at cheek (m/s)	0 ('no PEC'), 0.6, 1.0
nozzle diameter (inches)	2, 4
number of nozzles:	1, 2

72 tests took 3.5 hours, with 3 one-hour test phases of separate PEC configurations, and a 10-minute break time between each one-hour phase. The order of the PEC configurations in each phase was randomly alternated and balanced. The subjects were surveyed in the beginning, middle, and end of each phase. The last survey (at the 60th minute) is used to analyze the stable condition. The first survey is used for analyzing how comfort is restored after absence from PEC.

The survey questionnaires regarding thermal sensation, comfort, and PAQ appeared automatically on the subject's computer screen at scheduled times.

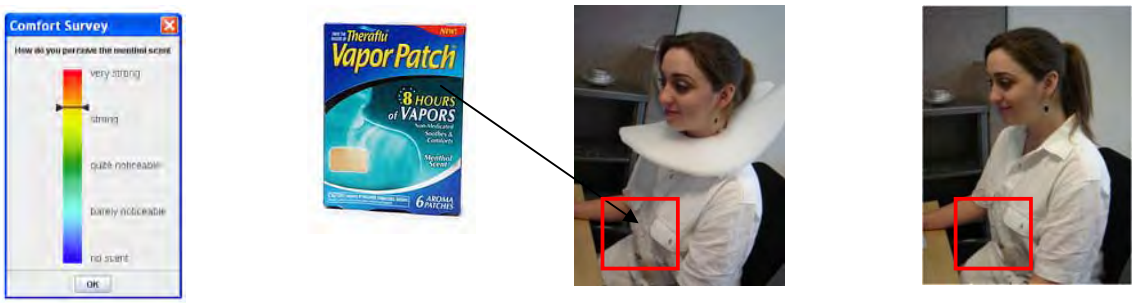
Absence at a low activity level. During the 10-minute break periods, the PEC systems were turned off and the subjects were asked to leave their workstations. They were encouraged to stretch their bodies and to involve in conversation with each other and with the researcher in charge of the test (Figure 2a). At the end of the 10-minute break, before resuming the next PEC test configuration, they answered a set of questions about thermal comfort and PAQ.

Absence with high metabolic exertion. In 18 tests, 15-minute breaks were used. Both at the beginning and end of a 15-minute break, the subjects climbed a step 40 times (Figure 2b), corresponding to climbing and down 5 stories. At the end of the break, they answered the

survey questions. Then they sat in the workstation with the PECS on, and repeated the survey immediately. They were then surveyed again five and fifteen minutes later.

Body odor/scent intensity tests. In the 18 tests with high-exertion breaks, subjects did not perceive an increase body odor after exertion. To examine the mechanism of body odor transported in the body's thermal plume, a cough-relief patch containing menthol, camphor, and eucalyptus oils was placed on the abdomen skin to add a perceptible scent (Figure 3). The subjects were surveyed about the scent intensity under three conditions: with and without a collar to deflect the body plume away from the nose, and then with the PEC providing air movement.

The scent test took 1 hour. After 15 minutes in the chamber, we applied the menthol patch to the subject. Each configuration lasted five minutes interchanging the order of the collar while adding the PEC (0.6 m/s) at the end. Five minutes later, they answered a survey about scent intensity and PAQ.



a. scent intensity question b. scent c. with collar d. no collar

Figure 3. Scent test

RESULTS

1. Whole-body thermal sensation and comfort with the PEC system

Whole-body thermal sensation and comfort are presented in Figure 4. The X axis represents the test configurations listed in Table 1. Without PEC, thermal sensation is close to 'warm'. At the low velocity setting (0.6 m/s at the breathing/cheek region) there is a slight but insignificant cooling effect. All the other configurations are between 'slightly warm' and 'neutral', and the differences are significant ($p<0.05$).

Comfort is improved by PECS from 'just uncomfortable' (the negative value for 'no PEC') to a range of comfortable values for all other configurations. It is interesting that although the thermal sensation cooled slightly and insignificantly under low air movement (0.6 m/s), comfort is significantly improved under the same condition. At the higher air movements, comfort is above 1 on the comfort scale. The highest comfort values are for the 2x4" nozzle configurations (both 1 m/s and user-controlled). Comfort is better for the 2x 4" than the 2x2" configurations, and user control does not provide an advantage for either size. The comfort for the two nozzles is significantly better than the comfort for the single nozzle.

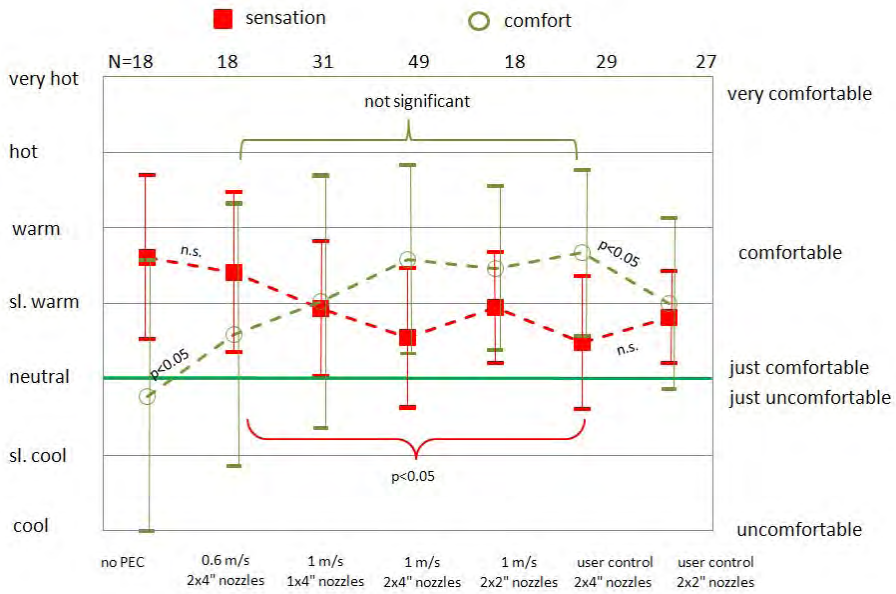


Figure 4. Whole-body thermal sensation and comfort (bar represents ± SD, n.s. represents not significant)

2. Comfort during and after break periods

After the 10 min break, subjects' sensations are between slightly warm and warm (Figure 5). Thermal comfort is reduced 1 scale unit (Figure 5), but is still on the comfortable side. Once the PECS is applied, comfort is immediately restored to above 1.

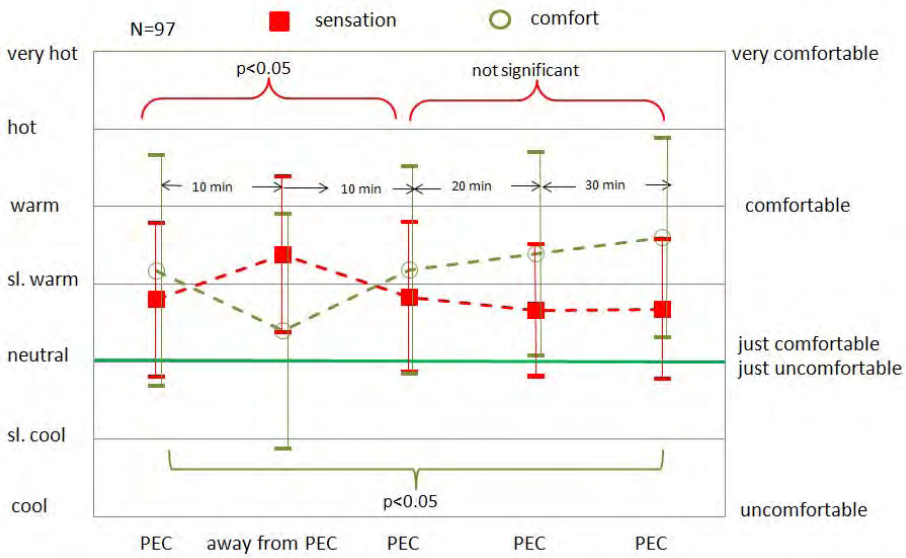


Figure 5. Thermal sensation after the 10-minute break

After the 15 minute break in which the subjects twice did 40 steps in the ambient condition of 28ºC, subjects are warm (sensation between warm and hot) and uncomfortable (Figure 6). Right after the PEC is applied, the sensation is close to be slightly warm and comfort goes up

1.5 scale units and is above zero. The two subsequent surveys show continued cooler thermal sensation and improvement to comfort.

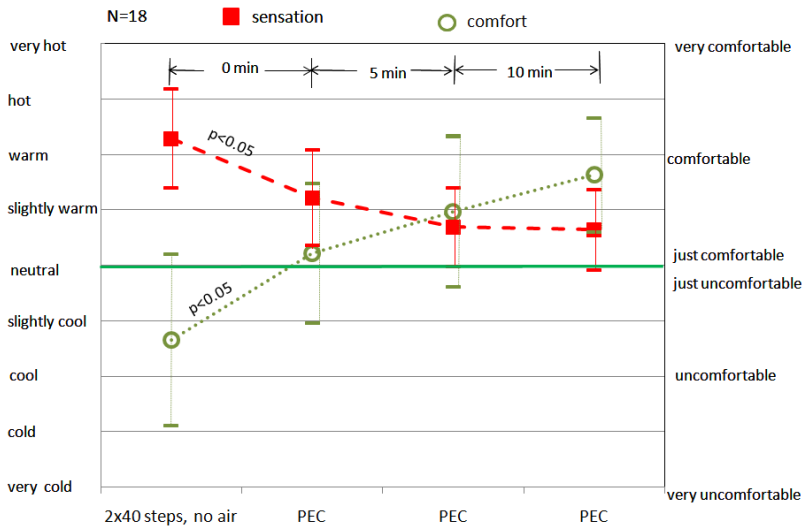


Figure 6. Sensation and comfort after exercise in the 15 minute break.

3. Perceived air quality (PAQ)

Perceived air quality is significantly improved by air movement at 28ºC. Without air movement, the PAQ was just unacceptable, but under all PEC configurations it was fully acceptable. This finding is similar to that of a previous PECS test (Arens et al. 2008). Thermal comfort determines PAQ rather than air temperature, as seen in Figure 7.

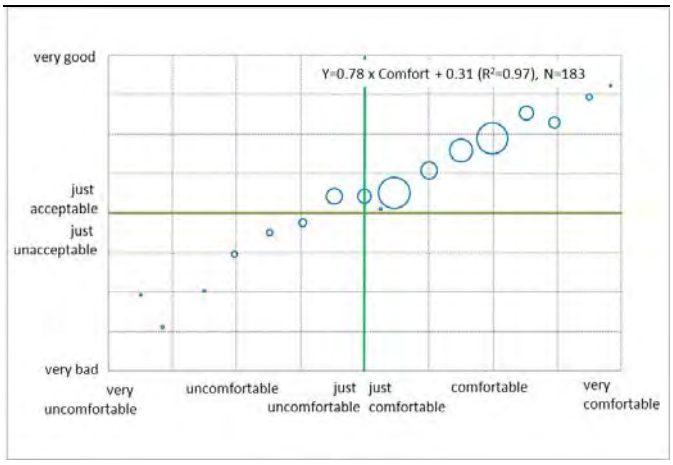


Figure 7. PAQ vs. thermal comfort (circle size represents the number of votes)

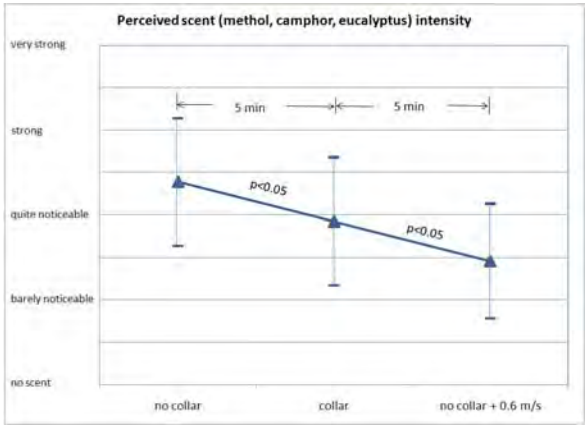


Figure 8. Sent intensity between collar, no collar, and air movement

Adding the menthol patch, significant differences are found for the scent intensity between the no collar configuration, the with-collar configuration, and the no-collar-with-low-air-movement at the breathing zone (Figure 8). The collar acts to reduce the scent intensity over no collar, and air movement reduced it more than the collar.

CONCLUSIONS

For thermal comfort questions:

- The low-power task-ambient conditioning system provides high levels of occupant comfort in a warm 28ºC (82.4ºF) environment.
- Comfort was maintained during the 10-minute break periods between PEC tests, in which subjects stood and engaged in conversation.
- PEC restores comfort immediately after a 15-minute period in which discomfort was induced by two intensive metabolic step exercises.

For PAQ questions:

- Air speed significantly improves PAQ, even at the lower speed of 0.6 m/s.
- Wearing a plume-deflecting collar significantly reduced the menthol scent intensity.
- PAQ and comfort are well correlated, with better PAQ at higher levels of comfort.

Throughout the tests, eye discomfort was not reported.

ACKNOWLEDGEMENTS

The authors gratefully acknowledge the Syracuse Center of Excellence CARTI project, which is supported by a grant from U.S. Environmental Protection Agency [Award No: X-83232501-0], and the support of the California Energy Commission Public Interest Energy Research (PIER) Buildings Program. They also thank Professor William Cain, UC San Diego, and Atze Boestra of Binnenmilieu, Rotterdam, for their help with developing the scent test, and Carlton Brown at Air Concepts Inc. for donating nozzles for the experiment.

We also fondly recall working with our enthusiastic and skilled collaborator, Professor Hiroshi Higuchi, who died November 21, 2010.

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Study Of A Personal Environmental Control System Using Opposing Airstreams

Author Block: Chonghui Liu, C&S Companies, Syracuse, NY; Hiroshi Higuchi, Syracuse Univ, Syracuse, NY; Edward Arens, Hui Zhang, UC Berkeley, Berkeley, CA

Abstract:
Abstract:
The goal of this joint project is to examine the physical performance of a Personal Environmental Control (PEC) system that uses opposing airstreams in order to control the microclimate around the human head, and its effect on people's thermal comfort and perceived air quality.
Both 4" (10.16 cm) and 2" (5.08 cm) diameter nozzles with respective exit velocity 2.0 m/s and 3.0 m/s (respective Reynolds numbers are 13,000 and 10,000) were studied with particle image velocimetry (PIV), hotwire anemometry and flow visualizations. Parallel human subject tests were conducted to validate their effects on people's perceptions of thermal comfort and indoor air quality.
The radial resultant jets generated from the PEC system are able to penetrate the free convection flow at a manikin's face, and possess a fluctuation frequency of 2 - 4 Hz. The PEC system can improve thermal comfort and perceived air quality, while no dry eye discomfort was detected.
Relevance:
A PEC system can provide improved thermal comfort and indoor air quality (e.g. decrease sick building syndrome, reduce the risk of contagious transmission etc.) while reducing overall energy usage (e.g., Bauman, 1994). Existing PEC systems tend to supply air from the front, which may cause little draft sensation (Fanger et al., 1988) but may cause dry eye discomfort (Wyon and Wyon, 1987; Wolkoff et al., 2005) depending on the velocity. Toftum (2005) and Mayer (1992) compared air approaching from different directions and found that air directed towards the side of the head was advantageous.
Uncertainty:
Both PIV and hotwire anemometry experiments were performed in duplicate and each analytical measurement was replicated. The combined uncertainty (including calibration, displacement of particle image, time interval, 3D effect, measurement position and time) of PIV measurements was 40.80 mm/s (relative uncertainty was 2.04%); and the relative expanded uncertainty (including calibration, linearization, A/D resolution, probe positioning, and temperature) of hotwire measurements was 4.9%. The uncertainty of the human subject tests was not quantitatively assessed, but it was estimated within 20% of the true values.
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Topic (Complete): Laboratory studies ; Human subjects ; Ventilation
Presentation (Complete):
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Manuscript (Complete):
: Full paper (6 pages max)

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Effect of a heated and cooled office chair on thermal comfort

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Abstract

A heated/cooled chair was evaluated for its effect on thermal sensation and comfort. Thirty college students participated in 150 1.75-hour tests. Two heated/cooled chairs were placed in an environmental chamber resembling an office environment. The chamber temperatures were set at 16, 18, 25 and 29 °C (60.8, 64.4, 77, 84.2 °F). During the tests the subjects had full control of the chair surface temperature through a knob located on the desk. An additional 64 tests with sixteen subjects were conducted at the same four temperatures but with regular mesh or cushion chairs in order to provide reference results for comparison.

Subjective responses about thermal sensation, comfort, and temperature satisfaction were obtained at 20 minute intervals and eight times before, during, and after a break period. The chair's energy consumption was monitored continuously. The results show that the heated/cooled chair strongly influences the subjects' thermal sensation and comfort, providing thermal comfort under all tested conditions, both warm and cool. The average power draw is 27 Watts at 16ºC (60.8 °F), and 45.5 Watts at 29ºC ambient conditions (84.2 °F).

Keywords: Heated seat, Cooled seat, Thermal comfort, Personal comfort system, PCS, Thermal sensation.

Introduction

Thermal discomfort is one of the major complaints from occupants in offices (Huizenga at al. 2006), but as proven by several researchers (Melikov et al. 1998, Tsuzuki et al. 1999, Akimoto et al. 2003, Zhang

et al. 2010), it is possible to reduce the dissatisfaction by using personal comfort systems (PCS). In the past, PCS systems have also been called task ambient conditioning (TAC) and personal environmental control (PEC). Zhang H. et al. (2010, 2011), Zhang and Zhao (2008), and Arens (2011) established that the use of PCS systems can broaden the acceptable ambient temperature range to 18-30 °C (64.4-86 °F). This implies that a building can be controlled with an extended thermostat deadband while still maintaining occupants' thermal comfort. Hoyt et al. (2009) studied the energy saving from extended setpoints compared with the typical range of 21.5-24 °C (71-75 °F). The saving is about 10% of total HVAC energy use for each degree Celsius increase or decrease in the setpoint (about 5% for each degree Fahrenheit).

A heated/cooled chair is a type of PCS that has been found to provide comfort. Watanabe et al. (2009) studied the influence of a ventilated chair incorporating two fans in the back and seat bottom to provide isothermal forced airflow for cooling. They found the chair enabled an acceptable ambient temperature of 30°C (86 °F). Kogawa et al. (2007) tested a ventilated chair in a real office. The chair had two air nozzles installed on the two armrests. It showed that the ventilated chair could keep occupants comfortable at 27ºC (80.6 °F). The chair cooled people up to 1 unit on the seven-point thermal sensation scale. Brooks and Parsons (1999) tested a car seat heated with encapsulated carbon fabric in cool environments. They reported improved overall thermal comfort when the ambient temperature was reduced below 20 °C (68 °F). Zhang et al. (2007) tested a car seat with surface that could be both heated and cooled. Their analysis indicated that the use of a heated/cooled seat would extend drivers' acceptable range of ambient temperatures 9.3 °C downwards and 6.4 °C upwards.

The purpose of this experiment was to test the effectiveness of a heated/cooled office chair at producing comfort in a realistic work environment with an expanded ambient temperature range.

Methods

The experiments were carried out at the Center for the Built Environment (CBE) at the University of California, Berkeley, between September 2011 and December 2011.

Heated/cooled office chair (named active chair). Two actively heated/cooled chairs, two regular cushion chairs and two mesh fabric chairs were used for the experiments. The active chairs are equipped with a thermoelectric device (TE) that can increase or reduce the seat and backrest surface temperatures

(Figure 1). The fabric that covers the chair has high conductivity to increase the heat exchange between the human body and seat surfaces. The heating and cooling power of the chair is controlled by a knob located on the desk (Figure 1). The measured power at maximum heating is 42 W, and 74 W at maximum cooling.

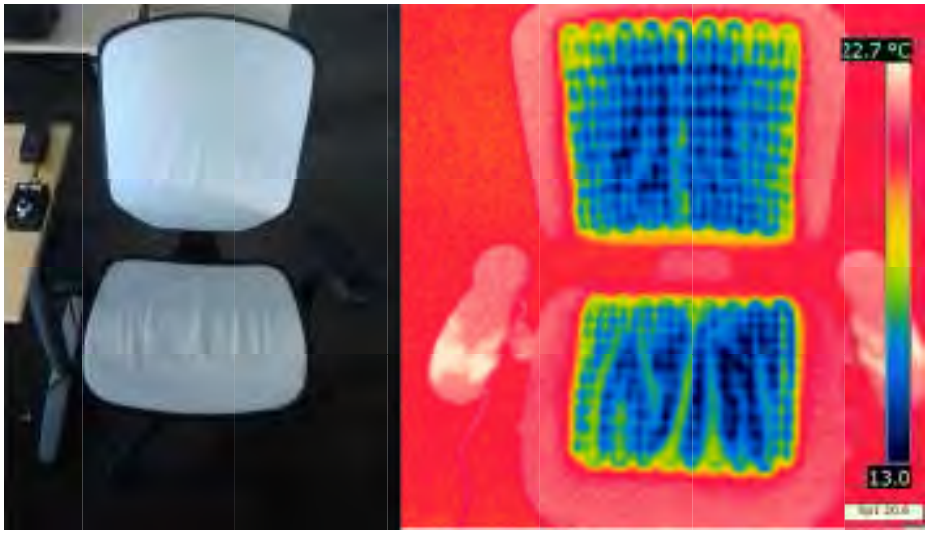

Figure 1. The active chair, control knob, and a picture of the chair with the IR camera.

Subjects and test conditions. The human subject tests obtained comfort responses and power consumption for winter (16, 18 °C) and summer (25, 29 °C) conditions. The relative humidity of the chamber was kept at 50% ±1%. The air velocity was less than 0.1 m/s (19.7 ft/min). Thirty subjects (16 females and 14 males) participated in each of the four test conditions using the active chair, for a total of 120 tests, plus an additional 30 tests at 29°C ambient temperature to study the effects of different chair pre-cooling levels. Also, tests were done with conventional chairs to provide reference conditions for comparison with the active chair. Sixteen subjects repeated the same test conditions with a commercial mesh chair (8 subjects) and a cushion chair (8 subjects), an additional 64 tests. Two workstations were installed in the Controlled Environmental Chamber at UC Berkeley so that two subjects could be tested at the same time. The chamber size is 5.5 x 5.5 m (18 x 18 ft), with windows on two sides, South and West. The windows are well shaded by fixed external shades. The temperature of the inner glass pane is controllable, and was kept isothermal with the interior. The room air temperature is controlled by 8 floor grill diffusers, and the air is exhausted through a 0.6 x 0.6 m (2 x 2 ft) ceiling return grill. Clothing was provided for the tests (0.5 clo for summer test conditions and 0.65 clo for winter test condition). During the tests the subjects were allowed to adjust the heating or cooling levels of the active chair using the control

knob. They were asked to perform computer related activities during the tests, including their own assignments.

Schedule for the tests. The tests took one hour and forty-five minutes. At the beginning of each test, the subjects sat for 15 minutes in a mesh chair to let their body to adapt to the chamber temperature. After this preconditioning time the subjects were asked to sit on the test chairs. The active chairs were pre-cooled/heated at an average power level. Upon sitting down, subjects had full control of the chair power to adjust the surface temperatures of the chair. The remaining part of the test was divided into two sessions by a ten-minute break. During the break period the subjects would stand, and in the middle of the break, they took 12 vertical steps on a 22cm tall step stool. This was to simulate a typical activity level when occupants are away from their desks in a real office. The survey questions automatically appeared on subjects' computer screen based on pre-set schedules (Figure 2).

Thirty pilot tests were conducted to verify whether pre-cooled/heated level affect the way subjects use the chair during the tests. Pilot tests results indicated that chair preconditioning level does not affect the way subjects use it during the test.

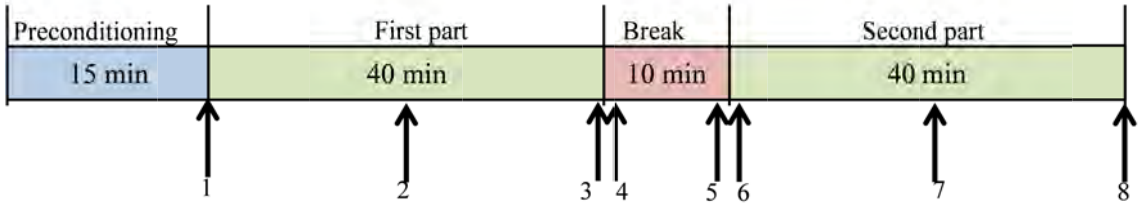

Figure 2 The test schedule. Arrows indicate times when the surveys are administrated.

Survey questions. The survey questionnaires included temperature satisfaction, thermal sensation, preferred thermal sensation and thermal comfort. The last three of these four questions were performed for the whole body, back region and gluteal region, to better understand the impacts of the two body parts involved in the heat exchange with the active chair on the whole body perception. Two survey question examples are shown in Figure 3. The scales are continuous.

Figure3 Satisfaction, thermal sensation, preferred thermal sensation and thermal comfort surveys.

Following Zhang H. et al. (2010), we used a paired thermal sensation scale (as shown in the right image in Figure 3). The paired sensation scale gives more information than the thermal preference scale (want to be warmer, no change, cooler) by itself.

Subject training session carried out prior testing. All subjects attended a one-hour training session prior to testing. The training session served to get the subjects familiar with the test procedure, the active chair, and the survey questions.

Measurements. The back and gluteal skin temperatures were monitored at 20-second intervals, using Hobo dataloggers. Room air temperature and humidity were measured at 1.1 m height. The active chair energy usage was measured every minute.

Results

This paper focuses on both stable and transient conditions. For the stable conditions we analyzed subjects' responses to the number seven survey (see Figure 2).

Whole-body thermal satisfaction. In the following analysis, the mesh chair, cushion chair and active chair are abbreviated with three letters, M, C and A, respectively (Figure 4). Every graph has a triangle representing statistical significance of differences between chairs. The red line with an asterisk on the side indicates that the difference between the two chairs is statistically significant ($p<0.05$). The statistical analysis was performed with a non-parametric method called a permutation test, using the program NPC Test R10. For more details about the program and the non-parametric method refer to Pesarin (2010).

In Figure 4 the temperature satisfaction is presented for the four room temperatures. There is a significant difference between the active chair and both the cushion and mesh chairs, for the cold and cool conditions (16 °C and 18 °C), and for the hot condition (29°C). There is no significant difference for the neutral room temperature (25 °C). For all the four temperatures the median and the average vote with the active chair is above the line that divides 'satisfied' and 'unsatisfied'. The satisfaction level with the active chair at 29ºC is just above the line, suggesting that this temperature is near the feasible maximum for the cooled chair. Under cool (16ºC and 18ºC) and warm (29ºC) conditions, subjects with the cushion and mesh chairs were not satisfied.

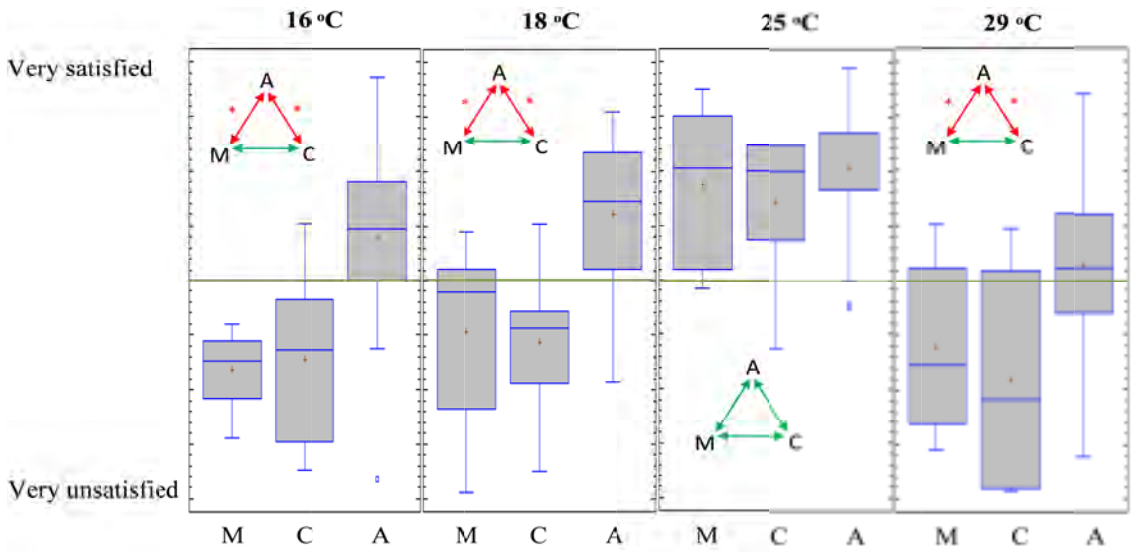

Figure 4 Whole-body thermal satisfaction

Whole-body thermal sensation. Thermal sensation for the four conditions is reported in Figure 5. As can be seen, the active chair keeps the average thermal sensation close to the neutral line. Again there is no statistical difference among the tests at 25°C. There is no statistical difference between the mesh chair and cushion chair for all 4 tested conditions. It seems that using a mesh chair does not provide any advantage in a hot environment or any disadvantage in a cold environment.

From Figure 5 it is also possible to notice that the distributions for the active chair (labeled with the A) are very similar under the four different conditions. We performed a non-parametric test to verify if the four distributions are different. The analysis indicates that there is no statistically significant difference for thermal sensation among the four thermal configurations if the active chair is used. This means that the

active chair keeps the user's thermal sensation constant over the range of temperatures analyzed in this study.

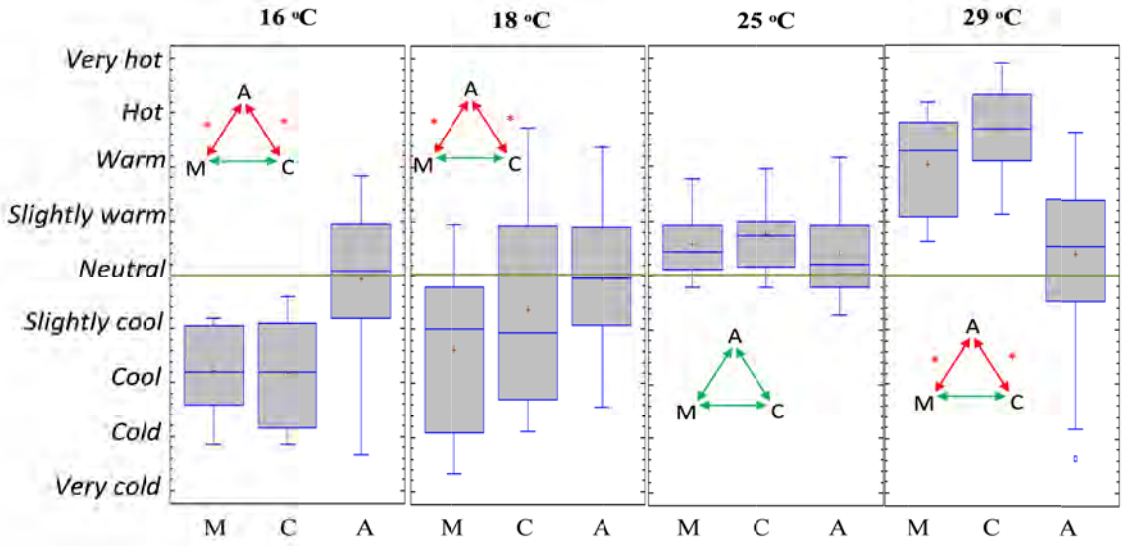

Figure 5 Whole-body thermal sensation

Whole-body preferred thermal sensation. There is only one significant difference in terms of preferred thermal sensation, between the active and cushion chairs at 18 °C. For all the other cases there is no significant difference between three types of chairs. In general, people's preferred sensations were around neutral, but we did see that subjects preferred to have a slightly warmer sensation for the mesh air at 16ºC temperature and a slightly cooler sensation for the cushion chair at 29ºC temperature (Figure 6), with both differences statistically significant.

Humphreys (2004) first used the paired scale for "preferred" thermal sensation and "thermal sensation". Humphreys and Hancock (2007) showed that there is a relationship between "desired" thermal sensation and "thermal sensation". From our results appears as if the use of the active chair fails to follow the expected correlation shown by Humphreys. This can be seen for the results at 16 °C presented in Figures 5 and 6. For this environmental condition the distributions for the whole-body "thermal sensation" are very different between active chair and mesh and cushion chairs (Figure 5), but there are no statistical differences for the whole-body "preferred" thermal sensation distributions between the three different chairs (Figure 6), as would be expected if there were a relation between "thermal sensation" and "preferred" thermal sensation. The same can be said for the 29 °C environment.

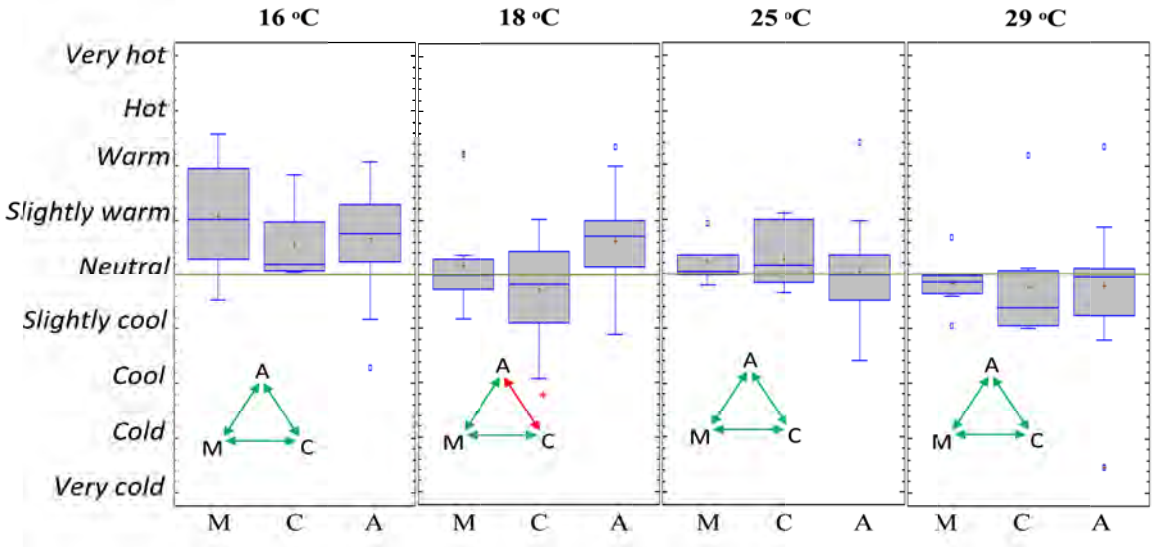

Figure 6 Whole body preferred thermal sensation.

Whole-body thermal comfort. The use of the active chair strongly influenced and improved thermal comfort for the 16, 18 and 29°C temperatures (see Figure 7). The increases are statistically significant between the active chair and the mesh and cushion chairs. With the active chair, the thermal comfort median and average are comfortable at these temperatures, while for other two chairs, they are uncomfortable. For the environment at 18 °C with the active chair, the entire population except one subject (3% of the population) is comfortable. The comfort levels are much higher in cool environments (16 and 18ºC) than in the hot environment (29ºC).

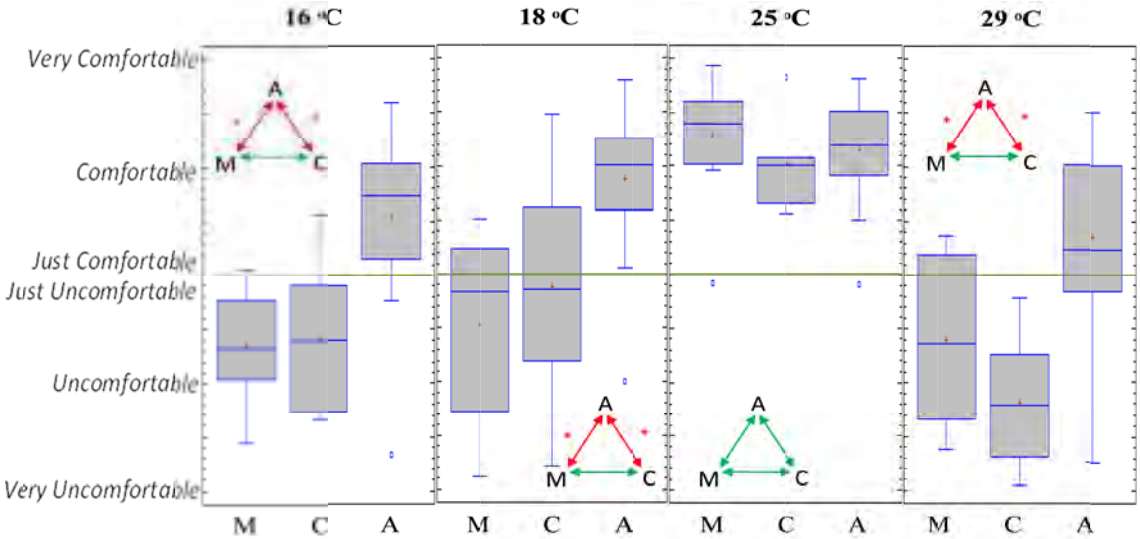

Figure 7 Whole body thermal comfort.

No significant difference was registered in whole-body thermal comfort for the three chairs in the 25 °C environment. Also in this case the active chair does not confer any advantage or disadvantage. No statistical differences are seen for the mesh and cushion chairs in any of the environments.

Others body parts. The survey questionnaires included questions for those body parts in direct contact with the chair. Questions about thermal comfort, thermal sensation and preferred thermal sensation were asked for the back and gluteal regions. The results are presented in Figures 8 to 13, following the scheme of the previous the charts.

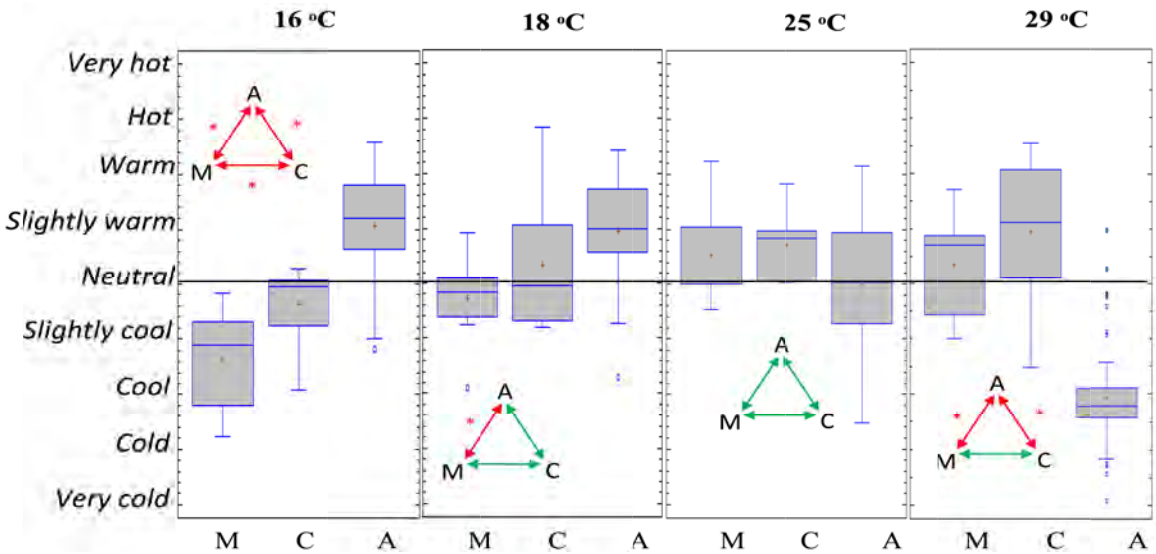

Figure 8 Back thermal sensation

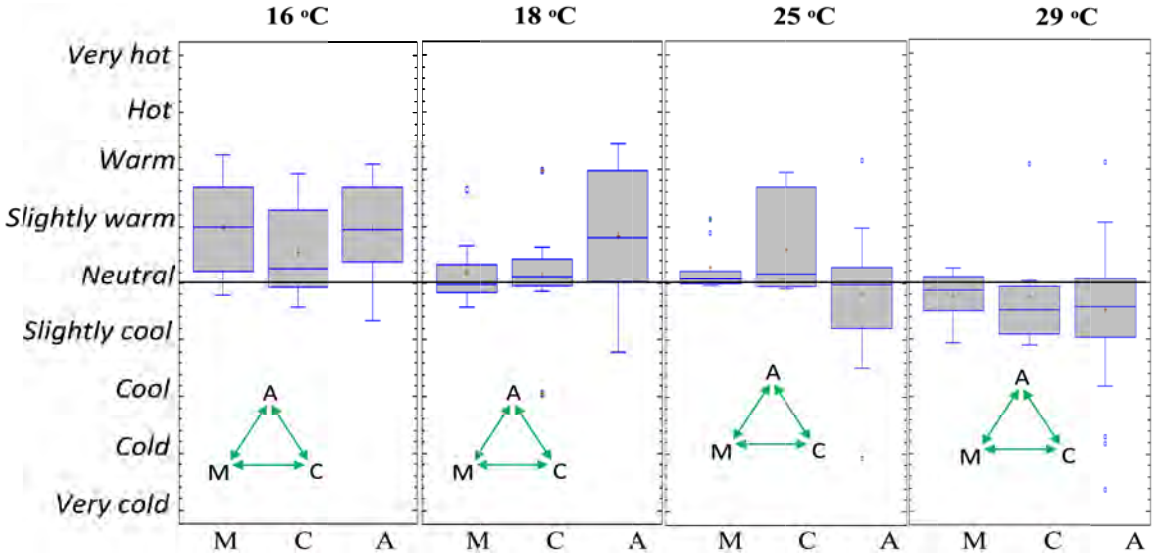

Figure 9 Back preferred thermal sensation

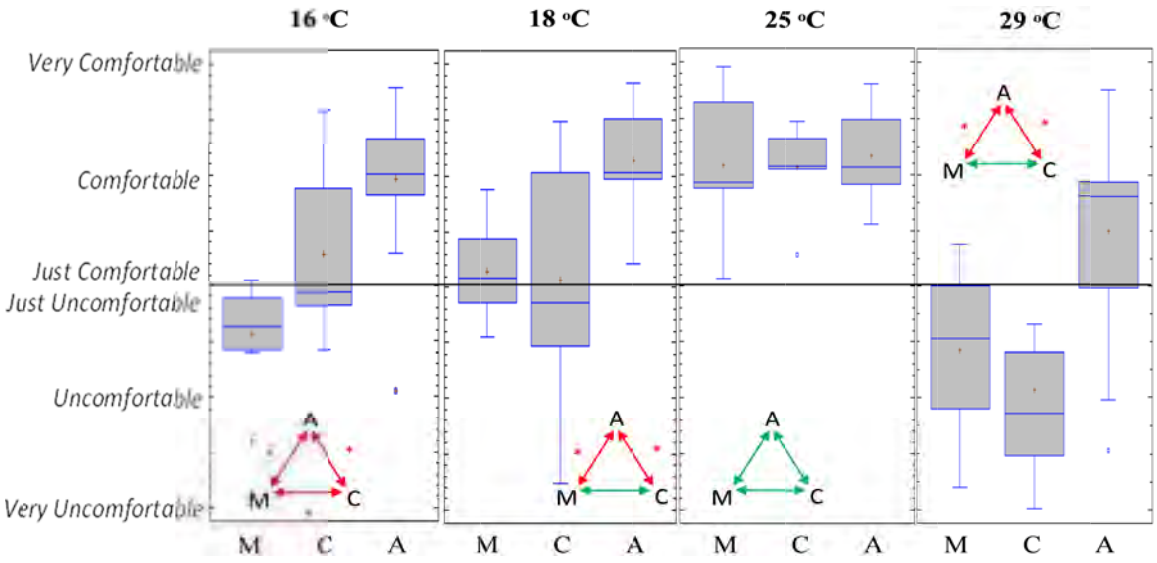

Figure 10 Back thermal comfort

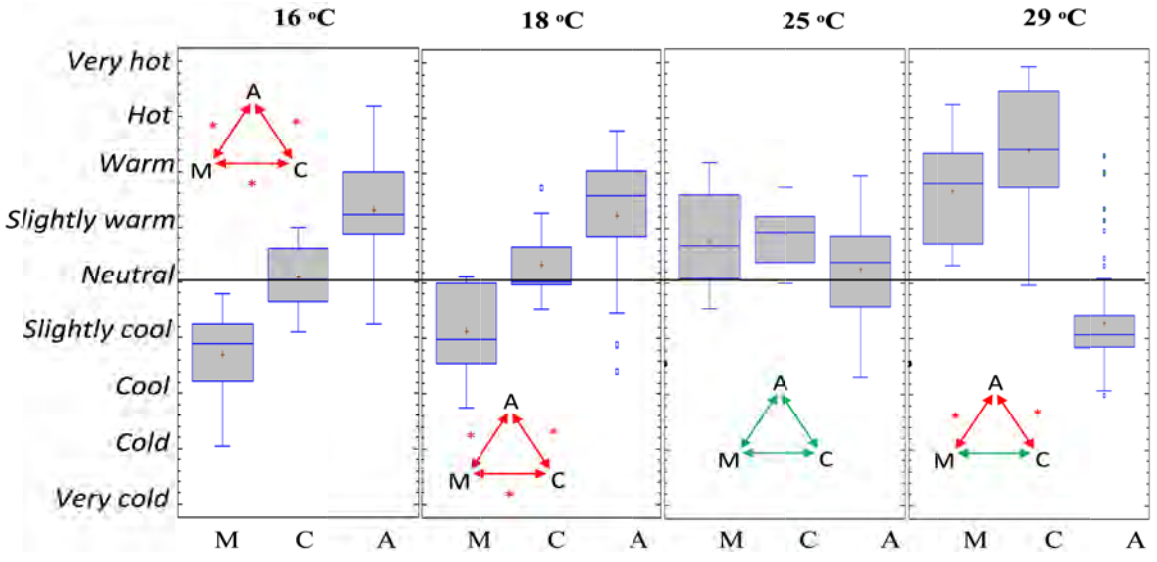

Figure 11 Gluteal region thermal sensation

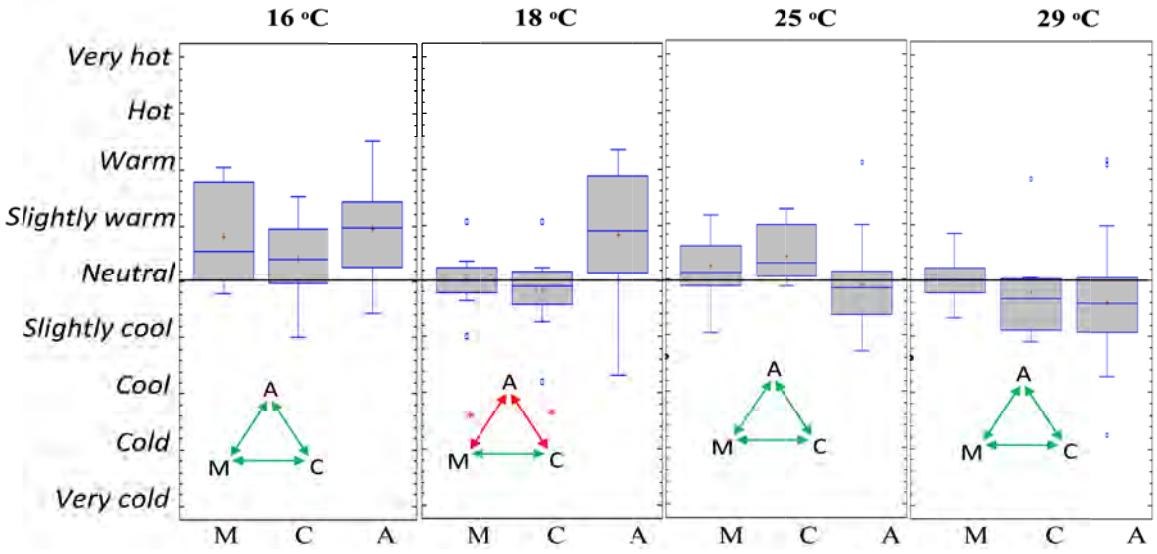

Figure 12 Gluteal region preferred thermal sensation

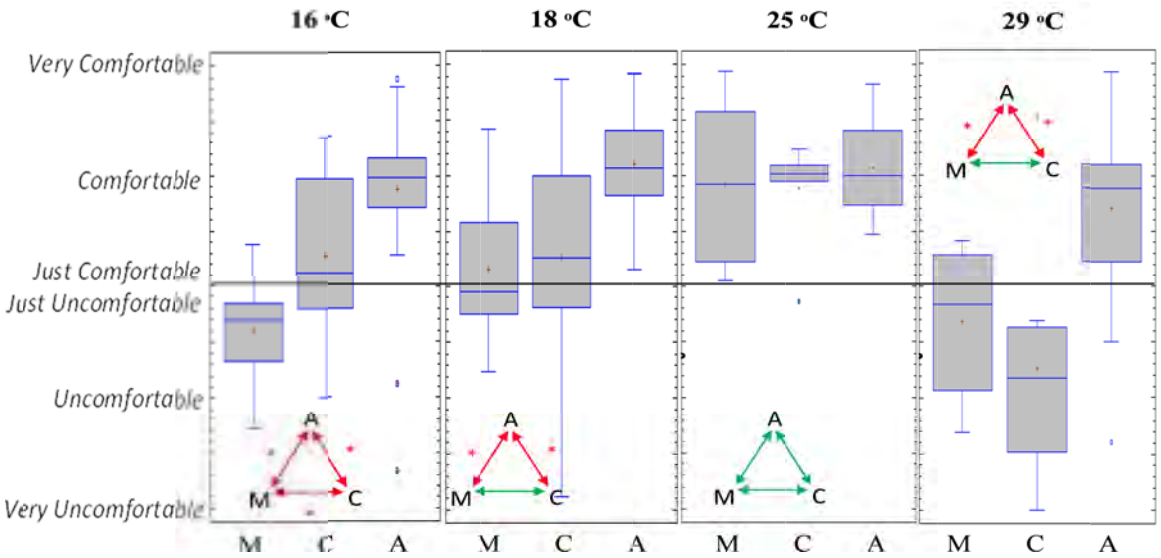

Figure 13 Gluteal region thermal comfort

Under neutral condition (25 °C) using the active chair affects neither thermal sensation nor thermal comfort.

The graphs indicate a difference in thermal comfort and perceived thermal sensation between mesh and cushion chairs for the test at 16 °C. This phenomenon was not recorded for the whole body. For the configuration at 29 °C, the statistical analysis between mesh and cushion chair never produced a p value lower than 0.05, but always very close to it. For the two body parts considered in this analysis it seems that

the cushion chair performs better under colder conditions than the mesh chair; again this result is not reflected by the statistical analysis of the whole body.

Transients. In Figures 14 and 15, temperature satisfaction, grouped by survey number as shown in Figure 2, is reported for 16°C and 29°C ambient conditions respectively. Surveys four and five (highlighted in green), are the surveys performed during the break. The reduction in temperature satisfaction during the break, when subjects had to stand up and take twelve steps, is clearly visible from the figures.

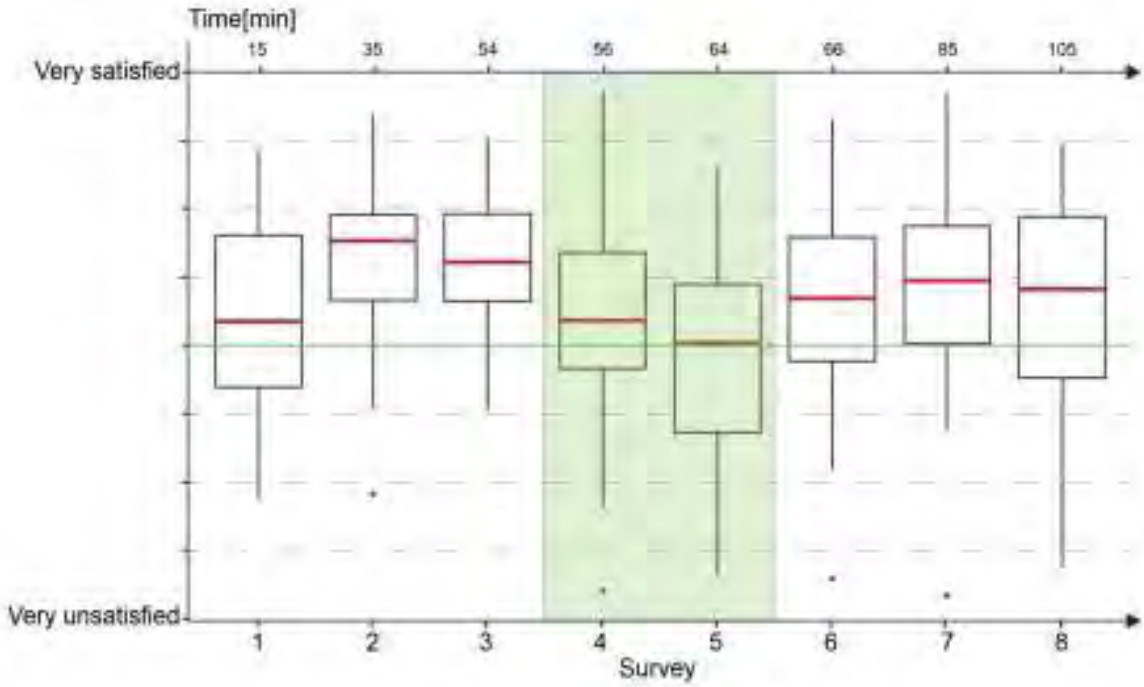

Figure 14 Temperature satisfaction, 16 °C

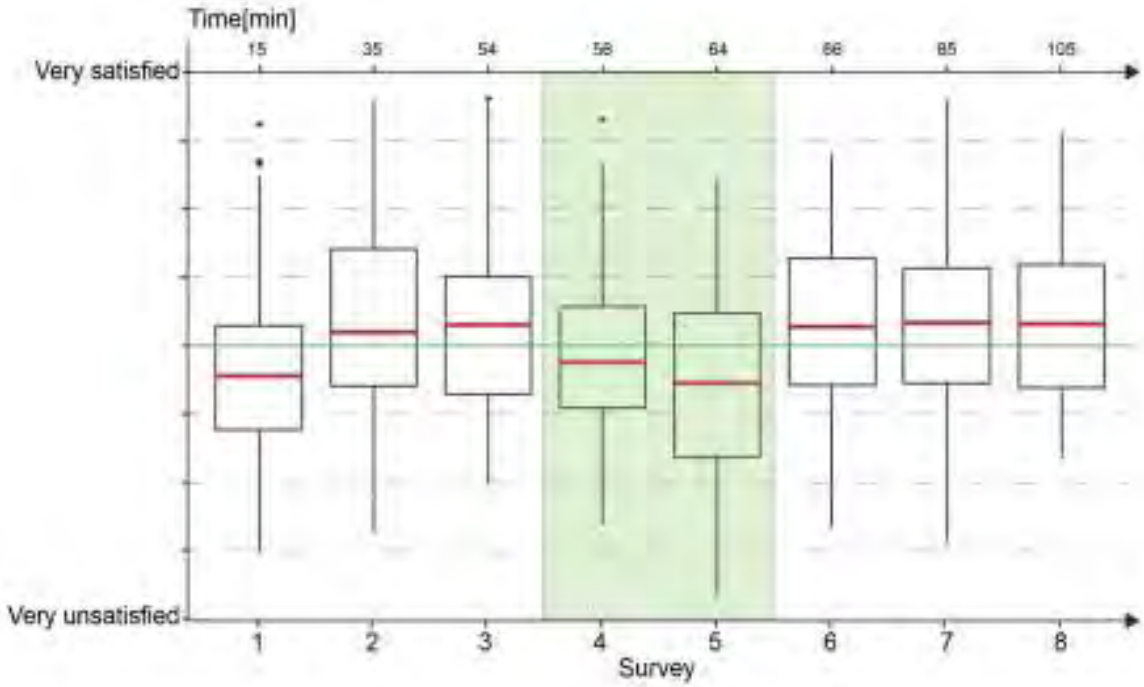

Figure 15 Temperature satisfaction, 29 °C

The active chair has an instantaneous effect on temperature satisfaction, as it can be noticed comparing survey 5 taken standing at the end of the break, and survey 6 taken one minute after the break, with the subjects in the chair again.

Energy consumption. As described above, subjects had freedom to adjust the power levels of the active chair, and the energy consumption was monitored every minute. The average energy consumption under stable conditions (last 10 minutes of every test) is reported in Table 1. The energy use in warm environments (29°C) is about double the amount used in cool environments (16 and 18°C).

Table 1. Average energy consumption under different room temperatures

Temperatures, °C (°F)	16 (60.8)	18 (64.4)	25 (77)	29 (84.2)
Average Energy Consumption [W]	27	23.5	16.5	45.5

Gender. No gender differences are shown based on the statistical analysis of the results. Energy consumptions for male and female at 29°C and 16°C conditions are reported in Figure 16. There is no statistical difference for the energy consumption (meaning the active chair surface temperature) between male and female. The distributions for 29 °C conditions are wider than for 16 °C for both genders, indicating large variations in the selected surface temperatures.

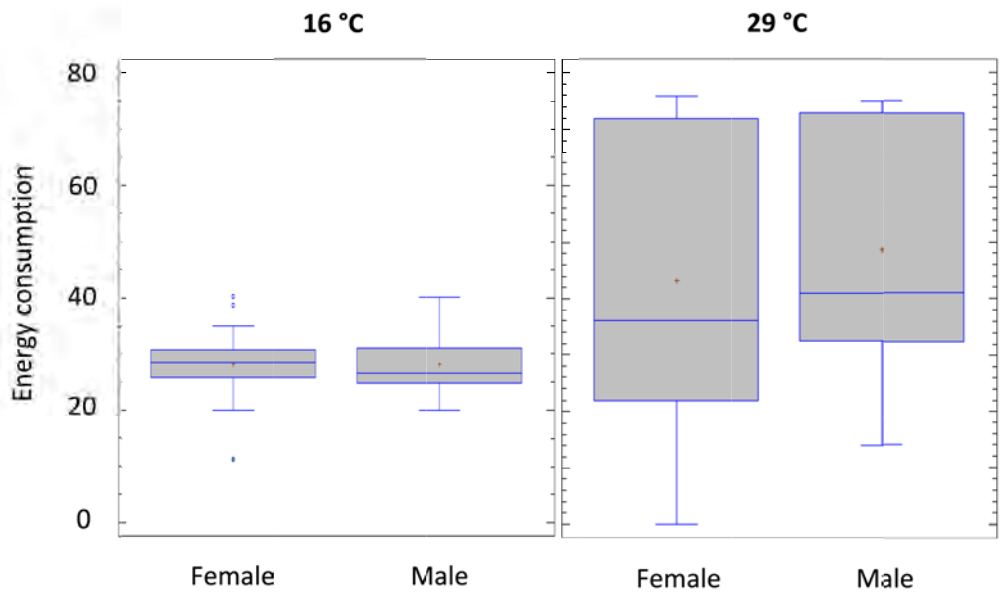

Figure 16 Chair energy consumption for male and female at 16°C and 29 °C

Discussion

1. The active chair performs better in a cold environment than in a hot environment. This result is in line with the findings of Zhang (2007).

2. With the active chair the average comfort levels were on the comfortable side of the scale (Figure 7), satisfying 80% of the subjects at 16 °C ambient temperature, but only 63% were satisfied at 29°C. Still, this is a big improvement compared to the mesh and cushion chairs, which satisfied only 25% of subjects.

   Zhang (2003) defined the "most influential group" of local body parts affecting thermal comfort as the back, chest, and pelvis. She showed that sensation from these body parts has a dominant impact on overall sensation. The active chair affects thermal sensation for two of the three body parts in this most-influential group. This explains the strong influence that the active chair has on increasing or reducing whole-body thermal sensation towards neutral.

3. It is interesting to notice that the use of the active chair shows a wider standard deviation for 29 C° (Figure 4, 5, 7). That might indicate that people have different feelings regarding the cooled seat. On exit surveys that we performed at the end of the tests, some subjects expressed their dissatisfaction

with the chair in cooling mode. The chair has the potential to cool down their bodies, but they did not like the feeling of a cold chair in a warm environment. Based on this result a more gentle system to cool down the back and bottom of the body might be advised. Another good indication for this phenomenon is the energy consumption distributions in Figure 16. The distributions for 29 °C condition are wider then for 16 °C. The feeling of a cold surface in touch with the body is so different among the subjects that some used the chair at full power and others did not use it at all. On the other hand everybody seemed to like the sensation of a warm surface in a cold environment.

4. The non-uniform environment created by the active chair modifies the correlation between "thermal sensation" and "preferred" thermal sensation. The effect of active chair on thermal sensation is significant, and its magnitude is visible in Figure 5. However, despite the influence on thermal sensation, there is no difference for the preferred thermal sensation among the three chairs. A possible explanation could be that in a non-uniform environment the "preferred" thermal sensation is not led by the whole body "thermal sensation", but it is more likely led by the thermal sensation of the most uncomfortable body part (i.e. feet in a cold environment or face in warm environment). We cannot clearly prove what affects the preferred thermal sensation in a non-uniform environment, but we can say that it is not related with the whole body thermal sensation, at least for the two extreme conditions, 16 °C and 29 °C.
5. Using the active chair does not give any benefits under neutral conditions (25 °C). This is true for the whole body as well as for single body parts.
6. The maximum chair energy consumption for cold condition (27W) is relatively low if compared with that of typical personal heaters, which are about 1000-1500W. This has a good potential for energy savings in buildings where this device (or other PCS devices) could be used. The potential energy saving related with a lower winter temperature set point (Hoyt et al. 2009) greatly overcomes the energy consumption of the chair.
7. Although the differences of thermal sensation and comfort between cushion chair and mesh chair for the back and gluteal region are statistically significant (Figure 8, 10 and 11, 13), there are no differences for the whole body thermal sensation and comfort (Figure 5 and 7). Under cold conditions

using a cushion chair instead of a mesh chair does not overcome cool or warm thermal sensations of other body parts like chest, feet and hands.

8. There is no any statistically significant difference in the way males and females use the active chair (Figure 16). Also, for all the questions covered by the survey we did not encounter any difference between males and females.

Conclusions

1. The active chair maintains subjects' thermal sensation around neutral under the tested conditions between 16 and 29°C.
2. The active chair maintains comfort at tested conditions. The improvements in comfort with the active chair are statistically significant at 16, 18, and 29°C, compared to the mesh and the cushion chairs.
3. The comfort levels in heating mode at 16 and 18°C room air temperatures are higher than with in cooling mode at 29°C room air temperature. The heated chair in cool environments appears more effective than the cooled chair in a hot environment.
4. The non-uniform thermal environment created by the chair does not support the previously observed correlation between whole body thermal sensation and preferred thermal sensation.
5. Under neutral temperature conditions (25 °C) there is no improvement resulting from the use of the active chair.
6. There is no statistical difference between the cushion and mesh chairs for any tested room temperature.
7. The average energy use of the active chair is 27W in heating mode and 45W in cooling mode.
8. The active chair can re-establish satisfaction within one minute after a break away from the chair, in which the subject stands and performs a light metabolic activity task.
9. There is no difference in the way females and males use this PCS device.

Acknowledgment

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Interim report March 2013

UFAD Energy Optimization Study #1

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ABSTRACT

Underfloor air distribution system (UFAD) is a mechanical air distribution strategy in which the conditioned air is primarily delivered to the zone from a pressurized plenum through floor-mounted diffusers. It has several potential advantages compared to conventional overhead (OH) mixing systems. However, most of the energy simulation programs widely used by the industry are not able to represent two distinct features of UFAD systems: room air stratification and the underfloor supply plenum. The situation has been improved with the development of a UFAD module in EnergyPlus. The Center for the Built Environment developed the modeling methods, tested them extensively, and conducted numerous studies of UFAD energy performance. This paper presents results from one of the sensitivity studies conducted by CBE with its development version of EnergyPlus.

INTRODUCTION

Underfloor air distribution system (UFAD) primarily delivers the conditioned air to the zone from a pressurized plenum through floor-mounted diffusers. 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, two important features of UFAD systems, which are room air stratification and the underfloor supply plenum, could not be properly represented by most of the energy simulation programs widely used by the industry. The situation has been 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. EnergyPlus/UFAD facilitates investigation of energy and comfort performance for a wide variety of design and operating conditions; the focus of the paper is on the sensitivity of energy performance to these factors. .

Alajmia et al. (2010) conducted the simulation study using EnergyPlus to investigate the energy benefit of UFAD compared to conventional ceiling based system. It is found that 30% of energy saving could be achieved. Anecdotally, others have reported similar savings. However, these studies do not provide information on the simulation details such as how thermal decay and stratification are addressed and how terminal units are sized. Thermal decay might not be taken into account during the simulation. Many of these studies have used "work-arounds" with mixed systems simulation tools such as eQuest.

The work reported here was supported by the California Energy Commission "Advanced Integrated Systems Technology Development" Project. The CBE sub-project under which this work was conducted is named "UFAD Optimization Study #1" nicknamed herein as "Opto1."

SIMULATION SOFTWARE

The authors developed the office building prototype described below for development, testing, and performance studies. CBE development versions based on EnergyPlus 3.1 and 6.0 of software were used for this a related studies. [Bauman et al. 2007], [Linden et al., 2009],[Webster et al. 2008, 2010, 2012], [Lee et al. 2011].

EnergyPlus is a relatively new building energy simulation program developed under support from the U.S. Department of Energy. (DOE 2010) Based on the combination of two predecessor programs DOE-2 and BLAST, it has greater capabilities than those two as well as many other programs. Among the unique features of EnergyPlus that makes it a good platform for simulating UFAD (and other advanced technologies) are discussed in the following.

ROOM AIR STRATIFICATION

Room air stratification (RAS) is a key characteristic of UFAD systems compared to conventional OH systems. Increased stratification is associated with improved energy efficiency (Linden et al., 2009) and is considered a key marker of well performing systems. To represent stratification, the room is divided into two fully mixed sub-zones as shown in Figure 1. EnergyPlus performs a heat balance on each sub-zone with the surface between the upper and lower layers in the room configured to be an "air surface" that is transparent to all radiation.

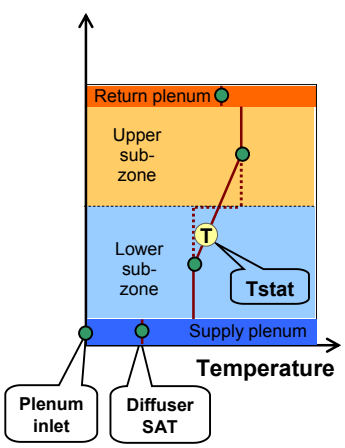

Figure 1. EnergyPlus two-layer UFAD model

There are a number of factors that influence the degree of stratification in UFAD systems and it is

important that these effects be captured in UFAD modeling. Both full scale and bench scale experiments were performed to develop the stratification algorithms. Details can be found in Bauman et al. (2007), Liu and Linden (2008), Webster et al. (2008), and DOE (2010).

In the studies report herein, the thermostat is assumed to be always in the lower layer. This is a simplification of how real systems work where the thermostat is typically located at 4 ft and stratification is not uniform, contrary to how it is shown in Figure 1. This also obscures somewhat the impact of different stratification levels causing lower average occupied zone temperatures and therefore requiring high thermostat setpoints to maintain the same average lower layer temperature; an estimate of the impact is shown in the Room Temperature Setpoints section below. However, maintaining the lower zone temperature constant at the control point makes comparison to overhead systems equivalent, and the error in energy performance by assuming the lower zone mixed at the control point is expected to be small (especially for comparisons and studied in this project).

SUPPLY PLENUM TEMPERATURE RISE

A major barrier to modeling UFAD systems has been the inability to model supply air plenums that account for the interaction with adjacent zones and the effects of heat gain to the supply air. Sometimes known as “thermal decay,” the temperature rise of the conditioned air due to convective heat gain as it travels through the underfloor supply plenum is an important phenomenon that must be taken into account in UFAD system modeling. Due to its significance, active research has been done by many researchers including Woods et al. (2008), Schiavaon et al. (2011), Bauman et al. (2006) and Schiavon et al. (2010). The authors have conducted extensive research including experimental and field studies and computational fluid dynamics (CFD) simulations to investigate the nature and impact of the temperature rise; results from these studies, show that it is too important to ignore. A comprehensive description of the phenomenon and its implication has been reported by Lee et al. [2011]. Unlike other energy simulation tools, EnergyPlus has the capability to model each underfloor plenum as a completely separate zone, performing all the heat, mass and energy balances to accurately simulate thermal decay. In addition, it enables the investigation of the effect of different supply plenum configurations on energy consumption.

RADIANT HEAT TRANSFER

EnergyPlus has the capability to perform a detailed heat balance on each surface including the effects of radiant exchange, to calculate the surface temperature in each time step, including radiant exchange between floor and ceiling via transparency of the boundary between the two room layers.,

FULLY INTEGRATED SOLUTION OF ZONE, SYSTEM AND PLANT

Unlike typical tools simulating the building loads (based on the assumed system capacity) and the HVAC system separately, EnergyPlus performs the system and plant simulation, and the air and surface heat balances simultaneously, allowing the real-time interaction among those different building components. This is important for realistic energy modeling, since the behaviour of all building components such as zone air and surfaces, chillers, fans, boilers, and pumps are highly interconnected with each other in each time step.

WHOLE BUILDING SIMULATION MODELS

A three-story prototype office building with a rectangular shape and aspect ratio of 1.5 was used.

The floor plate size is 40,000 ft^2 (total floor area is 60,000 ft^2) and each floor is composed of 4 perimeter zones, an interior zone and a service core, which represent approximately 28%, 56% and 16% of the floor area, respectively (see Figure 2). The floor-to-floor height is 13 ft and the return plenum height is 1.9 ft. The raised floor height is 16 in. Strip windows are evenly distributed (i.e., a "ribbon" window) in the walls and the window-to-wall ratio (WWR) is 38%. The constructions and the thermal properties of windows change based on each climate and they comply with ASHRAE 90.1 (2004). Design day simulations use ASHRAE 0.4% summer and 99.6% winter design conditions. An overview of the building model characteristics is presented in Table 1. Figure 2 shows a layout of a floor plate.

Table 1. Whole building model feature summary

Feature	Overhead	UFAD
Floor plate size, ft^2	40K	Same
Number of floors	3	Same
Floor to floor height, ft	13	Same
Return plenum height, ft	3.3	1.9
Supply plenum height, ft	NA	1.3
Skin/glazing	T-24/90.1/low-e	Same
Window/wall ratio	38%	Same
Room setpoints, Occ (Unocc), °F	75/70 (85/60)	Same
Internal loads: Lights, W/gft^2	1.0	Same
Equipment, W/Gft^2	0.8	Same
People, Gft^2/Person	201	Same

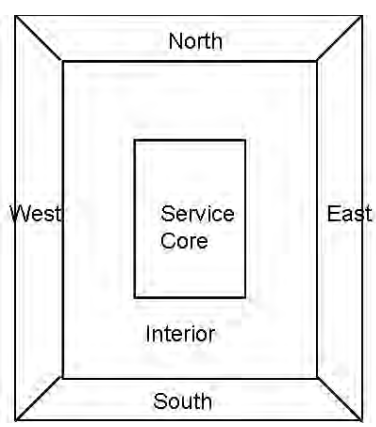

Figure 2. Floor plan of the building model

From 5:00 until 19:00 the system controls the internal air temperature to a cooling and heating temperature setpoint of 75°F and 70°F, respectively. During the night the system is switched off.

Infiltration was assumed equal to 0.065 cfm/ft^2) (flow per exterior surface area), constant for 24 hours. The minimum outdoor air flow rate was set to be 0.15 cfm/ft^2) (flow per gross floor area).

For zone terminal units of each zone, supply air is distributed through swirl diffusers in interior zones and linear bar grille diffusers in the perimeter zones. Perimeter zones are served by variable speed fan coils (VSFCU) where the fan is on (and heating coil is off) during cooling mode; during heating mode, the fan and the heating coil are on. Table 2 shows a summary of the terminal unit specifications. With some variation in terminal unit configurations, the system type modelled is very typical of current practice in the industry.

Table 2. Terminal unit specifications

	Overhead	**UFAD (VS-FCU)**
Zone terminal simulation type, Interior	Single duct VAV/Reheat	Single duct VAV (pressure controlled swirl diffusers)
Terminal unit simulation type, Perimeter	Single duct VAV/Reheat	Variable speed FCU (VS-FCU)
Minimum ventilation settings	0.4 cfm/sf, 30%, 0.15 cfm/sf (VAV heating)	0.15 cfm/sf[1]

As summarized in Table 3, the building is served by a single variable speed central station air handling unit (AHU) including an air-side economizer, a chilled water cooling coil, a hot water heating coil and a supply fan. The AHU fan is simulated as being controlled with constant supply air temperature (SAT) and a static pressure reset strategy. The central plant consists of a central centrifugal chiller with variable speed pumps and a two-speed cooling tower. A gas fired hot water boiler provides hot water to all heating coils.

All constructions, load and operating schedules, and equipment specifications were based on ASHRAE 90.1-2004, Appendix G specifications [ASHRAE 2004]; some changes (e.g., schedules were made simpler and outside air rates were drawn from CA Title-24-2005 specifications) to expedite analysis of results.

Table 3. Summary of HVAC system configurations

Parameter	**Overhead**	**UFAD**
SAT settings	57°F	57°F, 63°F
Supply fan design static	4.1 iwc	3.1 iwc
Economizer	Diff dry bulb (except MI)	Same
Minimum outside air rate	0.15 cfm/ft^2	Same
AHU heating coil	SF, MI = No, MN = Yes	Same

[1] This is the minimum setting, however the VSFCU is set to a minimum of 12% (consistent with the minimum fan speed of EMC drives on these units) and 0.15 cfm/ft^2 usually result in a lower fraction, ~3-15%. There also is a leakage flow of 7.5% (hardwired into the EnergyPlus code) in the deadband when the fan if off, however, there appears to a bug that fails to turn the fan off in these runs.

SAT reset	No	Same
SA pressure reset	Yes	Same
AHU return	Relief fan	Same
Chillers (screw)	2	Same
VS Pumps	Primary only	Same
Chilled water reset (OSA)	Yes	Same
Cooling tower	2-speed	Same
Boilers (forced draft)	2	Same
Hot water reset (OSA)	Yes	Same

RESULTS

Results shown below cover a range of UFAD vs. OH comparisons and sensitivity analyses for three climates (ASHRAE climate zones using TMY2 weather files); Miami (1A), San Francisco (3C), and Minneapolis (6A). These climates were selected to "bracket" the energy performance to covering a range of US climate extremes. All results show HVAC site energy EUI (kBtu/ft^2/yr), without central AHU heating coil except where noted. Supply air temperature for all overhead cases used was constant at 57°F unless otherwise noted.

UFAD VS. OH ENERGY COMPARISON

Figures 4 through 9 show results for each of the three climates studied. These charts show three cases of minimum terminal unit settings for OH and two SAT settings for UFAD. The savings range are shown only for the OH @ 30% minimum and "dual max" (i.e., VAV heating) cases since the 0.4 cfm/ft^2 case is an unrealistic and outmoded specification. Savings relative to the cases indicated by the star (*) are shown on the chart.

The dual max control logic is illustrated in Figure 3a below; the primary features are VAV heating and a low minimum setpoint that meets ventilation standards.

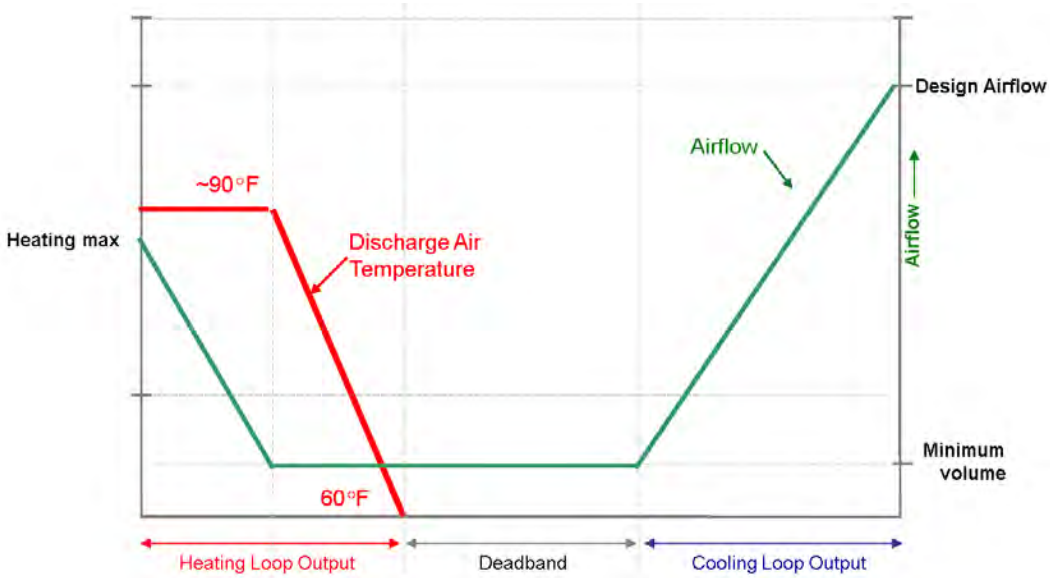

Figure 3a. Control diagram for OH VAV reheat terminal units (illustrating VAV heating)

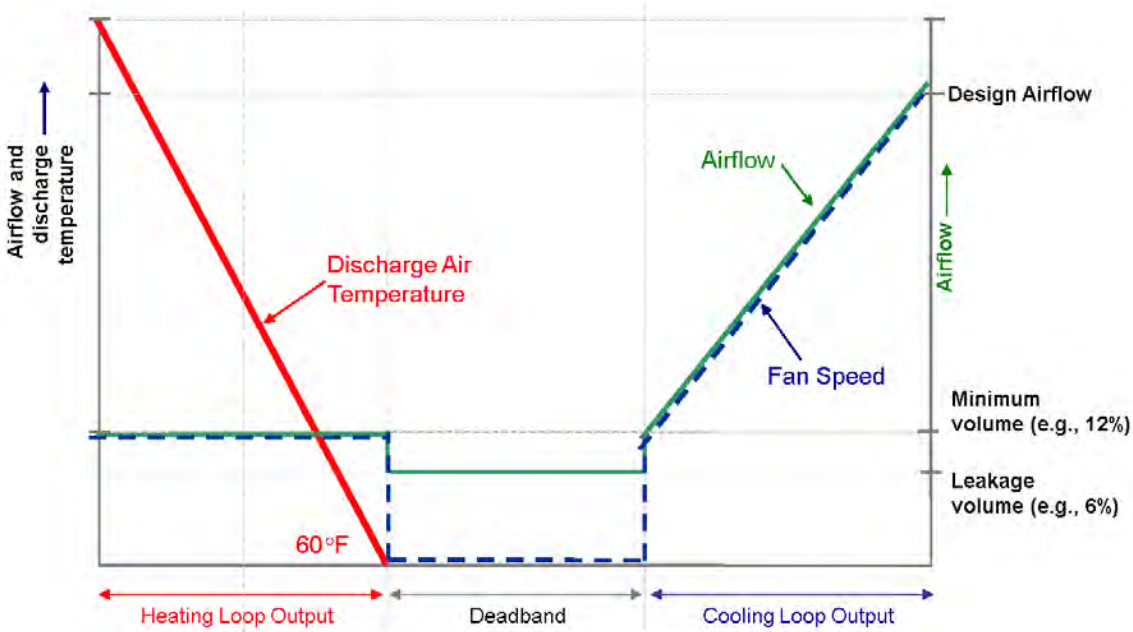

Figure 3b. UFAD VSFCU control logic

In San Francisco, an extreme mild climate, the savings shown in Figure 4 is negative (-13.8%) for the SAT57 (i.e., 57°F fixed supply air temperature) relative to the OH VAV heating case. This results from higher airflow and therefore cooling and fan energy for UFAD. However, comparing UFAD for SAT57 with SAT63, the effects of the economizer in mild climates is demonstrated; while airflow (and therefore fan energy) is greater for SAT63, cooling energy is substantially lower due to the effects of the economizer. Figure 5 shows a breakdown for this case which indicates that the majority of the cooling occurs during partial cooling when the economizer is open for this climate. Although the ratio of open vs. total cooling is roughly the same, the total cooling for UFAD (UFAD cooling savings of 30%) is lower because of more hours of economizer operation.

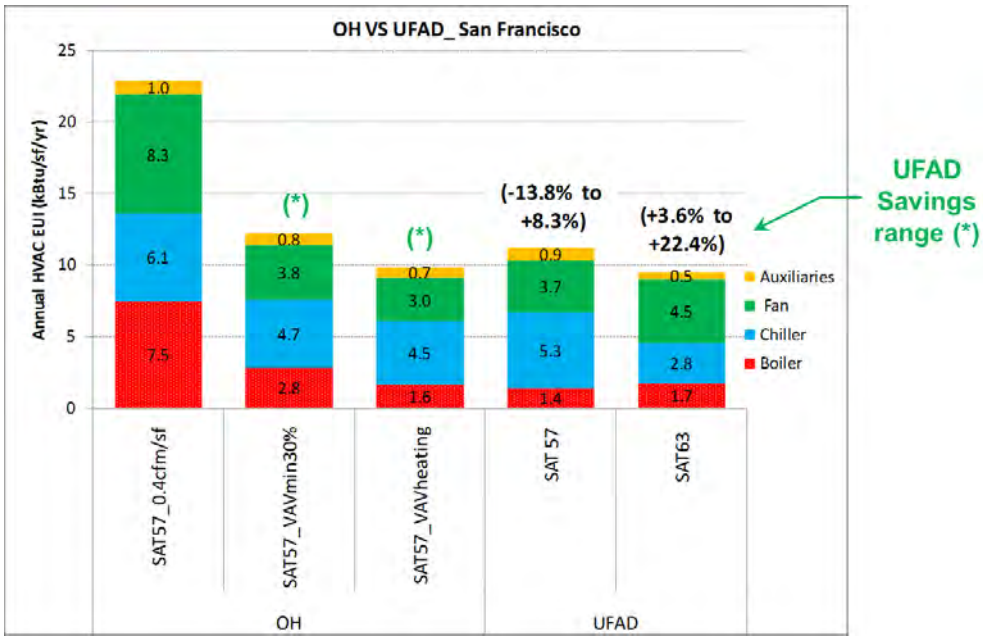

Figure 4. UFAD performance comparison, San Francisco

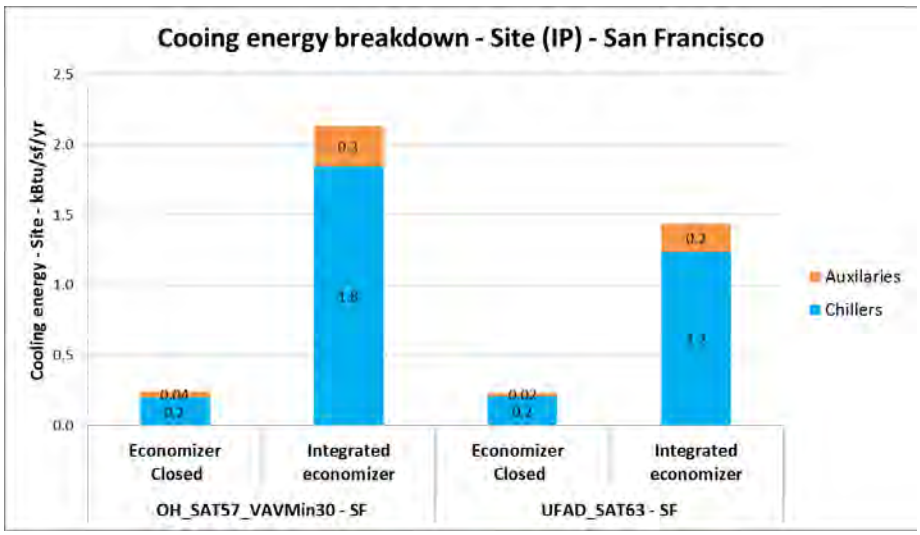

Figure 5. Cooling energy breakdown, San Francisco

For comparison purposes, Figure 5a shows a breakdown for the warmer climate of Sacramento.[2] In this case the primary cooling is just the opposite with most cooling occurring when economizer is closed, and with no cooling savings.[3]

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[2] Results from Opto2 study with a different model, and using SAT reset. [Bauman et. al. 2013]

[3] Other studies [Webster et. al. 2010] is different assumptions have shown a ~20% cooling savings for Sacramento

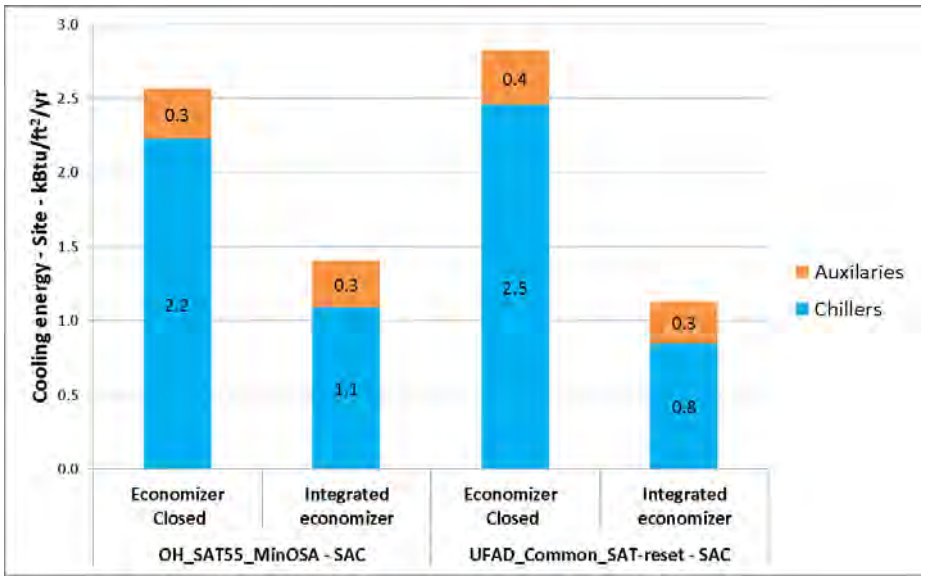

Figure 5a. Cooling energy breakdown for Sacramento

Figure 4 also shows a heating energy increase for SAT63. This results from an increase in reheat energy as shown in Figure 6. Reheat is increased due to the combined effects of terminal unit (FPB) entering temperature and airflow rate. The entering temperature at the FPB results from the temperature rise in the plenum; although it may seem like the entering temperature is proportional to the SAT delivered to the plenum so for example the difference would be ~6°F, there is a counterintuitive effect of airflow rate on temperature rise in the plenum. As airflow is increased for SAT63, the plenum temperature *rise decreases* suggesting that the entering temperatures for both SAT57 and SAT63 would tend to equalize. However, as shown in Figure A-1, entering temperatures are still higher for SAT63 than for SAT57, but the difference between them is reduced to between ~0.5 to 2.3°F in Figure A-1. When the airflow differences and resulting temperature differences in Figure A-1 are computed, reheat is shown to be greater for SAT63. For series systems, the airflow in the interior zone will also have an indirect effect on the FPB entering temperature.

The heating results also show the impact of lower reheat for UFAD; for the SAT at 57°F , the same as OH, UFAD heating energy is lower than OH due to the effects of temperature rise in the plenum which lowers reheat.

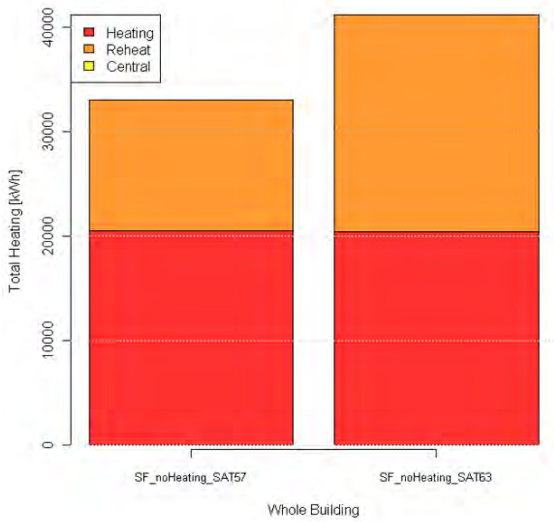

Figure 6. Heating breakdown for two SATs for San Francisco

Figure 7 show results for Miami, an extreme hot-humid climate. In this case, no economizer was used (fixed outside air only) due to the high humidity in this climate zone (per ASHARE Standard 90.1-2004). In this climate where an economizer cannot offset the effects of increased airflow required by UFAD, the savings are negative indicating that the limited energy benefits of UFAD in these types of climates.

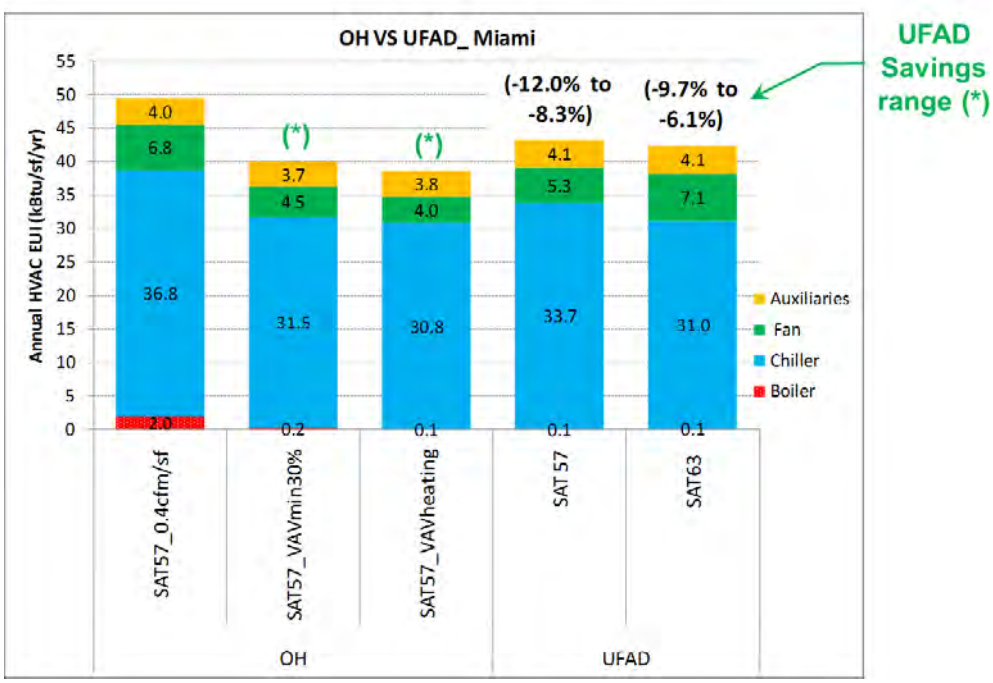

Figure 7. UFAD performance comparison, Miami

Figure 8 shows results for Minneapolis, an extreme heating climate. These results show the large impact of heating on the overall HVAC energy. A large component of the heating in this case results from energy consumption due to operation of the heating coil in the central AHU. This heating coil is required in cold climates to mitigate the effects of cold supply air temperatures in winter on interior zone comfort; i.e., there is no method for zone heating of interior zones in the types of UFAD systems modeled.

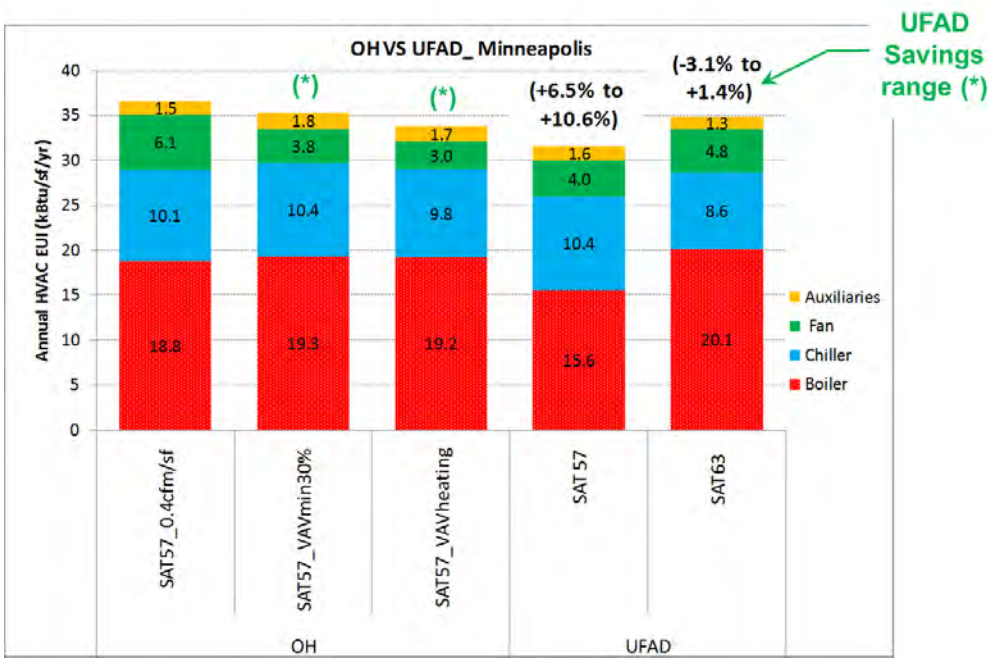

Figure 8. UFAD performance comparison, Minneapolis

The cause of the heating increase for the SAT63 case is due to the increase in central coil heating due to larger airflow rates compared to the SAT57 case as shown in Figure 9. Note that the reheat and heating load for this case is the reverse of that shown for San Francisco. In addition, Figure 9 shows a reduction in room heating load for SAT63; this may be due to the warmer plenum temperatures reducing the cool floor component of the heating. Figure A-2, shows that for SAT63 although the airflow is higher, the deltaT for SAT57 is less which results in lower reheat for SAT63; likewise, the room load is also shown to be smaller.

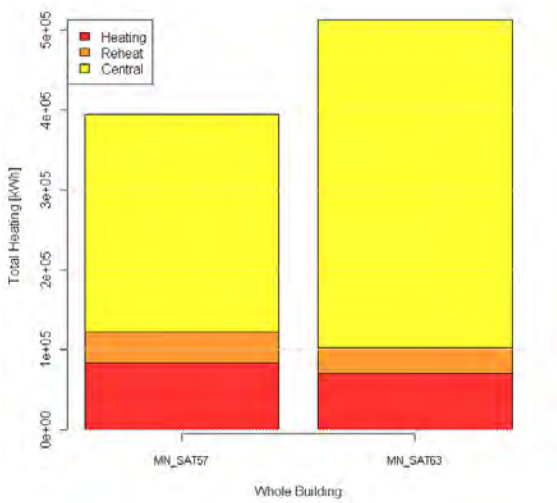

Figure 9. Heating breakdown for Minneapolis

IMPACT OF DESIGN AND OPERATING CONDITIONS

The following section describes the results from sensitivity studies aimed at understanding the impact of various design and operating choices.

Terminal unit minimum volume settings

Minimum volume settings for terminal units are a key factor in energy performance; primarily for heating and secondarily for cooling and fan energy. Figure 10 illustrates this impact for conventional overhead systems. While the authors believe the 0.4 cfm/ft^2 case is not realistic (and has recently been deleted from 90.1 standards), it serves to demonstrate the large impact it has on heating performance. The second case, 30% minimum, has been the default standard for many years. Likewise, at the other extreme, the reduction in heating shown in Figure 10 for San Francisco with dual max heating is significant, but not so in the other extreme climates. For example, in Minneapolis there are virtually no heating benefits (due to the central heating coil) but there are still secondary benefits. This indicates that dual max strategies in the milder climates can be an effective way to minimize total energy consumption. Cooling is reduced somewhat in the VAV heating cases due to the lower minimums in the deadband which reduces the AHU fan and cooling energy.

UFAD minimum volumes were set to achieve 0.15cfm/sf which yields fractions (in SF) of 3-13% depending on zone. However, the VSFCU has a minimum setting of 12% (during heating and cooling) and a leakage flow when the fan is off of 6%. There was no VAV heating in this study.

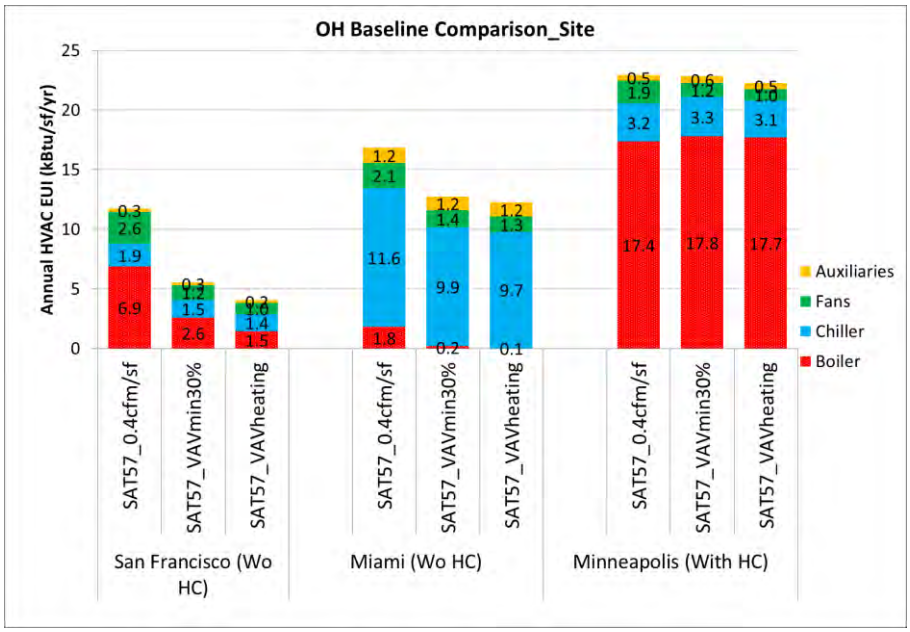

Figure 10. OH baseline impact of minimum volumes for three climates

Supply air temperature

Figure 11 reiterates what has been shown in previous figures, that SAT has an impact on energy consumption due to the effects of the economizer. However, the impact does depend on climate, more impact in mild climates less so in extreme cooling and heating climates due to other mitigating factors. For example, in Miami the cooling energy is lower for SAT63 which is counterintuitive to previous discussions showing that higher SAT increases airflow and therefore fan and cooling energy. In the case of Miami, fan energy does increase but cooling does not, which results from the fact that more latent cooling is done with SAT57 in this climate. Likewise, in Minneapolis, changes in heating dominate the total energy consumption for reasons explained earlier.

One effect of increased SAT which nominally allows increased use of the economizer is illustrated in Figure 12. The results for each climate demonstrate a “tradeoff” effect; as SAT increases, airflow increases causing fan and cooling energy to increase which is in turn mitigated by the effect of increased economizer use (which will vary with climate). This tradeoff reduces the impact of increased economizer use more than is commonly expected. However, Figure 12 shows that the net savings is positive in each of the three climates studied, although the overall difference (in electric energy) is small. In addition, other studies [Webster et. al. 2010] show that increasing SAT can have diminishing returns. For example, although not shown here, the energy use for San Francisco for SAT = 60°F is less than at 63°F.

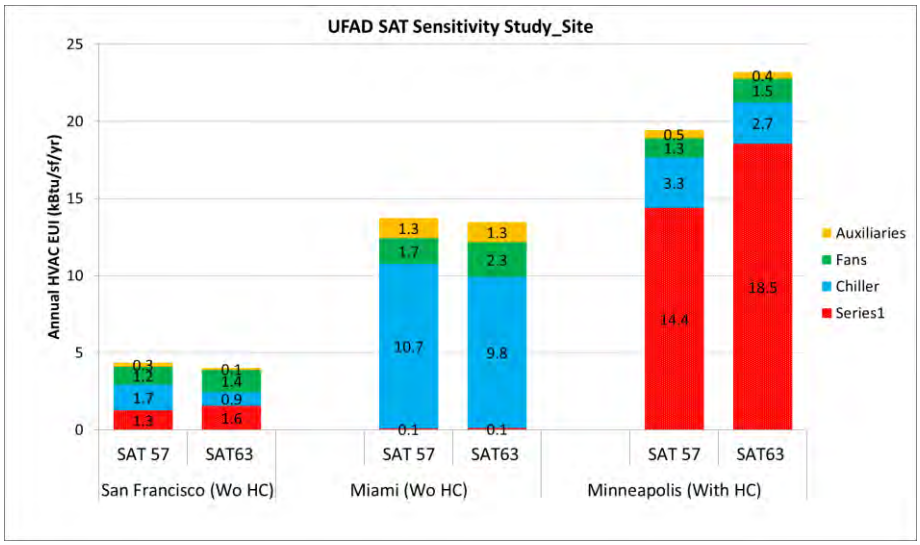

Figure 1. SAT sensitivity study, stacked

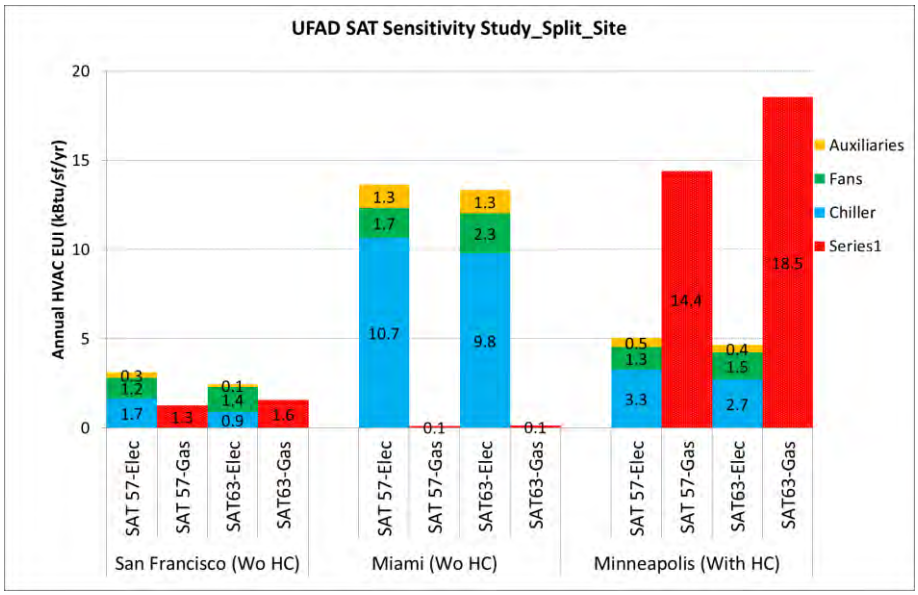

Figure 12. SAT sensitivity study for three climates, unstacked

Central heating coil

As discussed earlier, a central heating coil is essential in colder climates to mitigate overcooling of the interior zones in winter. Figure 13 illustrates what would occur if a central heating coil was used in a mild climate; heating energy would be significantly increased while cooling and fan energy are virtually constant.

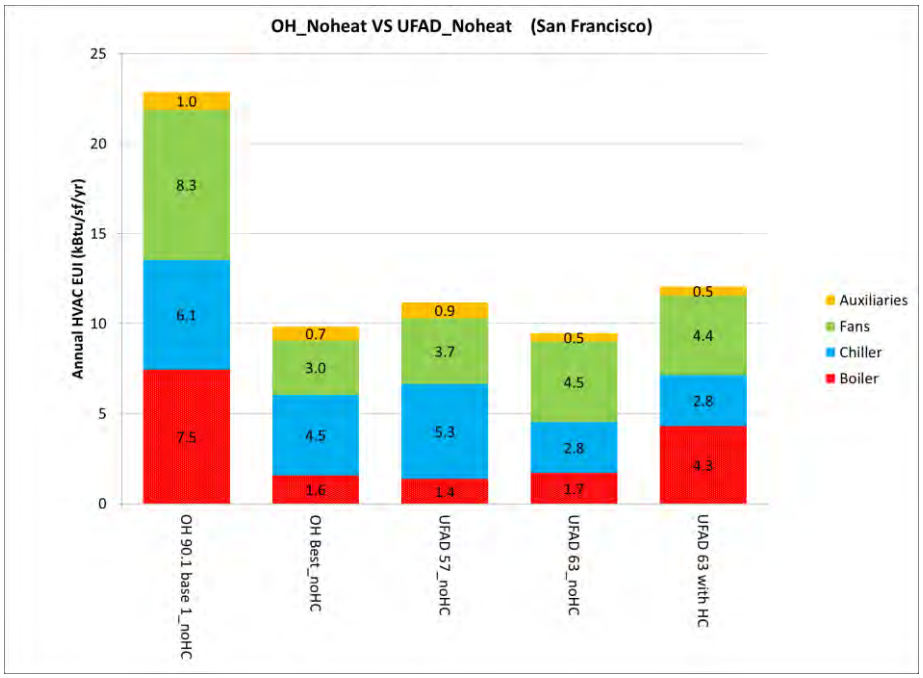

Figure 13. Effect of central heating coil, San Francisco

Plenum configurations

Two cases of plenum configuration were studied, parallel plenum air supply and insulated slab. In a parallel plenum configuration (as opposed to a series plenum configuration) air is supplied in parallel to both the interior and perimeter zones so that they each receive the coolest possible air. The expectation is that this will allow perimeter zones to perform better when the supply air entering the FPBs is lower. This does not eliminate temperature rise in the plenum, however, as a "fully ducted" arrangement would. (See Simbuild 2012 paper [Webster et. al. 2012] for a more thorough study of the impact of plenum configuration.) Therefore, the effects overall are small as shown in Figure 14; only the extreme cooling climate of Miami shows a benefit.

Theoretically, adding insulation to the underside of the supply plenum slab (i.e., at the ceiling of the return plenum for the floor below) should decrease the heat transfer into the supply plenum resulting in lower overall energy use. Figure 14 indicates that this has a negligible impact in all three climates studied. These results may be compromised to some extent by the fact for these runs a central heating coil was used. The reasons for the apparent counterintuitive result would require more investigation, but other studies by the authors indicate that the limited impact is may be due to that fact that the portion of heat transfer through the slab is smaller than expected, and/or the complex interactions in the system are altered in such a way to affect the fundamental behavior.

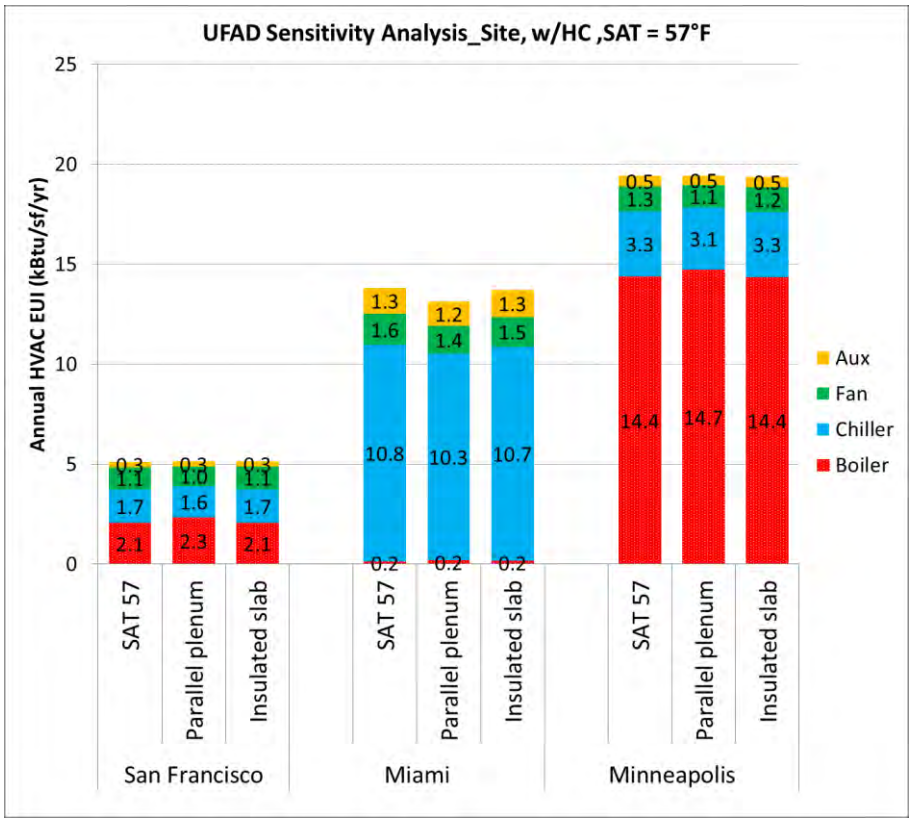

Figure 14. UFAD sensitivity analysis, plenum configuration (with central hearing coil)

Room air stratification

Room air stratification is one of the signature differences between conventional overhead and UFAD systems. It is illustrated in Figure 15. In this sub-study, the impact on energy due to increased stratification was studied.

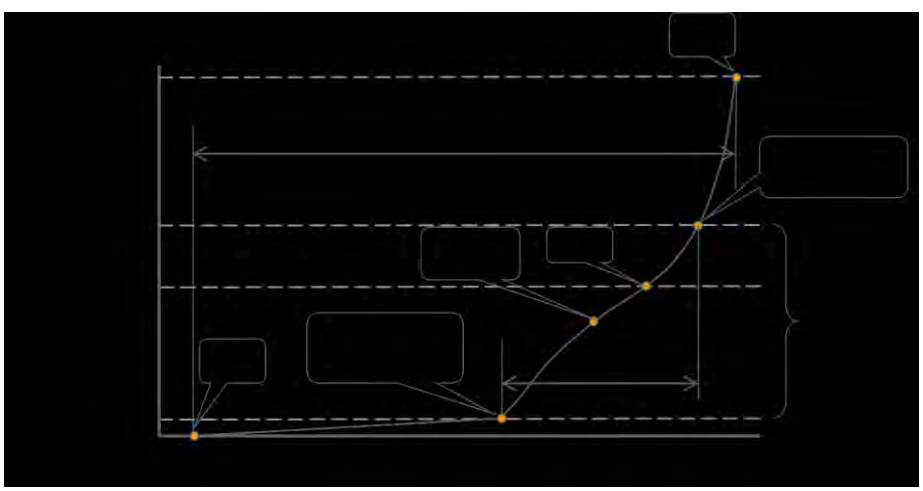

Figure 15. Illustration of room air stratification

Figure 16 shows the effect of increased stratification for two climates. Savings are shown relative to a baseline of SAT63 as indicated by the asterisk. Phi represents the degree of stratification, it is

the ratio of the deltaT of the lower zone to the room deltaT, it varies from 0 to 1; lower Phi means more stratification. Typical diffusers on the market today can deliver a Phi in the range of ~0.65 to 1.0, depending on the design. Most designs produce a phi at design flow of ~0.7-0.8; only horizontal discharge diffusers approach 0.6. Furthermore, in the author's experience, many designs, especially linear bar grilles, produce negligible stratification in practice. This can be addressed somewhat by increasing the number of diffusers as indicated by the 2xDiffusers case in Figure 16. Phi's lower than 0.6 are theoretically possible but have yet to be demonstrated. However, Figure 16 shows what the impact would be if they could be realized. As indicated by the savings values, the overall effects are greater in the mild climate of San Francisco than the hot climate of Miami. While the fan energy percentage savings are equivalent for each climate, the cooling energy savings for San Francisco are greater. This results from lower airflow (due to less heat gain in the lower (controlled) layer) as stratification increases. This suggests that the effect will also apply to colder climates as well. However, the effect appears to be less for warm climates as indicated in Miami; see Appendix B for a way to estimate this difference.

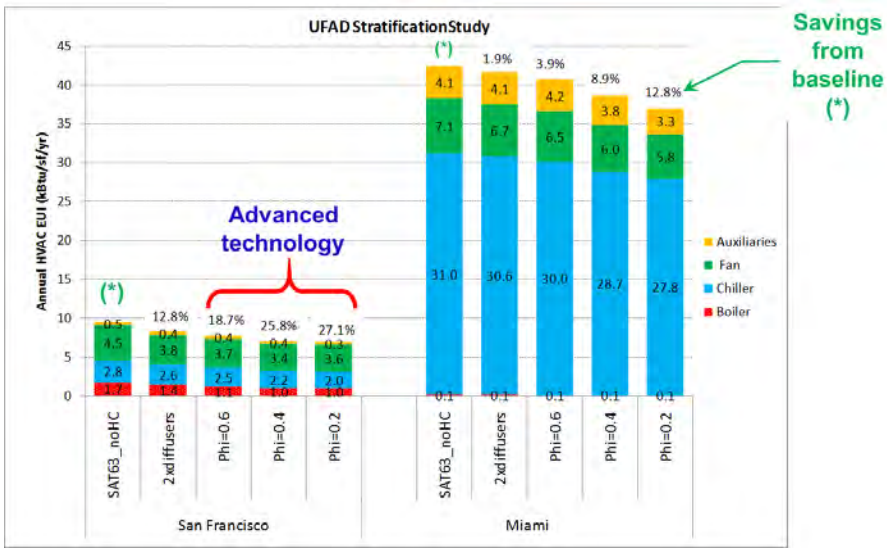

Figure 16. Effect of stratification

Room temperature setpoints

Figure 17 shows the impact of increasing room cooling setpoint from 75°F to an upper limit of 79°F. As discussed above, this provides an estimate of the impact of raising the setpoint to accommodate lower zone stratification. In this study, the design sizing was not changed for each run; i.e., it was assumed to be operating change where the system sizing was the same as the baseline. This has a small but beneficial effect stratification which would further reduce cooling and fan energy. Figure 17 shows, as expected, that cooling and fan energy are lower but the effect is greater in San Francisco (~18 %, 27% at 77 and 79°F, respectively) than in the other cities where is less than 10% (or negative as in Minneapolis) for 77°F.

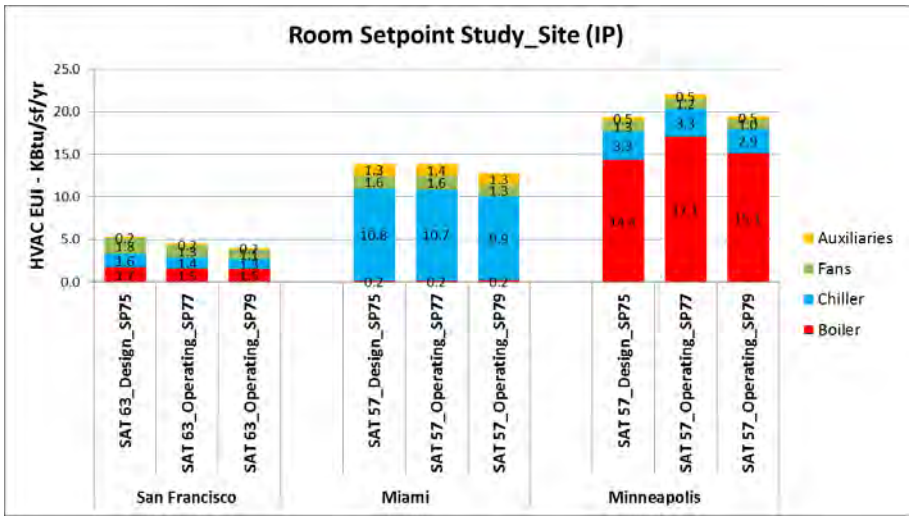

Figure 17. Impact of increasing room cooling setpoint.

Fan energy breakdown

Fan energy consumption is another important factor in overall energy use. UFAD systems of the type modeled here that use fan powered boxes, cause an increase in overall fan energy use as shown in Figure 18 where FPB energy is ~10-20% of total fan energy. This is partly due to the fact that the fan efficiencies for these units is so low; ~15% at peak. Also, in these studies the unit is turned off in the deadband so the energy use would be even greater if the units are not turned off in the deadband.

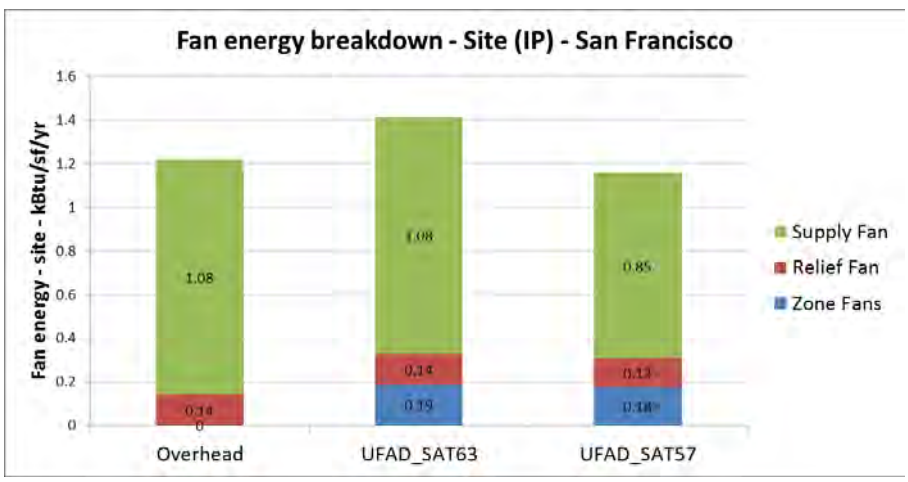

Figure 18. Fan energy breakdown, San Francisco

DISCUSSION

Highlights of the findings from this study are described in the flowing sections. These highlights are generalized findings based on the three climates studied and the system type simulated.

HEATING ENERGY SUMMARY

- Counterintuitive reheat effect – For series plenum arrangements in mild climates with low heating loads, reheat is increased when SAT is increased due to the impact. However, in cold climates, the reverse occurs; reheat is decreased as SAT increases. This results from

the way that VSFCU operate; the electrically commuted motor (ECM) drive, has a minimum speed of ~10-12% which results in a minimum flow corresponding to the size of the unit.[4]

- Central coil effects – Since UFAD systems have no independent way to heat the interior zones, central coils at the AHU are necessary to prevent overcooling the interior zones. This results in a large increase in the heating energy.
- OH vs UFAD reheat – Reheat for UFAD systems is significantly less than OH due to plenum temperature rise that increases entering temperature of the terminal unit.
- VAV heating – VAV heating using the dual max strategy significantly lowers heating and fan and cooling energy somewhat.

COOLING ENERGY SUMMARY

- Economizer "tradeoff" effect – Increased economizer use, a climate dependent factor, appears to reduce cooling energy in all but hot-humid climates. However, this requires increasing SAT setpoints which causes increased airflow and consequently fan energy. This is called the tradeoff effect which ends up reducing the effect of the cooling benefits of the increased SAT for UFAD systems. Furthermore, the impact of increased heating (due to increased reheat or central heating coil energy) further erodes the overall savings potential.

  This study did not simulate SAT reset, which is required in new standards and may have a significant impact on UFAD cooling and heating performance.

- Latent cooling – Latent cooling appears to have an impact on cooling energy for hot-humid climates. For example, in Miami where an economizer is not used SAT at 57°F would yield lowest overall energy except for the increase in cooling due to latent load.(i.e., at 57°F SAT cooling would normally be less than at 63°F but in Miami, it is greater.)

OTHER

- Stratification impact – Increased stratification has the potential to lower overall energy consumption by reducing all end use components; cooling, fan, and heating energy consumption. However, few diffusers being offered deliver this increased stratification potential in practice.
- Room cooling setpoints – Increasing room cooling setpoints is an obvious way to decrease energy use. Well-designed UFAD systems that create good stratification can allow this setpoint to be increased, contrary to OH systems. However, the effects appear to be greater in mild climates.
- Fan energy – For the type of UFAD system simulated in this study (i.e., FPB terminal units) total fan energy tends to be greater for UFAD even when supply fan energy is equivalent (e.g., OH 57°F SAT, UFAD 63°F) due to the impact of terminal unit fans that have been found to have very low efficiencies.

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[4] In these studies, "exact sizing" was used instead of "unit sizing" where the size is selected from available size ranges offered in practice, which could cause some changes in the results when SAT63 and SAT57 are compared. However, since the simulation zones represent an aggregation a series of smaller real zones, on average the roll-up sizing is probably fairly representative.

- Plenum configuration – Using a parallel plenum configuration or insulation on the underside of the supply plenum slab showed negligible differences from the baseline.

CONCLUSIONS

In this study, three climates were simulated for a series plenum, fan power box terminal unit system so the impact of design and operating factors could be studied over a range of extreme climates. UFAD energy performance was gauged by comparing to conventional overhead system applied to the same building model and designed and simulated according to ASHRAE 90.1-2004, Appendix G standards. The results are somewhat compromised by the discovery of anomalous behavior for the FPBs; the magnitude of this effect is known.

It is clear from the results that climate has a significant effect on performance due factors somewhat unrelated to UFAD; e.g., no economizer in Miami, and a central heating coil in Minneapolis. UFAD performance can be optimized the most by increasing SAT to maximize the use of the economizer, increasing stratification, and increasing room cooling setpoints. However, these strategies produce a tradeoff effect that tends to reduce the potential savings; e.g., increasing SAT tends to reduce cooling, but increase fan and reheat energy. Reducing minimum volumes at terminal units by using a dual max control strategy has been shown to decrease heating energy consumption.

Overall, this study suggests that UFAD energy performance is lowest in mild climates and where the design maximizes stratification. Increasing room setpoints can also have a beneficial effect in all climates, but it is more pronounced in mild climates.

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APPENDIX A – TERMINAL UNIT OPERATION DETAILS

The following figures show detail hourly outputs for the terminal units operated with SAT57 and SAT63. These figures help explain the counterintuitive effects on reheat and room heating load discussed above.[5] When the zone is in the deadband, the minimum airflow is ~0.2 and 0.3 m^3/s, SAT57 and SAT63 respectively, which is the leakage flow of 6% when the fan is supposed to be off (see footnote 1). The airflow appears to wander between the two minimums, which is likely exacerbated by the action of the deadband bug (see footnote 4), which complicates these analyses. The intended action is better represented in Figure A-2.

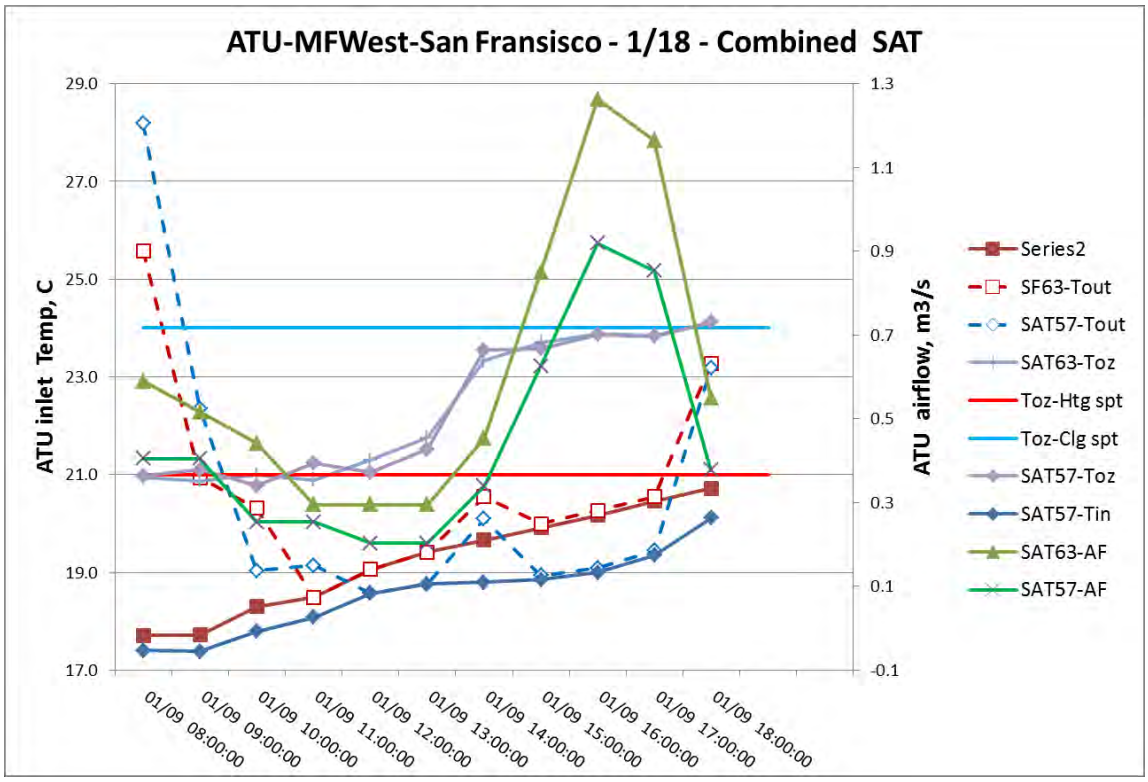

Figure A-1: Terminal unit performance comparison between SAT57 and SAT63, San Francisco

For Minneapolis, the VSFCU runs at its minimum volume (i.e., 12% of cooling design airflow consistent with minimum fan speed for units with ECM drives) – which is different for each SAT case due to the unit sizing differences) and then attempts to drop to its leakage flow at hour 15:00 or 16:00, depending on case. This explains why the reheat is greater; for SAT63, the airflow is greater airflow but the deltaT_rh (i.e., room temperature (Toz) minus the FCU entering temperature (Tin) is lower resulting in less reheat. It is also clear from this figure that VAV heating does not occur for the UFAD model.[6]

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[5] Also illustrated is the anomalous behavior when the zone is in the deadband; known as the "deadband bug." This result in heating being applied in the deadband when not called for. Another anomaly is shown at the end of the day when heating is taking place even though the room temperature is at the cooling setpoint.

[6] It may be that the VAV heating only is occurring between the leakage flow and the 12% minimum, which

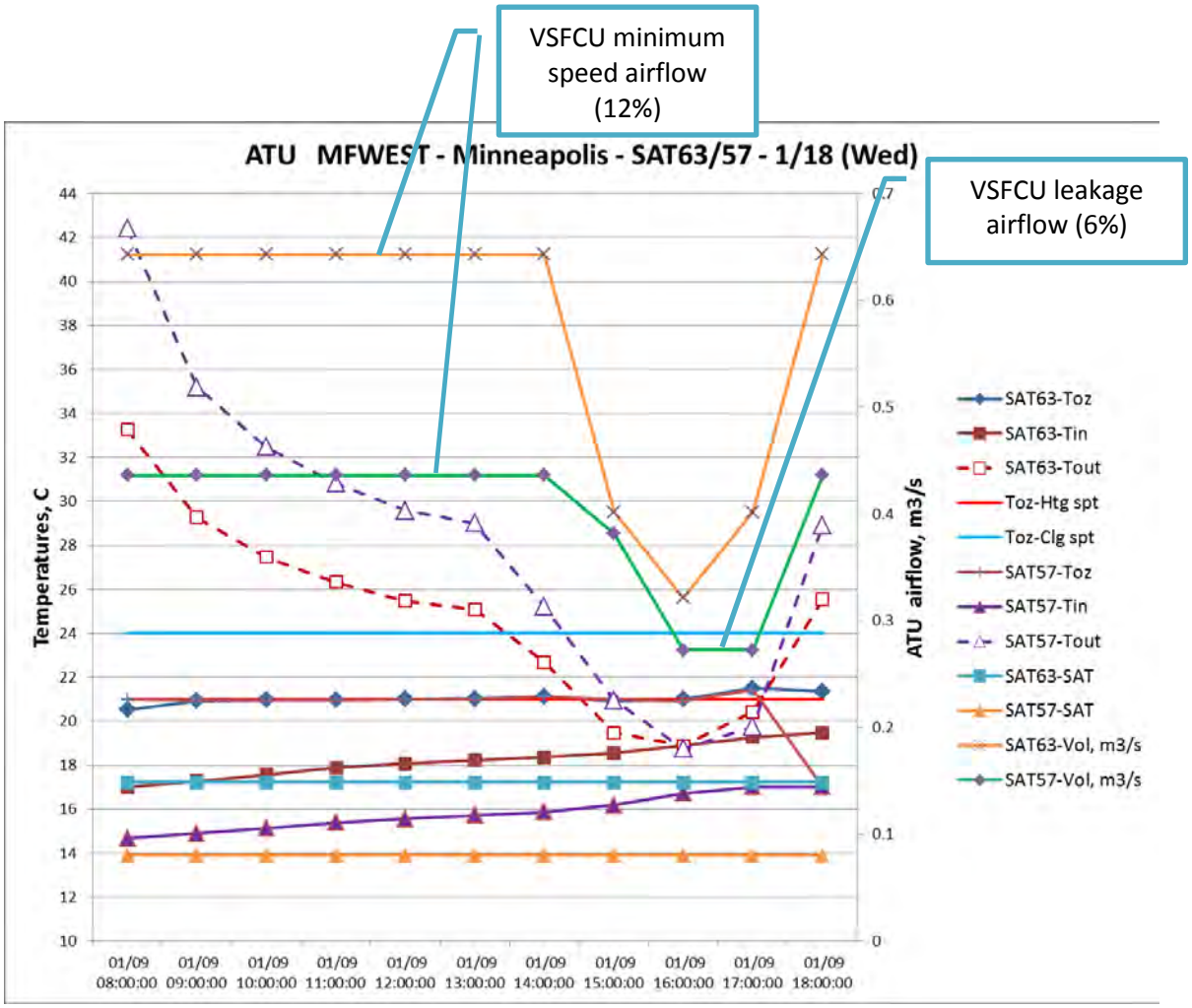

Figure A-2: Terminal unit performance comparison for SAT57 and SAT63, Minneapolis

APPENDIX B – EXAMPLE CALCULATION OF ENERGY SAVINGS DUE TO STRATIFICATION

Figure B-1 shows a standard peak airflow calculations used for sizing calculations for OH systems but modified to reflect the differences between OH and UFAD. Using this equation, the difference between UFAD and OH can be estimated by substituting typical values for the UFAD related parameters.

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would be in error. Further work clearly needs to be done on the UFAD VSFCU algorithms.

- Modify "well-mixed" zone airflow equation for UFAD:

$$CFM = \frac{UCLR * phi * ZF * Q_{clg-OH}}{1.1 \times (Btu/hr \cdot cfm \cdot °F) \times (T_{OZ} - T_{diff})}$$

- UCLR = (underfloor cooling load ratio): the ratio of UFAD to OH cooling loads (see design tool discussion)
- $Phi = (T_{OZ} - T_{diff})/(T_R - T_{diff})$ cooling load distribution in the room: the ratio of cooling load in lower occupied zone (OZ) vs total room cooling load
- ZF = Zone fraction - Zone/room portion of total cooling load (i.e., room split)
- Q_{clg-OH} is the total room cooling load of an equivalent OH system with the same heat gains
- Delta T = $(T_{OZ} - T_{diff})$ Room OZ temperature minus diffuser supply temperature (assumes OZ average temperature = control setpoint)

Figure B-1: Simplified equation for peak airflow calculations

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).

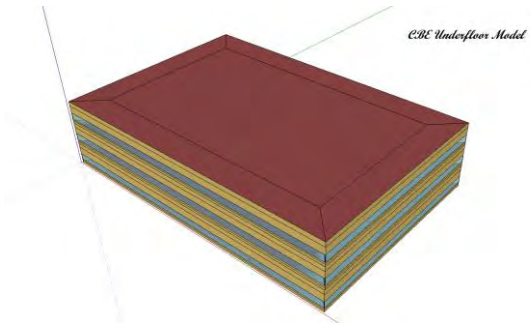

Figure 1. Illustration of building model and zoning

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.

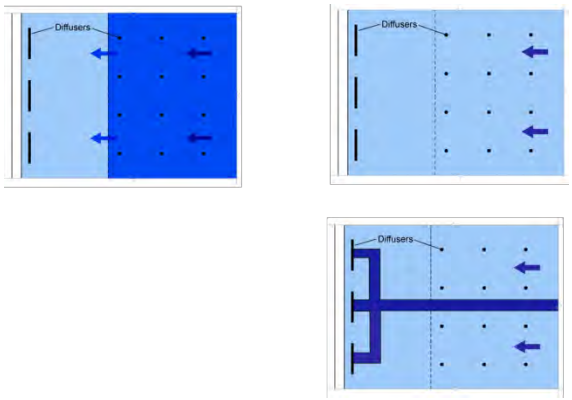

Figure 2. Supply plenum configurations; Clockwise, (a) Series, (b) common, and (c) ducted perimeter

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

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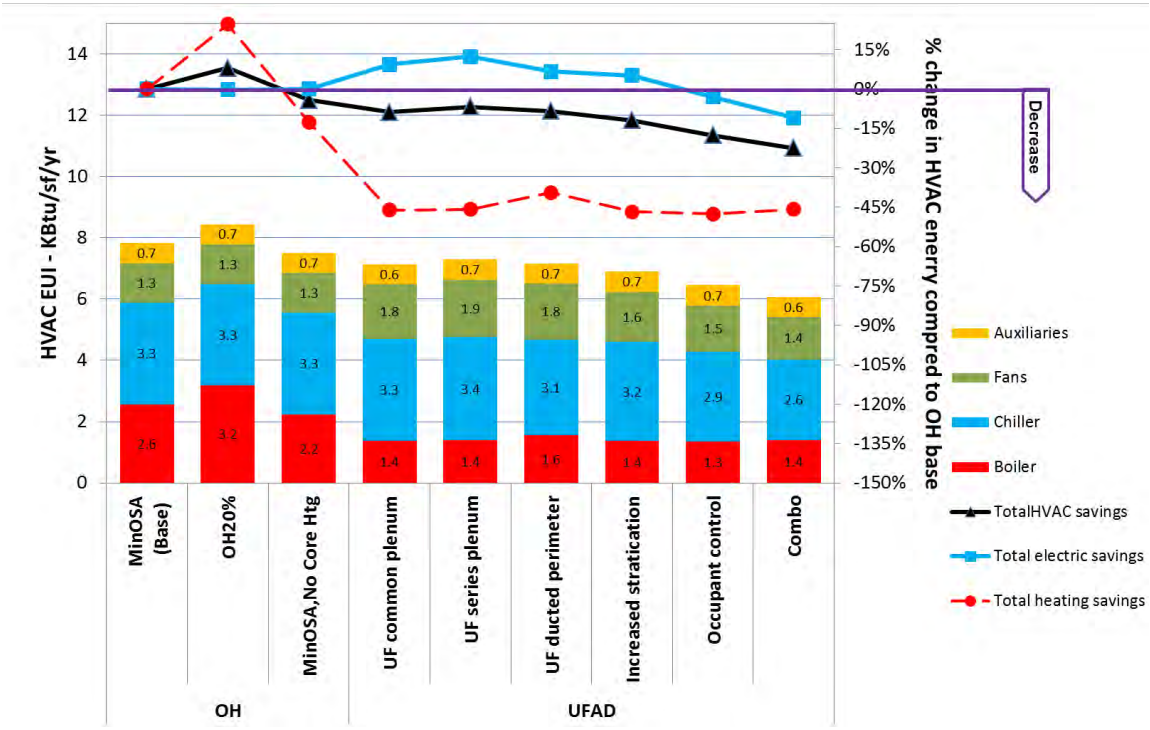

Figure 3: Energy performance comparison, Site HVAC EUI, Sacramento

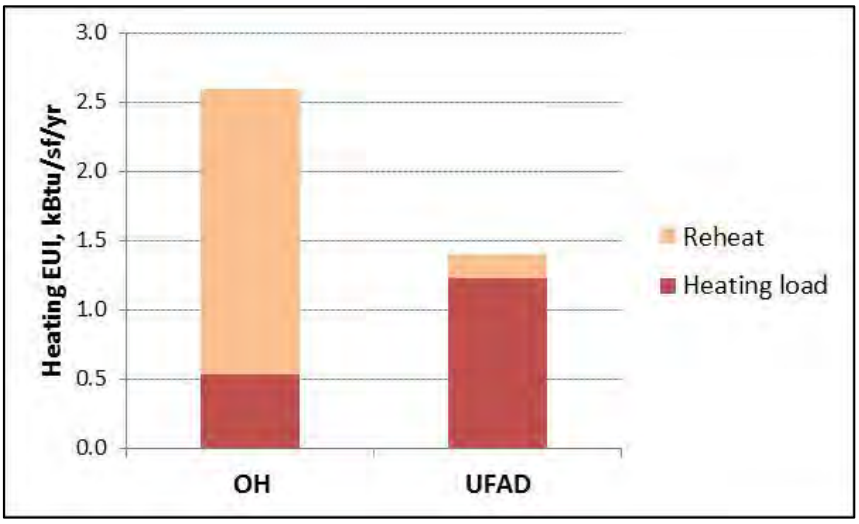

Figure 4: Example heating breakdown

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

INFLUENCE OF SUPPLY AIR TEMPERATURE ON UNDERFLOOR AIR DISTRIBUTION (UFAD) SYSTEM ENERGY PERFORMANCE

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ABSTRACT

Underfloor air distribution (UFAD) systems have received attention in recent years due to a number of potential advantages over conventional overhead (OH) systems. These potential advantages include increased layout flexibility, improved indoor air quality and thermal comfort, and energy savings. In particular, energy performance advantages have been difficult to evaluate analytically due to the lack of simulation tools that accurately model the complex heat transfer processes of stratification in the room and "thermal decay" (supply air temperature gain) in the underfloor plenum. Furthermore, the impact of key design and operating parameters cannot be easily determined without such tools. Fortunately, EnergyPlus v3.1 and beyond now contain validated calculation modules suitable to model these UFAD systems (U.S. DOE 2010).

Elevated supply air temperature is one of the distinguishing features of UFAD systems as compared to conventional overhead (OH) systems. In this paper EnergyPlus v3.1 simulations have been used to take a detailed look at the impact of variations in the air handling unit (AHU) supply air temperature (SAT) on the performance of one UFAD system commonly used in U.S. office buildings. The results indicate that raising design AHU SAT produces net savings in HVAC electricity consumption, even though cooling energy reductions trade off against fan energy increases; but the magnitude is climate dependent. However, heating energy (gas consumption) increases with increasing SAT, which tends to counterbalance decreases in electricity consumption. The paper includes a discussion of these somewhat counterintuitive findings.

INTRODUCTION

Underfloor air distribution (UFAD) is an innovative method of providing space conditioning to offices and other commercial buildings. Under cooling operation, properly controlled UFAD systems produce temperature stratification. UFAD systems supply cool air to the room from an underfloor plenum through vents, known as floor diffusers, located in a raised floor. In the most common application, cool air is forced upwards through supply diffusers by the pressure difference between the pressurized plenum and the space. In some cases, the airflow is fan-driven.

Previous research describes the UFAD modeling algorithms installed in EnergyPlus (Webster et al. 2008, Bauman et al. 2007a that make this study possible.

The potential energy savings of UFAD compared to conventional overhead systems are predominantly associated with two factors:

1. *Fan energy reductions*. Fan discharge pressures for UFAD systems can be lower than for OH due to reduced ductwork and thus reduced pressure drops in the supply distribution system. Differences in required airflow rate will also influence fan energy use. UFAD airflow rates are dependent on a variety of factors and design conditions. The amount of airflow required to satisfy the room cooling load will tend to decrease with lower SAT because of increasing heat transfer from the room to the underfloor supply plenum (thereby, reducing room cooling load). However, common practice is to use higher air handler SATs (typically 3-4°C (5-7°F) greater than OH) which tends to increase airflow by increasing the diffuser discharge temperatures. The increased diffuser temperature results from the dual effects of higher SAT and temperature gain (known as thermal decay, see below) in the supply plenum. The combined effect of all these factors produces UFAD cooling airflow rates that can be lower or higher than OH systems (Bauman et al. 2007b). For the

particular system type used for this study, perimeter fan coil unit (FCU) energy also adds to the air handling energy consumption.

2. *Cooling savings*. With the higher SATs commonly used for UFAD systems, there are increased opportunities for 'free cooling' savings from airside economizers in suitable climates.

Dynamic hourly models that can respond to changing operating conditions are required so that the impact on annual energy use of all the above factors can be accurately determined.

Differences in energy and demand performance between conventional overhead and UFAD systems were previously reported using the new UFAD capabilities of EnergyPlus (Linden et al. 2009). The present study examines one issue illuminated in that work; the tradeoff between cooling and fan energy use for UFAD systems in relation to changes in AHU supply temperature.

METHODS

This study uses a comparative, parametric approach where all building and system elements are the same and only the design AHU supply air setpoint is changed.

Description of EnergyPlus modeling

EnergyPlus is a building energy simulation program developed and supported by the U.S. Department of Energy (DOE). Based on a combination of two predecessor programs, DOE-2 and BLAST, it has greater capabilities than either one. The following key features make EnergyPlus an ideal candidate for simulating UFAD systems.

- *Room air stratification* – In UFAD systems stratification is a key feature that affects performance and thus must be modeled to accurately simulate these systems. Detailed heat balance calculations use a two-layer model of the room that is a simplified representation of the temperature stratification produced by UFAD systems under cooling operation. A semi-empirical algorithm predicts the distribution of heat gains between the lower and upper layers of the room and augments the heat balance calculations for each.
- *Simultaneous simulation of zone, system and plant*. EnergyPlus performs the system and plant simulation, and the air and surface heat balances (including radiant exchange) simultaneously. This is essential for realistic energy modeling of non-mixed systems like UFAD.

UFAD building model characteristics

This and other studies is based on a whole-building prototype created for EnergyPlus v 3.1[1] and run with California weather files for climate zone 12 (CZ12RV2.epw), herein referred to as Sacramento, and climate zone 3 (CZ03RV.epw) referred to as San Francisco. A summary of the prototype building characteristics follows.

Building geometry. The prototype is a three-story rectangular medium-sized office building with an aspect ratio of 1.5. Each 1,852 m^2 (20,000 ft^2) floor plate consists of four perimeter zones, an interior zone and a service core[2], representing approximately 40, 45 and 15 percent of the floor area, respectively. "Ribbon" windows with a 38% window-to-wall ratio (WWR) are used; other ratios can be simulated by changing the height of the window. Window properties and wall thermal characteristics adhere to ASHRAE 90.1-2004 performance rating specifications contained in 90.1 Appendix G. However, since real windows rarely match properties listed in the standard, the prototype contains specifications for real glazing systems with properties that are as close as possible to, but no worse than, the values listed in the standard. Key features of the building design can be found in Appendix A of Linden et al. (2009).

Supply plenum modeling. The prototype supports modeling of supply plenum effects by including supply plenum thermal zones below each room zone. Room zones receive cool supply air through these plenums. EnergyPlus performs a full heat balance on the underfloor plenum accounting for heat gain into the plenum from both the room above and return plenum from the floor below. This arrangement ensures accurate modeling of thermal decay (temperature increase in supply air as it passes through the plenum resulting from heat transfer to the plenum). The total floor-to-floor height and the occupied zone floor-to ceiling height (13 ft and 9 ft,

[1] For this study, a special development version of EnergyPlus v3.1 was used that incorporated a number of enhancements primarily to UFAD objects.

[2] The service core was not modeled in this project; no loads or schedules were applied. Occupancy, lighting, and equipment intensities need to be adjusted appropriately when comparing to other work where service cores are not included.

respectively) are equivalent to buildings with a conventional overhead VAV system but with return plenum height adjustments made to accommodate the supply plenums. For the bottom floor, ground temperature schedules were derived from the EnergyPlus slab simulator utility. For the supply plenum, airflow dependent convective heat transfer coefficients derived from work done by Bauman et al. (2007a) are used.

Internal Loads and Schedules. Table 1 lists internal loads assumed (based on total floor plate area). These values are similar to those found in the literature (NREL 2008) for other simulation studies.

Table 1. Internal load characteristics

Overhead lighting, W/m^2 (W/ft^2)	10.8 (1.0)
People sensible, W/m^2 (W/ft^2)	4.0 (0.37)
Peak occupancy, m^2/person (ft^2/person)	18.6 (201)
Equipment W/m^2 (W/ft^2)	8.1 (0.75)
Total, W/m^2 (W/ft^2)	22.9 (2.1)

Schedules for people, internal loads, and outside air ventilation are 8am to 5pm weekdays, and 8am to 12pm on weekends. The HVAC system operates from 7am to 10pm, and thermostat occupied setpoint schedules operate from 6am to 7pm. The HVAC system is off at night. Thermostats are set at 21.1°C/15.6°C (70°F/60°F) heating, 23.9°C/29.4°C (75°F/85°F) cooling for occupied/unoccupied hours.

HVAC Systems and Plant. Table 2 shows details of system and plant inputs. At the zone level, the UFAD system consists of swirl diffusers in interior zones and linear bar grille diffusers in the perimeter zones served by a variable speed series fan coil unit. The FCU shuts off when zone temperatures are in the heating-cooling dead band, but airflow of 6% of design volume is assumed to leak through the FCU when it is off. Consistent with currently available products, when the FCU is on, it starts at a minimum speed, which provides airflow of 12%. A conventional VAV reheat system serves the service core although no loads are assigned to this zone during this study.

The zones are served by a single variable speed central station air-handling unit including an economizer, chilled water cooling coil, and supply fan. The fan is a high efficiency airfoil where part-load operation is represented by a simulated static pressure reset strategy. Design static pressure requirements reflect an open plenum design where minimal ductwork is used. All simulations used the same design static pressure, which is assumed about 25% lower than OH primarily due to the elimination of distribution ductwork. (See Table 2.) The model simulates supply plenums in series, meaning that air is first delivered to the interior zone plenum and then to the perimeter zones. No central heating coil at the AHU is included to maintain comfort in interior zones, since it is not common practice in California climate types to do so. However, there are some comfort ramifications as discussed below.

Table 2. Summary of HVAC system configuration

HVAC	
AHU supply temp	13.9, 15.6, 17.2°C (57, 60, 63°F)
AHU fan design static press	775 Pa (3.1 iwc)
AHU fan design efficiency	63%
AHU part load shutoff[1]	125 Pa (0.5 iwc)
Relief fan design static	150 Pa (0.6 iwc)
Relief fan design efficiency	37%
Outside air rate	0.76 l/(s m^2) (0.15 cfm/ft^2)
System cycles at night	No
Zone Min airflow, % max	Opt[2]
Interior zone reheat	No
FCU design static pressure	125 Pa (0.5 iwc)
FCU design efficiency	15%
Plant	
Chiller (screw) design COP	5.26
Boiler design efficiency	80%
[1]Designates low shutoff pressure associated with static pressure reset [2]Opt = Optimized for 0.76 l/(s m^2) (0.15 cfm/sf) (CA Title-24)	

The central plant consists of dual central screw chillers with variable speed pumps and cooling tower using chilled water reset based on outside air temperature. Two gas fired, forced draft hot water boilers provide heating to all heating coils. The boiler simulation includes a special boiler curve that more accurately simulates turndown at low part load conditions. Hot water supply temperature is reset based on outside air temperature.

Sizing issues

For each parametric run, system components are re-sized. However, during this and other studies conducted by the authors it became obvious that the autosizing procedures used in EnergyPlus are not

always appropriate or are not even available for UFAD system components (e.g., determining the number of diffusers). In EnergyPlus, the user should explicitly specify the VAV box cooling design supply air temperature (SAT) for the sizing of each VAV box. With that information, EnergyPlus sizes each VAV box based on the peak cooling load determined from the design day calculation. In contrast to a conventional OH system, the actual diffuser discharge air temperature in a UFAD system is significantly different from the central air handler SAT due to the heat gain of the conditioned air in the underfloor plenum. The user needs to take this into account and specify the correct VAV box design SAT under each condition. Several methods to more accurately size system components were developed. For example, to support the use of low minimum ventilation rates, a special routine allows simulation of minimum ventilation airflows that UFAD systems can provide (as opposed to higher minimums used in OH VAV to prevent dumping during cooling and short-circuiting in heating). Design day runs were made first to determine zone design air volumes. Then these results plus minimum ventilation requirements were used to specify the minimum fraction for each zone, since the current EnergyPlus version only allows the user to specify the minimum fraction of each VAV box, not the minimum airflow. This technique ensures that all simulations have consistent ventilation rates. A calculation of the design number of UFAD diffusers was the focus of another method where design day results plus diffuser design volumes determines the number of diffusers used in the annual simulation. This ensures accurate simulation of stratification. In addition to determining the zone design airflow volumes, other HVAC components such as chillers and boilers were also sized based on the design day run results instead of relying on the EnergyPlus autosizing function.

Sensitivity of energy performance

In comparison studies, it is important to understand how representative the results are for other design and operating conditions. In the Linden study, the impact of building design parameters on the comparison of HVAC system energy use was assessed by a set of sensitivity runs. Figure 1 shows (from Linden et al. 2009) there is little impact on HVAC system performance *differences* (e.g., in this case the difference between OH and UFAD) due to window-to-wall ratio and internal load levels. These figures show results for Sacramento only, but are representative of results in other California climates.

RESULTS AND DISCUSSION

Cooling tradeoff analysis

In this study, the SAT leaving the air handler was fixed throughout annual operation but was varied parametrically from 13.9°C (57°F)[3], representative of OH system setpoints, to 17.2°C (63°F), a commonly employed setpoint for UFAD systems. All other parameters were unchanged. Figure 2 shows HVAC site energy consumption results in terms of the energy use intensity (EUI) for the three cases in two California climate zones and Figure 3 shows the gas/electric breakdown.

As shown in Figure 2, and contrary to expectations of the researchers, best design SAT performance appears to be climate dependent; 15.6°C (60°F) in Sacramento but 17.2°C (63°F) for San Francisco. From Figure 3 it is clear that there is a net electric savings as SAT is increased but more so for San Francisco than Sacramento; savings for both appear to level off as SAT is increased. It is also clear that this is due to a tradeoff between fan energy and cooling energy; increased SAT tends to lower cooling energy due to more economizer hours but fan energy is increased due to the higher cooling airflows required.

As Table 3 shows, the net tradeoff (sum of cooling, fan, and auxiliary energy) varies with climate. In warm climates like Sacramento, cooling and fan energy changes tend to offset each other as SAT is increased so there is little difference between SAT 15.6°C (60°F) and 17.2°C (63°F). In a mild, "good economizer" climate like San Francisco there are noticeable savings at SAT 17.2°C (63°F). However, as shown by Figure 3, heating energy increases with increasing SAT. This increase counterbalances the net electric savings producing the counterintuitive results shown yielding, for example, an optimum SAT for Sacramento of 15.6°C (60°F).

Heating energy analysis

The following describes how the heating system operation leads to higher heating energy use as SAT increases. Two major processes are occurring concurrently to produce the breakdowns shown in Figure 3:

[3] A SAT of 13.9°C (57°F) was used in an attempt to provide some consideration for the impact of OH system duct heat gain on airflow requirements. To keep consistent economizer performance, the same value was used for the UFAD base case.

- Zone heating loads decrease as SAT increases. This is due to increased concrete slab and raised floor surface temperatures caused by higher temperatures of the air passing through the underfloor plenum at higher SATs. However, as shown, this effect is relatively small.
- As shown in Figure 3, the reheat portion of the FCU heating energy increases as SAT increases (i.e., reheat is calculated from the FCU supply volume and the temperature difference between the room and supply air temperature entering the FCU). This occurs primarily because the minimum volume of the FCU (i.e.,12% of maximum design volume) is greater due to the larger size FCUs required to accommodate the higher cooling airflows at higher SATs.

  The reheat increases despite the fact that the temperature entering the FCU is higher as SAT is increased; the higher minimum volume dominates this tradeoff. The entering temperature is higher with increased SAT even though the thermal decay actually decreases; the temperature rise through the plenum is less as airflow increases with higher SAT, but not enough to lower the FCU entering temperature.

Comfort considerations

One method to assess relative comfort is to compare zone operative temperatures (average of zone dry bulb and mean radiant temperature) during occupied hours for different simulations. For UFAD, the temperature in the lower occupied region of the zone is used for the dry bulb component. Operative temperature is a necessary but not sufficient metric to measure occupant comfort completely since other parameters such as clothing level, metabolic rate, air velocity and humidity are not included. It also does not account for other factors such as any benefit assigned for occupant personal control, which may be applicable to UFAD systems in particular. However, for this study, it at least provides an indicator when applied to the zone where problems are most likely to arise; interior zones when operating near heating conditions (recall that interior zones have no terminal heating equipment).

Figure 4 shows operative temperature histograms for each SAT in each of the two climates. The curves show cumulative results on the right hand axis. If we assume that heating conditions occur at operative temperatures below about 21°C (equal to the heating dry bulb set point), lower SATs appear to affect comfort very little in San Francisco. However, in Sacramento they have more of an effect; 24% below 21°C for SAT =13.9°C (57°F), and 13% for SAT= 15.6°C (60°F), the optimum energy operating point. With this knowledge, designers can make a judgment about whether to install a central heating coil in the AHU to mitigate low interior zone temperatures or possibly rely on occupants controlling their diffuser to manage their comfort instead. Other work by the authors indicates that these coils can have a significant energy impact. In San Francisco, it appears that there is little risk of overcooling interior zone occupants, which confirms common practice in this area.

Further studies

Further studies may reveal variations on these results due to factors not studied such as the following:

- other system types and configurations (e.g., alternative plenum distribution strategies),
- improved performance from more or alternative diffusers types,
- different control strategies (e.g., SAT reset strategies; wider thermostat deadbands that allow credit for personal control),
- methods for, and impact of, improved ventilation effectiveness,
- operation in other more severe climates and in larger buildings.

The authors anticipate conducting studies such as these in the future.

CONCLUSION

The purpose of this study was to determine the impact of changes in AHU supply air temperature on UFAD system energy performance and to better understand how cooling, heating and fan energy "tradeoff" as SAT is increased.

Although limited to two California climate zones, the results indicate that there are net decreases in HVAC electricity (cooling, fans, and auxiliaries) consumption as SAT (using fixed setpoints) is increased, but the magnitude is climate dependent. In general, diminishing returns occur as SAT is increased since fan energy increases tend to balance cooling energy decreases (due to more economizer hours available at higher SATs). However, heating energy increases as SAT is increased. This results in varying optimum design SATs depending on climate.

The results also indicate that there may be interior zone comfort consequences to operating these systems without a central AHU heating coil, again

depending on climate. The optimum operating SAT is likely to be affected for those cases (e.g., cold climates) where a heating coil is required. It appears that designers need to consider this energy vs. comfort tradeoff carefully in cool and cold climates.

ACKNOWLEDGMENTS

This work was supported by the California Energy Commission (CEC) Public Interest Energy Research (PIER) Buildings Program under Contract 500-08-044 and the Center for the Built Environment (CBE) at the University of California, Berkeley.

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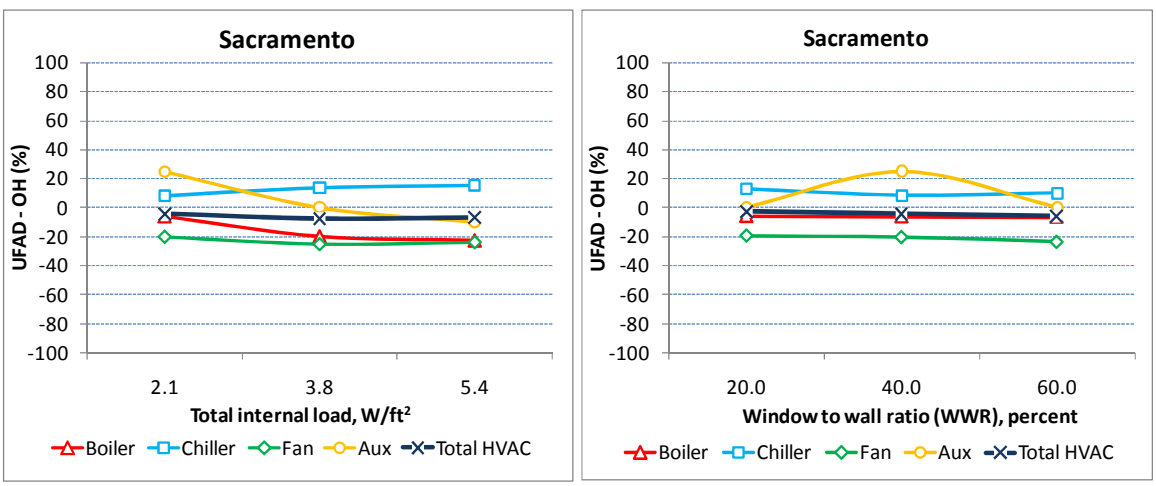

Figure 1. Example of sensitivity analysis for internal load and window-to-wall ratio for overhead to UFAD comparison (from Linden et al.)

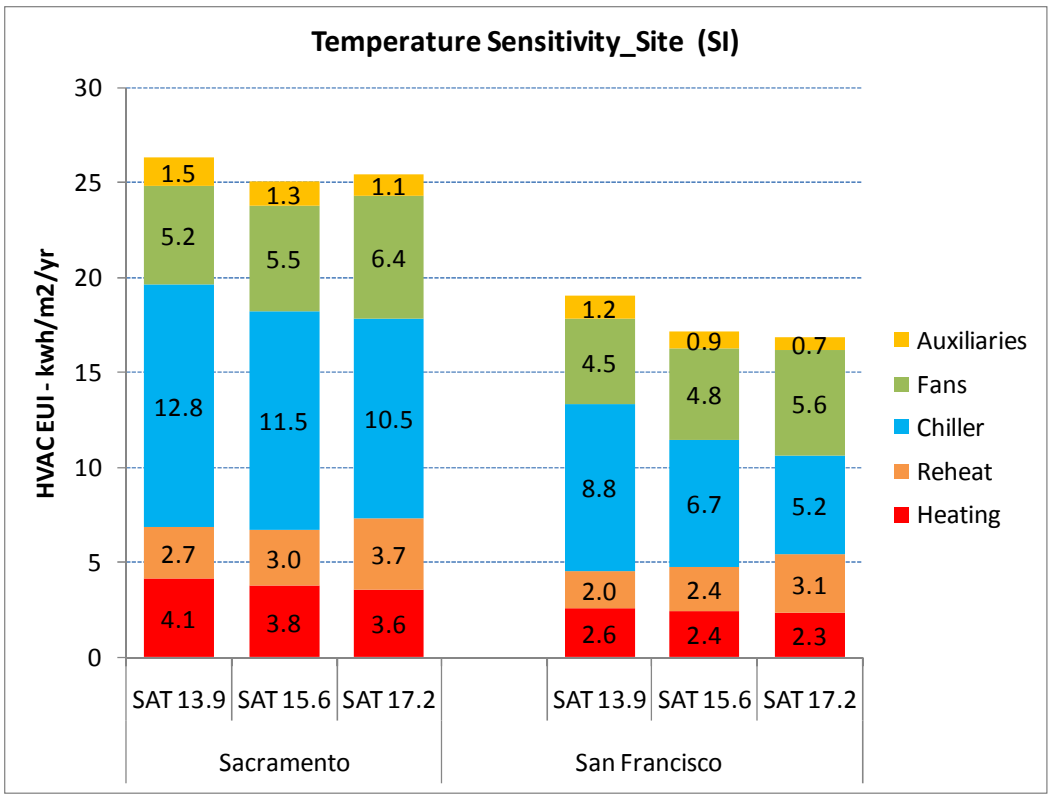

Figure 2. Impact of AHU supply air temperature in two climates

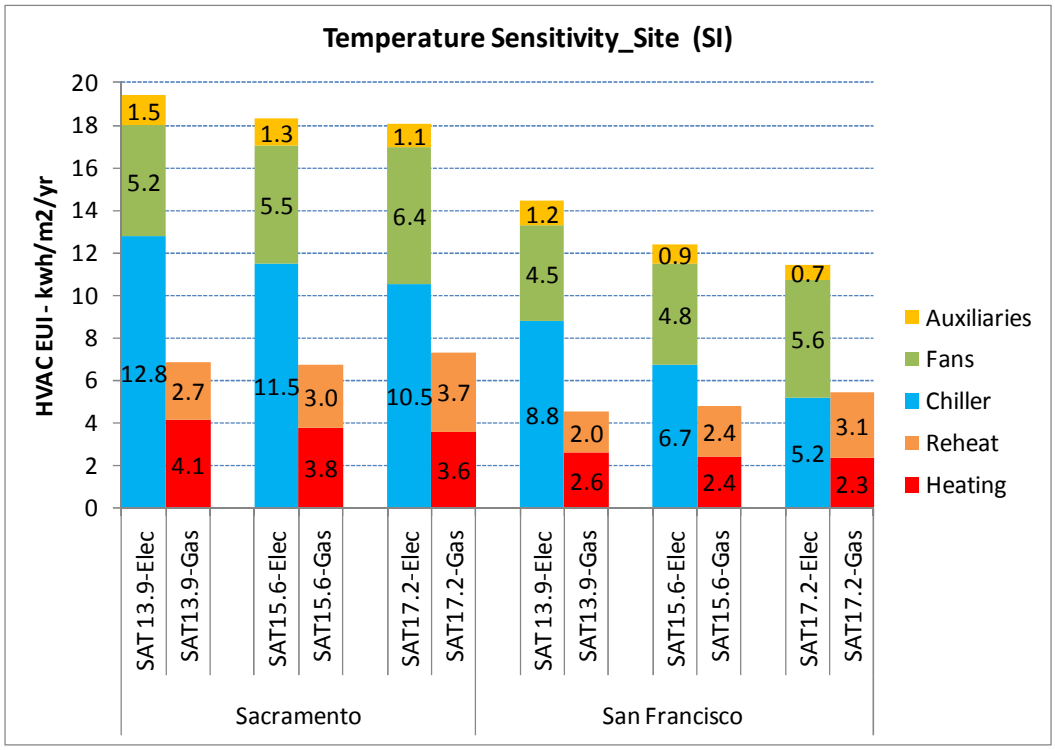

Figure 3. Gas/electric breakdown for variation in AHU supply temperature in two climates

Table 3. HVAC electric and gas breakdown (difference from result for SAT = 13.9°C)

kWH/m2-yr	Sacramento			San Francisco		
SAT	**13.9°C**	**15.6°C**	**17.2°C**	**13.9°C**	**15.6°C**	**17.2°C**
HVAC Gas EUI	6.9	6.7 (-0.1)	7.3 (+0.5)	4.6	4.8 (+0.2)	5.4 (+0.8)
HVAC Elec EUI	19.4	18.3 (-1.1)	18.1 (-1.3)	14.5	12.4 (-2.1)	11.4 (-3.0)
Total	26.3	25.0 (-1.3)	25.4(-0.9)	19.0	17.2 (-1.8)	16.9 (-2.2)
Change, %	--	-4.9%	-3.4%	--	-9.5%	-11.1%

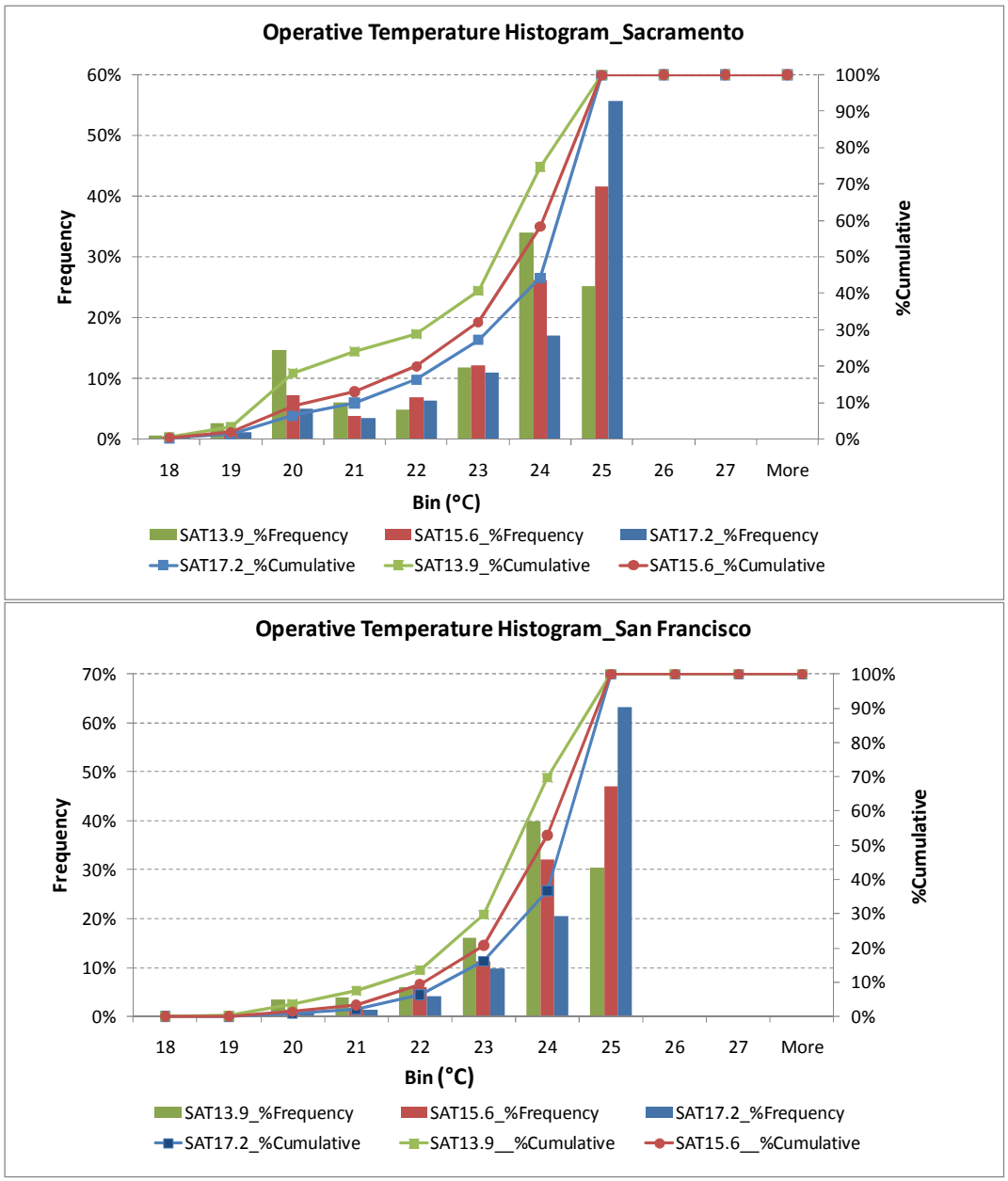

Figure 4: Comfort histograms

Internal report March 2013

Appendix 3.2.4

The New York Times Building End-use Energy Monitoring and Detailed EnergyPlus Modeling

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April 1, 2013

INTRODUCTION

This POE study at the New York Times (NYT) Headquarters was sponsored by U.S. DOE's Commercial Buildings Partnerships (CBP) Program, which encourages building owners and operators to collaborate with research staff at national laboratories and universities to explore energy-saving ideas and strategies in retrofit and new construction projects. It is hoped that some of the innovations used in the NYT Building can become a model and prototype for larger scale implementation and replication in new and existing buildings in New York and nationally. The primary purpose of the POE project was to conduct a detailed post-occupancy evaluation of The New York Times (NYT) Headquarters, a 52-story high performance office building in downtown Manhattan. The NYT Building has attracted interest nationally due to its design incorporating innovative technologies, including advanced external shading, automatic internal shading control, dimmable electric lighting, and underfloor air distribution (UFAD).

Researchers from the Center for the Built Environment (CBE) at UC Berkeley and Lawrence Berkeley National Laboratory (LBNL) collaborated on the study; CBE focused on evaluating the performance of the UFAD system, while LBNL studied the shading and lighting performance. Energy and comfort performance was carefully measured and documented for the building. The results have primary applicability to office buildings but can be useful in part to all commercial buildings. Results are presented in the final report by E. Lee [E. Lee 2012].

Alongside the measurement activities, the team developed a detailed EnergyPlus model for one tower floor, with source-code modifications made by the LBNL team to EnergyPlus that allowed a detailed Radiance model (used by LBNL separately for daylighting analysis) to be used to calculate the solar gains throughout the space at each simulation timestep. The model was then configured with detailed schedules for the occupancy, plug loads, lighting energy, etc., based on monitored

data – both that described in the CBP final report and the more detailed HVAC measurements described herein. The resulting model thus provides a very detailed description of some aspects of the energy behavior of that one tower floor. The model remains, however, a poor predictor of energy consumption. This report describes the details of the HVAC measurements and the EnergyPlus model, and discusses further steps required to increase the model's accuracy in terms of energy consumption.

DESCRIPTION OF THE NEW YORK TIMES BUILDING

The New York Times Building (Figure 1) was completed and occupied in 2007. The 52-story, 1.5 Mft^2 office tower, at 1,046 ft high, is tied with the Chrysler Building as the 4th highest in New York City. Architect Renzo Piano employed a unique double-skinned façade, featuring clear floor-to-ceiling glazing to provide views and transparency in combination with a second external skin made up of horizontal ceramic rods that serves as a sunshade. The building also incorporates advanced automatic internal shading and dimmable electric lighting system developed in collaboration with Lawrence Berkeley National Laboratory. The internal shades operate to further control solar loads while allowing natural daylight to replace the use of electric lighting as much as possible. The New York Times Company occupies floors 2-27, the first floor is retail and entrance lobby, and the upper floors are occupied by other tenants. Underfloor air distribution (UFAD) was installed on all floors occupied by the Times. As discussed below, CBE researchers installed their detailed wireless monitoring system on the 20th floor, which was selected to represent a typical tower floor of the high-rise NYT Building.

Figure 1. The New York Times Building

ENERGYPLUS MODELING

A detailed EnergyPlus v6.0 model of the 20th floor of The New York Times Building was developed, using a design-phase EnergyPlus v1.2 model constructed by NaturalWorks (Linden, 2006) as a starting point, but making major changes throughout. (In the end, the envelope material

specifications were the only parts that remained unchanged from that original model.) Detailed model specifications are provided in Appendix A (for everything excluding solar gains) and Appendix B (describing the Radiance-EnergyPlus link for solar gains). Salient features are described below.

The model consists of eight perimeter zones, four interior zones, and a service core as shown in Figure 2. The zoning used in the EnergyPlus model is a simplification of the actual zoning layout of the building, shown for comparison in Figure 3a. Note that this zoning also differs from the measured lighting zoning (Figure 3b), and the roller shade control zoning (Figure 3c) – the annual EnergyPlus schedules used for these thus had to be derived from the measured data to attribute the correct amount to each energy model zone.

Underfloor air distribution is modeled with the UFAD modeling routines in EnergyPlus v6.0 (EnergyPlus 2010)using inputs supplied by CBE corresponding to equipment types used in the building. Sizing was based on the actual sizing of zone equipment rolled-up to the simulation zones (as shown in Appendix A).

Due to the complexities of the interior and exterior shading devices, a novel approach to modeling solar gains on interior room surfaces was devised. Instead of modeling the shades directly in EnergyPlus, a detailed Radiance [Radiance 2010] model was constructed by researchers at LBNL (Radiance 2010). The output of the Radiance simulation, which models the building envelope, urban shadowing, and exterior and interior shades, was used to generate hourly schedules of solar gains to each interior surface including window glazing layers, internal shades and furniture. Data from the shading system was used in the Radiance model to ensure accurate accounting of the shading operations. In order to apply these custom-calculated solar gains to interior surfaces in the thermal model, special modifications to the EnergyPlus source code were made. A field specifying a schedule for incident solar radiation was added to the input definition of the “detailed building surface” object. Internally to EnergyPlus, the appropriate schedule value overrides the incident solar gains that would normally be calculated. Similarly, solar gains schedules were also added to the “detailed fenestration” object definition. Gains schedules derived from Radiance hourly output are now applied to the two layers of glazing for the fenestration. In addition, the Radiance model is also used to estimate solar gains on furniture, which are being applied in the EnergyPlus model using the “other equipment gains” object. These methods ensure that furniture is treated as internal surfaces that shade the raised floor, which is potentially important because of the effects of solar gains on thermal decay in the UFAD plenum.

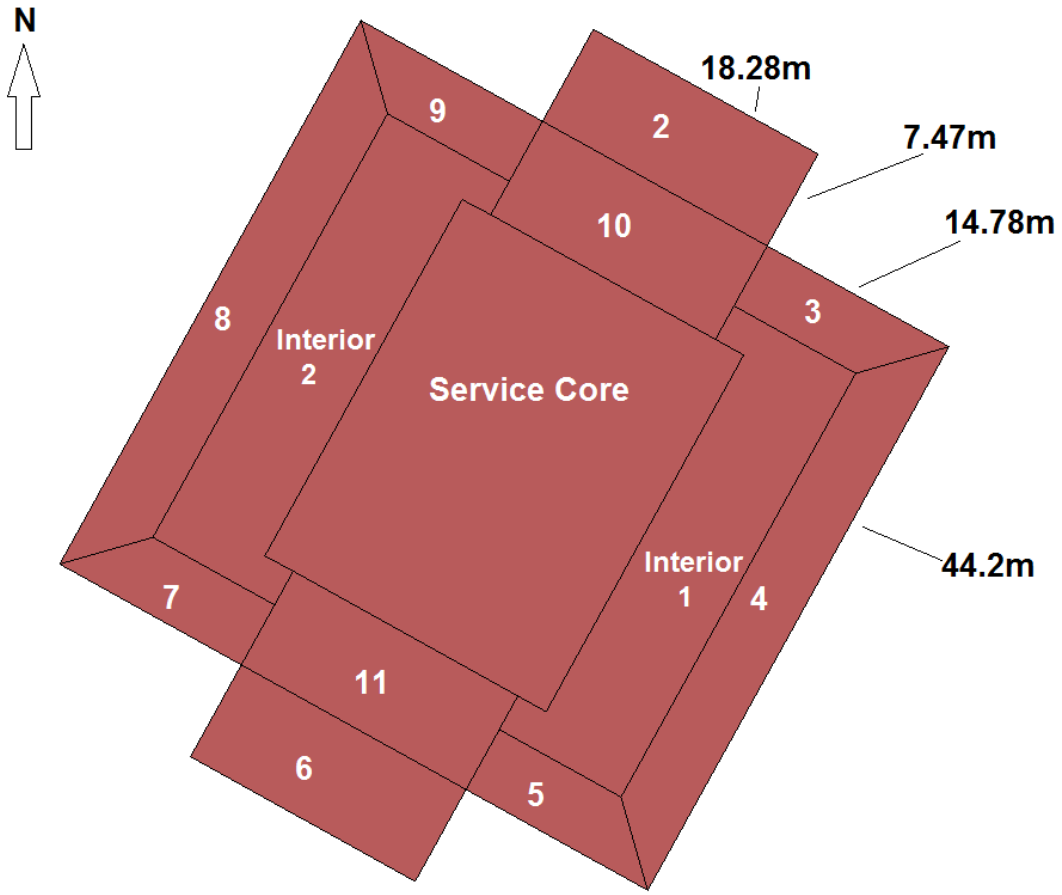

Figure 2. Simulation zoning configuration

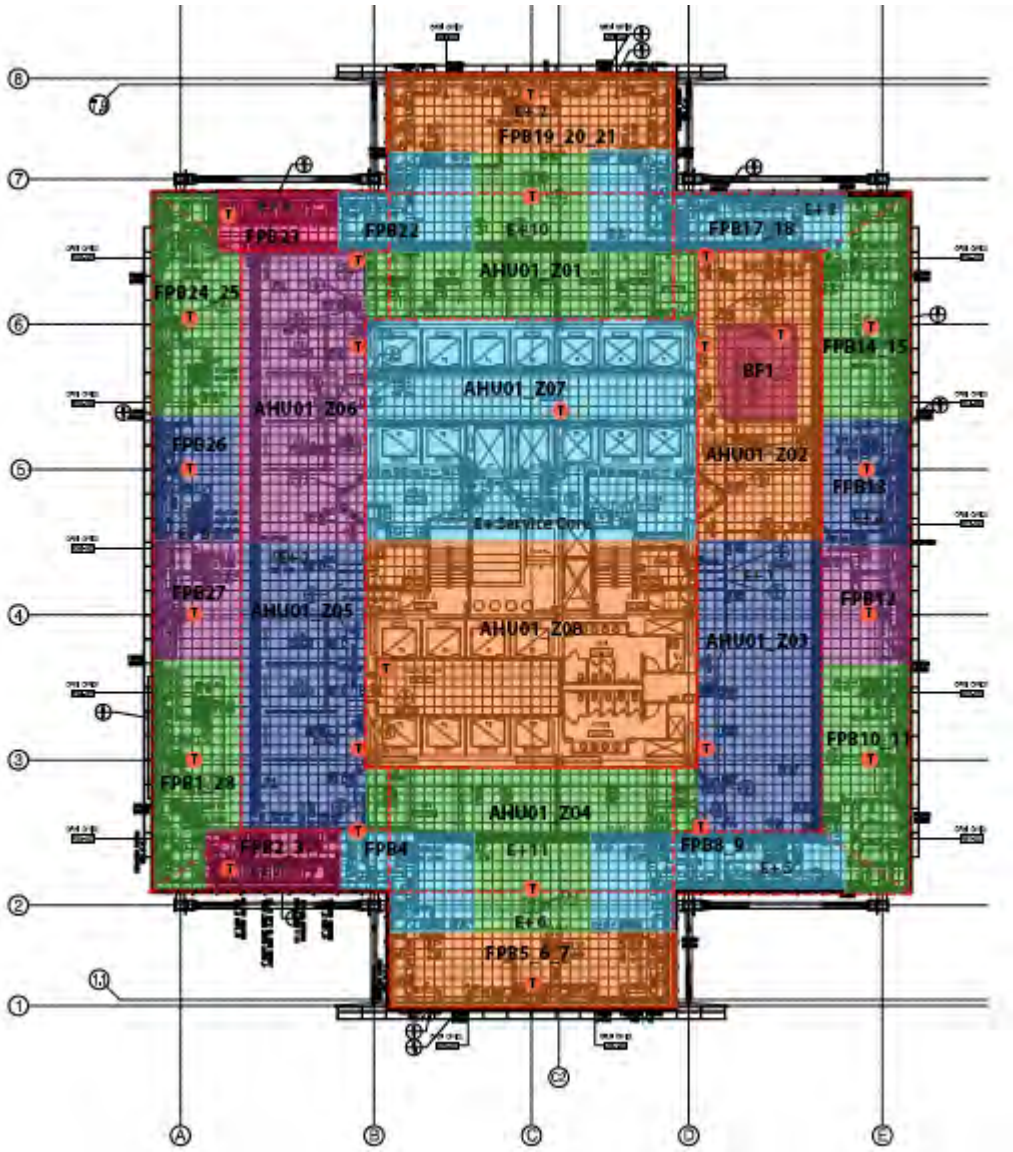

Figure 3a. Actual thermal zoning layout

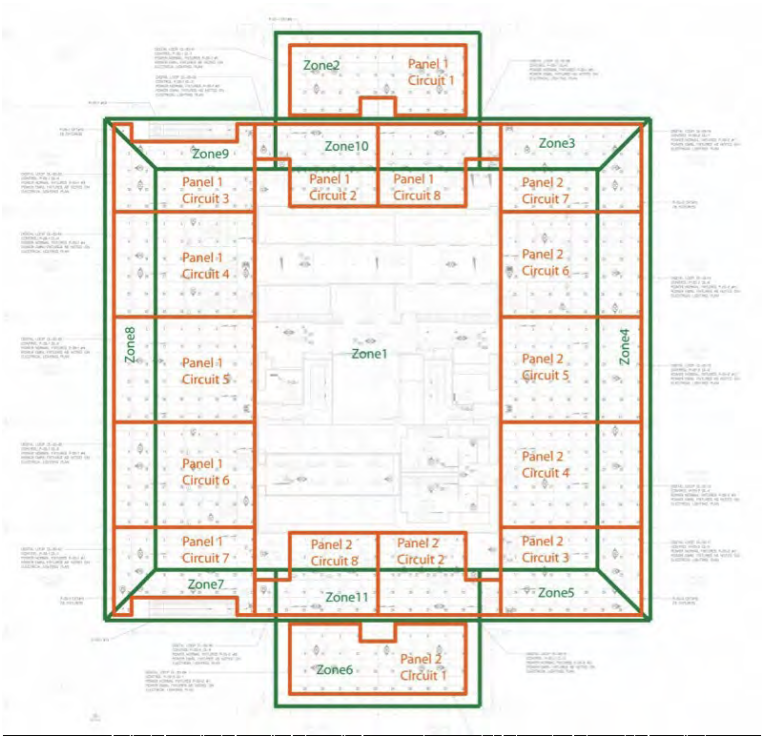

Figure 3b. Model thermal zones compared with the measured lighting zones

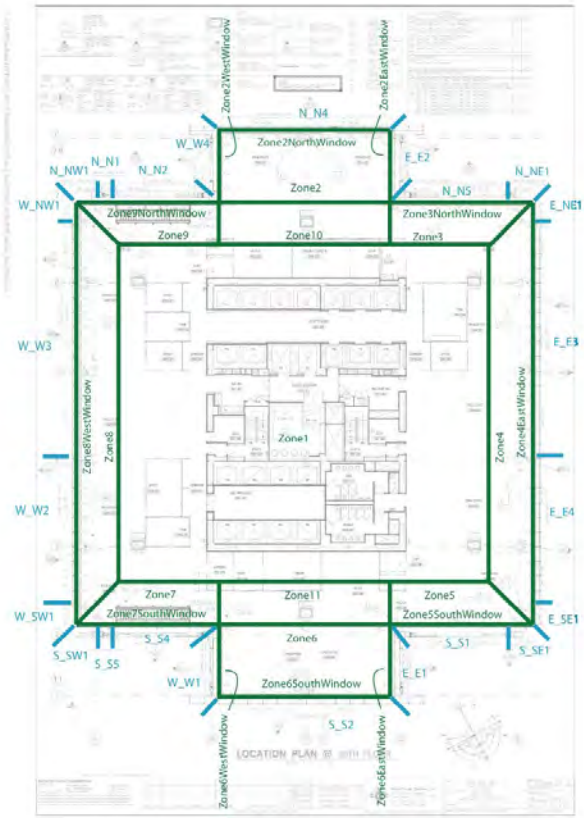

Figure 3c. Model thermal zones compared with the shading control zones

BUILDING PERFORMANCE DATA COLLECTION AND ANALYSIS

The goal of this work was to monitor and collect a variety of building performance data related to the UFAD system and to measure the end use energy consumption for a single typical floor of the building. Data describing the performance of the internal shades, dimmable lighting system, and prediction of solar energy gains using Radiance modeling was provided by LBNL researchers as discussed above. All of this data was used to configure schedules and some parameters in the EnergyPlus simulation model. The model's output was then to the measured end use data that was collected over a four month period in 2011.

DESCRIPTION OF FIELD MONITORING OF UFAD SYSTEM PERFORMANCE

During the first field visit (August 22-26, 2011), an array of wireless sensors was deployed to measure temperature, water flow, and power on the 20th floor of the NYT Building (see Figure 4). These wireless sensors continuously monitored conditions over the entire study period (August 25, 2011 – January 9, 2012) and transferred data to our data server in Berkeley in real time.

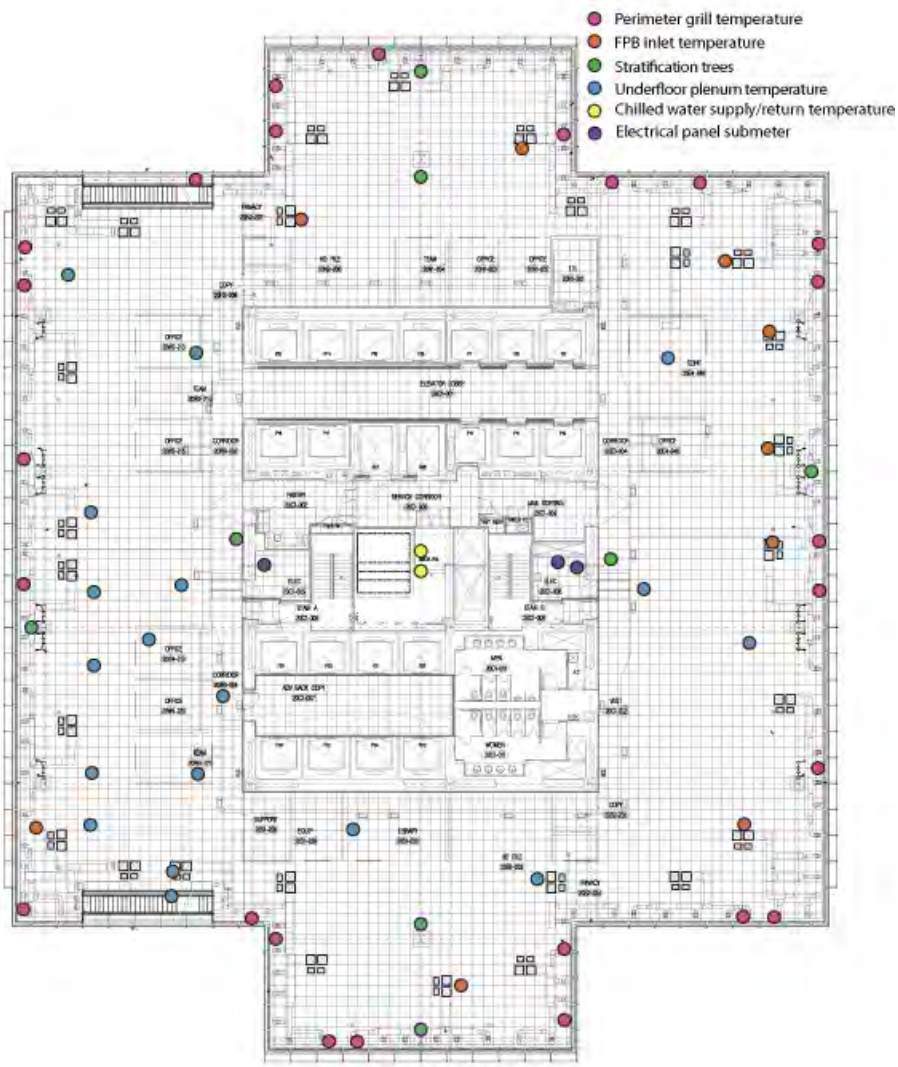

Figure 4. Map of wireless sensors deployed on 20th floor of NYT Building

Pictures of the monitoring setup are included in the next set of figures:

- Stratification trees – air temperature stratification at perimeter and interior locations of each building exposure (Figure 5)
- Stratification cart – air temperature stratification, floor/ceiling surface temperatures, and underfloor plenum pressure at various locations across the floorplate (Figure 6)
- Power meters – lighting, fan-powered-box, plug-loads, and whole floor consumption (Figure 7)
- Chilled water supply and return temperatures and water flow rate (Figure 8)
- Underfloor plenum temperature at selected locations across the floorplate (Figure 9)
- Fan-powered-box, FPB, flow calibration (Figure 10)

Figure 5. Stratification tree

Figure 6. Stratification cart

Figure 7. Power metering

Figure 8. Chilled water flow and temperature metering

Figure 9. Underfloor plenum temperature (fan-powered-box inlet temperature)

Figure 10. Fan-powered-box flow meter calibration

Figure 11 shows stratification profiles measured in the East interior zone (AHU_Z03) on 9/14/2011.

Figure 12 shows a comparison of these measured profiles (from portable cart) to the EnergyPlus simulations. Note that all the measured profiles are less than the setpoint (at 84 inches) indicating that the zone is overcooled; however the BMS data shows the thermostat readings (accounting for offsets) are very close to the 75°F setpoint. This is corroborated by the fact that the supply plenum static pressures were nearly at their 0.065 iwc limit for most of the day. This indicates that there is a considerable mismatch between the control points and the actual temperatures in the occupied portion of the zone, which may explain why there are some "too cool" complaints from occupants. In addition, the zone temperatures are "floating" instead of being centered on a control point. While the modeling shows some degree of correspondence with measured profiles (especially in the late afternoon) there are obvious differences that would have to be considered for full calibration. This results from the fact that the stratification modeling assumed the control point is in the lower layer (contrary to the actual case), with a setpoint of 71°F to replicate the measured average occupied zone temperature. [1]

Additional data was collected from the BMS and integrated with the wireless data; the combined set was used to create a measured end use breakdown and simulation model inputs as described in the following.

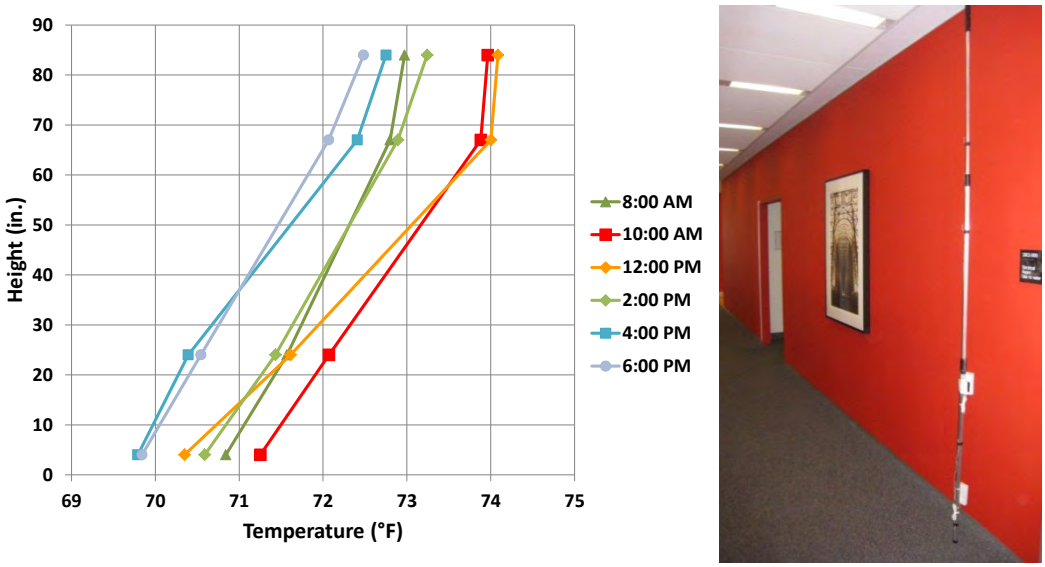

Figure 11a. East interior zone stratification pole and example hourly temperature profiles, 9/14/2011

[1] The 71°F setpoint in the lower occupied zone was based on a cursory review of actual measured average temperatures in the occupied zone on average for all interior zones.

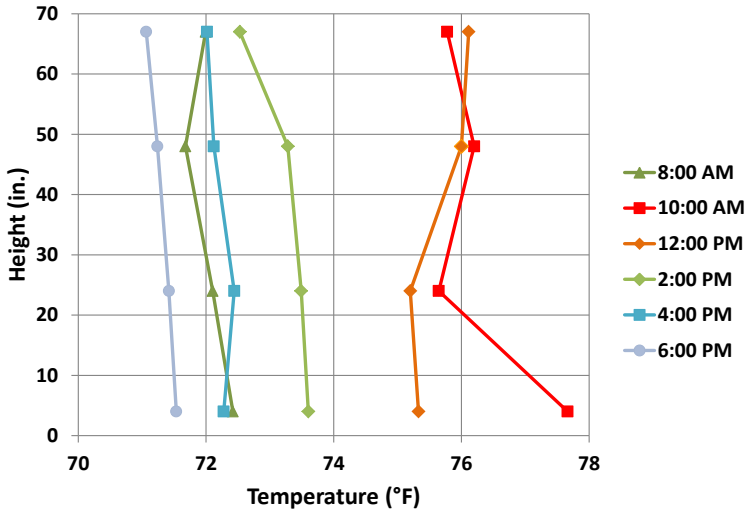

Figure 11b. East perimeter stratification profiles, 9/14/2011

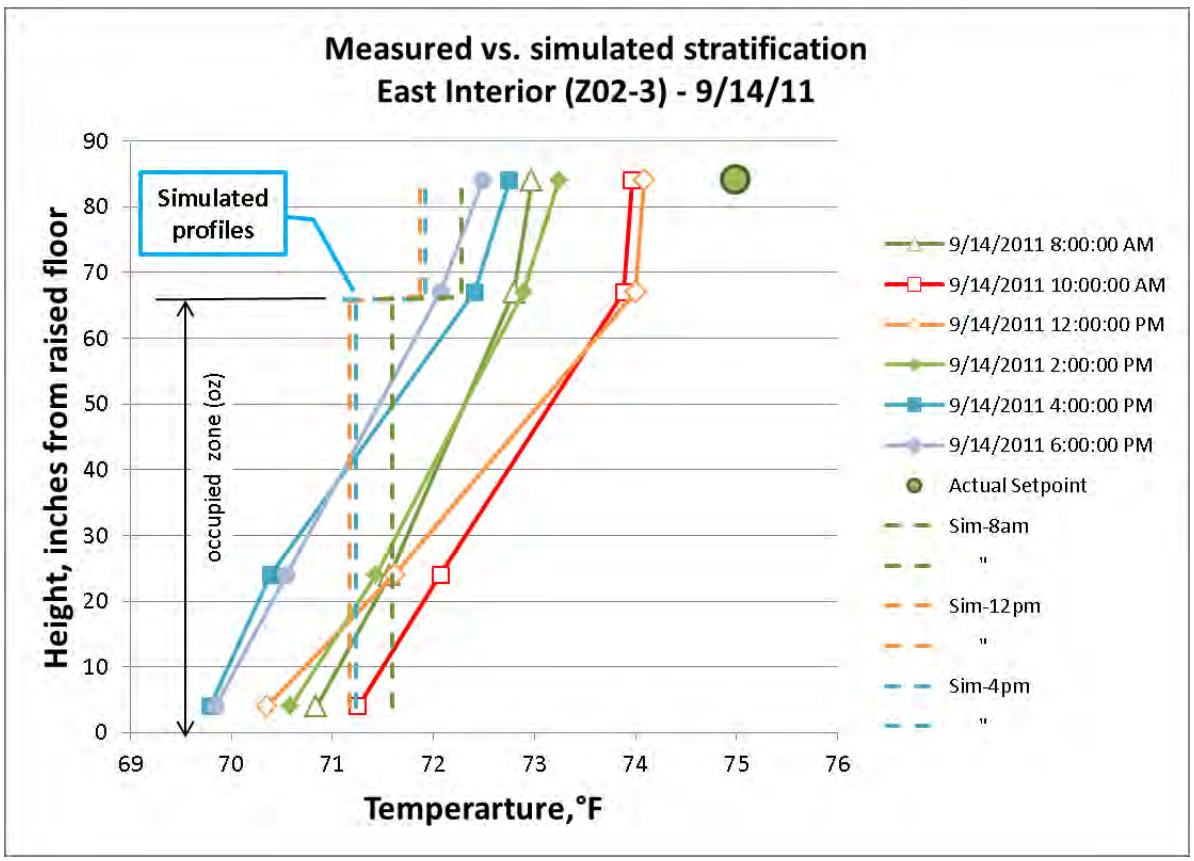

Figure 12. Measured vs. simulated stratification profiles for 9/14/2011

INTERNAL LOADS

Lighting and plug loads were monitored at the sub-panels providing a floor wide measurement. Weekday and weekend data from each month were used to make average daily profiles of lighting and plug loads for each month of measurements.

Lighting

Figures 13a and 13b present total floor measured lighting loads, which show remarkable similarity in the morning hours for weekdays, varying between ~0.3 to 0.4 W/ft^2. There are slight differences in summer vs winter behavior in the afternoons. Weekend loads indicate an irregular occupancy, but still substantially lower than for weekdays, as expected; ~ 0.06 W/ ft^2.

Simulation profiles derived from detailed, 15 minute lighting schedules based on the measurements from the open plan lighting system. These were augmented by additional lighting objects to represent estimated perimeter troffer lighting to perimeter zones, plus added interior lighting to account for service core lighting in the real building that is included in the floor level measurements but not in the open plan lighting system.

Occupancy loads for simulations were estimated from the as-built floor layout and input from operators.

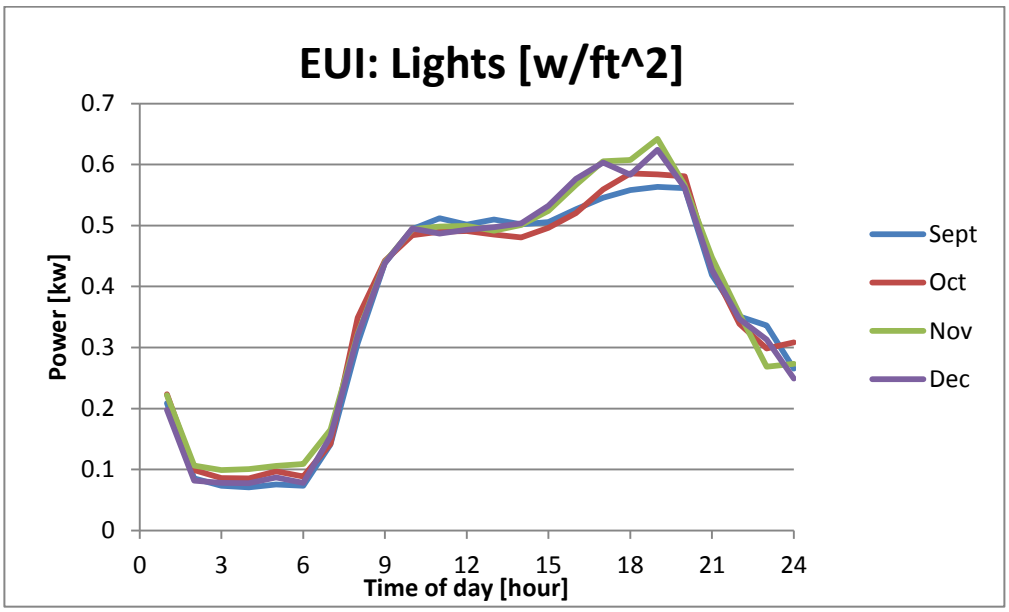

Figure 13a. Measured weekday lighting

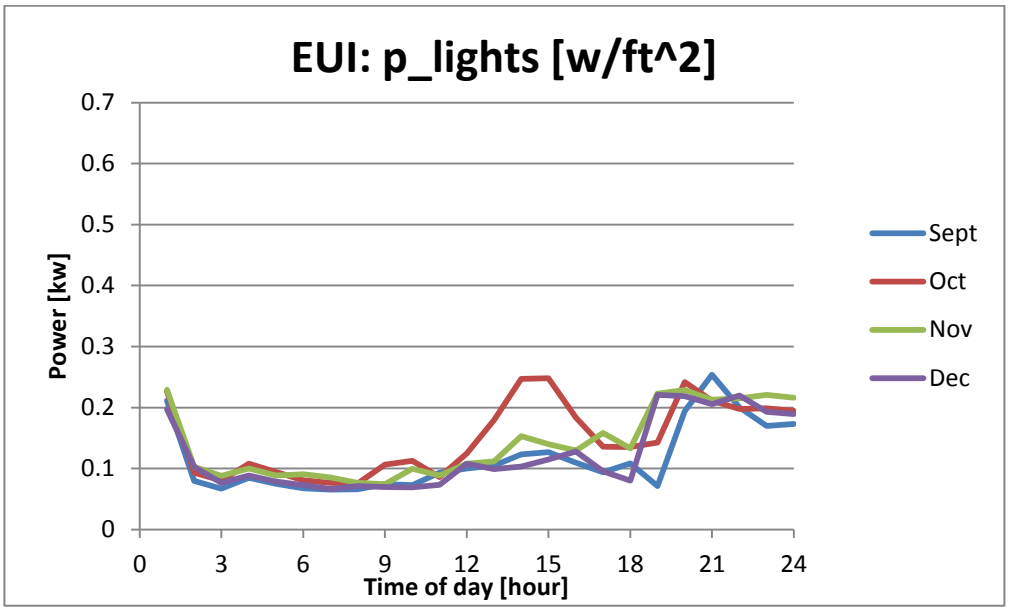

Figure 13b. Measured Saturday lighting

Plug loads

Plug load profiles shown in Figure 14 reveal a slight seasonal dependence with somewhat reduced loads in the winter; ~0.4 W/ ft^2 at peak, and 0.17 W/ ft^2 at nights and weekends. Equipment loads were modeled simply by using the standard equipment power density (EPD) model in Eplus, but using the fractions schedules from the measured data. We iterated on the EPD to match up the monthly EUI between measured and simulated.

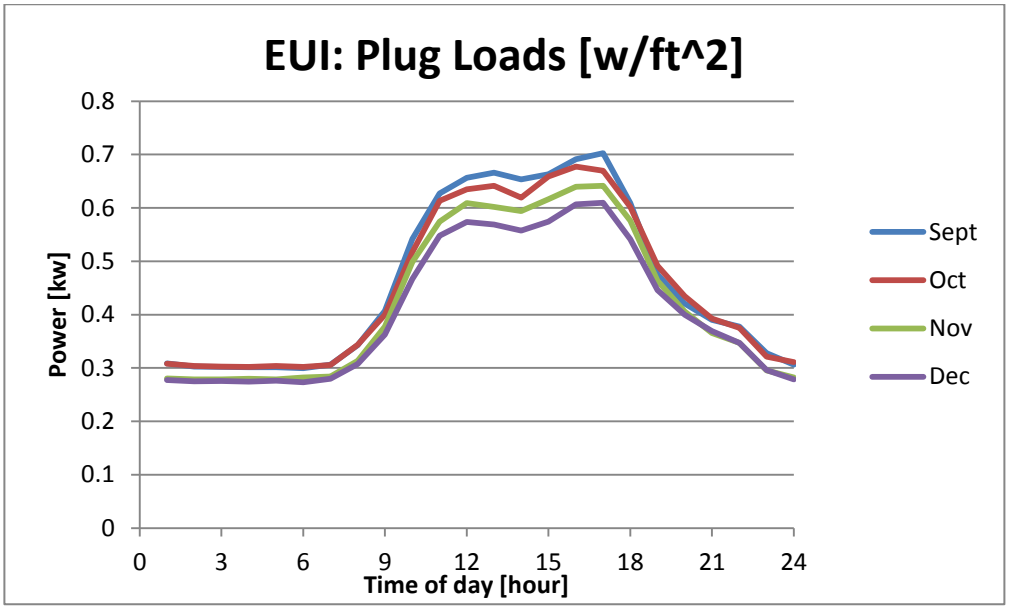

Figure 14a. Measured weekday plug loads

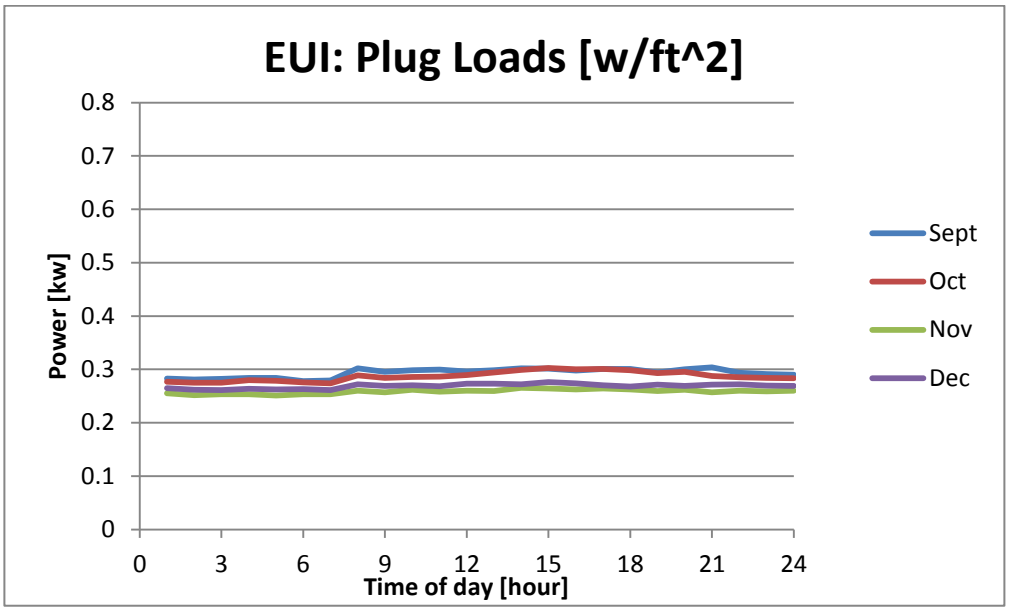

Figure 14b. Measured Saturday plug loads

HVAC SYSTEM MODELING

The following sections describe how the results of the monitoring were used to develop models for various HVAC components. A detailed summary of the modeling parameters appears in Appendix A.

AHU performance

Figures 15 shows load profiles for the dedicated air handler on the 20th floor where the weekend peak load is about 30% lower than weekday.

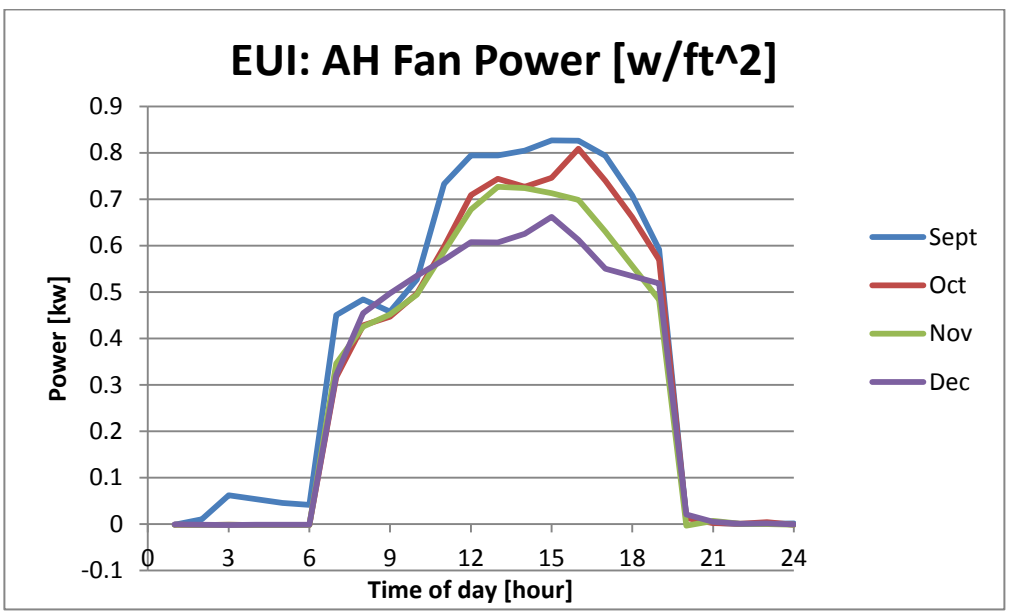

Figure 15a. Weekday AHU power

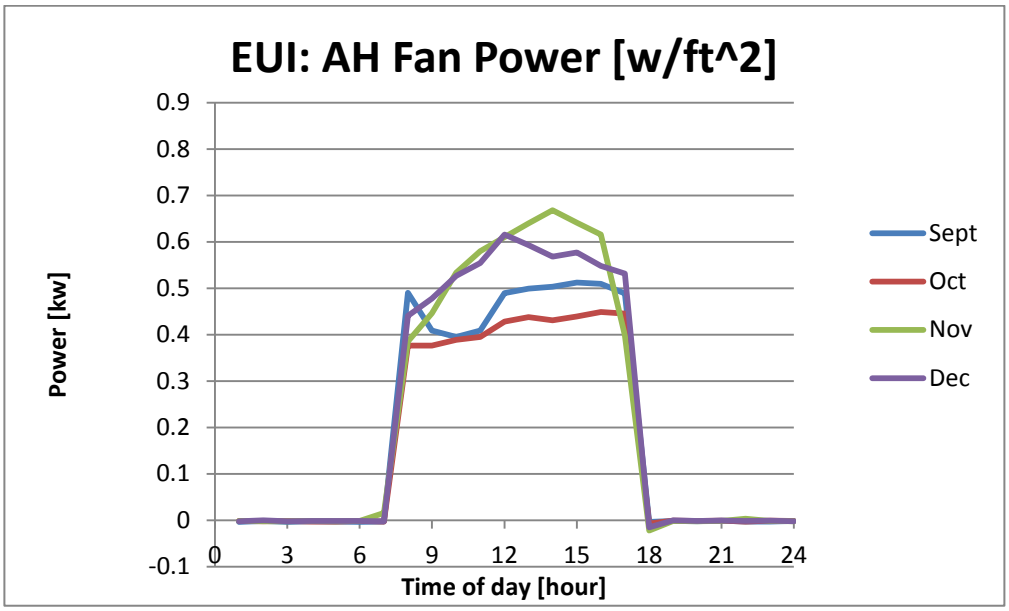

Figure 15b. Saturday AHU power

AHU airflow and power

Figures 16 shows results from analyses to determine airflow at the AHU. Fan energy was modeled based on an analysis of measured fan speed (BMS data) and power (wireless data) by plotting the maximum on the York Solutions fan performance chart (see Figure 16a); the resulting point, 31,900 cfm @ 34 Hp and 4.5 iwc, is close to the commissioning measurements and not too far from the design point.[2] Figure 16b shows a simplified fan performance map and the estimated operating curve based on additional measurements made at the site with a handheld anemometer and pressure gauge.

These methods were used because the original intention to use the measured heat extraction resulted in large scatter and a resulting indeterminate model as shown in Figure 16c. The wide scatter is assumed to result from uncertainties and possible inaccuracies in the measurements used for that calculation; e.g., MAT, SAT, CHWS, CHWR, and water flow.

The operating curve model from Figures 16a and 16b was used to derive the fan part load model of fraction fan airflow vs. fraction of full load power shown in Figure 16e which incorporates the drive efficiently model shown in Figure 16d. EnergyPlus fan modeling was derived from measured fan power, and measured peak airflow and static pressure shown in Figure 16a to obtain the peak efficiency of 70%. Estimated AHU fan flows were calculated from measured power using the model shown in Figure 16e. This model was also used for the part load model in the EnergyPlus simulations.

[2] The commissioning (Cx) report was used to determine the maximum fan speed (1250 RPM) vs. VFD speed reported by the BMS. Motor efficiency from the Cx report and an assumed model for the VFD was used to back calculate peak power fan power (34 Hp) from measured line power monitored by the wireless system. From this data a peak efficiency of 70% was calculated.

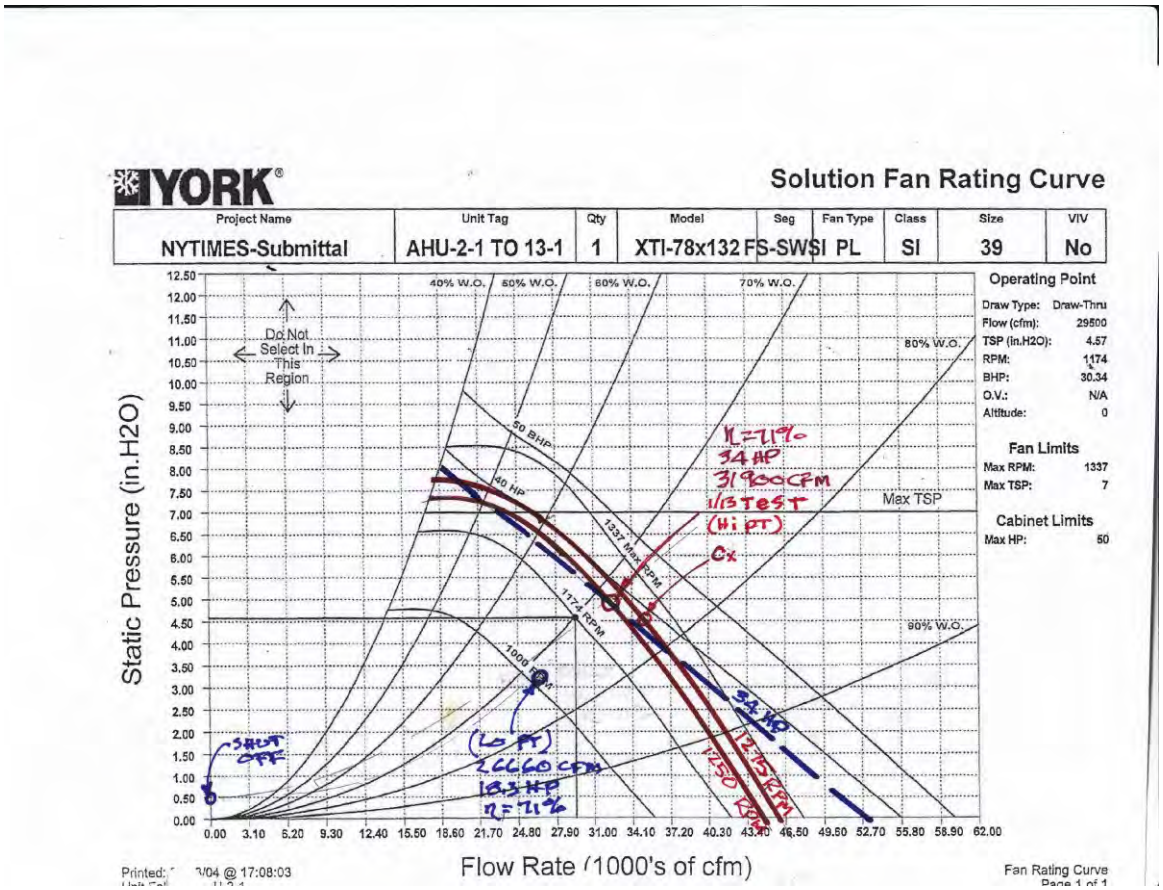

Figure 16a. York fan curves with superimposed peak speed and power curves

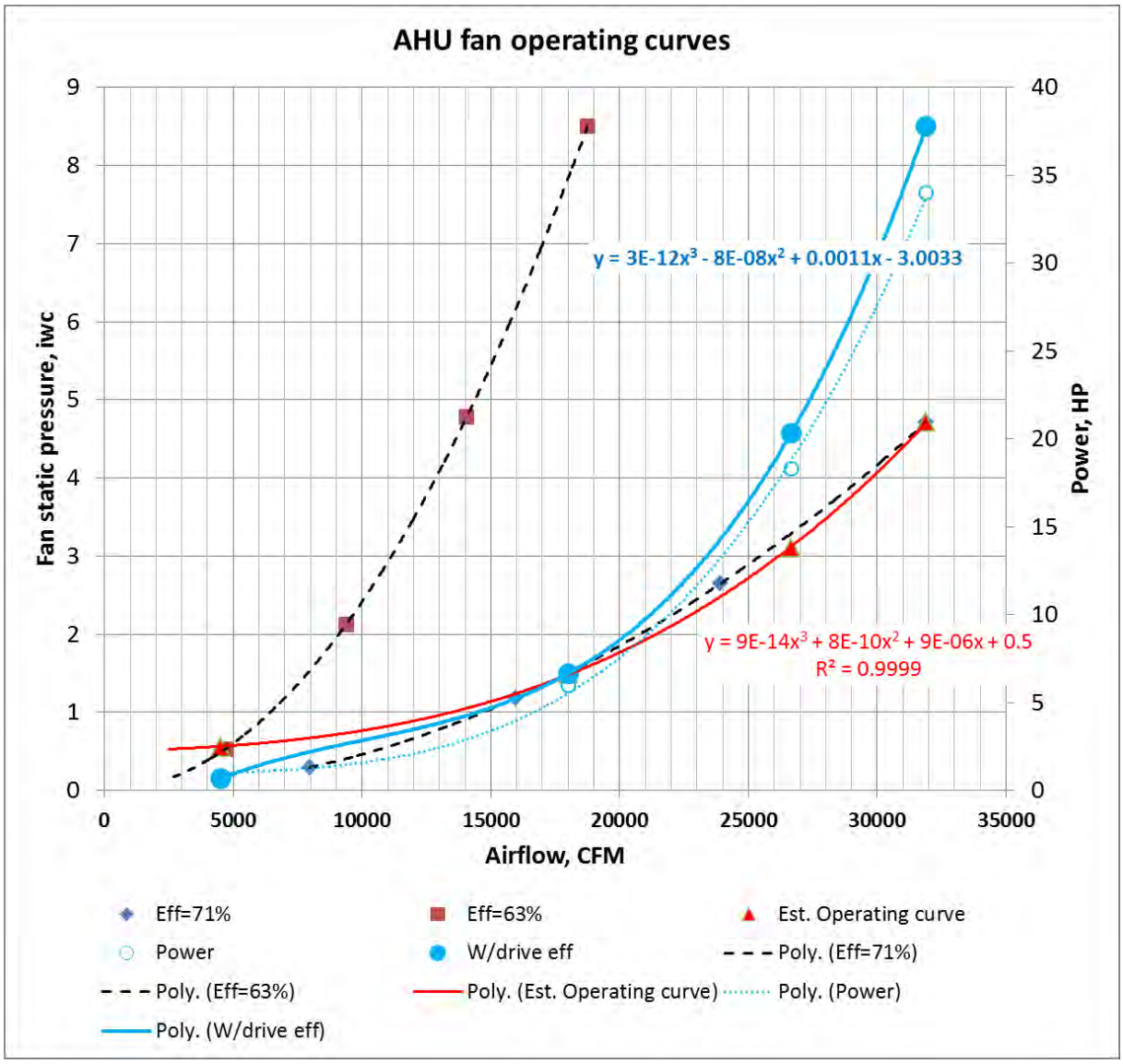

Figure 16b. Fan performance curves (used for air flow estimation and fan PLR)

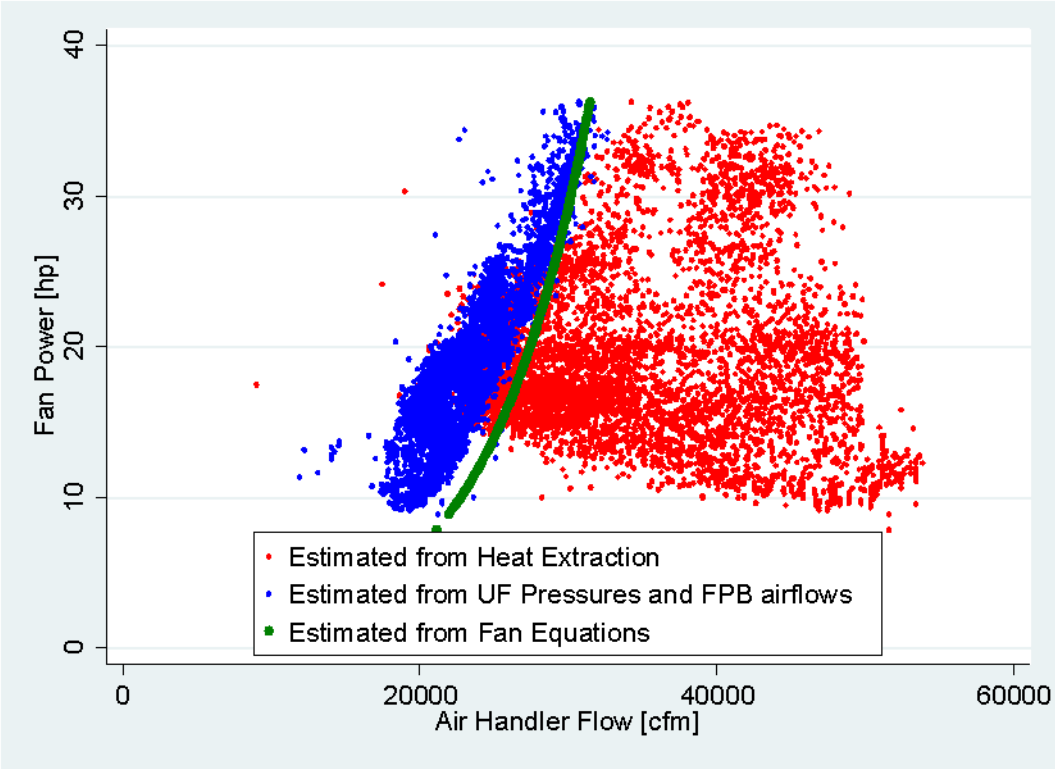

Figure 16c. AHU power vs. airflow data and models

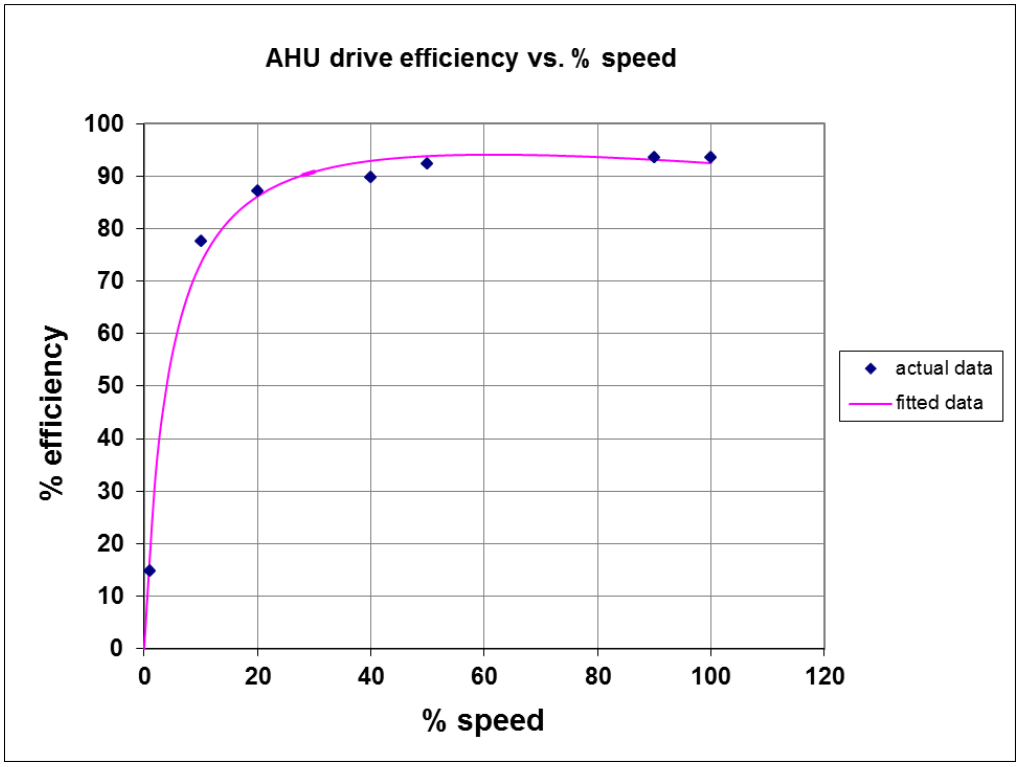

Figure 16d. AHU drive efficiency

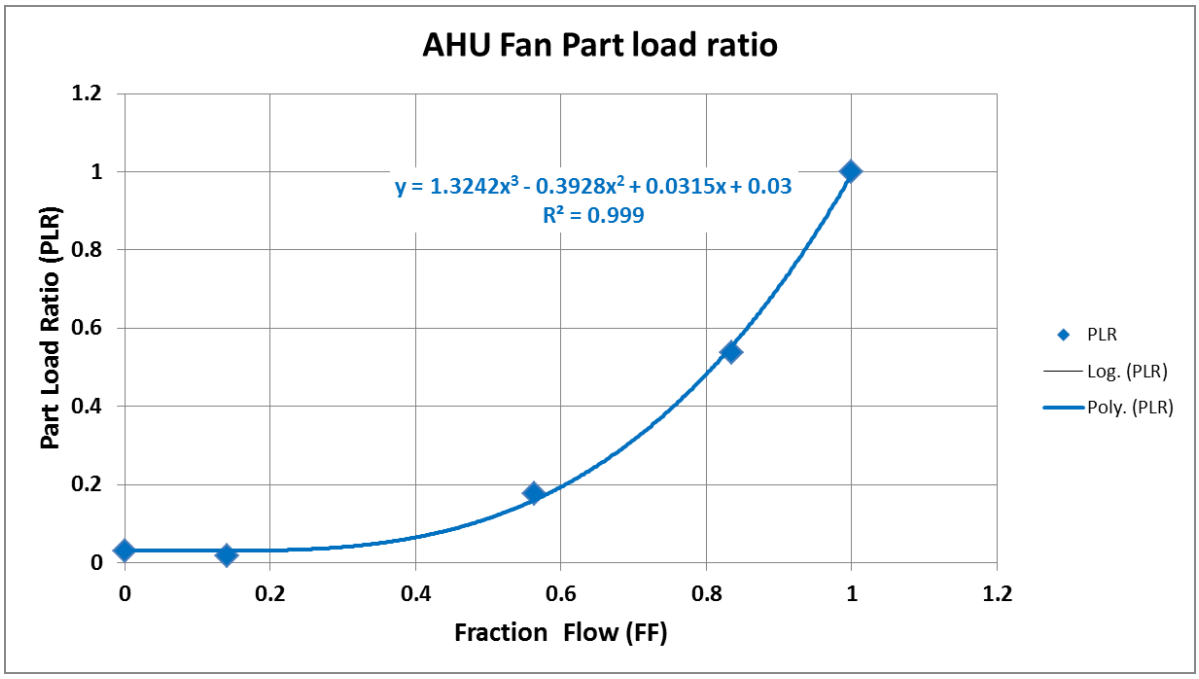

Figure 16e. Operating fan part load power curve (used for simulation)

Terminal units

Figures 17 shows that the perimeter fan powered boxes (FPB) total power decreases in colder months since the fans run a minimum volume during heating. Weekend usage is almost as high as in weekdays but with reduced hours.

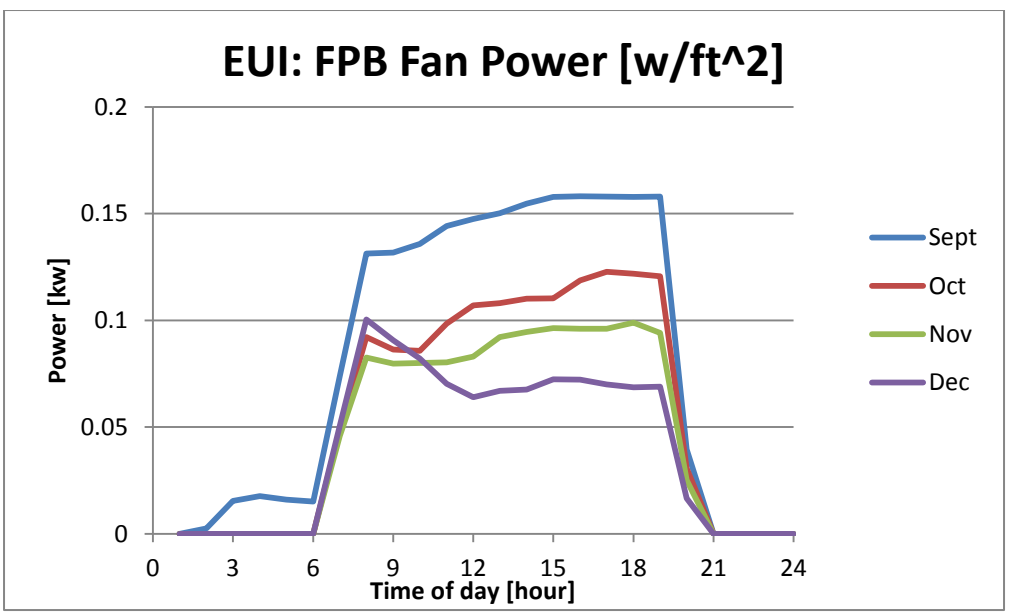

Figure 17a. Weekday FPB power, calculated from calibrated FPB speed vs. power model

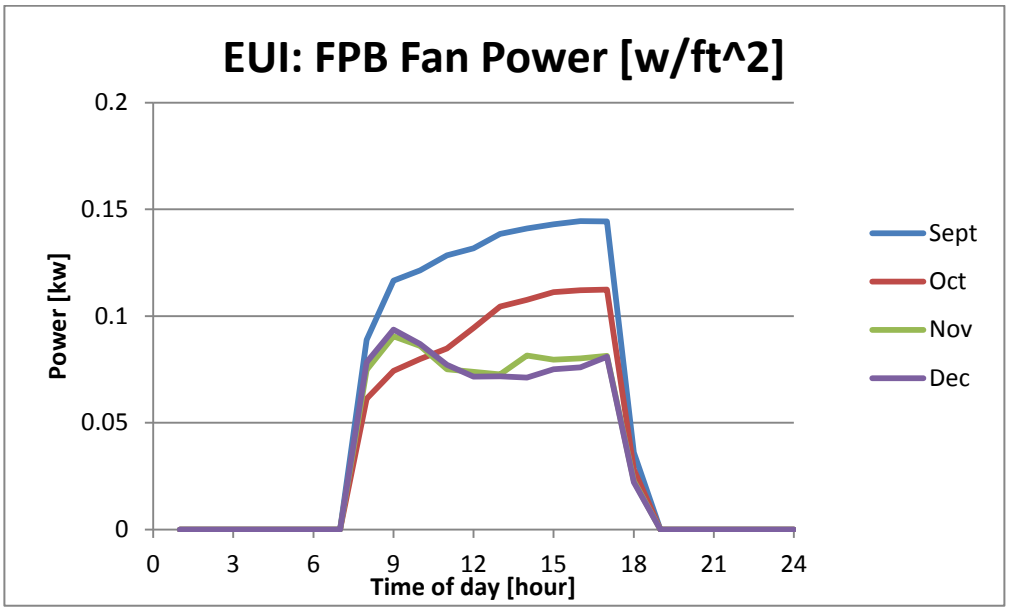

Figure 17b: Saturday FPB power

The flow grid calibrations for six FPBs were checked. These showed an error of up to 20%, but some boxes were low and others high. The average correction was less than 1% so no correction attempt was made to any FPB in the calculations of flow, power, and heat gains.

Figure 18 shows the power/flow relationship measured for one box at two different outlet configurations. Measurements were conducted initially with the perimeter diffusers fully open, then were repeated with about 60% of the diffuser area blocked. There was very little difference in the power flow relationship for these two conditions. This relationship was used for every FPB in subsequent calculations of FPB energy usage.

The original intention was to use this relationship in the calibrated EnergyPlus simulation to model the FPB power consumption in the form of a fraction of fan power vs. fraction of airflow while using the measured peak efficiency of 15% (calculated from measurements) and design fan static pressure of ~0.25 iwc. However, for some reason the UFAD FPB object did not work in the customized version of EnergyPlus. Therefore, a VAV terminal unit was substituted for the FPB and FPB energy was calculated by post processing (based on measured/calibrated airflow) to yield estimated fan power from the model shown in Figure 18; peak airflows for each zone were derived from actual sizing and maximum settings as shown in Table 1 below.

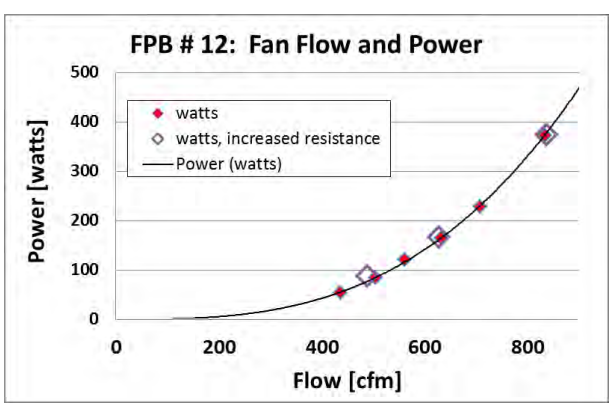

Figure 18: The power/flow relationship for FPB12; with and without increased resistance to flow.

Cooling and heating

Chilled water cooling energy

Figures 19a and b show heat extracted by the chilled water system (secondary cooling energy) as measured by the wireless monitoring system using an ultrasonic water flow meter. There is little variation between seasons except for September; weekend data shows lower use for both September and October. This is counterintuitive but one possible explanation, discussed more fully below, is if the DOAS supplies warmer air in the winter than in the summer months. Also, as indicated previously, there was considerable scatter in the water flow data.

As discussed below, these measurements were not used in the final analysis due to highly scattered data and other missing measurements; the alternative methods are discussed in the section on Comparison Results.

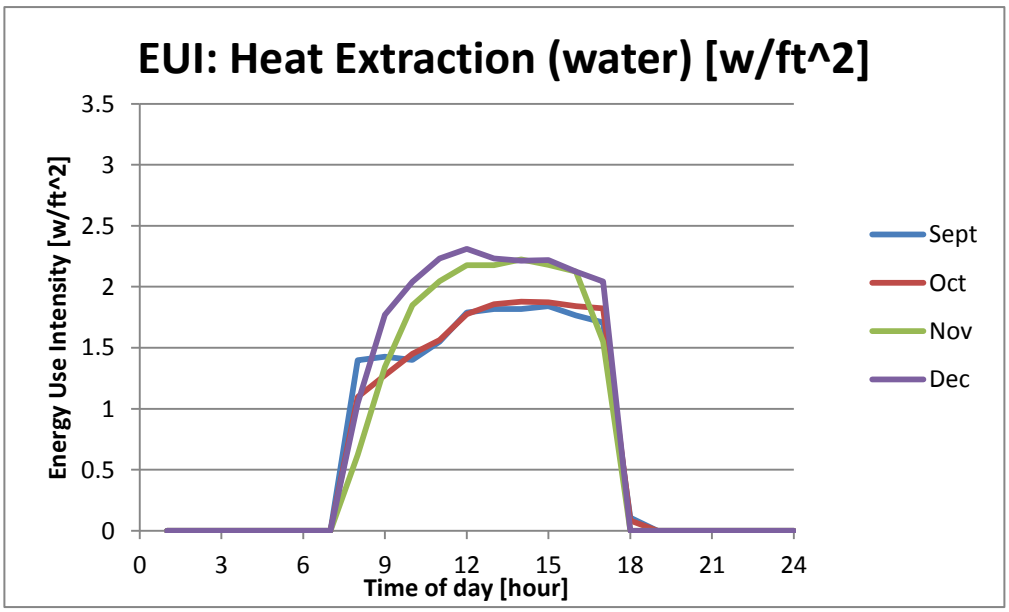

Figure 19a. Weekday measured cooling coil heat extraction (from chilled water flow and supply return temperatures (wireless sensing)

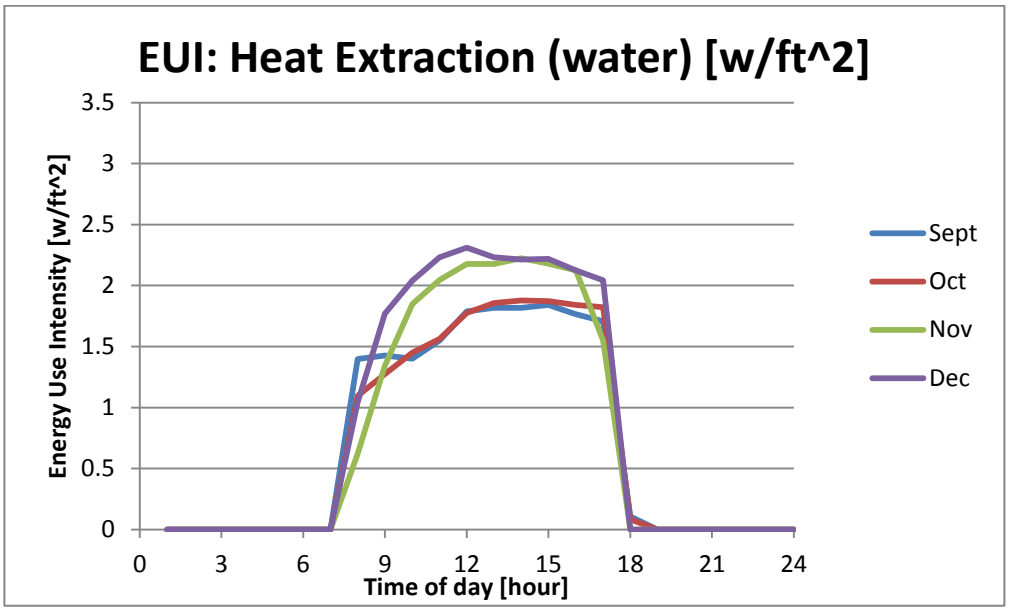

Figure 19b. Saturday measured cooling coil heat extraction

Heating

Figure 19c shows total measured heating by perimeter fan coil units in monthly average power, W/sf. This was derived from the measured airflow and measured entering and leaving temperatures at the fan coils, summed for all units when in heating mode. This represents the "secondary heating" energy, not the actual end use at the boiler. Similar calculations were made for the simulations results to ensure accuracy in the comparison.

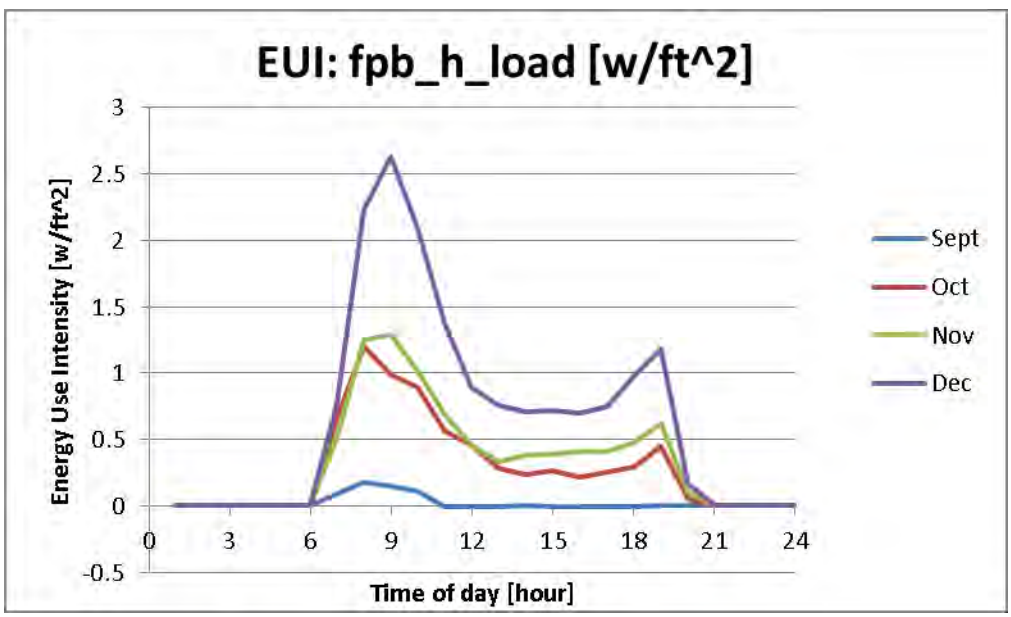

Figure 19c. Fan coil unit heating energy

Zoning sizing and stratification modeling

For perimeter zones, simulation zones' airflow rates were sized based on the actual equipment

sizes installed in the building modified by the maximum volume settings in the control system; these settings were largely the same as the design volumes. The detailed zone sizing was rolled-up to simulation zones as shown in Table 1. For interior zones VAV boxes were used as simulation objects with sizing based on estimates of the number of open diffusers and using typical supply plenum pressures to calculate overall maximum and minimum airflows, as shown in Table 2.

Table 1. Simulation zone airflow (AF) sizing summary, perimeter zones

Sims Zone	FPB	Design airflow (AF), cfm	Sim zone AF, cfm	Sim zone AF, m3/s	Zone length, m	Number of linear diffusers	check length, m
2	19	660	2370	1.12	27.76	**20**	18.1
	20	810					
	21	900					
3	17	810	1470	0.69	7.47	**6**	5.6
	18	660					
4	14	810	4140	1.95	44.2	**35**	31.6
	13	810					
	12	810					
	11	810					
	10	900					
5	8	660	1470	0.69	7.47	**6**	5.6
	9	810					
6	5	812	2522	1.19	27.76	**21**	19.2
	6	900					
	7	810					
7	2	750	1960	0.93	7.47	**8**	7.5
	3	550					
	4	660					
8	1	900	5040	2.38	44.2	**42**	38.4
	28	810					
	27	810					
	26	810					
	25	810					
	24	900					
9	22	660	1410	0.67	7.47	**6**	5.4
	23	750					

Table 2. Simulation zone airflow (AF) sizing summary, interior zones

Sim Zone	Actual zone #	Des # of swirl diffusers	2/2013 Est. # of OPEN swirl diffusers	ATU des AF; zebra, cfm	ATU des AF, cfm	Min AF zebra @30% cfm	AF @ 0.065, # open, Oper max cfm	AF @ 0.065, # open, Oper max m3/s	AF @ 0.01, # open, Oper Min cfm	Oper. %Min
Interior 1, E	Z02+Z03	69	38	9650	5175	2895	3193	1.51	1085	34%
Interior 2, W	Z05+Z06	54	39	9675	4050	2903	3277	1.55	1114	34%
Interior 11, N	Z01	19	14	4875	1425	1463	1176	0.56	400	34%
Interior 10, S	Z04	17	14	4325	1275	1298	1176	0.56	400	34%

Room air stratification was modeled using custom parameters representative of the Titus diffusers installed in the building along with the number of diffusers as shown in Table 1 and Table 2.

Controls settings

Room thermostat settings were modified from actual as discussed below; these are single setpoints (same for heating and cooling) that are seasonally (or operator) adjusted. Actual seasonal schedules are shown in Table 3.

Table 3. BMS occupied thermostat settings

	Thermostat Setpoints, °F	
	Summer	Winter
Interior*	75	76
Perimeter	73	74

**For thermostat at 84 inches*

Thermostat setpoints were adjusted for the UFAD model to reflect the stratification that was measured yielding modeling settings lower than BMS settings for interior zones[3], but not for perimeter zones due to the lack of stratification there.

Supply air temperature was scheduled according to the average of the monthly profiles shown in Figure 20.

[3] Interior zone thermostats are mounted at 84 inches; simulations control the occupied zone (below 67 inches).

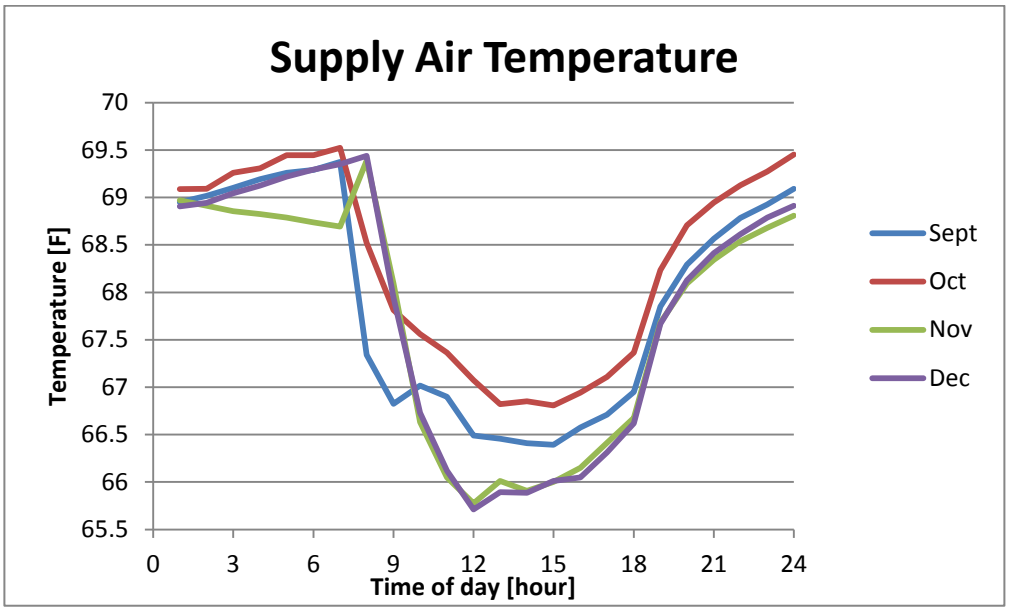

Figure 20. Measured AHU supply air temperatures

Outside air modeling

Outside air is supplied to each floor of the building with a dedicated outside air system (DOAS) that conditions the air to manage humidity and temperature delivered to the floors. Figure 21 shows a diagram of the system. At the floor level (at least for the 20th floor) the outside air volume is constant at 2600 cfm (based on BMS readings). During testing the DOAS was operated at constant dew point discharge temperature of 46°F year around. Unfortunately, the DOAS operating data was not logged. Furthermore, modeling this system configuration is not readily available in EnergyPlus and resources availability militated against creating a workable model. This complicates the analysis of cooling energy since the coil consumption is not representative of cooling loads (loads in the spaces). Consequently, the approach used to assess cooling energy had to be modified (see below).

By modeling both zero outside air and the true minimum outside air (2600 cfm) using actual weather file outside air temperatures (instead of DOAS temperatures) the cooling coil energy can be estimated since the actual contribution will be somewhere in between these two poles.

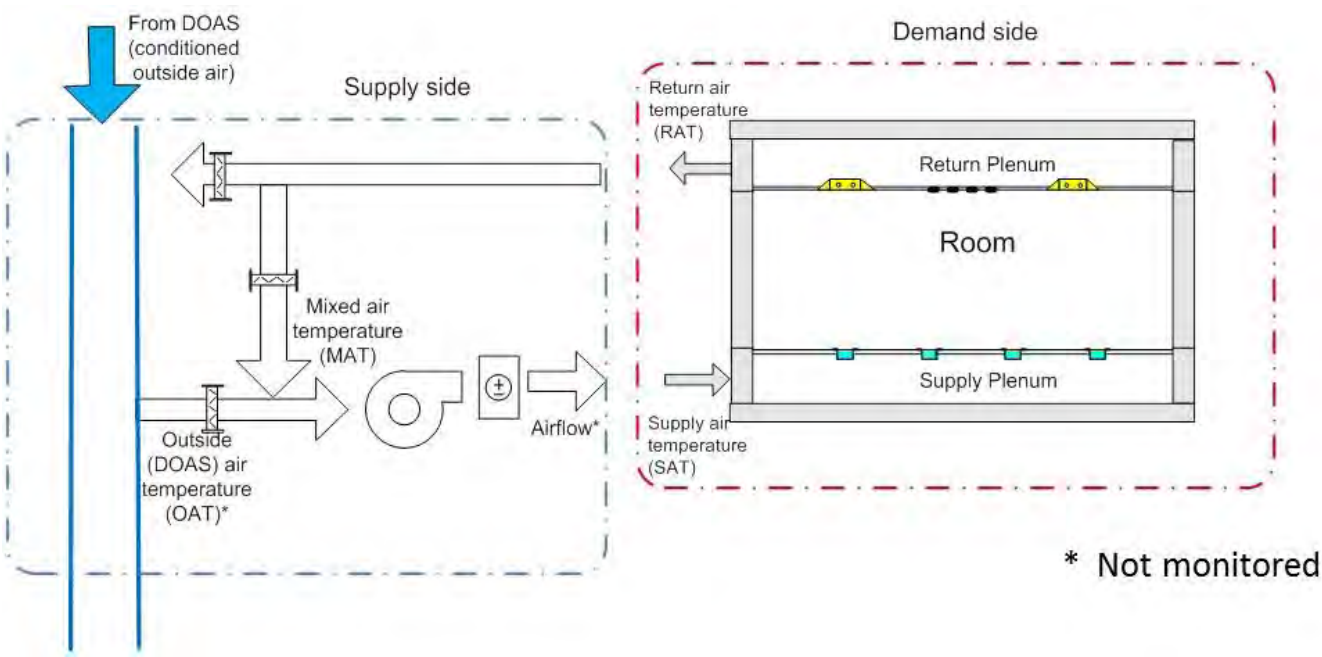

Figure 21. Diagram of DOAS system configuration

COMPARISON RESULTS

End use breakdown comparison

Figures 22 and 23 show comparisons of end use breakdowns for two simulated cases using actual local weather data from WeatherAnalytics; 1) No Outside air (No OSA), and 2) fixed minimum outside as measured (2600 cfm). These results emphasize impact on secondary cooling (cooling coil) performance; all other components are virtually the same in both simulations.[4] These figures show the following:

- Lighting and plug loads are modeled accurately as expected
- Cooling performance, based on floor cooling energy[5], is somewhat different
- Heating performance, based on floor level heating,[6] shows a greater difference than cooling
- Fan energy is significantly different but as explained below is realistic given the error in cooling energy

Each of these topics (other than lighting and plug loads) is discussed in more detail in the following sections.

[4] Heating and fan energy are shown with "corrected" values, see discussion below

[5] "Floor cooling" energy as described below represents sensible secondary cooling energy without conditioning outside air; i.e., cooling loads at the gross floor level. It is not end use chiller energy.

[6] "Floor level heating" closely represents the sum of fan coil heating, but is not end use boiler energy.

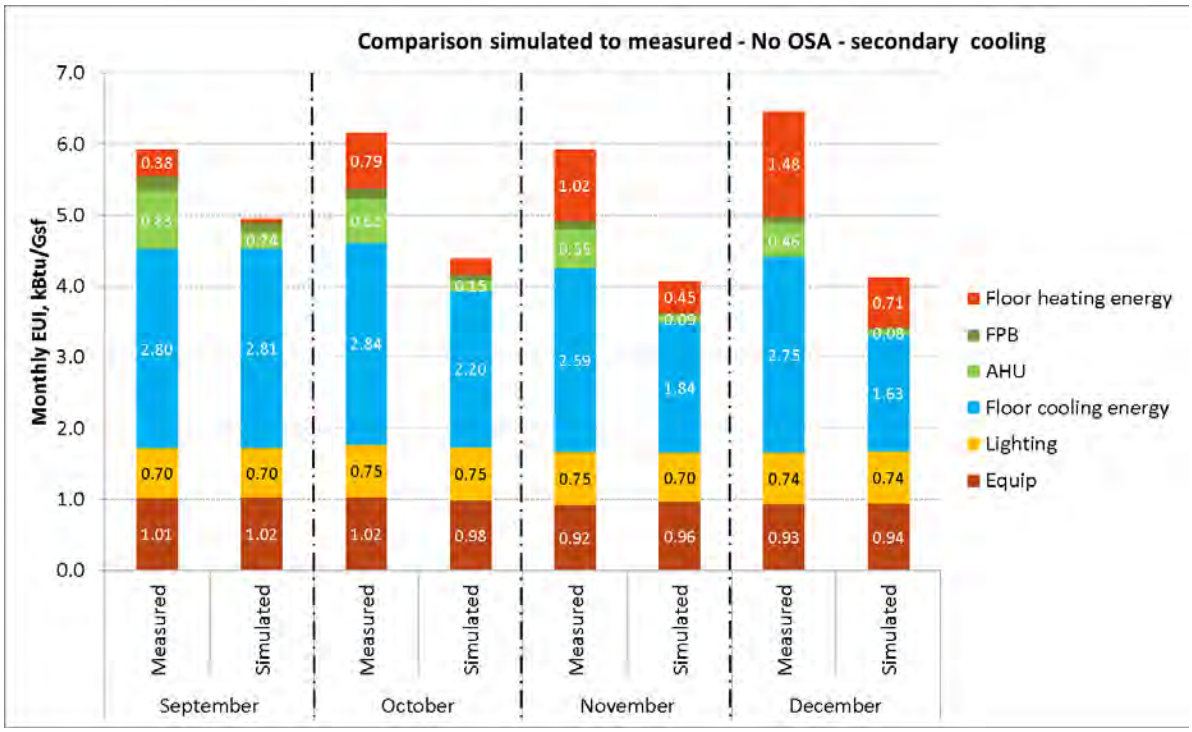

Figure 22. End use breakdown, measured vs. simulated with ***no outside air*** *and* ***secondary cooling*** *and* ***heating***

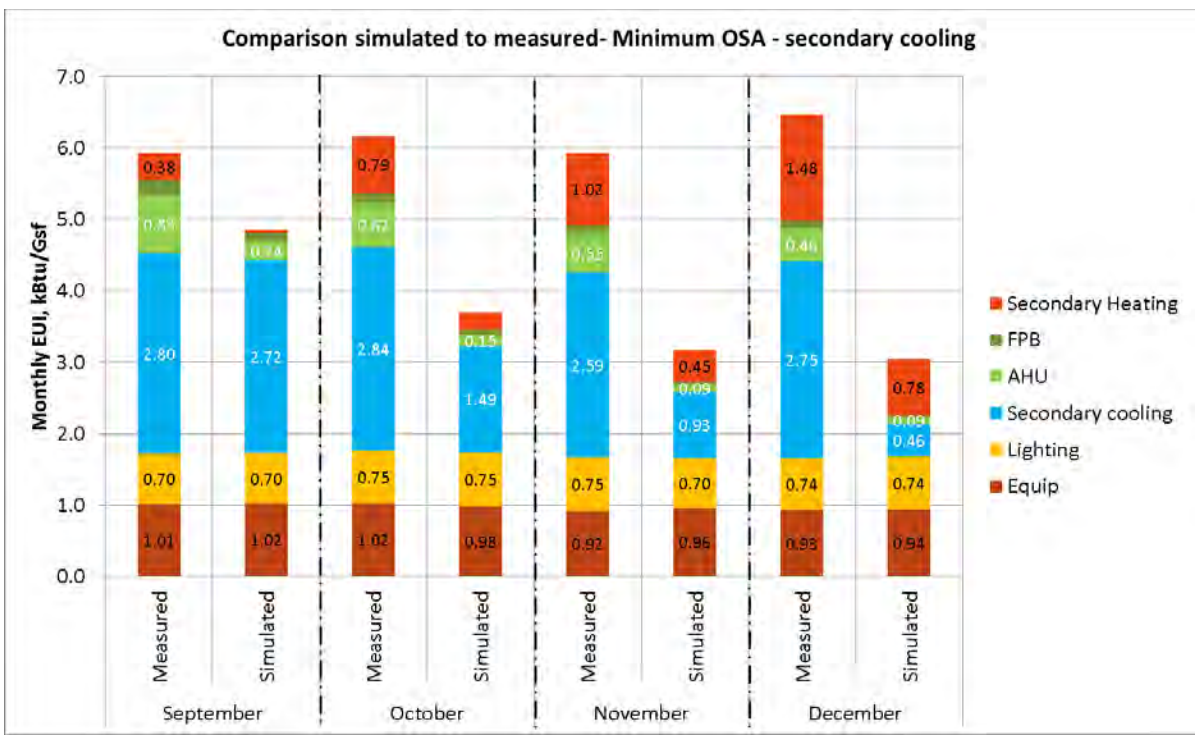

Figure 23. End use breakdown, measured vs. simulated ***with minimum 2600 cfm outside air*** *and* ***secondary cooling*** *and* ***heating***

Cooling comparison methods

Our original intent was to base cooling energy comparisons on secondary cooling (i.e., cooling coil energy) which removes the issues of plant simulations which would be overly complicated to model accurately and would add to the uncertainties. However, Figure 22 shows measured cooling energy to be relatively constant for the four testing months. Simulated secondary cooling for the case of zero outside air is less than 1% greater for September and about 40% lower for December. Figure 23 shows simulated scondary cooling for minimum outside air (2600 cfm) to be equal to (September) or significantly less than measured. As discussed above under outside air modeling, the actual system uses a DOAS system for the outdoor air supply which conditions the air in an unknown way. Comparing results for zero and minimum "real" outside air a wide difference is shown in simulated cooling energy while measured energy is somewhat constant. This suggests that DOAS system is actually *warming* the outside supply air in the cooling months, and cooling it in the warmer months, thus yielding relatively constant energy use over the four months. Therefore, another method for comparing measured vs. simulated at a more fundamental level is called for.

Measured cooling methods

Secondary cooling comparisons are complicated by the outside air issues discussed above. Simulated secondary cooling performance will never match measured since the UFAD model assumes "real" OSA when a fixed volume is simulated as opposed to the actual system configuration using a ~25% DOAS conditioned air. A better comparison can be made based on the "demand" side that represents the cooling required at the floor or space level, herein generally called the "Floor or space cooling energy." Also, the demand side cooling is divorced from the cooling coil performance thus the outside air rate is irrelevant so only the *zero* outside air simulations are compared to the equivalent measured results. Furthermore, it is obvious from the cooling coil psychometrics (not shown here) that the cooling coil operates as a "dry coil" meaning that the analysis can focus on sensible loads for the comparisons. As shown below, the energies calculate by this method are "net" cooling or heating, meaning only the airflow used to meet cooling (or heating) loads are summed (which is different than what the cooling coil supplies). The same calculations are performed on the EnergyPlus results as described below.

Floor cooling energy represents the total energy required to cool the entire floor plate which includes the impact of supply and return plenums. Figure 24 shows the thermal envelope for this analysis.

Space cooling energy represents the energy required for cooling the space at each hour, which is driven by the net heating gains. This method has the additional benefit of being a closer representation of the load modeling performed by EnergyPlus. Figure 25 shows the thermal envelope for this case.

Excess cooling/heating

The NY Times floorplate is designed with both swirl and linear bar grilles in the perimeter zones. The swirl diffusers are supplied via the interior zone controls and the linear bar grilles are supplied by fan coil units. However, EnergyPlus cannot readily simulate UFAD zones with two types of diffusers simultaneously. The impact of swirl diffusers in perimeter zones is twofold: (1) cooling stratification modeling cannot be accurately represented since the models were developed from testing where only perimeter liner bar grilles were supplying the zone; and (2) the extra cooling will cause heating loads to be larger; i.e., cool air enters the zone while the fan coils are in heating mode. The measured net cooling and heating demand calculations are "corrected" in an attempt to estimate this effect by subtracting or adding the contribution of the swirl diffusers from the

overall measured heating, measured cooling, and AHU fan power (due to projected lower airflow) when perimeter zones are in heating mode. This correction attempts to better align the measured results to how the simulation works, although it should be considered only an approximation.

Detailed equations for each of these calculations are described in the following sections.

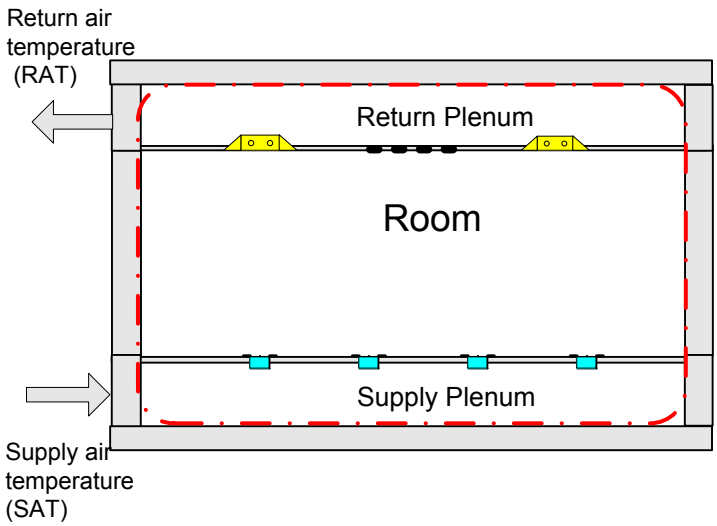

Figure 24. Illustration of thermal envelope for *floor* net cooling/heating demand calculations

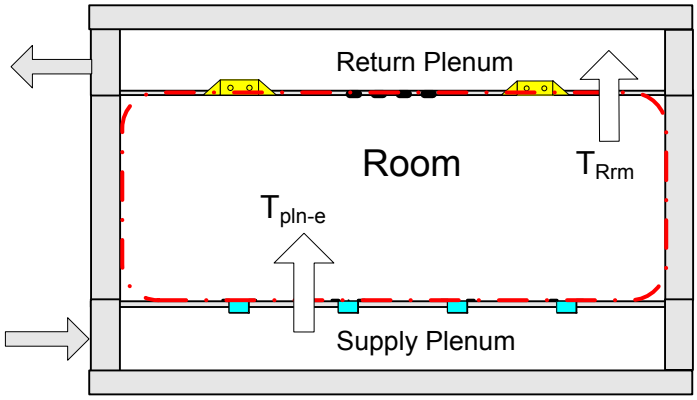

Figure 25. Illustration of thermal envelope for *space* net cooling/heating demand calculations

Floor cooling energy (monthly)

For this calculation the sum of the airflows for all the FPBs in heating mode plus the swirl diffuser excess is subtracted from the AHU airflow which is then multiplied by the floor level RAT/SAT difference to arrive at the net cooling energy for each hour which is then summed for all operating hours in a month. This represents the amount of energy devoted to satisfying cooling loads at the floor level.[7]

[7] Another way to view the floor level is that it represents what the HVAC system supplies to meet the overall

$$E_{FC-tot} = C * [CFM_{AHU} - \sum_{fpb_Zn@htg} (CFM_{fpb} + CFM_{diff})] * (RAT - SAT)$$

Where,

$CFM_{fpb} = fan\ powered\ box\ airflow$
$CFM_{diff} = perimter\ zone\ swirl\ diffuser\ airflow$
$CFM_{AHU} = Air\ handler\ air\ flow$

$C = Constant$

$RAT = AHU\ return\ air\ temperature$
$SAT = AHU\ supply\ air\ temperature\ (air\ highway\ temperature$
$fpb_Zn@htg = \ summed\ for\ all\ perimeter\ zones\ in\ heating\ mode$

Space (net) cooling energy (monthly)

In this calculation the (net) energy use at the level of the space load is calculated as shown in the following equation where the airflow to the interior spaces plus FPBs in cooling mode are summed hourly for a month. Note that the "excess cooling" airflow (CFM_{diff}) is added to the cooling to account for the fact that there are two paths for cooling airflow to perimeter zones while the simulations have only one to meet the load demands. The result of this calculation represents the amount of energy spent at the space level to satisfying cooling loads.

$$E_{SC-tot} = C * \sum_{Zn_I} CFM_{int} * \left(T_{R_{rm}} - \bar{T}_{pln-s}\right)$$

$$+$$

$$C * \sum_{fpb_Zn@clg} (CFM_{fpb} + CFM_{diff}) * \left(T_{R_{rm}} - \bar{T}_{pln-s}\right)$$

Where,

$CFM_{fpb} = fan\ powered\ box\ airflow$
$CFM_{diff} = perimter\ zone\ swirl\ diffuser\ airflow$
$CFM_{int} = interior\ zone\ airflow\ (calculated\ by\ total\ AHU\ airflow\ minus\ FPB$
$total\ airflow\ proportioned\ by\ number\ of\ diffusers\ in\ the\ intrior\ zone)$

$C = Constant$

$\bar{T}_{pln-s} = average\ plenum\ temperature\ in\ zone$
$T_{R_{rm}} = Room\ return\ temperature\ (\ top\ of\ stratification\ tree)$
$Zn_I = indicates\ summed\ for\ all\ \ interior\ zones$

$fpb_Zn@clg = \ summed\ for\ all\ perimeter\ zones\ in\ cooling\ mode$
$fpb_Zn@htg = \ summed\ for\ all\ perimeter\ zones\ in\ heating\ mode$

---

demands in the building; space loads and other elements such as supply plenum and return plenum; i.e., not what the cooling coil supplies

Measured heating methods

As for cooling, heating was calculated at the "Floor" level and the more detailed "Space" level; these equations are described below.

Floor heating energy (monthly)

The following calculation is performed at each hourly time step for all operating hours for each month.

$$E_{FH-tot} = C * \sum_{fpb_Zn@htg} CFM_{fpb} * \left(T_d - \bar{T}_{pln-s}\right)$$

$$-C * \sum_{fpb_Zn@htg} CFM_{diff} * (T_{R_{rm}} - \bar{T}_{pln-s})$$

The latter term is the "excess cooling" caused by swirl diffusers discharging into the perimeter zone while heating therefore exacerbating the heating load. The excess is subtracted from the measured data to make it roughly equivalent the simulations that does not model this simultaneous cooling.

Space heating energy (monthly)

$$E_{SH-tot} = C * \sum_{fpb_Zn@htg} CFM_{fpb} \times \left(T_d - T_{R_{rm}}\right)$$

$$-C * \sum_{fpb_Zn@htg} CFM_{diff} \times (T_{R_{rm}} - \bar{T}_{pln-s})$$

Where,

$CFM_{fpb} = fan\ powered\ box\ airflow$
$CFM_{diff} = perimter\ zone\ swirl\ diffuser\ airflow$

$C = Constant$

$T_d = FPB\ discharge\ temperature$
$\bar{T}_{pln-s} = average\ plenum\ temperature\ in\ zone$
$T_{R_{rm}} = Room\ return\ temperature\ (\ top\ of\ stratification\ tree)$
$Zn_P = indicates\ summed\ for\ all\ perimeter\ zones$
$fpb_Zn@htg =\ summed\ for\ all\ perimeter\ zones\ in\ heating\ mode$

Simulation cooling methods

Hourly simulation reports were used to derive total monthly energies via an Excel macro. This process simply "rolled up" reported or computed values to the monthly level while filtering for occupied hours only for all days the system was operating to match the schedules of the measured data.

Floor cooling energy

Floor energy was computed using the AHU supply and return temperatures and AHU airflow, using the following output parameters.

- SUPPLY FAN INLET NODE:System Node Volume Flow Rate Standard Density
- RELIEF AIR NODE:System Node Temp (i.e., RAT)
- AIRLOOP OUTLET NODE:System Node Temp (i.e, SAT)

From the hourly total fan volume above, the airflow for all FPBs (REHEAT AIR OUTLET NODE:System Node Volume Flow Rate Standard Density) in *heating mode* was subtracted to yield the net cooling energy for the floor.

Space cooling energy

In this case, each zone cooling hourly load was used and summed for all operating hours; the following output parameter was used:

- ZONE NAME:Zone/Sys Sensible Cooling Rate

Floor heating energy

Floor heating represents the total FPB heating coil energy (therefore including reheat); the following report variable from the Meters file was used and summed for all operating hours each month:

- ZONE NAME:HeatingCoils:EnergyTransfer

Space heating energy

For space level heating, the zone sensible heating was computed by summing the following variable for all operating hours during the month:

- ZONE NAME:Zone/Sys Sensible Heating Rate

Comparison findings

Comparison metrics

ASHRAE Guideline 14, (Section 5.2.11.3; Modeling Uncertainty) [ASHRAE 2010] was used as a benchmark to evaluate the results. This calculation uses the coefficient of variation of the root mean square error (CVRMSE) and normalized mean bias error (NMBE) between the monthly simulated and measured energy use. However, it can only be used as a rough indicator since it was devised for whole building energy, not individual end uses; but as a standardized error measurement it can serve as a rough indicator. What probably cannot be relied upon in this instance is the compliance figures which are based on whole building energy; they are included here for reference purposes only.

End use breakdown comparison

Figure 30 shows the results of the end use breakdown after corrections were applied (compare to Figure 22).

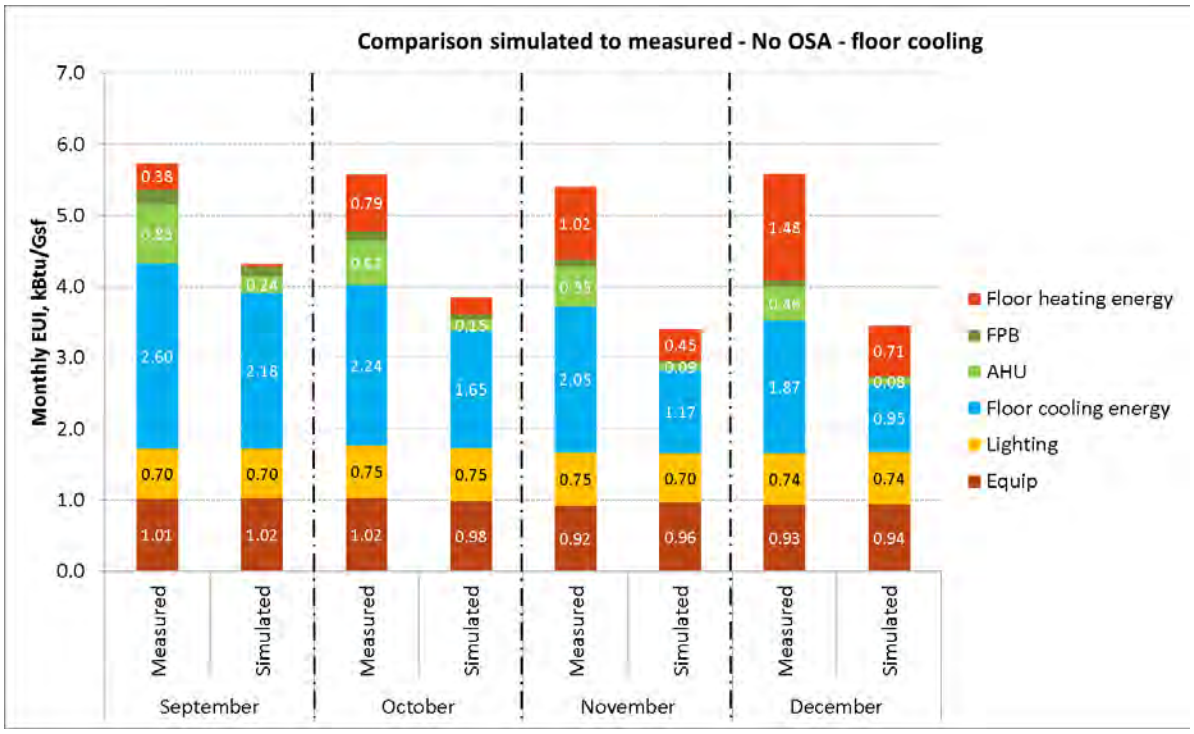

*Figure 26. Monthly end use breakdown based on **floor cooling** and **heating energy**, zero outside air*

Floor cooling comparisons

Figure 27 and Table 4 show summarize findings from the floor cooling energy comparisons. The Guideline 14 compliance ranges are shown for reference only. Errors exist on both measurement and simulation side due to differences in assumptions for each and this was only a preliminary step in the calibration process.

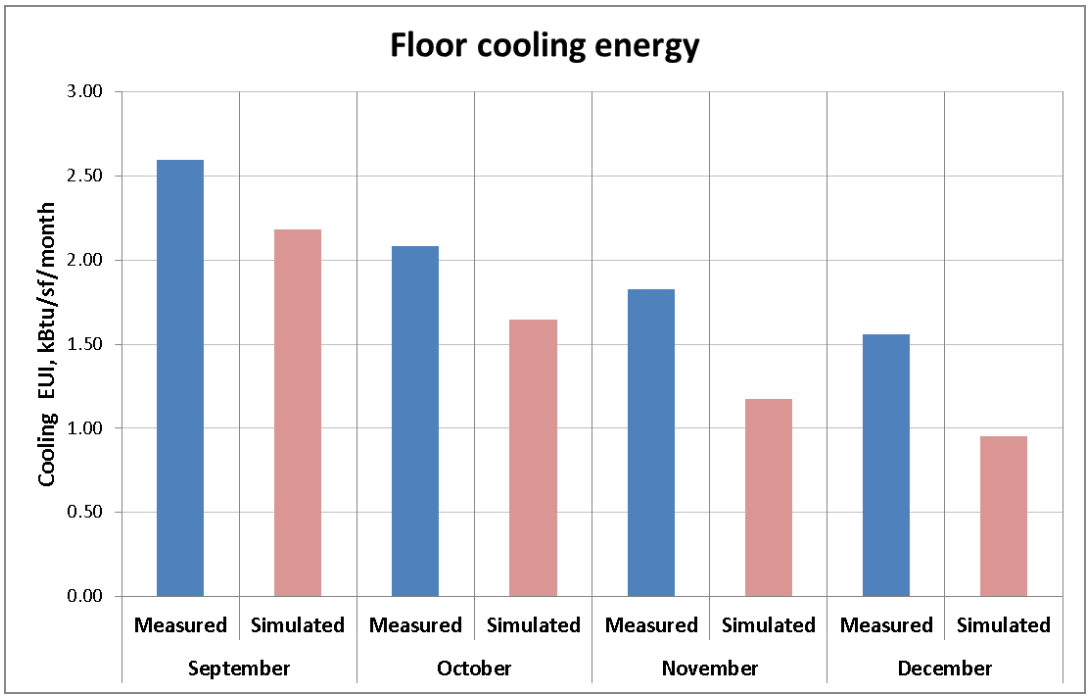

Figure 27. Floor cooling energy comparison, measured vs. simulated with zero OSA

Table 4. Error analysis for floor cooling energy comparison

Floor Cooling Energy, No OSA					
	Measured	**Simulated**	**RMSE**		
September	2.60	2.18	0.17		
October	2.08	1.65	0.19		
November	1.83	1.17	0.43		
December	1.56	0.95	0.37		Compliance
			30.9	CVRMSE, %	15%-30%
			19.2	NMBE, %	5% -10%

Space cooling comparisons

At the space level, Figure 28 and Table 5 show that errors are greater at this level than for the floor level. However, the trends are consistent and the total error not much different than for the floor level case. This suggests that loads' modeling is the primary cause of the difference from measured and that modeling of the supply and return plenums has not introduced much additional error. Part of this error can be attributed to the measurements processing since a number of assumptions had to be made to calculate the cooling at the space level. In this case measurement uncertainties tend to compound due to difficulties in estimating the supply air temperatures to the zones and aggregating actual zones to simulation zones. The excess cooling factor also plays a role here; the interplay of simultaneous heating and cooling (in the real building) and our attempt to deal with it could lead to differences from the simulations. On the modeling side, all of the UFAD models (room stratification, supply plenums) assume fully mixed subzones which are idealized representations of real systems.

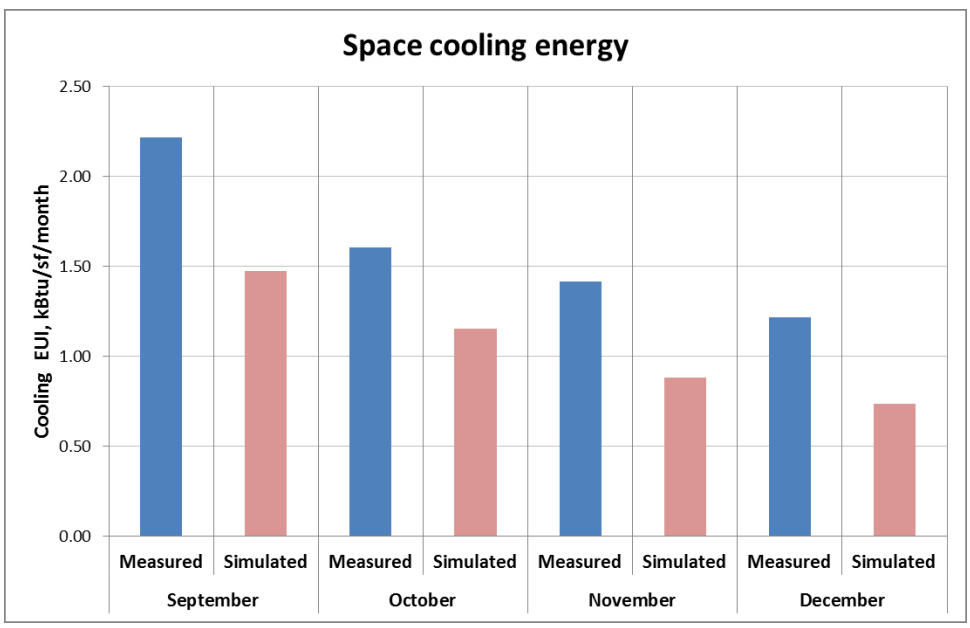

Figure 28. Space cooling energy comparison, measured vs. simulated with zero OSA

Table 5. Error analysis for space cooling energy comparison

Space cooling energy, No OSA					
	Measured	**Simulated**	**RMSE**		
September	2.22	1.47	0.55		
October	1.61	1.15	0.21		
November	1.42	0.88	0.28		
December	1.22	0.74	0.23		
			40.4	CVRMSE, %	15% max
			26.3	NMBE, %	5% max

Heating comparisons

The figure (Figure 29) and table (Table 6) below show a large difference in heating performance between measured and simulated (including allowance for the excess cooling provided by interior swirl diffusers in the perimeter zones). In this case the measured results reflect the performance of all the FPBs; the heating for each box was measured using calibrated airflows from the BMS, and measured inlet vs. discharge temperature difference. The inlet and discharge temperatures were derived from wireless mote readings using motes closest to the each of the FPB inlets.

The large differences in these results suggest that the simulated loads do on accurately represent the real case. Previously the authors have observed some inconsistency in the outputs for UFAD simulations that may (once fully analyzed) help explain the difference[8] but other factors may also have an impact (e.g., assumption of closed blinds in simulations and zone control differences).

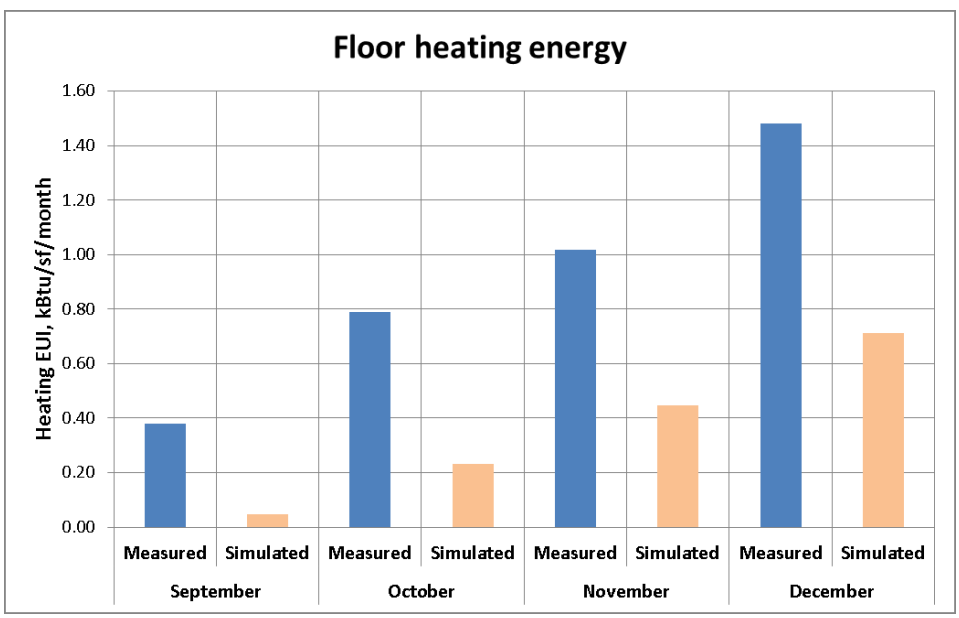

Figure 29. Floor heating comparison, measured vs. simulated with zero OSA

[8] This source of error is currently (early 2013) being investigated by CBE. However, another error has been confirmed (aka deadband bug) for the UFAD variable speed fan coil model that would overestimate the heating, but that model was not used in the NY Times modeling.

Table 6. Error analysis for floor heating energy

Floor Heating Energy, No OSA					
	Measured	**Simulated**	**RMSE**		
September	0.38	0.05	0.11		
October	0.79	0.23	0.31		
November	1.02	0.45	0.33		
December	1.48	0.71	0.59		Compliance
			72.9	CVRMSE, %	15%-30%
			48.7	NMBE, %	5% -10%

Space heating comparisons

The space heating results are similar to the floor heating; i.e., errors are comparable. This might indicate that load simulations for heating are deficient and therefore are a primary source of the floor level heating errors. However, given the uncertainty on the measurement side it would be hard to be sure about this conclusion. More certain (relatively) is that there is error in the estimate of "excess cooling" used to better align the measurements with simulated results given the inability to simulate the real case but this contributes relatively little to the overall differences.

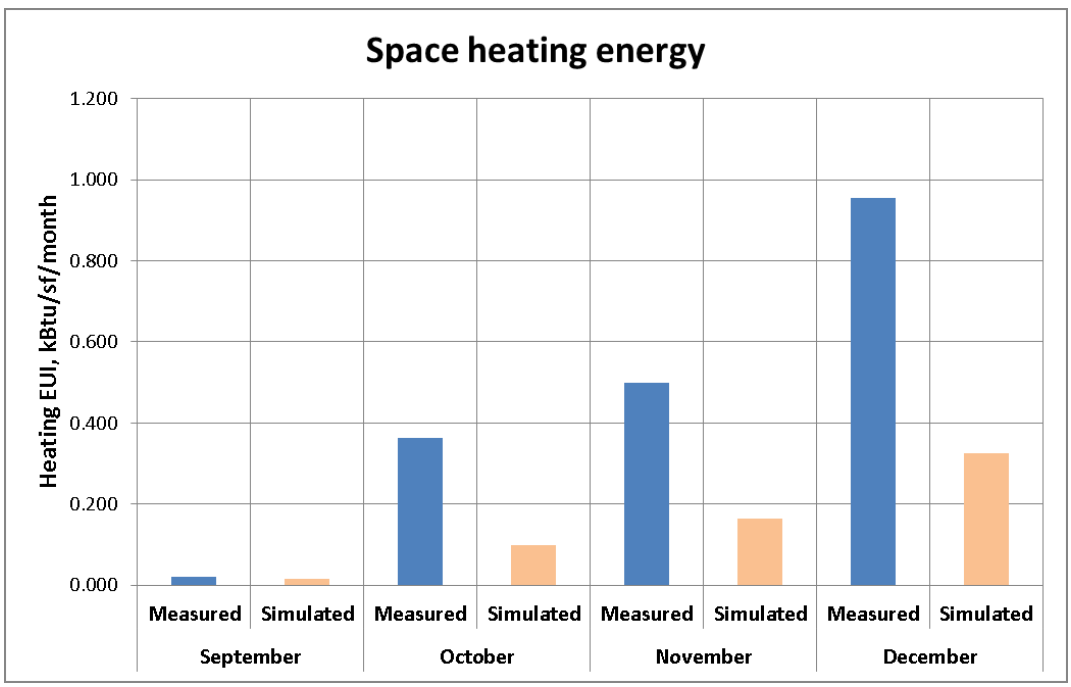

Figure 30. Space heating energy comparison, measured vs. simulated with zero OSA

Table 7. Space heating error analysis

Space heating energy, No OSA					
	Measured	**Simulated**	**RMSE**		
September	0.020	0.014	0.000		
October	0.363	0.099	0.070		
November	0.499	0.163	0.113		
December	0.954	0.326	0.394		Compliance
			95.6	CVRMSE, %	15%-30%
			41.9	NMBE, %	5% -10%

AHU fan power comparisons

As shown in Figure 31 and corroborated by Table 8, simulated fan energy is much less than measured. We have high confidence in the fan models as explained above and the much lower fan energy results from the cubic relationship between airflow and fan power. Figures 32 show monthly distributions for measured and simulated airflows confirming the difference expressed in the cooling results. Assuming the cooling load results are primarily caused by lower airflow this suggests that simulated cooling loads on average about 20% less than actual.

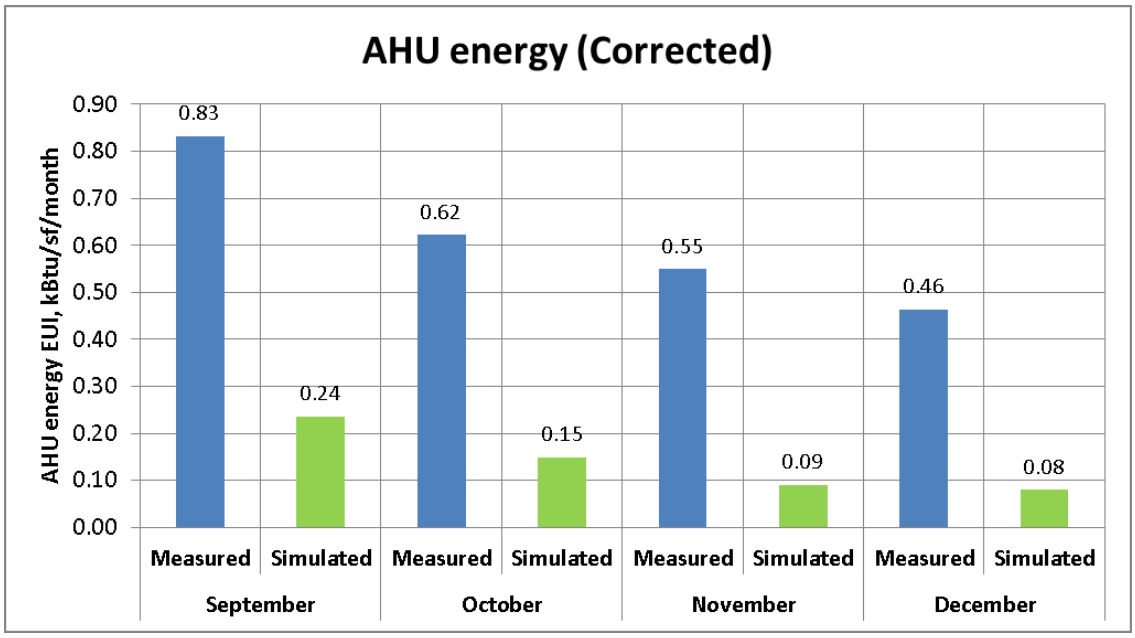

Figure 31. AHU energy comparison, measured vs. simulated with zero OSA

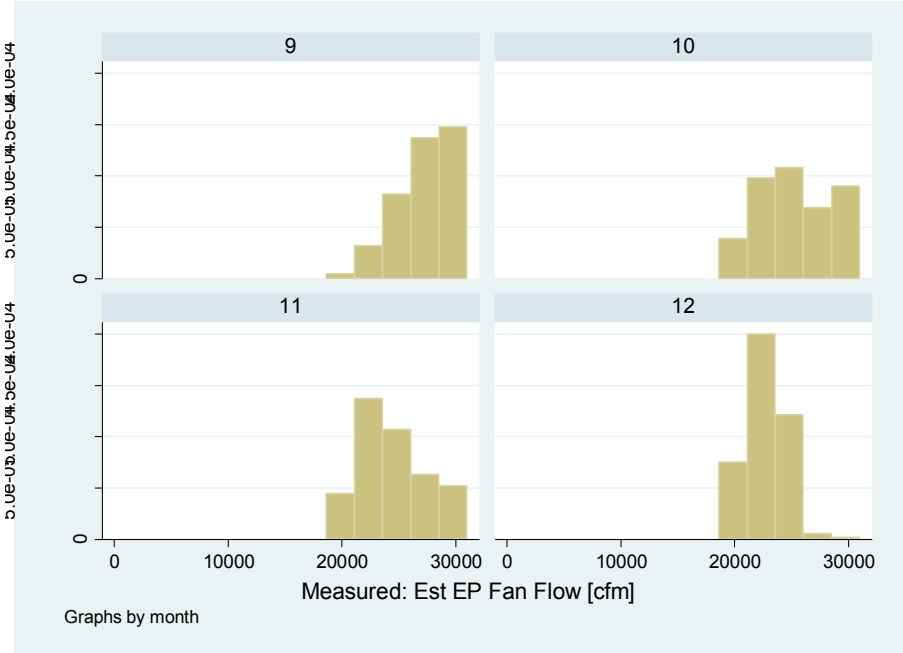

Figure 32a. Measured airflow histogram (based on fan model and measured fan power)

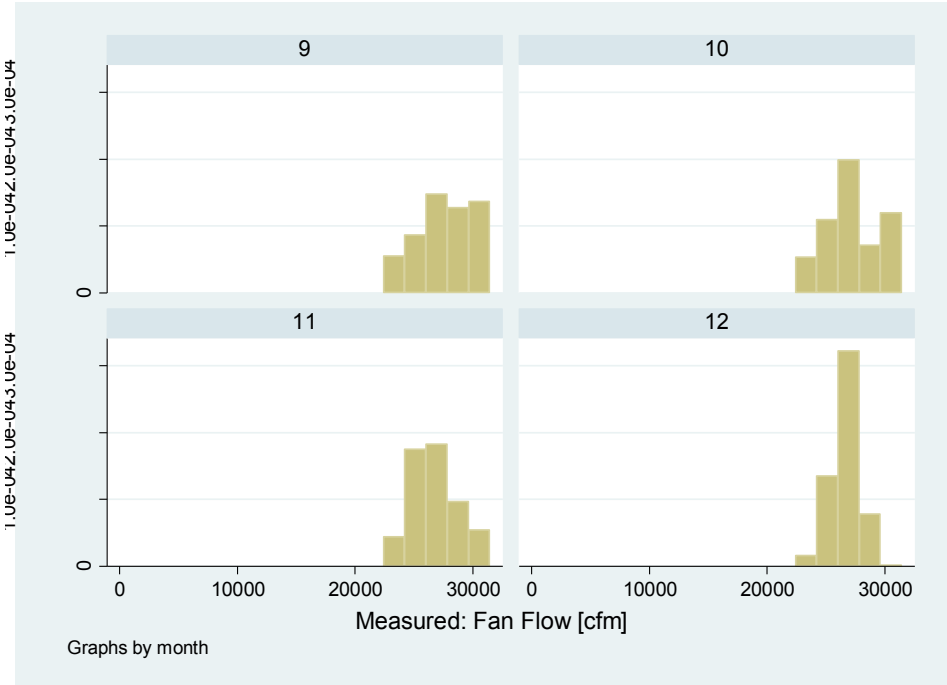

Figure 32b. Simulated airflow histogram (from EnergyPlus results))

Table 8. Error analysis of AHU fan power for corrected cooling airflow

AHU Energy (Corr), No OSA					
	Measured	**Simulated**	**RMSE**		
September	0.83	0.24	0.35		
October	0.62	0.15	0.22		
November	0.55	0.09	0.21		
December	0.46	0.08	0.15		Compliance
			90.6	CVRMSE, %	15%-30%
			50.7	NMBE, %	5% -10%

DISCUSSION

As shown in tables above errors between simulated and measured conditions range from modest (floor cooling energy) to large (space heating and AHU fan) when using Guideline 14 calculation methods.

Considerable effort was expended to thoroughly analyze these results in an attempt to make simulated vs. measured cases as similar as possible and to fully vet the measurements used in the calculations for accuracy. However, overall the analysis was handicapped by a deficient number of critical measurements and the need to employ "work arounds" in an attempt to make apples-to-apples comparisons. And there will always be a certain number of unknowns in the operation or measurements that were not captured in the simulation. While the fan energy errors look large, they are to be expected from the level of the floor cooling errors which result from lower simulated airflows; when the cubic effect of fan power vs. flow is considered the AHU energy error is expected. The cooling errors indicate that the simulated cooling loads are lower than actual loads but various methods could be applied to refine the modeling as discussed below in the review of assumptions section.

It should be emphasized that comparisons of these results to others such as CBECS (EIA 2003) or other case studies is not straightforward or recommended due to the following issues:

- Cooling – the cooling energy analysis in this report uses sensible cooling at floor and space level (not cooling coil or secondary cooling level); without considering the cooling contributed by the DOAS system (including virtually all of the latent cooling) results will not be comparable to other cases.
- Heating – UFAD systems significantly reduce reheat energy, so comparison to non-UFAD buildings will not constitute a valid comparison.
- Internal loads – plug loads may differ markedly from other buildings since only one floor was measured and ancillary loads such as server rooms, elevators, etc. are not included. Lighting loads due not include exterior lighting and task lighting appears under plug loads.
- Energy totals – the results from this study are only for a four month, mid-season period, care should be exercised when comparing to annual results from other studies.

Uncertainties

A detailed analysis of uncertainties is beyond the scope of this project, but the following lists a number of issues that most likely contribute in some way to the errors and could be investigated further during a full calibration study.

- Measured values come from two sources, BMS data and the wireless monitoring system.

The wireless sensors were calibrated except for power meters which we assume to be accurate. The BMS sensors appear to have been checked (but not calibrated in detail) during commissioning. We assume the VFD variables are also accurate. Overall, we believe the measurements of variables of primary significance are reasonably accurate, with one exception; the water flow meter for chilled water flow was suspect due to wide scatter and inability to rationalize with other checking calculations (see Figure 16c). However, the flow meter data was not used in our final analysis.

- Service core airflow was not measured or estimated which results in more uncertainty on the measurement side; i.e., this represents airflow unaccounted for in the comparison which would bias the measured fan energy and cooling air flow toward the high side (overestimating)
- The actual fan curve from which the AHU airflow was derived could not be obtained. We assume they were derived from model testing of a given class of fans so there are inherent errors of unknown magnitude. We have increased confidence in the AHU model since we obtained two sets of spot measurements of AHU airflow and static pressure rise across the fan during February 2013 using a handheld anemometer and digital manometer. Likewise, the fan power peak operating point is reasonably close to that measured during commissioning. As shown above, these measurements line up reasonably well with the fan performance curves, also showing that the fan operating curve is nearly cubic. Also, independent checks using estimated airflow from diffusers based on static pressure measurements compared to the derived fan curve from spot measurement s. Further investigation would be required to better understand the large error for fan power.

  The degree of confidence is highest for fan powered terminal units measurements; everything was measured with instruments of known calibrations that CBE provided. The resulting field measured fan efficiency is very comparable to other measurements CBE has made on these types of units.
- Simulated cooling loads rely heavily on solar gains and the interaction with the blinds; the net gains were derived from Radiance complex fenestration calculations. The LBNL team verified the Radiance model, but for the visual spectrum only. Still this provides some measure of confidence in the results. Assuming it is accurate implies that there may be other issues related to the modeling of cooling loads that would require more in depth study to uncover.

  The same can be said for heating; anecdotal evidence suggests that heating calculations in EnergyPlus may result in under predicting heating loads, especially for UFAD simulations using the air manager routines.

Review of assumptions and findings

There will always be a certain degree of difference between measured and simulated performance due to assumptions made in simulation and aligning those with other assumptions made in processing the measured data and to account for real differences between the model and actual building operation as discussed in the following. Embedded in these items are tips for adjustments that could be made during a full-calibration study.

- Zone aggregation – aggregation of the real zones are not perfectly aligned with the simulation zones. The differences in operation between perimeter zones (some are cooling while others may be heating) in the real building are not captured in the simulation

because the zones are simulated on an exposure basis. On the other hand, the amount of total heating and cooling is captured in the measured data, which indicates differences between simulated and measured may exist and could only be resolved by using the actual zone layout and comparing on a zone by zone basis.

- Perimeter zones – although the interior walls of the perimeter zones in the EnergyPlus model are opaque (which is not reflective of the open plan space), Radiance modeled these without the interior walls so solar gains that penetrated into the interior zones were captured realistically. Also despite the fact that the setpoints for the perimeter were lower than for interior in both summer and winter mode, the actual occupied zone temperatures were similar as shown in Figures 11. The temperatures for the perimeter in winter were set higher than is typically done to maintain comfort; it is likely that the simultaneous cooling (from swirl diffusers, see below) caused this and the noticeable (see Figure 11b) over-cooling in the afternoon. This indicates that further work could be done on tuning the simulation model to account for these effects. (Perhaps EnergyPlus could be modified further to allow dual paths for airflow to the zone.)
- Interior diffusers in the perimeter zone – the perimeter zone model does not include swirl diffusers that exist in the building (and are controlled by the interior zone thermostats); effects were estimated using an offset (excess cooling) in the processing of the measurement results. However, the heat transfer impacts of simultaneous heating and cooling may have secondary effects that may be difficult to model.
- Impact of blinds – past research has shown that blinds can affect the stratification level in perimeter zones; this effect was not modeled since the perimeter zones are nearly always mixed due to the high throw diffusers being used. However, the blinds were assumed to be always down in the simulations; the impact on heating and cooling loads (e.g., transmission losses/gains) is a factor that could be studied in a full calibration effort.
- Weather files match up – although a local actual weather file was used the local conditions at the building could have varied.
- Controls – EnergyPlus simulates exact control while real controls do not; several cases of controls hunting and other anomalies (e.g., room setpoints) were observed. Also, it is unknown how operators modified controls settings during the monitoring period, although we assume it was small since they were asked not to.
- Infiltration assumed – infiltration rates were assumed to be nominal values typically used in these types of simulations; there was no investigation as to what they might be based on the construction characteristics of the Times building.
- Stratification (interior) - As shown by Figure 12, the actual zone temperatures measured by the UFAD cart were used to model stratification and zone temperatures which biases the simulated cooling energy high; i.e., if actual setpoints and thermostat readings were used, the simulated airflows and cooling loads would be lower causing further discrepancy between measured values. Better matching between the actual temperatures and their hourly variation could be achieved using scheduled setpoints. Since the simulated stratification appears less than measured, stratification modeling could be improved by modifying the number of diffusers along with using scheduled setpoints. The estimated number of open diffusers could also be improved with a detailed survey in the actual building.

Lessons learned

- One of the most significant obstacles encountered in this study was the inability to properly measure (and simulate) the DOAS system. This indicates that more time be spent up front to fully understand the design and operation of the system(s) before the measurement list is finalized.
- The analysis of cooling loads at the cooling coil was complicated by a lack of data about the operation of the DOAS system. If this data was available, it would serve as a secondary check on the net cooling load calculations.
- It became obvious during this study that measurements of humidity should be made so that comparisons can be made on an enthalpy basis as opposed to a sensible energy basis; but given that the cooling coil appears to run dry, this is assumed to be a minor problem in this instance.
- Accurate fan airflow is critical to an analysis of this sort; extra effort should be made to monitor airflow directly after suitable calibration of an airflow sensor. Likewise, fan static pressure across the fan should be made directly.
- Studies of this detail (and importance) should be adequately funded and staffed to allow more detailed analyses to be made.

SUMMARY AND CONCLUSIONS

Although considerable effort was expended to attend to the details of end use models on the measurement and simulation side, the results were not considered to be close enough to qualify as a "calibrated simulation." The most effort went into ensuring the internal loads were accurately input (solar and internal loads) and that HVAC models were well represented based on field measurements. The final results comparing monthly measured vs. simulate energy use at the basic floor level loads shows an average ~20% difference between measured and simulated for cooling energy. This difference results in an average fan energy difference of ~50% due to the fan power cubic effect. Heating showed ~50% average difference. However, considering that this study is just the first phase of a more detailed calibration, these results are to be expected. A next phase would investigate problems with loads modeling for cooling and especially heating with consideration of the uncertainties outlined above. Resolution of the discrepancies would require parametric simulations and parameter identification methods to bring the results into compliance with Guideline 14 standards.

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APPENDIX A – MODELING SPECIFICATIONS (NOT INCLUDING THE RADIANCE-ENERGYPLUS LINK FOR SOLAR GAINS CALCULATIONS)

ENVELOPE AND INTERNAL SURFACES THERMAL PROPERTY DEFINITIONS

Tables A1-A6 describe the constructions and materials definitions. Note that most of these were taken directly from the NaturalWorks model.

Table A1: Construction layer definitions

	Outermost layer			
Exterior Wall	Aluminum 1mm	Mineral Wool 2in	Aluminum 1mm	
Exterior Roof / Floor	HW Concrete 12cm	FireProof		
Interior Ceiling	Plasterboard 12mm			
Floor (over UFAD plenum)	Steel 1mm	Air 3.8cm	Steel 1mm	Carpet
Interior Wall	Plasterboard 12mm	Mineral Wool 3in	Plasterboard 12mm	
Window	VE-2M on Starphire 5.7mm	WinGas Air 12.7mm	Starphire 5.7mm	Mechoshade

Table A2: Massive opaque material definitions

	Plaster-board 12mm	Steel 1mm	Alum-inum 1mm	HW Concrete 12cm	Mineral Wool 2in	Mineral Wool 3in	Mineral Wool 20cm
Roughness	Smooth	Smooth	Smooth	Medium Rough	Very Rough	Very Rough	Very Rough
Thickness {m}	0.012	0.001	0.001	0.12	0.0508	0.0762	0.2
Conductivity {W/m-K}	0.16	44.96	221.38	1.72	0.047	0.047	0.047
Density {kg/m3}	784.9	7688.86	2739.15	2242.58	50	50	50
Specific Heat {J/kg-K}	830	410	890	830	837	837	837
Thermal Absorptance	0.9	0.9	0.9	0.9	0.9	0.9	0.9
Solar Absorptance	0.15	0.2	0.2	0.65	0.15	0.15	0.15
Visible Absorptance	0.15	0.2	0.2	0.65	0.15	0.15	0.15

Table A3: No-mass opaque material definitions

	FireProof	Carpet

Roughness	Rough	Rough
Thermal Resistance {m2-K/W}	0.175	0.217
Thermal Absorptance	0.9	0.9
Solar Absorptance	0.6	0.6
Visible Absorptance	0.6	0.6

Table A4: Glazing material definitions (Note: thermal properties only, solar properties overridden by Radiance)

	VE-2M on Starphire 5.7mm	Starphire 5.7mm
Thickness {m}	0.0057	0.0057
Infrared Transmittance at Normal Incidence	0	0
Front Side Infrared Hemispherical Emissivity	0.84	0.84
Back Side Infrared Hemispherical Emissivity	0.04	0.84
Conductivity {W/m-K}	1	1

Table A5: Interior shading material definitions (Note: thermal properties only, solar properties overridden by Radiance)

	Mechoshade
Thermal hemispherical emissivity	0.9
Thermal transmittance	0.04
Thickness {m}	0.002
Conductivity {W/m-K}	0.5
Shade-to-glass distance {m}	0.1
Top opening multiplier	0.5
Bottom opening multiplier	0.5
Left-side opening multiplier	0.5
Right-side opening multiplier	0.5
Air flow permeability	0.03

Table A6: Air gap definitions

	Air - 3.8cm	WinGas Air 12.7mm
EnergyPlus material type	Material:AirGap	WindowMaterial:Gas
Gas Type		AIR
Thickness {m}		0.0127
Thermal Resistance {m2-K/W}	0.165	

INTERNAL LOADS AND SCHEDULES

For each of the internal loads (lights, plugs and people), EnergyPlus requires a multiplier (e.g. EPD,

LPD) (representing the design or peak load) and a relative schedule (usually fractions, between 0-1, but does necessarily have to be, can be greater than 1). The multiplier and relative schedule go hand in hand, one is meaningless without the other. For simplicity when dealing with monitored data, we can set the multiplier to 1, making the schedule carry the full meaning. The 'schedule' is not repeated in this model, it is specified hourly (or every 15-minutes) for the entire length of the simulation, except for auxiliary lighting input (see below).

Lighting

Lighting power consumption has been monitored for every lighting zone for the full calendar year of 2011, and has been organized into average values for each 15-minute increment.

The mapping between the lighting power in each lighting zone and the lighting power attributed to each thermal zone in the EnergyPlus model is as follows: The empirical lighting power data was split according to the new thermal zones. A table of ratios representing this correspondence was created that assigns different portions of the lighting zones to the thermal zones; EnergyPlus schedules were created with the result of this mapping.

Note that EnergyPlus has various ways of reading schedules. The simplest is through the Schedule:File object, but this only allows for hour increments. For a 15-minute schedule, it needs to be written as either Schedule:Compact object, or as a collection of Schedule:Day, Schedule:Week and Schedule:Year objects. Python scripts were written to convert 15-min csv data into text files with the necessary syntax for EnergyPlus, using both of the possible ways. The Schedule:Compact approach uncovered a bug in the EnergyPlus implementation – it is being fixed for the next release – that will not allow this object's definition to be long enough to include 15-minute increments for a full year. So the 15-minute schedules are included in the EnergyPlus file as a collection of Schedule:Day, Schedule:Week and Schedule:Year objects (in very long text files).

Auxiliary Lighting	• For lighting other than the main lighting system described above, there is service core lighting and perimeter troffer lighting that are not included in the detailed lighting schedules. These were input using the standard schedules objects using hourly schedules derived from lighting panel field measurements profile; the estimated LPD used was iterated to match the measured monthly energy use. • Main lighting is ceiling fixtures with 50% lights to return and was modeled as such.

Plug loads

Plug loads derived from field measurements and plug load measured profile; EPD iterated to achieve measured monthly energy use, and apportioned to the zones on an area basis.

Occupancy

The measured plug load schedule was used for occupancy and peak occupancy values were based on correspondence with the building operations personnel.

HVAC CONFIGURATION

The UFAD system is modeled as a set of connected thermal zones. Supply air from the air handling unit is sent to the core underfloor plenum zone, from there it travels out to the perimeter

underfloor plenum zones and up into the office zones and finally into the overhead plenum before returning to the air handling unit. The air handling consists of an outside air mixing box, a supply fan, and a cooling coil. In reality the coils are fed by water-side loops that include many other floors' coils and the plant equipment of chillers (with condenser water loops and cooling towers) and boilers. Plant models for chilled water and hot water were used with efficiency values close to those specified for the Times plant equipment.

Air paths

The section illustration in Figure A1 and the line diagram in Figure A2 show the air flow paths, with the underfloor plenum and office zones in series.

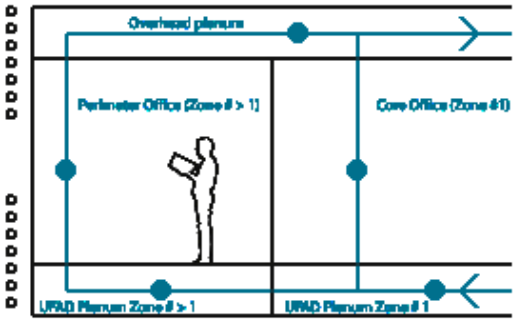

Figure A1: Diagram of UFAD airflow path

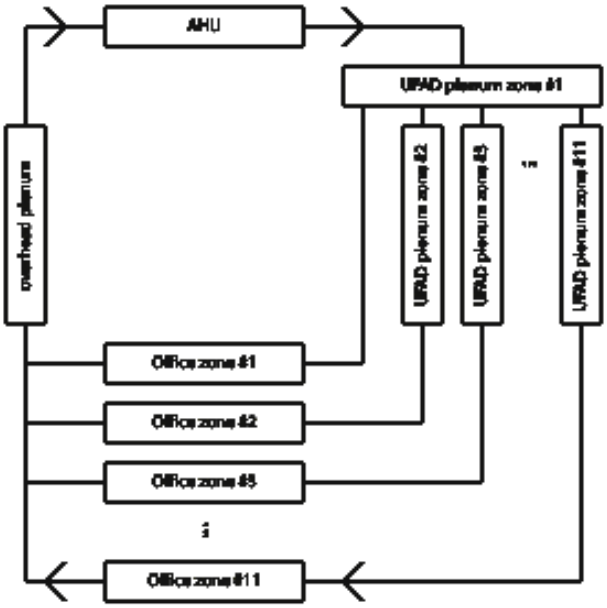

Figure A2: Air paths through UFAD zones, office zones and overhead plenum

Air handler and water side

Figure A3 illustrates the air handler configuration. Tables A7 and A8 list details of the supply fan and cooling coil descriptions. There was no heating coil since the actual system did not have one. The mixing box uses a fixed outside air rate (zero or minimum) but the real dedicated outside air system used in the Times system was not modeled. (See discussion of finding sections.)

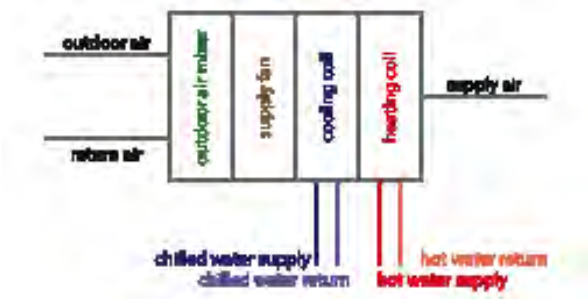

Figure A3: Air handler configuration

Air handlers (AHU) and Fan powered boxes (FPB)	• Central AHU: 70% efficiency, measured part load curves • FPB; 15% efficient fans with cubic part load curves derived from measured data • No economizer

Cooling	Central chilled water system. (This system was added to allow servicing of the heating and cooling loads; the chiller end use energy was not used.)
Heating	• Higher discharge temperatures (110°F) due to heating from floor • Minimum volume set at 30% based on design specs • UFAD system has no heating in interior zones.
Sizing	• All sizing based on design or measured data

Controls

Controls settings	• Zone thermostats: See Table 3 • AHU SAT: 66°F, no reset • Min VAV box settings: ▪ Interior – 30% ▪ Perimeter – 30% minimum and "dual max" heating @ 72% of max cooling • Schedules; system, thermostats and internal loads; derived from operator input and plug/lighting load profiles: ▪ System/thermostats: Weekdays, 6am-7pm; Saturday, 7am-5pm; Sunday, 8am-10pm

WEATHER FILE

The weather file used in simulations is a 2011 actual year epw weather file for New York City. The file was downloaded using the proprietary online service from weatheranalytics.com.

APPENDIX B – SOLAR GAINS CALCULATIONS – COMBINING RADIANCE AND ENERGYPLUS

As noted, there was a previously-constructed Radiance model of the 20th floor that the research team could use to accurately model the hourly solar gains on each of the window layers, floor surfaces and zone furniture in the EnergyPlus model. This allows for much more detailed consideration of the external and internal shades and the furniture geometry and reflectivities. The areas associated with each of the surfaces in the EnergyPlus model were mapped onto the Radiance model, and the same hourly weather file (based on actual measured weather data) was used in the Radiance annual calculations as in the EnergyPlus calculations. An image of the interior of the Radiance model is shown in Figure B-1.

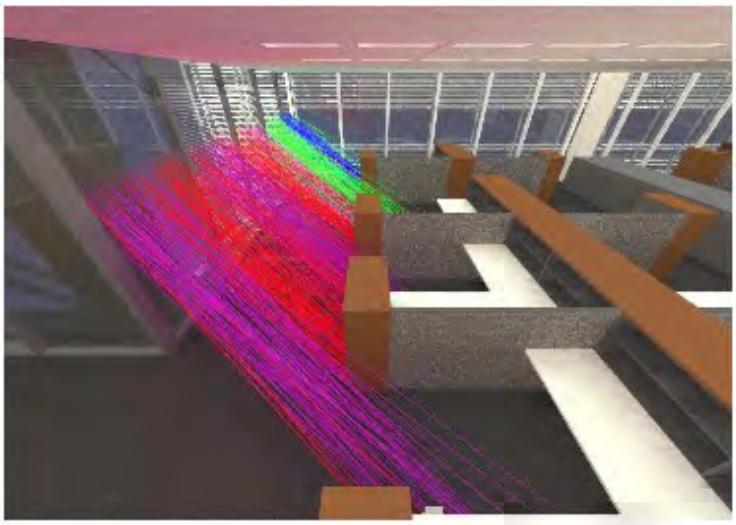

Figure B-1. An interior view of NY Times Radiance model, copied from Fig 5 of an internal report by Nouidui et al.). "There are four colors denoting rays that strike either the floor or furniture in the core or perimeter zones."

The methods whereby this Radiance-to-EnergyPlus connection works are described in detail in an internal report by Nouidui et al (2011). It essentially amounts to a small change in the EnergyPlus source code to associate schedules to the fenestration surface and opaque surface objects and to overwrite the solar gains values in the surface heat balance subroutines. Figure B-2 shows a diagram of the heat gains components in a surface heat balance calculation for an inside surface of an opaque wall (the components for the outside surface and for each layer in a window are similar). Scheduled values, taken from the Radiance outputs, are used to overwrite whatever EnergyPlus solar calculations do for the "shortwave radiation from solar sources" (but not internal sources) component in Figure 8.

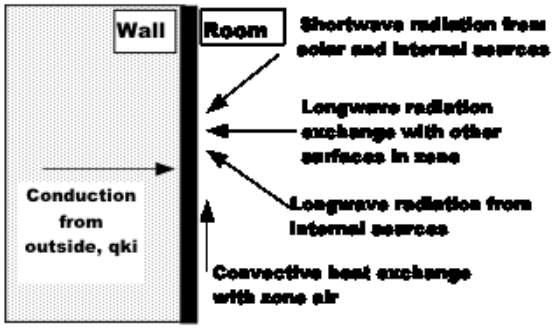

Figure B-2. Inside heat balance control volume diagram, copied from Fig 16 (p. 61) of EnergyPlus Engineering Reference (EnergyPlus 2010)

Output from the Radiance annual simulations are hourly schedules of the solar gains. These are formatted as csv files and read by EnergyPlus through Schedule:File objects, and referenced in the model's FenestrationSurface:Detailed and BuildingSurface:Detailed objects. The furniture gains calculated by Radiance are also formatted as csv files and read by EnergyPlus through Schedule:File objects. These schedules are used in the EnergyPlus model as internal sensible heat gains directly to the zone air node through OtherEquipment objects.

Note that the annual Radiance simulations include the measured shade positions. Each of the shade groups can be at one of any five height levels, and can vary at sub-hourly intervals. This model takes the average shade height over the hour, rounded to the nearest full position.

Notes on interior shades

The interior shades are included in the EnergyPlus model to capture their thermal impact. They are one of the layers on the window model that receives solar gains through the schedules from Radiance. Their hourly position in the EnergyPlus model thus does not affect the solar gains (as long as they are not completely up when there is solar gain scheduled onto them). It does, however, affect the IR transmission, convection heat transfer from the window to the zone, and may affect zone air stratification in the room air model (using CBE's code for this, if the monitored data shows any stratification, which it may not). They are currently scheduled to be always down in the EnergyPlus model. Schedules can be made based on the measured values, using average hourly shade heights (averaged across a surface, which may include multiple shade groups or the surface definitions can be changed so that there are multiple window surfaces per façade to line up with the shade groups.) This would also require changing the window surfaces from being a single floor-to-ceiling surface to being five surfaces floor-to-ceiling, each with a deployable shade – EnergyPlus shade controls allow them to be only fully up or fully down. Doing so would split the intra-window convective loop and thus decrease the calculated heat transfer between panes. The trade-off is thus between this decrease in accuracy and the decreases in accuracy in the simpler alternative of keeping the shades always down in the model – those being inaccuracies in the convective transfer from window to zone and in the IR calculations. Neither configuration is ideal; the much simpler option of 'always down' was kept for this study.

Glazing/solar gains	• WWR 76%, SHGC 0.30, U-value 2.43 W/m^2-°K , Tvis=0.53 • Solar gains derived from Radiance solar gain model impinging on all internal surfaces
Shading	• Outdoor scrim/ shade • Indoor automated roller shades • Neighboring buildings

APPENDIX C – WIRELESS MONITORING SYSTEM

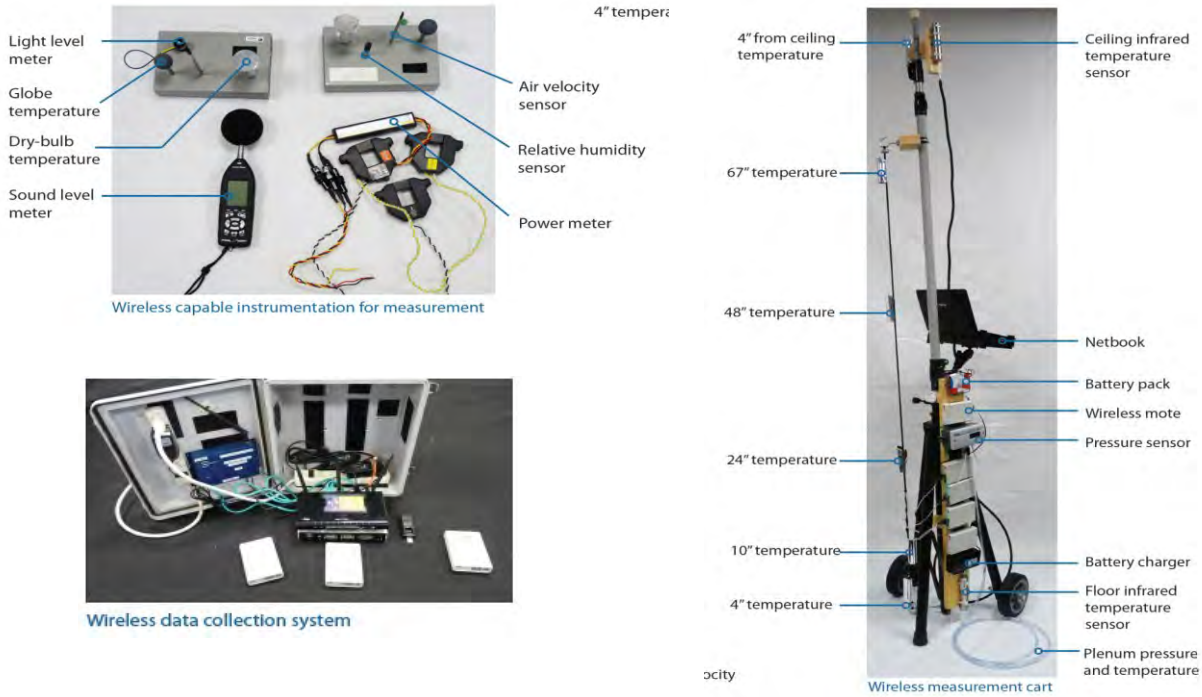

Table B-1: Wireless monitoring system measurements sensor list

Sensor	Device	location
IRT ceiling	Mobile cart, stratification	cart
4" from ceiling	Mobile cart, stratification	cart
pressure	Mobile cart, stratification	cart
SAT	Mobile cart, stratification	cart
67"	Mobile cart, stratification	cart
24"	Mobile cart, stratification	cart
48"	Mobile cart, stratification	cart
4"	Mobile cart, stratification	cart
IRT floor	Mobile cart, stratification	cart
10"	Fixed, stratification	cart
84"	Fixed, stratification-1	W20
67"	Fixed, stratification-1	W20
24"	Fixed, stratification-1	W20

Sensor	Device	location
4"	Fixed, stratification-1	W20
67"	Fixed, stratification-2	AE17
48"	Fixed, stratification-2	AE17
24"	Fixed, stratification-2	AE17
4"	Fixed, stratification-2	AE17
67"	Fixed, stratification-3	O2
48"	Fixed, stratification-3	O2
24"	Fixed, stratification-3	O2
4"	Fixed, stratification-3	O2
24"	Fixed, stratification-4	O6
4"	Fixed, stratification-4	O6
84"	Fixed, stratification-4	O6
67"	Fixed, stratification-4	O6
67"	Fixed, stratification-5	A22
48"	Fixed, stratification-5	A22
24"	Fixed, stratification-5	A22
4"	Fixed, stratification-5	A22
84"	Fixed, stratification-6	H20
67"	Fixed, stratification-6	H20
24"	Fixed, stratification-6	H20
4"	Fixed, stratification-6	H20
67"	Fixed, stratification-7	O37
48"	Fixed, stratification-7	O37
24"	Fixed, stratification-7	O37
4"	Fixed, stratification-7	O37
67"	Fixed, stratification-8	O34
48"	Fixed, stratification-8	O34
24"	Fixed, stratification-8	O34
4"	Fixed, stratification-8	O34
diffuser	Mote, temperature	C18
diffuser	Mote, temperature	B9
diffuser	Mote, temperature	C21
FPB14 inlet	Mote, temperature	AC11
chilled water return	Mote, temperature	
chilled water supply	Mote, temperature	
chilled water flow	Mote, Ulrtasonic flowmeter	
MPP-20	Mote, power meter, FPB	
P-20-1	Mote, power meter, lighting	
L-PP-20-1	Mote, power meter, plug loads	
P-20-2	Mote, power meter, lighting	
L-PP-20-2	Mote, power meter, plug loads	
perimeter grill SAT	Mote, temperature	AC33
FPB9,10	Mote, temperature	AB33
perimeter grill SAT	Mote, temperature	AE19
FPB12,13	Mote, temperature	AE20
perimeter grill SAT	Mote, temperature	AE8
FPB14,15	Mote, temperature	AE9

Sensor	Device	location
perimeter grill SAT	Mote, temperature	J4
FPB21,22	Mote, temperature	J3
perimeter grill SAT	Mote, temperature	A9
FPB24,25	Mote, temperature	A8
Air highway SAT east/central	Mote, temperature	X21
Air highway SAT south/west	Mote, temperature	H25
Air highway SAT northwest corner	Mote, temperature	I9
Air highway SAT north/east	Mote, temperature	X10
perimeter grill SAT	Mote, temperature	A24
FPB27,28	Mote, temperature	A25
perimeter grill SAT	Mote, temperature	M38
FPB5,6	Mote, temperature	L38
FPB12 inlet	Mote, temperature	AC19
FPB13 inlet	Mote, temperature	AC15
FPB15 inlet	Mote, temperature	AB9
FPB22 inlet	Mote, temperature	K7
FPB5 inlet	Mote, temperature	K34
FPB10 inlet	Mote, temperature	AB30
FPB6 inlet	Mote, temperature	Q36
diffuser	Mote, temperature	C30
Air highway SAT - private office, mote on damper	Mote, temperature	M30
diffuser	Mote, temperature	T32
diffuser	Mote, temperature	AB23
diffuser	Mote, temperature	T5
Air highway SAT mote on damper	Mote, temperature	G12
diffuser under table	Mote, temperature	Y12

APPENDIX D - BMS MONITORING POINTS

Table D-1. BMS monitored sensor list

Point Name	Units	Description
AHU01.COA	ppm	CO2 Outside air
AHU01.CRA	ppm	CO2 Return air
AHU01.CRA.DIFF	ppm	CO2 difference between return air and outside air
AHU01.CRA.SP	ppm	CO2 Return setpoint
AHU01.DE	0 = Off, 1 = On	Exhaust damper
AHU01.DP	0 = Off, 1 = On	Purge damper
AHU01.DSP	iwc	Static pressure (lower of DSP.N and DSP.S)
AHU01.DSP.N	iwc	Static pressure north
AHU01.DSP.S	iwc	Static pressure south
AHU01.DSP.SP	iwc	Static pressure setpoint
AHU01.ES.STS		
AHU01.FPB.MODE		
AHU01.HR	%	Return humidity
AHU01.NGT.CLG.SP	F	Night cooling setpoint
AHU01.NGT.COOL	0 = Disabled, 1 = Enabled	Night cooling purge
AHU01.NGT.HEAT	0 = Disabled, 1 = Enabled	Night heating purge
AHU01.NGT.HTG.SP	F	Night heating setpoint
AHU01.OA.CAV.DMPR.CMD	%	Constant air volume outdoor air damper command
AHU01.OA.CAV.FLOW	cfm	Constant air volume outdoor air flow
AHU01.OA.CAV.FLOW.SP	cfm	Constant air volume outdoor air flow setpoint
AHU01.OA.CAV:AIR_VOLUME	cfm	Constant air volume outside air air volume (calculated based on MEC)
AHU01.OA.CAV:DMPR_COMD	%	Constant air volume outdoor air damper command
AHU01.OA.CAV:OCC.UNOCC	0 = UNOCC, 1 = OCC	
AHU01.OA.CAV:ROOM_TEMP	F	
AHU01.SAF.AF	0 = Off, 1 = On	Filter alarm
AHU01.SAF.HP	0 = Off, 1 = On	High pressure alarm
AHU01.SAF.VFD.AMPS	A	Supply air fan VFD amperage
AHU01.SAF.VFD.CMD.STS	0 = Stop, 1 = Start	Supply air fan VFD CMD status
AHU01.SAF.VFD.SPD	%	Supply air fan VFD speed CMD
AHU01.SAF.VFD.SS	0 = Stop, 1 = Start	Supply air fan VFD start/stop

Point Name	Units	Description
AHU01.TM	F	Mixed air temperature
AHU01.TR	F	Return air temperature
AHU01.TS	F	Supply air temperature
AHU01.TS.SP	F	Supply air temperature setpoint
AHU01.Z01.D	%	Zebra Damper
AHU01.Z01.DSP	iwc	Static Pressure
AHU01.Z01.DSP.SP	iwc	Static pressure setpoint
AHU01.Z01.ENABLE	0 = Disabled, 1 = Enabled	Zebra damper control
AHU01.Z01.TH	F	Zone Temperature
AHU01.Z01.TH:OFFSET	F	Zone temperature offset (THV - TH)
AHU01.Z01.THV	F	Zone overcall temperature
AHU01.Z02.D	%	Zebra Damper
AHU01.Z02.DSP	iwc	Static Pressure
AHU01.Z02.DSP.SP	iwc	Static pressure setpoint
AHU01.Z02.ENABLE	0 = Disabled, 1 = Enabled	Zebra damper control
AHU01.Z02.TH	F	Zone Temperature
AHU01.Z02.TH:OFFSET	F	Zone temperature offset (THV - TH)
AHU01.Z02.THV	F	Zone overcall temperature
AHU01.Z03.D	%	Zebra Damper
AHU01.Z03.DSP	iwc	Static Pressure
AHU01.Z03.DSP.SP	iwc	Static pressure setpoint
AHU01.Z03.ENABLE	0 = Disabled, 1 = Enabled	Zebra damper control
AHU01.Z03.TH	F	Zone Temperature
AHU01.Z04.D	%	Zebra Damper
AHU01.Z04.DSP	iwc	Static Pressure
AHU01.Z04.DSP.SP	iwc	Static pressure setpoint
AHU01.Z04.ENABLE	0 = Disabled, 1 = Enabled	Zebra damper control
AHU01.Z04.TH	F	Zone Temperature
AHU01.Z05.D	%	Zebra Damper
AHU01.Z05.DSP	iwc	Static Pressure
AHU01.Z05.DSP.SP	iwc	Static pressure setpoint
AHU01.Z05.ENABLE	0 = Disabled, 1 = Enabled	Zebra damper control
AHU01.Z05.TH	F	Zone Temperature
AHU01.Z06.D	%	Zebra Damper
AHU01.Z06.DSP	iwc	Static Pressure
AHU01.Z06.DSP.SP	iwc	Static pressure setpoint
AHU01.Z06.ENABLE	0 = Disabled, 1 = Enabled	Zebra damper control
AHU01.Z06.TH	F	Zone Temperature
AHU01.Z07.D1	%	Zebra Damper

Point Name	Units	Description
AHU01.Z07.D2	%	Zebra Damper
AHU01.Z07.DSP.SP	iwc	Static pressure setpoint
AHU01.Z07.TZ	F	Zone Temperature (not overcall)
AHU01.Z07.TZ.SP	F	Zone temperature setpoint
AHU01.Z08.D	%	Zebra Damper
AHU01.Z08.ENABLE	0 = Disabled, 1 = Enabled	Zebra damper control
AHU01.Z08.TZ	F	Zone Temperature (not overcall)
AHU01.Z08.TZ.SP	F	Zone temperature setpoint
BF01:AIR_VOLUME	cfm	Air flow rate
BF01:CTL_STPT	F	Control setpoint
BF01:CTL_TEMP	F	Control temperature
BF01:FAN	0 = Off, 1 = On	Fan on/off
BF01:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
BF01:HEAT.COOL	0 = Heat, 1 = Cool	Booster fan heat or cool mode
BF01:HTG_VLV1_POS	%	Heating valve position
BF01:ROOM_TEMP	F	Room temperature
FPB01:AIR_VOLUME	cfm	Air flow rate
FPB01:CTL_STPT	F	Control setpoint
FPB01:CTL_TEMP	F	Control temperature
FPB01:FAN	0 = Off, 1 = On	Fan on/off
FPB01:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB01:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB01:HTG_VLV1_POS	%	Heating valve position
FPB01:ROOM_TEMP	F	Room temperature
FPB02:AIR_VOLUME	cfm	Air flow rate
FPB02:CTL_STPT	F	Control setpoint
FPB02:CTL_TEMP	F	Control temperature
FPB02:FAN	0 = Off, 1 = On	Fan on/off
FPB02:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB02:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB02:HTG_VLV1_POS	%	Heating valve position
FPB02:ROOM_TEMP	F	Room temperature
FPB03:AIR_VOLUME	cfm	Air flow rate
FPB03:CTL_STPT	F	Control setpoint
FPB03:CTL_TEMP	F	Control temperature
FPB03:FAN	0 = Off, 1 = On	Fan on/off
FPB03:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB03:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB03:HTG_VLV1_POS	%	Heating valve position

Point Name	Units	Description
FPB03:ROOM_TEMP	F	Room temperature
FPB04:AIR_VOLUME	cfm	Air flow rate
FPB04:CTL_STPT	F	Control setpoint
FPB04:CTL_TEMP	F	Control temperature
FPB04:FAN	0 = Off, 1 = On	Fan on/off
FPB04:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB04:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB04:HTG_VLV1_POS	%	Heating valve position
FPB04:RMTEMPV	F	Room temperature (overcall temperature)
FPB04:ROOM_TEMP	F	Room temperature
FPB05:AIR_VOLUME	cfm	Air flow rate
FPB05:CTL_STPT	F	Control setpoint
FPB05:CTL_TEMP	F	Control temperature
FPB05:FAN	0 = Off, 1 = On	Fan on/off
FPB05:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB05:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB05:HTG_VLV1_POS	%	Heating valve position
FPB05:ROOM_TEMP	F	Room temperature
FPB06:AIR_VOLUME	cfm	Air flow rate
FPB06:CTL_STPT	F	Control setpoint
FPB06:CTL_TEMP	F	Control temperature
FPB06:FAN	0 = Off, 1 = On	Fan on/off
FPB06:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB06:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB06:HTG_VLV1_POS	%	Heating valve position
FPB06:RMTEMPV	F	Room temperature (overcall temperature)
FPB06:ROOM_TEMP	F	Room temperature
FPB07:AIR_VOLUME	cfm	Air flow rate
FPB07:CTL_STPT	F	Control setpoint
FPB07:CTL_TEMP	F	Control temperature
FPB07:FAN	0 = Off, 1 = On	Fan on/off
FPB07:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB07:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB07:HTG_VLV1_POS	%	Heating valve position
FPB07:ROOM_TEMP	F	Room temperature
FPB08:AIR_VOLUME	cfm	Air flow rate
FPB08:CTL_STPT	F	Control setpoint
FPB08:CTL_TEMP	F	Control temperature
FPB08:FAN	0 = Off, 1 = On	Fan on/off
FPB08:FAN_AOV1	volts (0-10)	Analog output voltage, FPB

Point Name	Units	Description
FPB08:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB08:HTG_VLV1_POS	%	Heating valve position
FPB08:RMTEMPV	F	Room temperature (overcall temperature)
FPB08:ROOM_TEMP	F	Room temperature
FPB09:AIR_VOLUME	cfm	Air flow rate
FPB09:CTL_TEMP	F	Control setpoint
FPB09:FAN	F	Control temperature
FPB09:FAN_AOV1	0 = Off, 1 = On	Fan on/off
FPB09:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB09:HTG_VLV1_POS	volts (0-10)	Analog output voltage, FPB
FPB10:AIR_VOLUME	cfm	Air flow rate
FPB10:CTL_STPT	F	Control setpoint
FPB10:CTL_TEMP	F	Control temperature
FPB10:FAN	0 = Off, 1 = On	Fan on/off
FPB10:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB10:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB10:HTG_VLV1_POS	%	Heating valve position
FPB10:RMTEMPV	F	Room temperature (overcall temperature)
FPB10:ROOM_TEMP	F	Room temperature
FPB11:AIR_VOLUME	cfm	Air flow rate
FPB11:CTL_STPT	F	Control setpoint
FPB11:CTL_TEMP	F	Control temperature
FPB11:FAN	0 = Off, 1 = On	Fan on/off
FPB11:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB11:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB11:HTG_VLV1_POS	%	Heating valve position
FPB11:ROOM_TEMP	F	Room temperature
FPB12:AIR_VOLUME	cfm	Air flow rate
FPB12:CTL_STPT	F	Room temperature
FPB12:CTL_TEMP	F	Control temperature
FPB12:FAN	0 = Off, 1 = On	Fan on/off
FPB12:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB12:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB12:HTG_VLV1_POS	%	Heating valve position
FPB12:RMTEMPV	F	Room temperature (overcall temperature)
FPB12:ROOM_TEMP	F	Room temperature
FPB13:CTL_STPT	F	Room temperature
FPB13:CTL_TEMP	F	Control temperature
FPB13:FAN	0 = Off, 1 = On	Fan on/off

Point Name	Units	Description
FPB13:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB13:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB13:HTG_VLV1_POS	%	Heating valve position
FPB13:RMTEMPV	F	Room temperature (overcall temperature)
FPB13:ROOM_TEMP	F	Room temperature
FPB14:CAL_AIR	0 = Off, 1 = On	Put FPB into airflow calibration
FPB14:CTL_STPT	F	Room temperature
FPB14:CTL_TEMP	F	Control temperature
FPB14:FAN	0 = Off, 1 = On	Fan on/off
FPB14:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB14:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB14:HTG_VLV1_POS	%	Heating valve position
FPB14:RMTEMPV	F	Room temperature (overcall temperature)
FPB14:ROOM_TEMP	F	Room temperature
FPB15:AIR_VOLUME	cfm	Air flow rate
FPB15:CTL_STPT	F	Control setpoint
FPB15:CTL_TEMP	F	Control temperature
FPB15:FAN	0 = Off, 1 = On	Fan on/off
FPB15:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB15:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB15:HTG_VLV1_POS	%	Heating valve position
FPB15:ROOM_TEMP	F	Room temperature
FPB16:AIR_VOLUME	cfm	Air flow rate
FPB16:CTL_STPT	F	Control setpoint
FPB16:CTL_TEMP	F	Control temperature
FPB16:FAN	0 = Off, 1 = On	Fan on/off
FPB16:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB16:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB16:HTG_VLV1_POS	%	Heating valve position
FPB16:ROOM_TEMP	F	Room temperature
FPB17:AIR_VOLUME	cfm	Air flow rate
FPB17:CTL_STPT	F	Control setpoint
FPB17:CTL_TEMP	F	Control temperature
FPB17:FAN	0 = Off, 1 = On	Fan on/off
FPB17:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB17:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB17:HTG_VLV1_POS	%	Heating valve position
FPB17:RMTEMPV	F	Room temperature (overcall temperature)
FPB17:ROOM_TEMP	F	Room temperature

Point Name	Units	Description
FPB18:AIR_VOLUME	cfm	Air flow rate
FPB18:CTL_STPT	F	Control setpoint
FPB18:CTL_TEMP	F	Control temperature
FPB18:FAN	0 = Off, 1 = On	Fan on/off
FPB18:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB18:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB18:HTG_VLV1_POS	%	Heating valve position
FPB18:ROOM_TEMP	F	Room temperature
FPB19:AIR_VOLUME	cfm	Air flow rate
FPB19:CTL_STPT	F	Control setpoint
FPB19:CTL_TEMP	F	Control temperature
FPB19:FAN	0 = Off, 1 = On	Fan on/off
FPB19:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB19:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB19:HTG_VLV1_POS	%	Heating valve position
FPB19:RMTEMPV	F	Room temperature (overcall temperature)
FPB19:ROOM_TEMP	F	Room temperature
FPB20.CAL.TIMER		
FPB20:AIR_VOLUME	cfm	Air flow rate
FPB20:CTL_STPT	F	Control setpoint
FPB20:CTL_TEMP	F	Control temperature
FPB20:FAN	0 = Off, 1 = On	Fan on/off
FPB20:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB20:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB20:HTG_VLV1_POS	%	Heating valve position
FPB20:ROOM_TEMP	F	Room temperature
FPB21:AIR_VOLUME	cfm	Air flow rate
FPB21:CTL_STPT	F	Control setpoint
FPB21:CTL_TEMP	F	Control temperature
FPB21:FAN	0 = Off, 1 = On	Fan on/off
FPB21:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB21:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB21:HTG_VLV1_POS	%	Heating valve position
FPB21:RMTEMPV	F	Room temperature (overcall temperature)
FPB21:ROOM_TEMP	F	Room temperature
FPB22:CTL_STPT	F	Control setpoint
FPB22:CTL_TEMP	F	Control temperature
FPB22:FAN	0 = Off, 1 = On	Fan on/off
FPB22:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB22:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode

Point Name	Units	Description
FPB22:HTG_VLV1_POS	%	Heating valve position
FPB22:ROOM_TEMP	F	Room temperature
FPB23:AIR_VOLUME	cfm	Air flow rate
FPB23:CTL_STPT	F	Control setpoint
FPB23:CTL_TEMP	F	Control temperature
FPB23:FAN	0 = Off, 1 = On	Fan on/off
FPB23:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB23:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB23:HTG_VLV1_POS	%	Heating valve position
FPB23:ROOM_TEMP	F	Room temperature
FPB24:AIR_VOLUME	cfm	Air flow rate
FPB24:FAN	0 = Off, 1 = On	Control setpoint
FPB24:FAN_AOV1	volts (0-10)	Control temperature
FPB24:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB24:HTG_VLV1_POS	%	Fan on/off
FPB24:RMTEMPV	F	Room temperature (overcall temperature)
FPB24:ROOM_TEMP	F	Analog output voltage, FPB
FPB25:AIR_VOLUME	cfm	Air flow rate
FPB25:CTL_STPT	F	Control setpoint
FPB25:CTL_TEMP	F	Control temperature
FPB25:FAN	0 = Off, 1 = On	Fan on/off
FPB25:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB25:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB25:HTG_VLV1_POS	%	Heating valve position
FPB25:RMTEMPV	F	Room temperature (overcall temperature)
FPB25:ROOM_TEMP	F	Room temperature
FPB26:AIR_VOLUME	cfm	Air flow rate
FPB26:CTL_STPT	F	Control setpoint
FPB26:CTL_TEMP	F	Control temperature
FPB26:FAN	0 = Off, 1 = On	Fan on/off
FPB26:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB26:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB26:HTG_VLV1_POS	%	Heating valve position
FPB26:RMTEMPV	F	Room temperature (overcall temperature)
FPB26:ROOM_TEMP	F	Room temperature
FPB27:AIR_VOLUME	cfm	Air flow rate
FPB27:CTL_STPT	F	Control setpoint
FPB27:CTL_TEMP	F	Control temperature
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Point Name	Units	Description
FPB27:FAN_AOV1	volts (0-10)	Analog output voltage, FPB
FPB27:HEAT.COOL	0 = Heat, 1 = Cool	Heating or cooling mode
FPB27:HTG_VLV1_POS	%	Heating valve position
FPB27:RMTEMPV	F	Room temperature (overcall temperature)
FPB27:ROOM_TEMP	F	Room temperature

Interim report April 2013

Radiant cooling system design and performance analysis

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April 29, 2013

ABSTRACT

Thermally activated building systems (TABS) are gaining popularity as a potentially energy efficient strategy for conditioning buildings. These systems can use large surfaces for heat exchange, and the temperature of the cooling water can be only a few degrees lower than the room air temperature. This small temperature difference allows the use of alternative cooling sources, for example, indirect/direct evaporative cooling, to possibly eliminate refrigerant cooling to reduce energy consumption. In addition, TABS allow the potential to reduce the electric power demand of the building if a night time precooling strategy is used. This research has investigated the application range of using slab-integrated hydronic radiant cooling (TABS) with a cooling tower providing chilled water as the primary way of conditioning the building. The objectives of this study are the following: 1) quantify the climatic limits of using evaporative cooling (cooling tower) for radiant ceiling slab system; 2) identify design options to expand the application; and 3) provide climate based advice for system design and operation.

INTRODUCTION

Thermally activated building systems (TABS) are gaining popularity as a potentially energy efficient strategy for conditioning buildings. These systems can use large surfaces for heat exchange, and the temperature of the cooling water can be only a few degrees lower than the room air temperature. This small temperature difference allows the use of alternative cooling sources, for example, indirect/direct evaporative cooling, to possibly eliminate refrigerant cooling to reduce energy consumption. In addition, TABS allow the potential to reduce the electric power demand of the building if a night time precooling strategy is used. This research has investigated the application range of using slab-integrated hydronic radiant cooling (TABS) with cooling tower providing chilled water as the primary way of conditioning the building. The objectives this study are the following: 1) quantify the climatic limits of using evaporative cooling (cooling tower) for

radiant ceiling slab system; 2) identify design options to expand the application; and 3) provide climate based advice for system design and operation.

We took the following approaches in this study: 1) survey of radiant system design experts to understand the current practices and design issues and limitations; 2) using energy simulation to evaluate the thermal comfort performance of the design.

RADIANT SYSTEM DESIGN SURVEY OF EXPERTS

In order for the study to best serve the interests of the design industry, we conducted a survey to get feedback from design practitioners, manufacturers, and top researchers who are experienced with radiant systems while we are defining our study scope. This survey served to provide practical design and control information and to ensure the simulation models were configured to represent design practice to the extent possible. It is our hope that the study results can be useful for a range of applications of radiant slab cooling systems.

The survey was designed to have two parts. The first part lists many of the key fixed system design parameters that we believe represent good design practice. We don't expect the simulation results to be highly sensitive to changes in these parameters and have therefore decided to keep them fixed during our simulation study. And the survey was used to make sure these assumptions are consistent with design practice.

The second part lists the important parameters that we focus our attention on and the proposed range over which they will vary. The goal was to understand the important system design and operational parameters.

Summarized below are some key findings from the survey.

Radiant system types and applications

- Compared to embedded radiant ceiling system, radiant floor system systems are the most common types, and people have more experience designing it successfully. One person mentioned that among all the embedded radiant projects, only 5% are radiant ceiling system. Another survey respondent said he usually designed radiant ceiling based system when the building is 5 stories and up, and design floor based system if it is under 5 stories.
- Radiant floor systems are a popular application in large rooms with high ceilings and when they can be used for the absorption of solar loads.
- TABS are used when it can be part of a common floor/ceiling system

Radiant system design specifications

- <u>Tube depth:</u> The depth of the hydronic tubing in the slab depends on construction and technique, and whether exploitation of the slab thermal inertia is considered. Also construction concerns are important: 2" depth of the tubes in the concrete is a code requirement in Canada to allow the minimum 1.5" concrete coverage to the reinforcing bars in the slab to meet fire ratings. When designing radiant ceiling system, the normal practice is to tie the tubes for a radiant ceiling system to the tops of the bottom layer of reinforcing bars.
- <u>Pipe diameter:</u> The tubing diameter is a function of the size of the radiant zones, floor plate size and economics: on smaller radiant slab systems 1/2" or 5/8" tubing is used, and there is no requirement for large zones. It is common to use 3/4" tubing for larger floor

plates for additional GPM and increased loop length; this can minimize the number of manifold cabinets and sizes of the manifold cabinets.

- Tube spacing: The spacing between tubing that is generally used is anywhere from 6” to 12” on center and this spacing is defined by the bend radius of the particular tube diameter being used, and what average slab surface temperature is desired, i.e., where the maximum cooling effect is desired, then tighter tube spacing is used to get a very consistent slab surface temperature, and where the cooling load/output is less critical and a minor amount of thermal striping is tolerable, then 12” spacing is still feasible for economic reasons.

Ventilation system design

- The minimum capacity of the ventilation system is usually determined by requirements of IAQ and humidity control, whichever is highest. If minimual ventilation is used, chilled beams are one of the alternatives to provide additional cooling in high load spaces.
- The capacities of the air system are often increased as supplemental cooling if radiant system capacity is not adequate. This is especially true for perimeter zones.

System operation

- Radiant loop water temperature differential range is normally between 3-5 °C.
- Pump operation: in the radiant circuit, constant flow is usually used. Pulse width modulation is optional that can minimize run-time on pumps. The slow response time of slabs allows for pump on/off operation based on slab temperature sensor setpoint deadband.
- Condensation control: dew point sensors are used for making sure that supply water temperature is control to be 1-2°C higher than dew point temperature in the space.

Plant design and operation

- A cooling tower can be used for supplying cold water under suitable climatic conditions. Ground source heat pumps are another alternative that is often considered. However, more conventional design is seen in practice because of concerns about the limited capacity of cooling towers and the first costs of installing ground source heat exchangers. A chiller is the most frequently used source for chilled water. One operational strategy mentioned is to use the same chiller to supply warmer temperature chilled water to pre-cool the slab at night, and during the day to treat the primary air with colder temperature chilled water if dehumidification is needed. This strategy is considered as economically feasible and still being able to exploit the energy efficiency potential of supplying warm water.
- One rule of thumb for operating cooling towers for precooling is when night time outdoor temperature during summer falls below 63°F (17°C) and the wet-bulb temperature is lower than 59°F (15°C) , the cooling tower at night is a viable cooling source for slab cooling.

Load control strategies

- Controlling (shading) solar heat gain is important in the success of a radiant cooling project.

CLIMATIC APPLICATION RANGE OF TABS WITH EVAPORATIVE COOLING SOURCE

EnergyPlus v7.2 was used for the simulation study in Sacramento, San Francisco, Phoenix, and Atlanta. For each climate zone studied, a single floor medium office-building was simulated. Minimum ventilation air was provided in the baseline model by a dedicated outdoor air system (DOAS) with proper humidity control. For some climates where the evaporative cooling + radiant cooling system alone may not be able to ensure thermal comfort for the hottest periods, we explored options such as expanding the thermal comfort zone by increasing air movement with personal fans or increasing the cooling capacity of the ventilation system. For some severe climates, such as Phoenix, other design options were investigated.

CLIMATE ANALYSIS

The four climates we selected for this study are San Francesco, Sacramento, Atlanta and Phoenix. They represent climatic conditions ranging from mild to hot and with humidity level ranging from dry to relatively humid. Figure 1 plots the annual dry-bulb temperature and wet-bulb temperature ranges for the four climates.

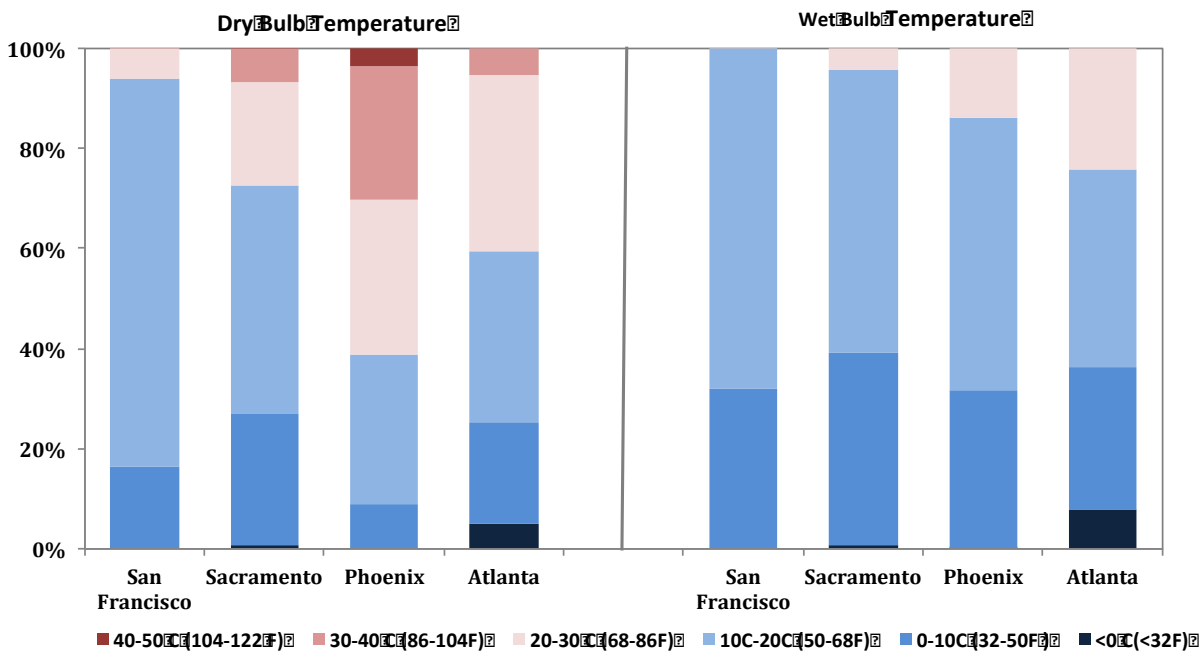


Figure 1: Dry-bulb and wet-bulb temperature ranges of the four selected climates

MODEL SPECIFICATIONS

For each climate, we developed a baseline EnergyPlus model, which is based on the prototype medium office buildings developed by DOE. The model envelope constructions are compliant with ASHRAE 90.1 2010 requirements (ASHRAE 2010a). One improvement in the prototype building was the shading systems. The survey results and literature study indicated that one key component of a successful TABS project is to control the solar heat gain. Since this study aims to evaluate the application potential of TABS integrated with evaporative cooling, shading systems

are designed to the extent possible to minimize direct solar heat gain. Table 1 summarizes the baseline building model specifications, and Table 2 summarizes the simulated baseline radiant system specifications.

Table 1: Model specifications

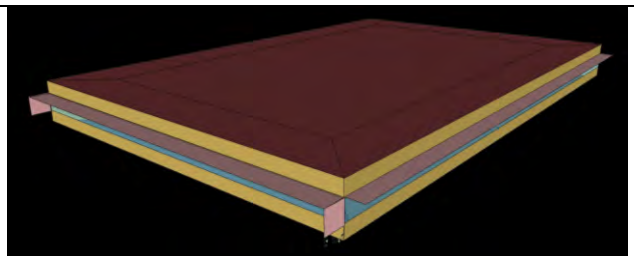
Items	Descriptions
Model image	
Building and Internal load	Single floor 5-zone model compliant with 90.1-2010
Solar control	Exterior fixed overhang for all façade, Exterior operable blind for west and east facade
Radiant system	Radiant ceiling TABS
Air system	Dedicated outdoor air unit with heat recovery
Cooling source	Cooling tower
Cold supply water control	Max (wetbulb temperature + 2°C, room dewpoint+1.5 °C)
Humidity control	Zone humidity ratio at 0.012 lb/lb
Radiant system operation	24 hours or when cooling tower is available

Table 2: Radiant system modelling specifications

Item	Description
System type	Concrete embedded radiant ceiling system
Tube depth	2” (0.0508 m) below concrete surface
Tube diameter	5/8” (0.0158m)
Tube spacing	6” (0.15m)
Radiant loop water temperature differential	5.4°F (3°C)

DESIGN AND CONTROL OPTIONS EVALUATED

For all climates, we started the analysis with the baseline model, and for some climates where the base design alone is not able to ensure thermal comfort for the hottest periods, we explored options such as expanding the thermal comfort zone by increasing air movement with personal fans, increasing the cooling capacity of the ventilation system, and alternative radiant cooling technology, i.e. lightweight embedded surface radiant cooling systems. Table 3 summarizes the design options evaluated.

Table 3: Evaluated system design and operating strategies

Notation	Strategies	Climates
Base	Radiant slab system + air system with design flow rate for ventilation and humidity control purposes	SF, Sac, Atl, Phx
Precooling only	Nighttime precooling only by utilizing thermal mass storage effect.	SF, Sac
Elevated Air	Enlarge thermal comfort zone by elevating air movement	Sac, Atl, Phx
EnhancedAirSys	Size of air system is increased to provide additional cooling capacity if radiant slab is inadequate	Sac, Atl, Phx
ESCS	Use lightweight embedded surface cooling systems instead of heavyweight TABS	Phx

Note: SF= San Francisco; Sac = Sacramento, Atl= Atlanta; Phx = Phoenix

Base design

Minimum air flow rate was provided in the base model by a dedicated outdoor air system (DOAS). The minimum air flow rate was determined by ventilation and humidity control requirements. DX coil was modeled for the DOAS system. Therefore, the base air system design has the capacity to supply air temperature as low as 55°F if necessary. To minimize energy consumption, enthalpy heat recovery system was also modeled. Evaporative cooling, i.e., cooling tower, was provided 24 hours per day when the following conditions were met: there was cooling demand from the space and the water provided by the cooling tower was more than 3°F lower than room air temperature. Condensation is very less likely to happen with the cold water supplied from a cooling tower and with dehumidification capability from the DOAS system.

Precooling-only strategy

In this design option, cold water is only available for 10 hours (from 8pm to 6am) at nighttime to pre-cool the slab. The night time cooling setpoint was set to 68°F, which means during the precooling period, the valves for each radiant slab zone were independently controlled to meet this set point. This strategy was always implemented together with other design options in the study.

Elevated air motion

The recently updated ASHRAE Standard 55 (ASHRAE 2010b) indicates that under warm conditions the thermal comfort zone can be enlarged with elevated air motion, which can be achieved via ceiling fans, personal fans or natural ventilation. The energy advantages of extending the thermal comfort zone has also been demonstrated (Hoyt et al. 2009). Since TABS + evaporative cooling are likely not able to guarantee stable indoor thermal conditions, we investigated the potential of using elevated air motion to enable the application of TABS.

Enhanced air system

In this design option, instead of supplying only minimum air for ventilation and humidity control, the cooling capacity of the air systems were increased to serve as supplemental cooling. The magnitude of enhancement depends on the amount of supplemental cooling needed for different climates.

Embedded surface cooling systems (ESCS)

According to the REHVA guidebook on radiant systems (2007), water-based embedded surface cooling systems (ESCS) are one kind of radiant systems with pipes embedded in plaster or gypsum board or cement screed, and they are thermally decoupled from the main building structure (floor, wall and ceiling) by the use of thermal insulation. They are used in all types of buildings and work with heat carriers at relatively high temperature for cooling. Compared to the TABS, which have pipes embedded in the building structure (slab, walls) to take advantage of the thermal storage capacity of the building structure, lightweight ESCS have less thermal storage potential, but can have slightly higher cooling capacity because the achievable slab surface temperature can be closer to water temperature due to smaller thermal resistance between pipes and building surfaces. Figure 2 shows the schematic of the ESCS system and TABS.

Figure 2: Schematic of lightweight embedded surface cooling system and TABS systems

THERMAL COMFORT EVALUATION

For evaluation of thermal comfort, we want to consider not only the total number of hours that the zone operative temperatures are outside of the thermal comfort zone, but also the severity of deviation from the comfort zone. In order to do this, the Method C PPD weighted criteria proposed in EN 15251 Appendix F (CEN 2007) was adopted here for long-term thermal comfort evaluation.

In this method, the time during which actual PMV exceeds the thermal comfort boundaries is weighted by a factor that is a function of the PPD.

Calculate weighting factor, wf, as shown below:

$$wf = \begin{cases} 0, & if \quad PMV_{limit_low} < PMV_{actual} < PMV_{limit_high} \\ \dfrac{PPD_{actual}}{PPD_{pmvlimit}}, & if \quad PMV_{limit_low} > PMV_{actual}, or\ PMV_{actual} > PMV_{limit_high} \end{cases} \tag{1}$$

Calculate the overall percentage of exceedance as the product of the weighting factor and the time for a characteristic working period during a year.

$$Exceedance_{warm}\% = \frac{\sum wf.time\,,\ if\ PMV_{actual} > PMV_{limit_high}}{\sum total\ characteristic\ working\ time}, \tag{2}$$

$$Exceedance_{cold}\% = \frac{\sum wf.time\,,\ if\ PMV_{actual} < PMV_{limit_low}}{\sum total\ characteristic\ working\ time} \tag{3}$$

In this study, PMV_{limit_low} =-0.5, and PMV_{limit_high} =0.5 were used.

In Appendix G of the same standard, the recommended threshold for acceptable deviation is that the percentage of exceedance in rooms representing 95% of the total occupied space is not more than 5% of occupied hours of a day, a week, a month and a year.

EXPANDED THERMAL COMFORT RANGE USING AIR MOTION

This section explains the impact of elevated air motion on expanding the thermal comfort zone. In *Figure 3*, we show the thermal comfort conditions of one of the hottest days in the cooling season in Sacramento. The left chart is the operative temperature profile over the day. Also shown are the lines that bound the thermal comfort zones. 79°F is the thermal comfort high limit corresponding to still air conditions (0.15 m/s air movement), 0.5 clo, and 0.012 humidity ratio. At this condition, PPD, the predicted percentage dissatisfied, reaches well above 10%. We can increase the thermal comfort high limit to 84°F when air movement is at 0.8 m/s. In the right chart, we show the PPD profiles for the same day for both design scenarios. We can see that in the late afternoon, PPD value without elevated air movement goes higher than 20% limit, but when air movement is provided, the PPD stays well below the 10% limit.

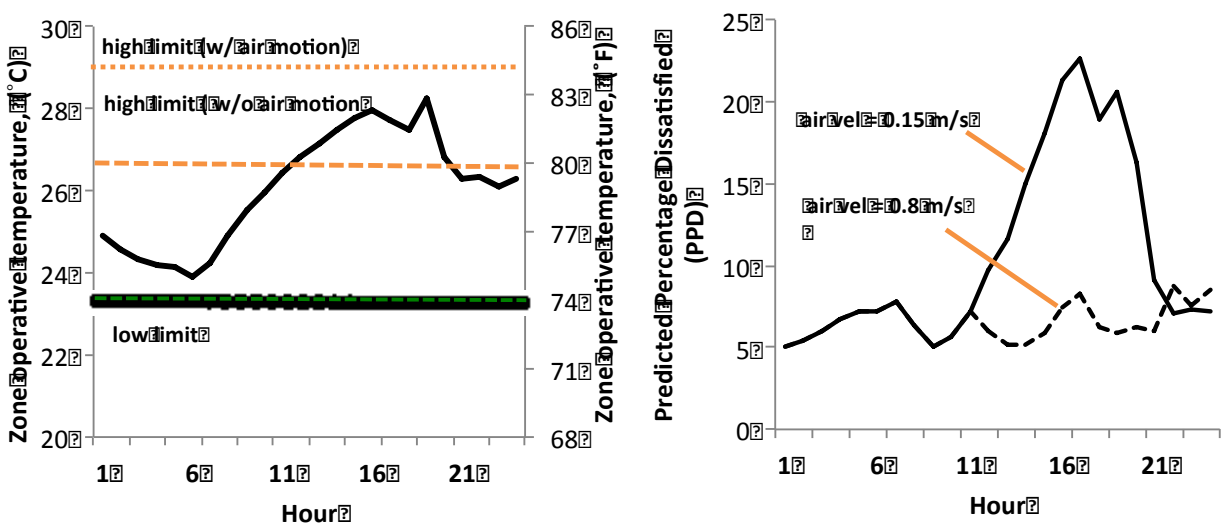


Figure 3: Example showing expanding thermal comfort range with air motion

RESULTS

The goal of this study was to investigate the application potential of using evaporative cooling as the primary cooling source for TABS, so we focused our analysis on warm discomfort during the cooling season.

San Francisco

San Francisco has a very mild climate with an average wetbulb temperature at 55 °F during the cooling season, and 100% of the time the wet-bulb temperature stays below 68 °F. For this climate condition coupled with well-designed shading system for the building, cooling demand was minimized in the simulated model. For San Francisco, we only evaluated the base design with pre-cooling only option, and the thermal comfort performance is quite satisfactory with hot exceedance level at 0%.

Sacramento

Sacramento features a warm and dry summer season with more than 10% of the time in a year having dry-bulb temperatures higher than 86 °F and an average wetbulb temperature at 60 °F during the cooling season. Weather analysis shows that the average diurnal wetbulb temperature difference during the cooling season is about 15 °F, and this indicates that Sacramento has a great potential for the precooling strategy.

For Sacramento, we investigated the following design options: Base case, base design + precooling only, with air system having 50% enhanced cooling capacity, with air system having 50% enhanced cooling capacity + precooling only. And for the all these four design options, we evaluated the thermal comfort performance of the system under two scenarios: with and without elevated air motion.

Figure 4 presents the thermal comfort results of all design cases. The red dash line is the 5% exceedance high limit required in EN 15251-2007.

Figure 4 indicates that if cooling tower can be made available for 24 hours a day, the base design, TABS with minimum ventilation air, can achieve acceptable thermal comfort performance. If cooling was provided only at night by pre-charging the slab, the hot exceedance level is 5.8%, which is higher than the 5% threshold. However, if elevated air motion can be provided to the space, the exceedance level can be pulled down to 0.17%.

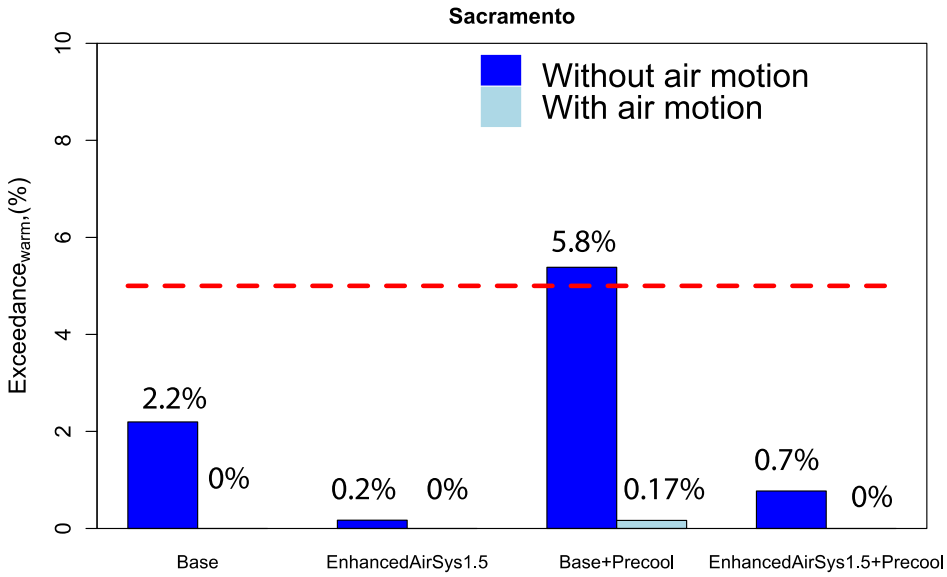


Figure 4: Exceedance of weighted PPD too warm for Sacramento

Atlanta

The climatic condition of Atlanta is warm and humid with more than 20% of the time having outside wetbulb temperatures higher than 68°F during the entire year. For Atlanta, we investigated the following design options: base case and air system with 50% enhanced cooling capacity. And for all options, we evaluated the thermal comfort performance of the system for the cases with and without elevated air motion.

Figure 5 presents the thermal comfort results of the design options for Atlanta.

First, we can see that the base design created the level of hot exceedance level at 40.8%, which is way above the accepted high limit. However, if elevated air motion can be provided to the space, the hot exceedance level can be reduced to 4.8%. This means during those discomfort hours, a large portion of the time has temperatures between 79 °F to 84 °F, a temperature range that can be satisfied by increasing air movement in the space.

A second option to improve the design was to increase the cooling capacity of the air system by increasing design airflow rate for 50%. This can reduce the hot exceedance level to 6.4%, which is still too high.

The last option we evaluated was to enhance cooling capacity of air system plus elevated air motion. This design can almost eliminate the hot discomfort for the whole cooling season.

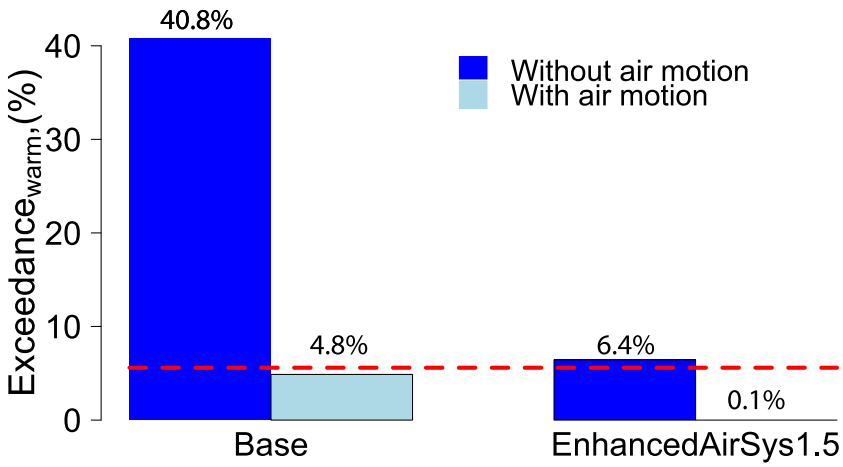


Figure 5: Exceedance of weighted PPD too warm for Atlanta

Phoenix

Phoenix's climate features a hot and dry cooling season with more than 30% of the time in a year having dry-bulb temperatures higher than 86 °F and an average wetbulb temperature at 63 °F during the cooling season. Weather analysis shows that the average diurnal wet-bulb temperature difference is about 6 °F and the differences are only 3°F during the hottest days. This indicates that precooling using evaporative cooling is not effective, because the cooling capacity of the cooling tower varies little between day and night times.

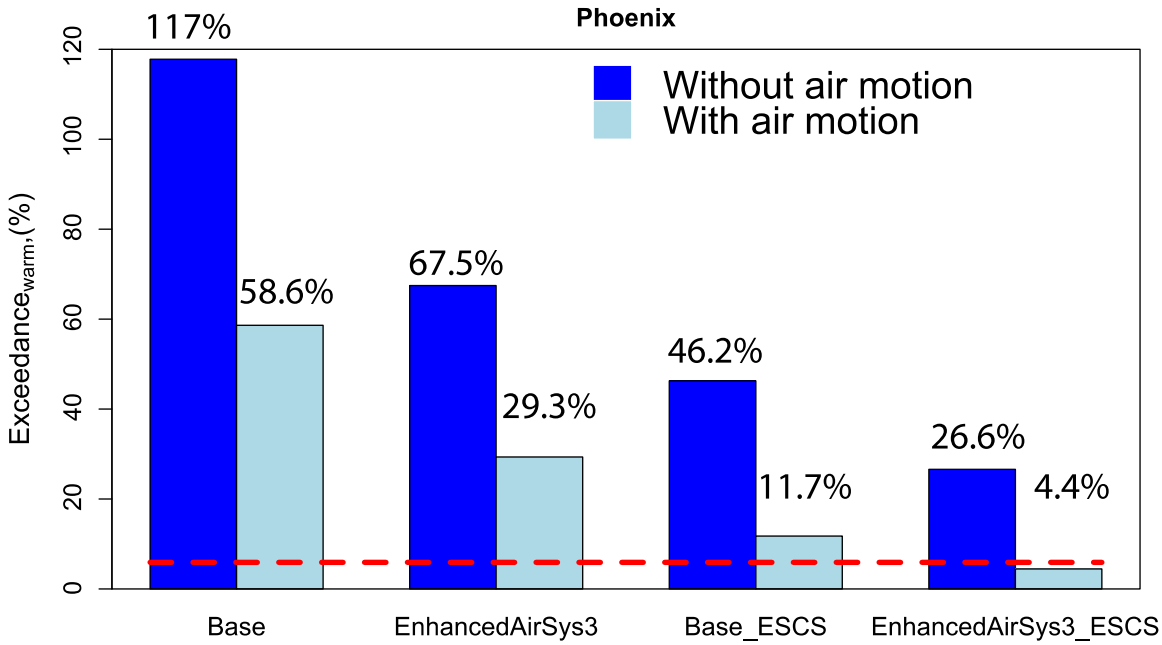


Figure 6 presents the thermal comfort results of the design options evaluated for Phoenix. We can see that most of the design options evaluated cannot satisfy the thermal comfort requirement. With the base design the level of hot exceedance level is 117%. The value can go higher than 100% because it is weighted PPD and 117% implies severe deviation from the thermal comfort range. If the cooling capacity of the system is augmented by triple the minimum air flow rate, the hot exceedance level is still 67.5%. If elevated air motion can be provided to the space, the exceedance level can be reduced to 29.3%, which is still much higher than the 5% high limit.

An alternative design option we evaluated was replacing the TABS with lightweight embedded surface cooling systems (ESCS), sometimes also called the plaster panel system. Compared to the TABS, the ESCS have higher cooling capacity because the thermal resistance between the radiant cooling surface and water pipe is smaller. As indicated in the climatic analysis, the cooling tower capacity varies little during day and night, so it is more effective to maximize the cooling capacity of the radiant system during occupied hours by minimizing thermal resistance between pipes and room space. The last design option employed the ESCS systems plus an air system with design cooling air flow rate three times the minimum requirement. This design alone cannot satisfy the thermal comfort requirement, but has reduced the discomfort level to 26.6%, and if elevated air motion is provided, the discomfort level can be further reduced to 4.4%.

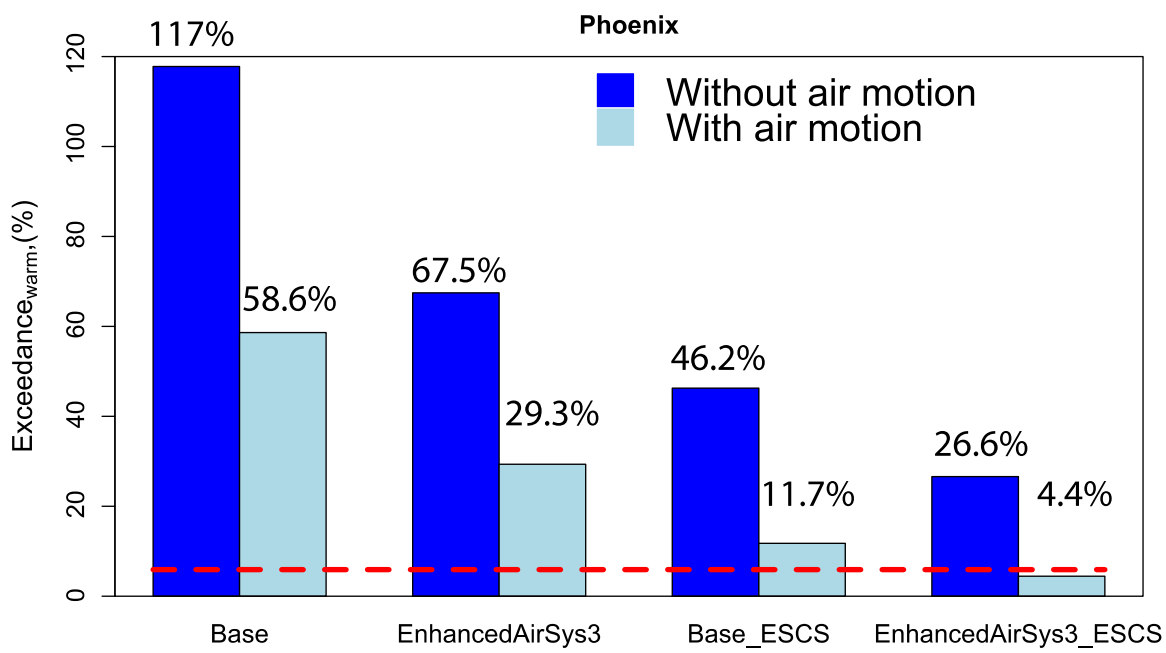


Figure 6: Exceedance of weighted PPD too warm for Phoenix

CONCLUSIONS

The conclusions from the simulation study are the following:

- In general, elevated air motion can dramatically reduce the hot discomfort level for most of the design options and climates.

- Evaporative cooling can be used as the only cooling source for the TABS in San Francisco. Hot discomfort can be eliminated by only precooling the slab.
- Sacramento is a good candidate for implementing a precooling strategy with TABS. This is because the cooling capacity of the cooling tower varies significantly during day and night due to the large (average 15°F) diurnal wet-bulb temperature difference.
- In Sacramento, hot thermal discomfort can be reduced from 5.8 % to 0.17% by providing elevated air motion for the base design using precooling only strategy.
- The base design option in Atlanta creates a 40.8% hot exceedance level. However, with elevated air motion, the hot exceedance level can be dramatically reduced to 4.8%. For Atlanta, another design option evaluated is to enhance the cooling capacity of the air system by increasing the design air flow rate to 1.5 times the minimum ventilation flow rate. This can reduce the hot exceedance level to 6.4%, and with elevated air motion, hot discomfort can be eliminated.
- Precooling with evaporative cooling is not effective in Phoenix, because the cooling capacity of the cooling tower varies little between day and night times during hot summer days.
- For Phoenix, using evaporative cooling as the primary cooling source for TABS cannot satisfy the thermal comfort requirement unless the cooling capacity on the air side is significantly enhanced. However, the use of an ESCS systems plus an air system with design cooling air flow rate tripled the minimum requirement can reduced the discomfort level to 26.6%, and if elevated air motion is provided, the discomfort level can be further reduced to 4.4%.

One final note is solar load control is crucial to the success of all the design options. All the results above are based on simulating the building models that have complied with ASHRAE Standard 90.1 2010 and the buildings were configured to be very well shaded to minimize solar heat gain.

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ENERGY SAVINGS FROM EXTENDED AIR TEMPERATURE SETPOINTS AND REDUCTIONS IN ROOM AIR MIXING

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INTRODUCTION

Large amounts of energy are consumed by air-conditioning systems to maintain tight control of air temperature in rooms--a narrow range of temperature excursion from neutral, and a uniform temperature in the ambient space. However, both field and lab studies are showing that neither narrow range nor uniformity is really necessary for providing occupant comfort. Data from several large field studies shows occupants accepting a much wider temperature range than is typically applied in practice (Arens 2009). In addition, if occupants have access to a personal environmental control (PEC) system, the acceptable ambient temperature range can be further extended, to as much as 18-30ºC (Zhang H 2009, Amai 2005, Zhang Y 2008). By targeting specific body parts, PEC systems produce equivalent comfort using much less energy than is needed to condition the entire ambient space.

Energy is also required consumed in fans and mixing diffusers to produce uniform room temperature conditions. To assure complete mixing, diffuser manufacturers specify minimum supply volumes that are as high as 50% of maximum volume. Some engineers have been successfully operating buildings well below these volume minima, and research is now underway to quantify the acceptability of their non-uniform environments to their occupants. Substantial fan energy savings are possible, and recent changes to energy standards (ASHRAE 2009) have begun to require a lower minimum.

This paper simulates the energy savings possible from through these two approaches to providing comfort in less tightly controlled spaces. Their savings are substantial and justify looking into how they might be incorporated into building design and operation.

METHODS

EnergyPlus simulations were performed to examine the energy implications of larger range of thermostat setpoints and low minimum supply volumes. A conventional HVAC building was simulated in the climates of San Francisco (mild coastal climate), Miami (hot and humid), Phoenix (hot and dry), and Minneapolis (hot in summer, cold in winter).

The building is based on a large office prototype developed by the U.S. Department of Energy. It is 111 x 278 ft with four stories and a basement, and each floor plate is 31,000 ft^2. The window to wall ratio is 0.4. Each floor consists of four perimeter zones of 15ft depth, and an interior zone (Figure 1). The floor-to-ceiling height is 13ft, and the floor-to-floor height is 15ft.

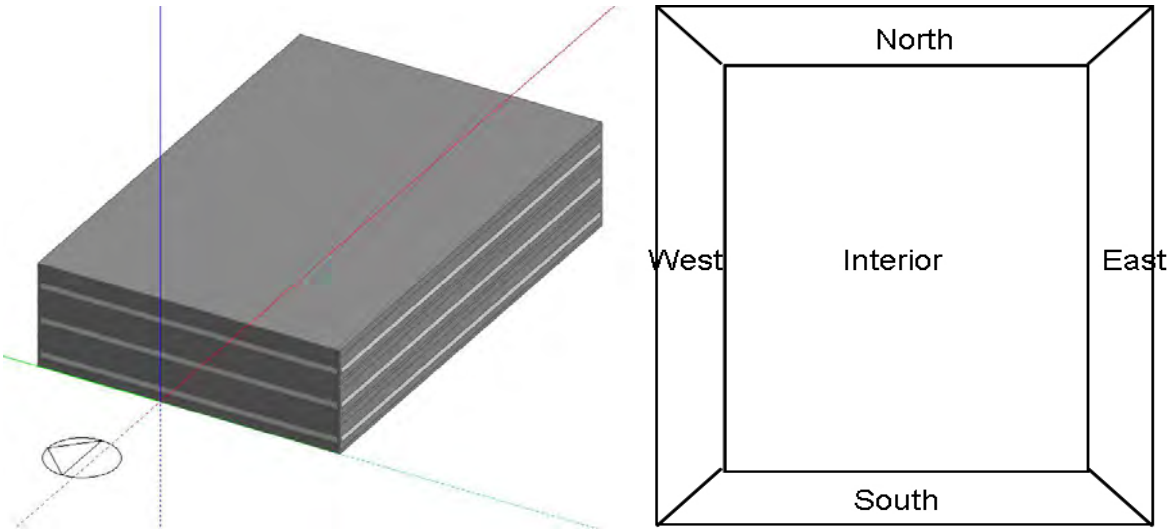

Figure 1. Configuration of the simulated building with perimeter and interior zones.

The model's internal load schedules for lighting, plug loads, and occupancy are typical for large offices. Peak power density for lighting is 1W/ft^2, peak plug load is 0.75W/ft^2, and peak occupancy is 200ft^2/person.

The HVAC system is variable air volume (VAV) with reheat. There are four air handling units, one for each floor. A central boiler and centrifugal chiller provide the heating and cooling. Infiltration peaks at 0.25 air changes/hour when the ventilation system is off.

In the US, buildings are commonly designed using 21.5 – 24ºC (71 – 75ºF) as the temperature setpoints. In this study, this is considered the base case. The annual energy use intensity (EUI) changes caused by moving the cooling setpoints from 24ºC to 25, 26, and 28ºC were simulated independently with the heating setpoint fixed at 21.5ºC. Similarly, the EUI changes from moving the heating set point from 21.5ºC to 20.5, 19.5, 18.5, and 17.5ºC were simulated with the cooling setpoint fixed at 24ºC.

The minimum supply air volume of the VAV terminal units, as well as the heating and cooling setpoints, were parameterized to carry out this study. The HVAC system components were autosized to optimally meet design loads, so that the predicted annual EUI values include both effects of optimizing system size/efficiency, and of changes to the hours and intensity of operation.

RESULTS

A. Widening room air temperature setpoints

The annual energy savings compared to the base case are presented in Figure 2 for Minneapolis, San Francisco, Phoenix and Miami. Increasing the cooling setpoint by one degree from 24ºC to 25ºC results in energy savings of 7-15%. When the setpoint is expanded to 28ºC, the energy

savings for three locations (except in Minneapolis) reaches 35 – 45%. In Minneapolis, the annual number of cooling hours is low so the potential for accumulated cooling savings at the highest setpoint is small.

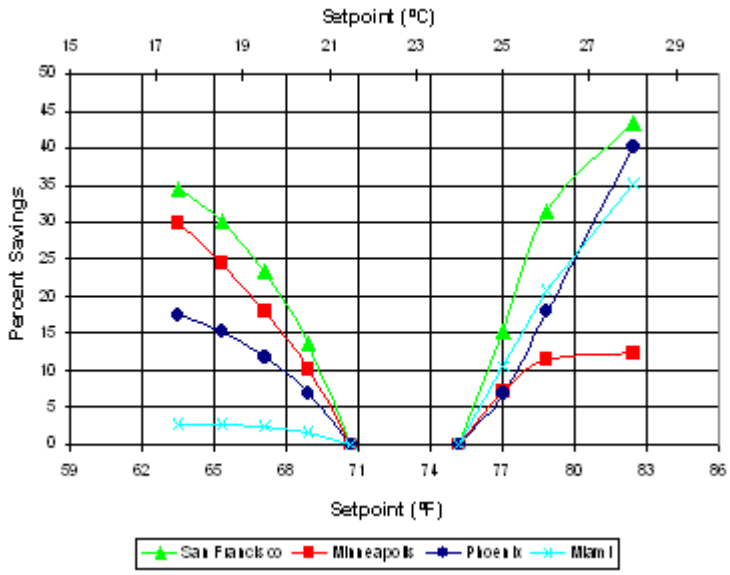

Figure 2. Percent energy savings for widened air temperature setpoints relative to conventional setpoint range in San Francisco, Miami, Phoenix, and Minneapolis.

Decreasing the heating setpoint by one degree from 21.5ºC to 20.5ºC results in energy savings of 7-14%, except in Miami. With the setpoint as low as 17.5ºC, the total energy savings reach 17 – 35%. In Miami, the heating load is small year-round so there is not much potential for energy savings.

When both heating and cooling setpoints are expanded together, the savings can be estimated by adding the savings from each side.

B. Lowering the minimum supply volume

Figure 3 compares lowering the minimum supply volume from the base value of 30% to 20% and 10%. This comparison is done for San Francisco only. At the base-case design temperature setpoints (21.5 – 24ºC), lowering the minimum supply volume to 20% and 10% saves 17% and 27%. At the expanded cooling setpoint (28ºC) or heating setpoint (17.5ºC), the energy savings reach 40 – 60% and 33 – 40%, respectively.

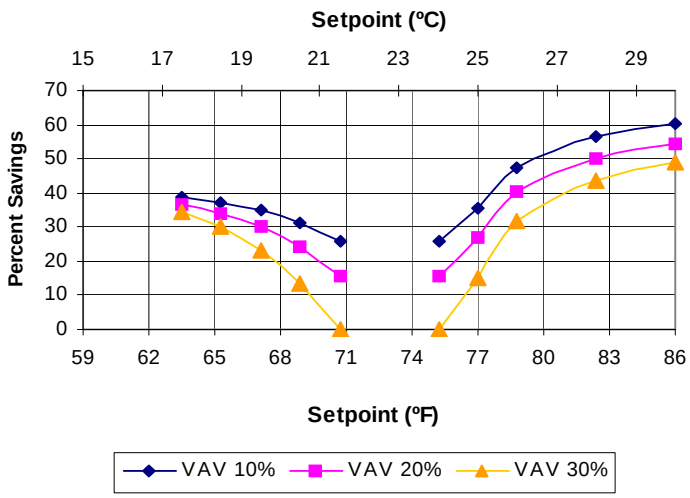

Figure 3. Annual energy use for the prototype in San Francisco with VAV minimum fractions at 10%, 20%, and 30%.

CONCLUSIONS

In a series of EnergyPlus simulations, we investigated the consequences of widening the indoor temperature range between thermostat settings for heating and cooling. The enlarged temperature range reduces energy use by lessening the cooling and heating loads in two ways. First, as a result of fewer heating and cooling hours, and second, as a result of a decrease in the magnitude of the difference between the setpoint and the outdoor temperature. The saving is about 10% for each degree Celsius increase or decrease in the setpoint.

In practice it may be possible to reduce the airflow minimum of VAV terminal units while still satisfying ventilation and mixing requirements. Performing the same setpoint-range simulations as above with VAV minimum fractions at 10%, 20%, and 30% of maximum capacity, minimum fractions of 10% and 20% produced ~25% and ~16% savings relative to energy use in the building with a minimum fraction of 30%.

The substantial savings justify further examination of these approaches in research, and their further application in energy and environmental standards and in practice.

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Assessing thermal comfort near glass facades with new tools

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1 ABSTRACT

Transparent facades impact the thermal comfort of occupants in multiple ways. This paper addresses such impacts both for generic glazing and for complex fenestration systems such as shades and blinds. The first step in the process of evaluating comfort for a specific façade is to calculate the solar radiation load on the occupant. The second step is to assess the person's physiological reaction to the radiation and how this reaction influences thermal sensation and perception of thermal comfort.

A newly developed tool (*SoLoCalc*) uses bi-directional scattering functions to characterize the radiant transmission of complex fenestration systems. The output is then linked to an advanced physiology and comfort model. Although this paper describes its use in evaluating glass facades, the tool applies equally to buildings with smaller window-to-wall ratios, or to skylights. A case study of a real construction project shows how the tool enables a new and comprehensive approach to assessing the impact of the building envelope on user comfort.

2 INTRODUCTION

Using glass as a dominant material for the building envelope offers great design opportunities for architects. In recent years the possibilities have been extended through the use of structural glazing, spandrel panels and a huge variety of coatings, films and frits. For the beholder from outside, successful glass architecture demonstrates transparency while for the occupant inside, the large transparent area may provide generous daylight and views. However, less successful designs can be seen to have shades and blinds closing off most of the window area, sacrificing view and daylight to avoid visual glare and occupant discomfort. In addition, large glazed areas often lead to excessive energy use compared to more traditional constructions with smaller window-to-wall ratios.

While a number of measures have been introduced to keep the energy consumption of glass architecture reasonable, the thermal comfort in the space behind the glass facade has typically not been thoroughly addressed. It is often considered to be the duty of the HVAC-system to provide comfortable indoor conditions – a task that may require great energy use to fulfill. With the attempt to build "greener" buildings, engineers aim more and more for low-energy technologies such as radiant cooling, natural ventilation, and personal environmental control [Zhang et al., 2009], and move away from uniformly air-conditioned spaces. It is within this trend that more attention is being paid to the influence of the facade on the comfort of occupants.

In order to assess the thermal conditions of a person sitting or standing in the perimeter zone of a large glazed area, one has to take into account the effects of the facade on air temperature, air movement, long wave radiation and solar load. The room geometry, transparent surface area, and the occupant's position in the room are as important as a number of physiological parameters. This is not addressable by traditional comfort models, all of which either do not take into account the physical presence of the occupant or treat the body as a cylinder or sphere. However, newer multi-segment-models such as the *Berkeley Comfort Model* [Huizenga et al., 2001] do describe the human body in detail and calculate its thermal state as it interacts with its surroundings. The human's perception of the thermal conditions is predicted by empirically determined models that relate skin temperatures and a person's thermal sensation and comfort.

For warm climates or in buildings where the need for cooling energy is predominant, it is essential to reduce the solar load through shading systems. The most favorable way to block the sun is through shading devices external to the glass that are integrated in the architectural design. Other, though less effective ways include blinds interior to the glass or within the glazing unit. For all these systems, the calculation of glass thermal performance values using the assumption of a normal solar incidence, as is commonly done for specular glazing, is no longer acceptable. That is why in recent years, substantial efforts have been made in the physical description of three-dimensional solar transmission through complex fenestration systems [Kohler, 2009].

Despite the progress that has been made in the fields of comfort assessment and solar transmission there has been no tool available that takes all influences on the occupants' heat balance into account: evaporative and convective heat transfer, long-wave radiation with the interior surroundings, and diffuse and direct solar radiation. Such all-embracing consideration will be possible in the future with an approach that combines current state-of-the-art models.

3 ASSESSING THERMAL COMFORT WITH A MULTI-SEGMENT MODEL

The significance of the building envelope for the overall energy balance of a building is well understood. The impact of the facade on thermal comfort is often less investigated because the still most frequently used metrics for thermal comfort, PMV and PPD[1], simplify boundary conditions and do not consider either solar load or the physical presence of the occupant. Therefore some of the main influences on how a person perceives conditions in the room are neglected within this simplified assessment.

3.1 Physiology model

Due to the complex geometrical relationship between facades and occupant, a more detailed model to represent the human body and the thermal impact on the occupant is

1 PMV: Predicted Mean Vote, PPD: Predicted Percentage of Dissatisfied [Fanger, 1970]

necessary. In this paper a mathematical multi-node thermo-regulatory model is used that is based on experimental and numerical work over the last 50 years from [Stolwijk, 1971], [Tanabe et al, 1998], [Huizenga et al., 2001] and others. It describes the physiological processes through a complex set of equations in which the human body is represented through 16 body parts, each of them consisting of a core layer, a muscle layer, a fat layer and a skin layer. Layers are connected to each other through conduction and body segments are linked via a blood flow model.

Within the body there are heat sources and heat sinks: the core layers of chest and back are cooled by the breathing of ambient air, and muscle layers generate metabolic heat under work activity and when shivering. The outer layer connects the body to the ambient conditions. The main heat transfer processes at the outer layer are the convective, longwave radiative, evaporative heat transfer, and the absorption of impinging solar.

A heat balance is established during every time step of the transient simulation and solved numerically; the model then outputs body temperatures for each time step as a result. If the heat balance of the body is not neutral, the resulting skin and core temperature emit signals to the hypothalamus. The modeled human body reacts then with shivering or sweating and with dilation or constriction of blood vessels where the effects on the heat balance are taken into account within the next time step.

3.2 Sensation and comfort model

Several human subject tests have shown that there is a strong relation between skin and core temperatures and thermal sensation and comfort. These relationships have been put into equations [Zhang, 2003] and implemented into the *Berkeley Comfort Model* software to derive sensation and comfort from the physiological predictions of the model [Zhang et al., 2009].

The outputs of the sensation and comfort algorithms are thermal sensation (in terms of being warm or cold) and thermal comfort (in terms of feeling comfortable or uncomfortable). Both metrics range from -4 (very cold or very uncomfortable) to +4 (very hot or very comfortable). They are given for each of the 16 body segments (local sensation / local comfort) as well as for the whole body (overall sensation / overall comfort).

The local sensation output for one body segment depends on the calculated skin temperature with high skin temperatures leading to warm or hot local sensations and low temperatures leading to a cool or cold sensation. The definition of high or low skin temperatures refer to setpoint values implemented in the program. Some body parts are more susceptible to heat than others, e.g. the head. The relationship between the local sensation for the 16 body parts and the overall sensation is complex.

In contrast the correlation between local sensation and local comfort for one body part is straightforward. The further the local sensation metric is from neutral (neutral conditions equal 0 for the sensation metric), the less comfortable one perceives the thermal condi-

tions for that specific body part. The most uncomfortable body parts determine the overall perception of thermal conditions: if e.g. only the head and two other upper body parts show a significant increase in skin temperature, thus high values for sensation, the conditions for these body parts are perceived as uncomfortable and the person will overall feel uncomfortable.

4 IMPACT OF GLASS FACADES ON THERMAL COMFORT

The impact of glazed surfaces on the thermal comfort of occupants can be significant for two reasons: their transparent property allows solar radiation to enter the room, and the glazing's inner surface temperature, which may be very different from the surface temperatures of other interior surfaces, causes longwave radiant heat exchange and convective heat flows in the adjacent space. Thus the glazed area influences the body's heat balance through convection, long-wave and short-wave radiation.

(Note: The following does not look at potential condensation on cold surfaces or other issues (daylight, acoustics) which are not directly related to thermal comfort.)

4.1 Effects of convection

In heating climates or during winter in mixed cooling / heating conditions the inner surface temperature of a glass facade or window is usually several degrees below the room air temperature. Warm air moving along the cold surface will be cooled through convective heat transfer. As its density increases, the cooled air follows the gravitational force and moves towards the floor. This eventually causes a continuously rolling air flow with a draft of cold air in the foot region that can lead to an uncomfortable feeling of cold feet.

The effects of inner surface temperatures on local air movement and air temperatures can be determined by direct measurement or by computational fluid dynamics software. The results can be incorporated into the calculation of the person's thermal state by inputting the corresponding heat transfer coefficients and air temperatures for different body parts into the multi-segment model [Voelker, 2011].

The same approach of first determining convective heat transfer coefficients and temperatures and then using them as input parameters can be used for other possible convective effects in the room, such as natural ventilation, mechanical ventilation (w or w/o air conditioning) and drafts due to cracks in the window / frame seal or to other parts of the facade that are not airtight.

4.2 Longwave radiation heat exchange between occupant and facade

All surfaces in a room emit longwave radiation and so does the occupant via her outer surface. Important for the body's heat balance is the difference of the radiation emitted to the ambient and the energy received from the surroundings (including reflective radiation). The parameters that influence this heat balance are the surface temperatures to the fourth power, the emission coefficients, and the geometry.

Compared to the inner surface temperatures of partitions, ceiling, and floor, glass facades tend to have significantly lower temperatures under heating conditions (cold outdoor temperatures with little solar radiation). In the presence of solar radiation they can show significantly higher temperatures than the inner walls especially when using solar coatings, frits or tinted glass due to the heat absorption in the glass.

To assess the influence of longwave radiation to and from glass facades on comfort it is convenient to use an advanced calculation tool such as *Window6*[2] in order to determine the surface temperatures at the inside of a glazing systems (w or w/o shading).

How much heat is exchanged between two surfaces not only depends on the emissivity of the surfaces and their temperatures but also on the geometry determining how much one surface can see of another. In the case of the occupant of a room it is critical how far the person is sitting (or standing) from the considered surface. If the surface temperatures exceed a certain comfort range, the only way to maintain a comfortable heat balance is to move further away so that the facade is seen less. The mathematical description of this influence is expressed by view factors [Howell, Siegel, 1972].

4.3 Solar radiation on occupant

Solar radiation that is transmitted through the transparent (or translucent) part of the facade and that hits the occupant in the room produces a heat load on the human body. The main part of the radiation energy is absorbed on the surface of either the skin (in the case of nude body parts such as face or hands) or the fabric (in the case of clothed body parts) and leads to a temperature increase of those outer layers. The values for solar absorption on the skin range from 0.6 to 0.9 and from 0.3 to 0.9 for common clothing. Following the general rules of heat transfer, the increased temperature of the outer layer reduces the heat flux from the inner core to the outside, meaning that the body cannot dispose of additional heat as readily as before.

4.3.1 Diffuse and direct radiation through the facade

The glazing industry provides a huge variety of coatings and films in glazing systems to address the issues of heat transfer and solar load through windows and glass facades. The optical properties of coatings and films in combination with specific substrates are

2 http://windows.lbl.gov/software/window/window.html

available from the International Glazing Database (IGDB) that provides more than 3800 entries in its current version (IGDB 21.0 as of February 2012). The transmission and reflection values in the IGDB are given over the entire solar spectrum, allowing the transmission of a particular glazing system to be calculated at one wave length and then integrating the spectral data to the integral values: visible transmittance (Tvis), transmittance over the whole solar spectrum (Tsol) and solar heat gain coefficient (Tsol + secondary heat gain through conduction, convection and radiation). These performance indices are calculated for a normal incidence angle of the sun.

Although some of the available coatings provide very low SHGC values, other options of reducing solar load are often more favorable for a specific project. These options include exterior shades (horizontal or vertical, operable or fixed etc.), blinds in the gap of an insulating glazing unit or double skin facade, frits on the glass and others. These measures are dominated by diffuse components where the direct ray is no longer transmitted in the same direction as when it impinged on the facade but is redirected in other directions and/or scattered.

Unlike in specular glazing systems[3], the assumption of normal incidence nearly always leads to wrong results when applied to shading systems. For example, horizontal slats will not improve a calculated SHGC coefficient for the system when assuming an incidence angle of 0°, but in practice horizontal slats are very efficient at blocking sun rays at high solar altitudes.

The significance of the solar incidence angle for the transmission of shading systems is obvious. LBL introduced the method of bi-directional data for these so called complex fenestration systems in the algorithms of “Window6” [Kohler, 2009]. The output of these algorithms, the “Bi-Directional Scattering Distribution Function (BSDF)”, is used in the newly developed calculation tool *SoLoCalc* described in Section 5. The BSDF contains information about the hemispherical distribution of transmitted solar radiation into the room for any given incidence angle. The transmitted radiation contains direct components (if any) as well as the diffuse components. (For more information see also: Christian Kohler: Simulation of Complex Glazing products, BEST3, April 2012)

4.3.2 Perception of diffuse and direct radiation

Although they both represent a heat load on the body, there are differences in the perception of diffuse and direct radiation. A uniformly distributed (diffuse) radiation on the body increases the local sensation of all body parts to a similar extent. The result is an overall warm sensation while the overall comfort can still be acceptable (depending on intensity and duration of the conditions).

3 Specular glazing: transmitted solar radiation follows the incident direction, without diffusing or scattering effects

Local direct radiation (with usually higher intensity than diffuse radiation) on only a few body parts raises skin temperature locally. That leads to high values of local sensation for the irradiated body parts while the local sensation of the non-irradiated segments is not influenced. The overall comfort rating is likely to be more uncomfortable because the most uncomfortable local segment dictates the overall comfort.

Figure 1 shows the prediction of overall sensation and overall comfort as well as for local comfort (see color scale in the middle referring to the color of the body parts) for the same heat load impinged on the body under steady-state conditions. In the picture on the right side, a heat load of 80W hit head, hands, the left upper arm and parts of the upper body.

In the picture on the left side, the same heat flux was distributed uniformly over the person. The results, calculated with the *Berkeley Comfort Model* software, show very uncomfortable values for head and hands in the case of direct radiation which leads to an overall comfort score of -2.8 that is equivalent to very uncomfortable. The person who receives the same heat flux (80W) in form of diffuse radiation feels less warm (overall sensation 1.5 compared to 3.0 for direct radiation) and more comfortable (overall comfort -1.1 compared to -2.8).

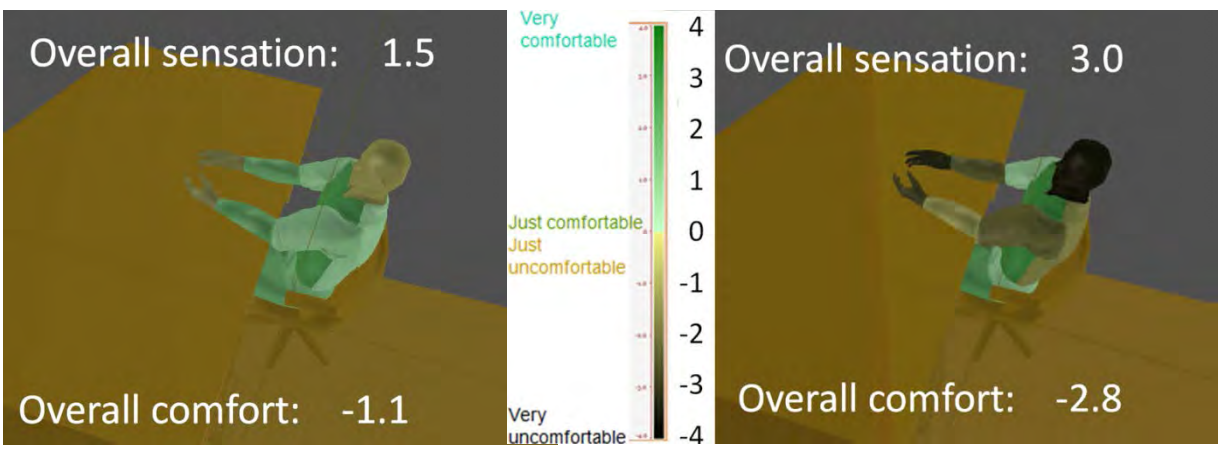

Figure 1: *Sensation and comfort for a solar heat load of 80 W calculated with the "Berkeley Comfort Model"*

Left: *diffuse radiation uniformly distributed over the whole body*
Right: *direct radiation on head, hands and upper body*

5 THE SOLAR LOAD CALCULATOR "SOLO CALC"

To account for diffusing or light redirecting elements in the facade, a new tool was developed as part of a research project within the Center for the Built Environment [Hoffmann et al., 2011].This tool, called *SoLoCalc*, calculates the solar load onto occupants by making use of data for three-dimensional transmission through complex fenestration systems as available in the *Window6* software.

5.1 Bi-directional scattering distribution function

SoLoCalc is based on bi-directional scattering distribution functions (BSDF). This information is provided from *Window6* in form of a matrix where the columns represent the hemispherical incidence angle of the sun ("outer hemisphere") and the rows represent the hemispherical transmission of solar radiation into the room ("inner hemisphere"). This concept allows for an easy access to the necessary data during the simulation.

SoLoCalc calculates the incidence angle on the facade for the location of the building, hour and day of the year and for the orientation of the facade. This incidence angle (expressed in spherical coordinates) gives the information which column of the BSDF matrix has to be chosen. The three-dimensionally transmitted energy has then to be attributed to the body parts of the person sitting (or standing) behind the facade. These values can be derived from the rows of the BSDF matrix.

While the coupling of the outer hemisphere to a given solar incidence angle is straightforward, the linking of the transmitted energy to the occupant needed a new approach in *SoLoCalc*.

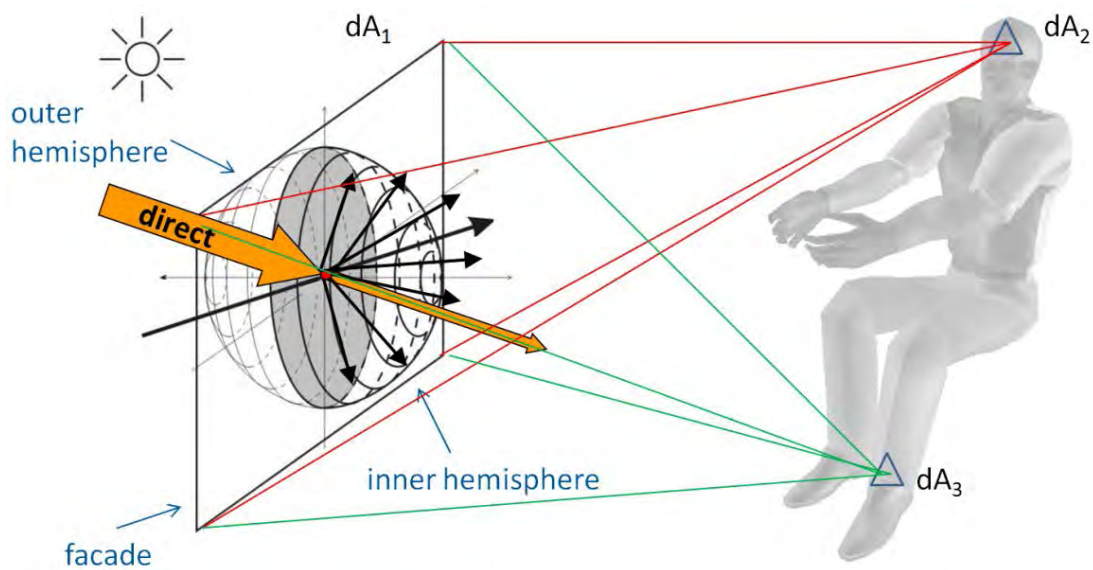

Figure 2: Linking the 3-D transmission to the solar load on certain body parts

5.2 Application of the viewfactor method to shortwave radiation

The approach that was chosen in *SoLoCalc* is to use viewfactors (as described above for longwave radiation exchange) for solar radiation. In general, the use of viewfactors in the calculation of radiative heat transfer is justified where the emitter can be considered as uniformly diffuse, which is only true for ideally diffusing systems. Nevertheless, in our case the viewfactor method is applicable for not-ideal scattering systems due to the incremental nature of the data. In the BSDF files the inner hemisphere is subdivided into a substantial number of “bins” (e.g. 145 bins for a “full size matrix” in *Window6* [Klems, 1994]). Each bin corresponds hereby to a defined solid angle on the unit sphere. The viewfactor method treats the emitting surface as a uniform diffuse emitter for this solid angle. The emitted heat flux is the amount of solar load transmitted in this particular direction.

For the chosen approach it is necessary to subdivide the geometrical description of the occupant (called “manikin”) into small plane polygons where a group of polygons represents a body segment. The currently used manikin consists of 1356 polygons to form the 16 body segments. It is also necessary to subdivide the facade into partial areas with an adequately chosen discretization.

For the viewfactor calculation we currently use the open source software *View3D*[4] to handle the number of surfaces and the amount of blocking [Walton, 2002] but other software tools that provide viewfactor calculation would be possible as well. The viewfactors between the manikin polygons and the façade are necessary input values for the algorithms implemented in *SoLoCalc*.

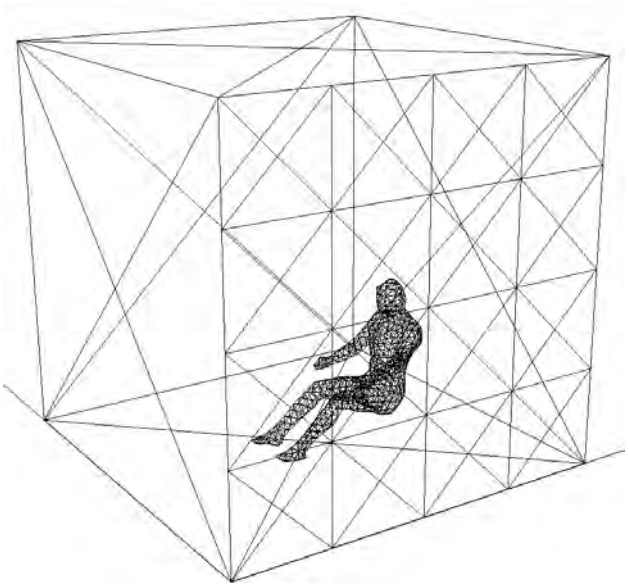

Figure 3: Meshing of manikin (1356 polygons) and facade (64 polygons)

4 http://view3d.sourceforge.net/

6 CASE STUDY

The case study presented here was initiated as a collaborate project between ZGF Architects, Portland and CBE. More details about the project, a hospital building in Denver, can be found in an associated BEST3 paper by Mark Perepelitza: Building Enclosure Performance Research - Applications in Professional Practice.

With the help of the following example the importance of the described approach will become obvious: the necessity for the detailed calculation of solar load onto the occupant because of the variation of sensitivity for different body parts, the concluding overall perception of the conditions and finally the comparison of how comfortable the occupant would feel for different options such as solar coatings, frits and/or shades.

The case study looks at a specific room geometry and specific occupant location within a building in Denver, Colorado. The results shown here were calculated for a patient room at the South façade of the building, with a person (visitor or patient) sitting in a chair at a distance of 9 feet from the glazing parallel to the façade, looking towards the West.

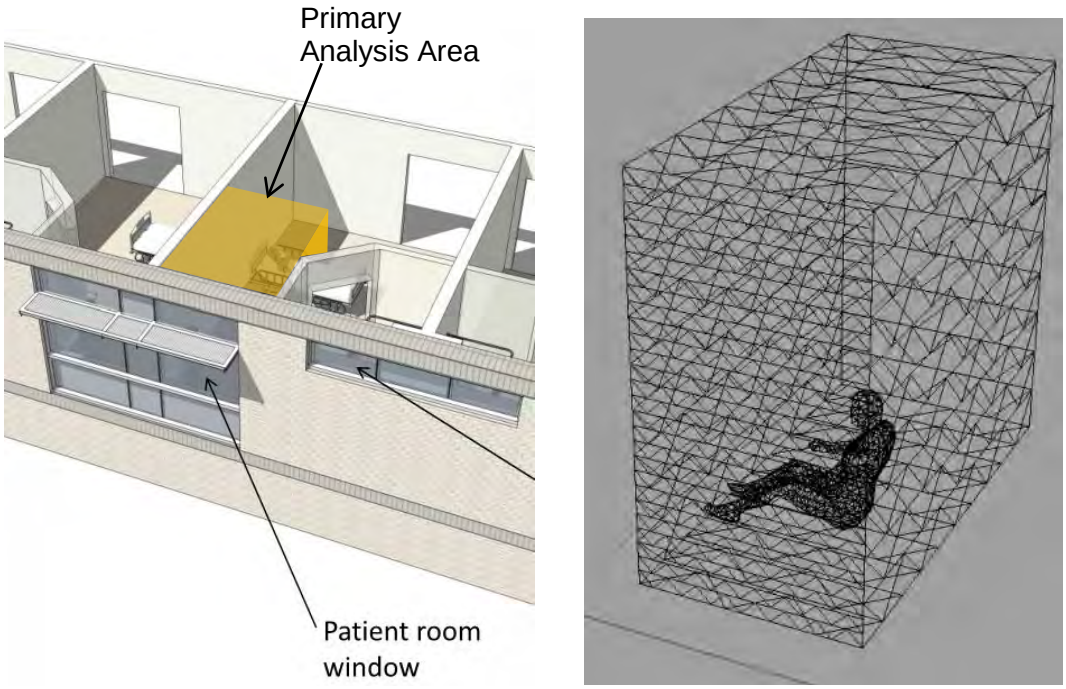

Figure 4: Denver hospital patient room (see also Mark Perepelitza: Applications in Professional Practice)

An extremely clear summer day was chosen as boundary condition with a peak radiation of 985 W/m^2. In order to make the decision of which would be the necessary measures to minimize thermal discomfort, the situation was modeled with and without exterior shading.

6.1 Solar load on the body

Due to the relative narrow room width, the number of hours when direct sunlight hits the person through the façade is limited. In the case of a solar coating without exterior shade, the person receives direct radiation for the first time around 11 am with the left lower extremities being irradiated (Fig. 5). The upper body is irradiated during the peak hours of solar load between 1 and 2 pm. In the late afternoon hours after 3 pm when the sun moves towards the West, the direct radiation that hits the body decreases.

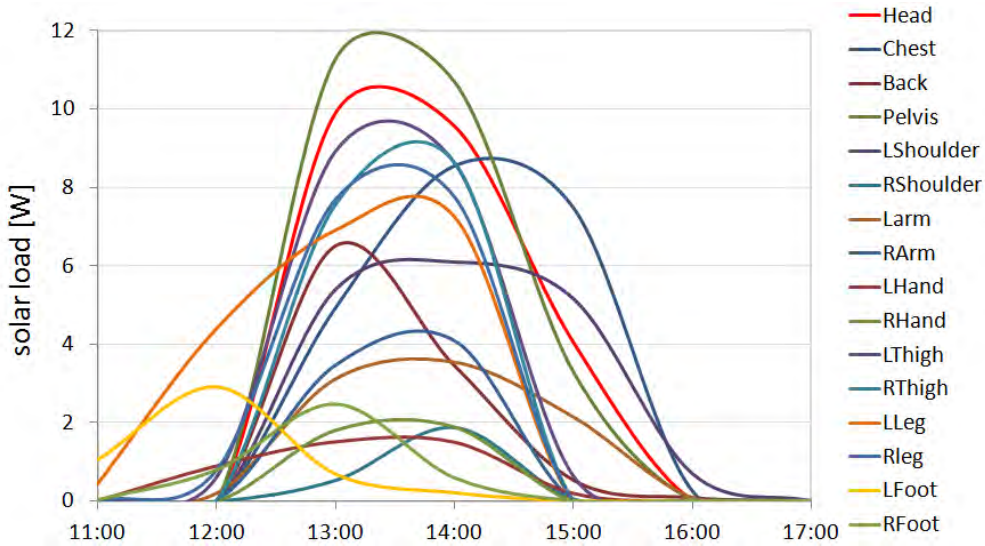

Figure 5: Solar load on body parts for a solar coating with SHGC = 0.33

If an exterior shade is mounted in addition to the solar coating of the glazing (Fig. 6), not only the overall solar load is significantly reduced, the received radiation is also distributed more evenly over the different body parts due to the diffusing effect of the exterior shade, and the period of time of irradiation is reduced.

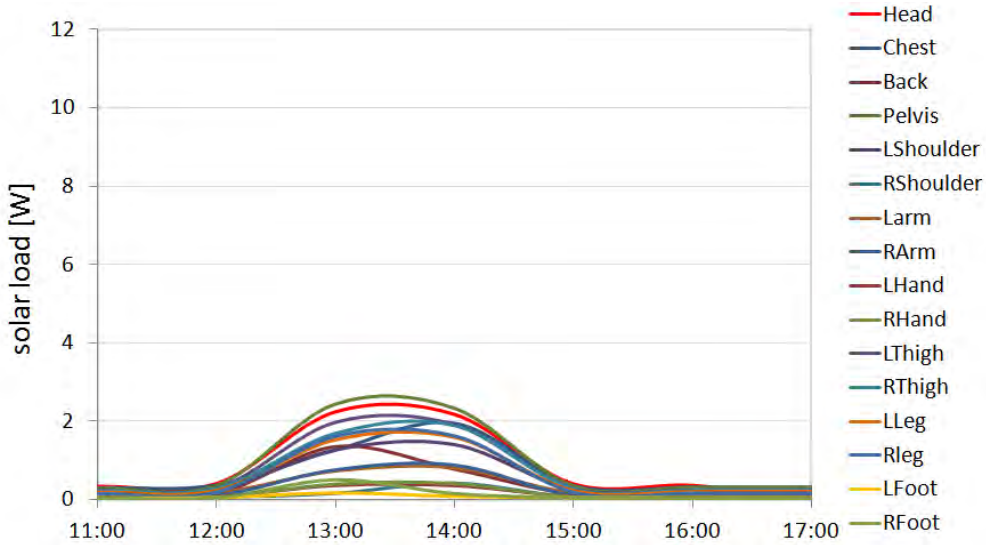

Figure 6: Solar load on body parts for the solar coating plus the exterior shade

6.2 Sensation and comfort

In Fig. 7a and 7b the dark red line shows the overall sensation of the occupant for the case without exterior shade. The metric “overall sensation” represents the vote that a normalized person would give if asked for the perception of temperature conditions in the room.[5] At noon the predicted vote would be “slightly warm”, from 1 pm to 3:30 pm the conditions would be perceived as “hot” before the overall sensation rating drops to “warm” at 4 pm and to “slightly warm” at 5 pm.

The other curves represented in the Fig. 7a and 7b correspond to the metric “local sensation”, i.e. how warm a specific body part is perceived. When comparing the overall sensation to the local sensation curves, it becomes obvious that the overall sensation is not equal to the mathematical average (mean) sensation: body parts have different impacts on the overall sensation.

In Fig. 7a, the highlighted local sensations are those of the left leg and left foot. These are the body parts that receive most of the direct radiation before noon. As the lower extremities are less sensitive to heat, their influence on the overall sensation is not very strong.

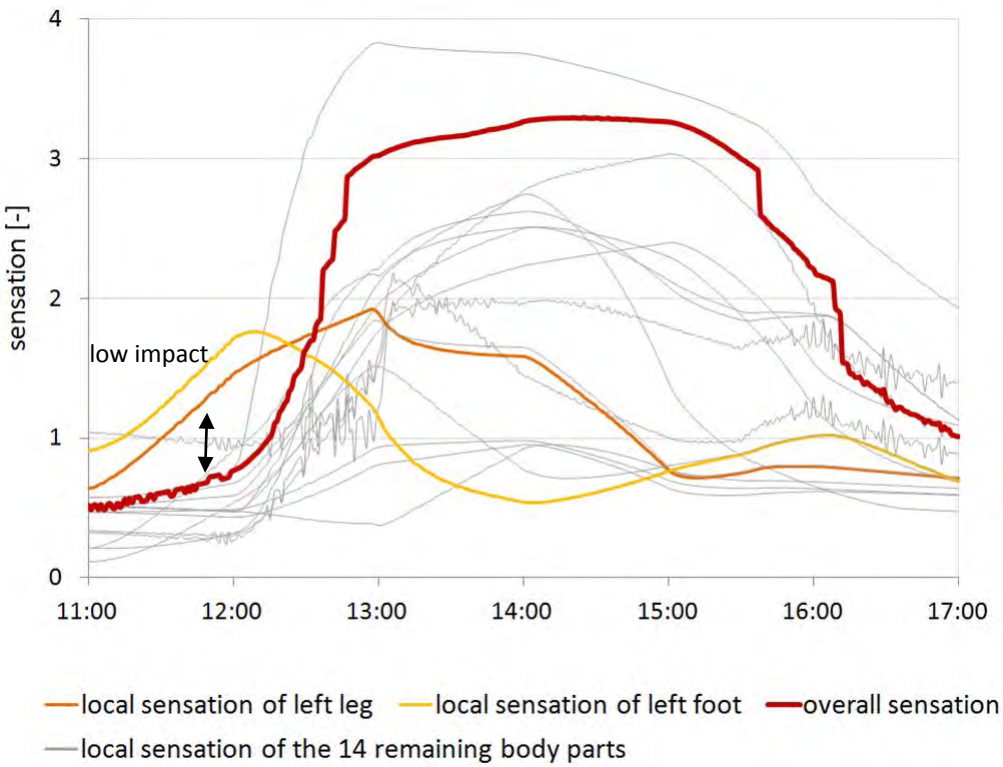

Figure 7a: Local and overall sensation for solar coating without exterior shade, left leg and foot highlighted

5 The modeling of the occupant’s thermal state starts a couple of hours earlier (e.g. around the time when the person enters the office and gets started with sedentary work) and it is assumed that prior to the occurrence of direct radiation the person is in a thermally neutral state, i.e. she or he is neither too hot nor too cold, i.e. overall sensation = 0.

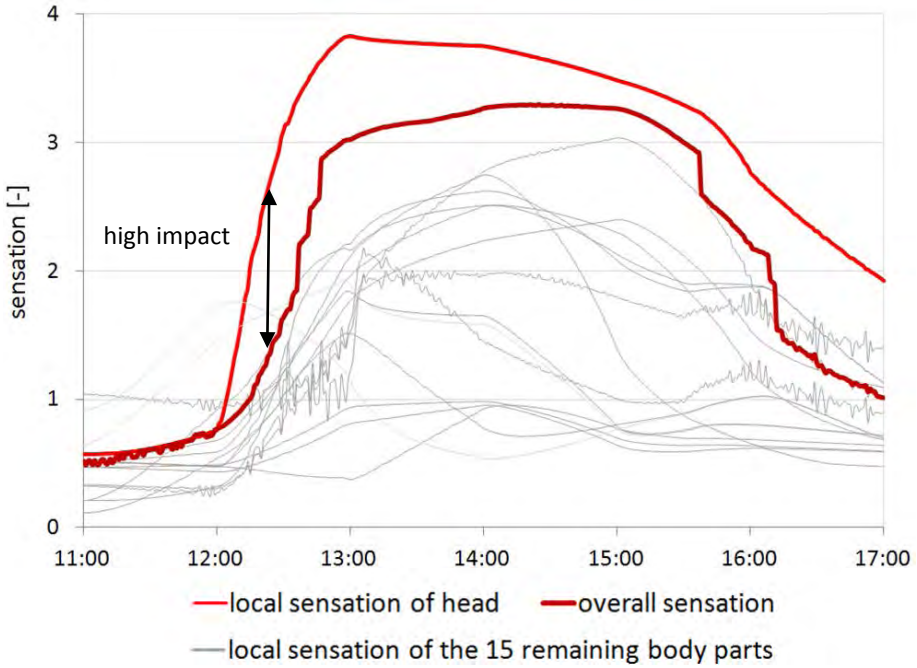

Figure 7b: Local and overall sensation for solar coating without exterior shade, head highlighted

In contrast to the low influence of the lower extremities, Fig. 7b shows how strongly the head reacts to the solar radiation and how the overall sensation follows the slope of the head's local sensation.

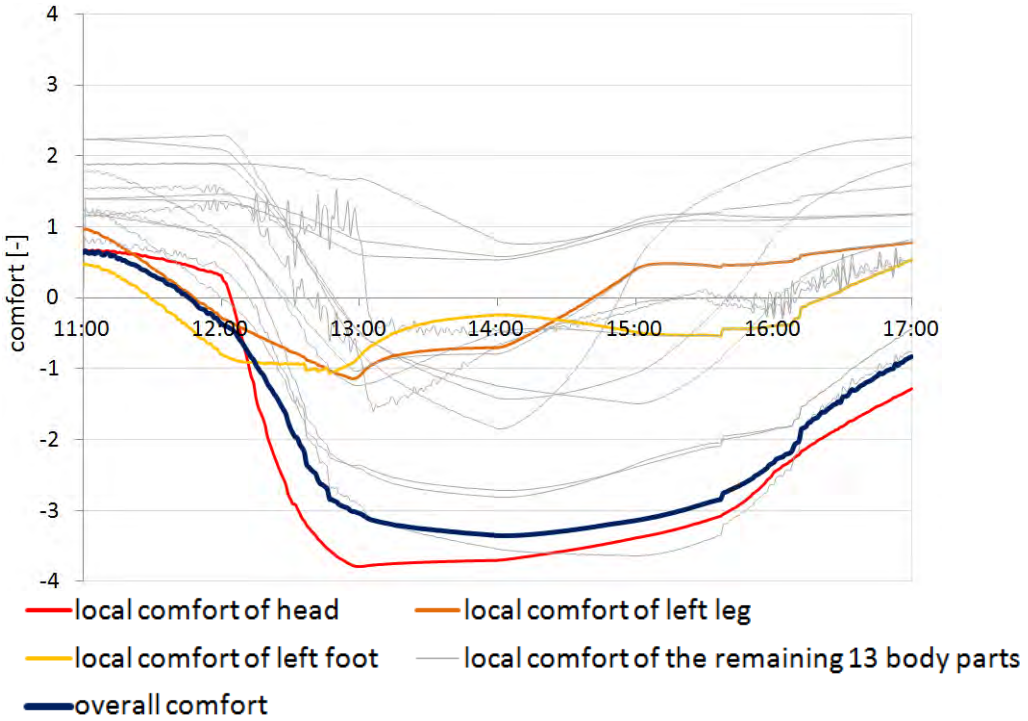

Figure 8: Local and overall comfort for solar coating without exterior shade, head, left leg and left foot highlighted

Fig. 8 shows how local and overall sensation translates to local and overall comfort. Any comfort value above -0.5 corresponds to the vote “comfortable” or better. Although Fig. 7a shows the left leg and left foot experiencing a warm sensation the occupant does not become uncomfortable. The head, which is more sensitive to heat, not only experiences a sensation of “very hot” in the presence of direct radiation (Fig. 7b) but it also becomes “very uncomfortable” (Fig. 8). The most uncomfortable body parts determine the overall perception of comfort; hence the overall vote drops to “very uncomfortable” at 1 pm and only reaches a comfortable level again at 5 pm.

6.3 The decision making process – comparing options

While the results presented in 6.2 demonstrate the process of assessing comfort based on a detailed solar load calculation, it is usually of greater interest to know how different project options compare to each other in terms of comfort. In this case study the main question to answer was whether an external shade as shown in Fig. 4 is needed to provide a satisfying comfort level, or if the use of a solar coating alone is sufficient.

In order to assess the effects of the solar coating alone without exterior shade, a base case with a clear glass IGU (w/o coating) was used as a reference (black lines).

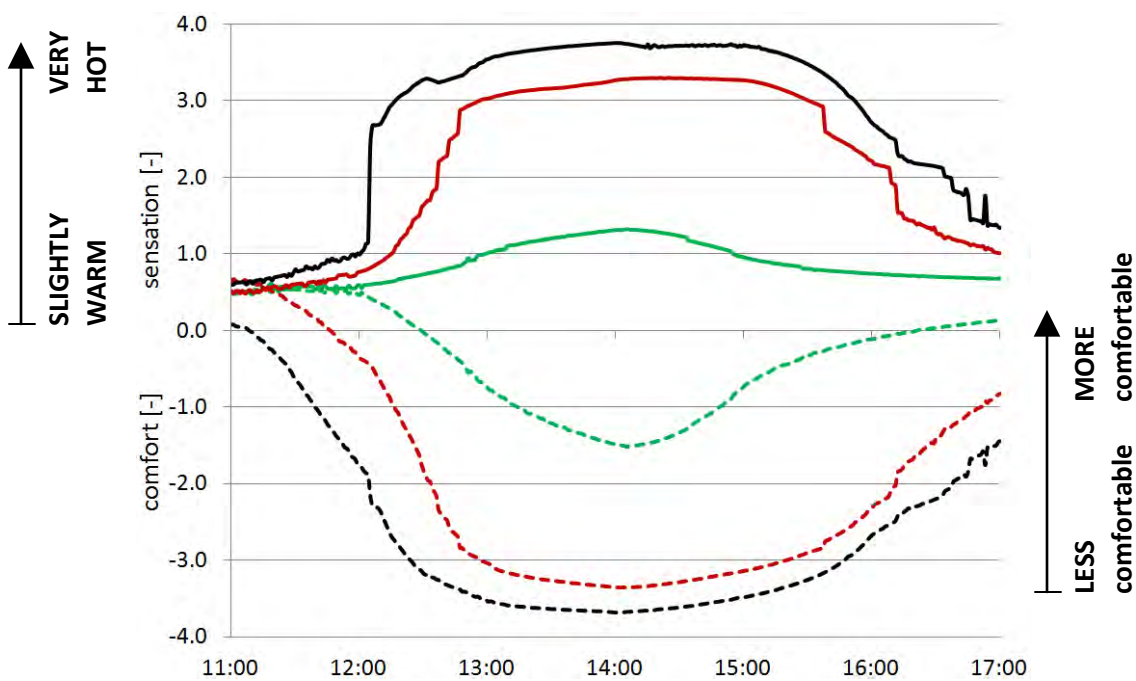

Figure 9: Overall sensation (bold) and overall comfort (dashed) for different options

Black: Clear glass IGU
Red: HP solar coating
Green: HP solar coating + exterior shade

While a person sitting in the patient room without an exterior shade would feel hot for most of the afternoon, adding the shade would make the person feel only “slightly warm” and therefore more comfortable. Although the solar coating with an SHGC of 33 % can reduce the heat load into the room substantially compared to a non-coated IGU, it does not reduce the thermal discomfort significantly. The direct radiation that hits the occupant on sensitive body parts such as the head leads automatically to a perception of the room conditions as very uncomfortably hot.

6.4 Discussion of the results

The presented results of the case study are the first ones in a series of calculations which will be published in a future paper in the Journal of Building Physics. In this paper a single, extremely clear summer day with maximum radiation was chosen in order to demonstrate the methodology and the outcome. In the next step, different sky conditions including more cloudy days and different seasons (fall, winter, spring) will be assessed.

As with all modeling, each of the used simulation tools is subject to assumptions which imply a certain uncertainty in the results. As these assumptions do not change within one set of simulation runs, numerical approaches are generally well-suited for parametric studies and the comparison of different options, but they may be less reliable in terms of absolute values. Comparing simulation results with observations in the field will increase the confidence when interpreting numerical values.

In this paper the focus lies on the assessment of thermal comfort. Other determining areas such as the general load on the HVAC system and the influence on daylight conditions in the room have to be investigated separately. Although a solar coating alone might not be favorable in terms of thermal comfort because it permits direct radiation in the perimeter zone, it is still efficient to reduce the solar load in the room and might be sufficient when no occupants are significantly exposed to the sun. On the other hand a certain shading system might be rejected for its impact on the daylight conditions in the room though it provides a thermally comfortable indoor climate. Which solution to choose and which trade-offs to make always depends on the specifics of the buildings and the priorities of the client.

7 CONCLUSIONS

The presented example (Section 4) and the case study (Section 6), as well as results from previous studies show that the comfort level in a building is higher when direct sun impinging on the occupant is avoided. Diffusing shading systems improve best the quality of the indoor conditions, especially in buildings with large transparent areas and unfavorable orientations, because they reduce direct radiation on sensitive body parts and local discomfort.

The newly developed software tool *SoLoCalc* together with a multi-segment physiology model and the corresponding comfort equations allows quantifying the effects of different shading systems, solar coatings, frits etc. on the occupants' perception of thermal comfort in the room. The innovation of *SoLoCalc* is the use of bi-directional scattering distribution functions for complex fenestration systems (exterior and interior venetian blinds, coatings, frits etc) as provided by the *Window6* software.

As shown in the case study, it is now possible to rank options for a given project by running parametric studies of solar load, thermal sensation and comfort while accounting for all important project conditions such as the geographic location of the building, the facade orientation, room geometry, window-to-wall ratio and the occupant's position in the perimeter zone.

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- Lawrence Berkeley National Laboratory (LBNL)
  – *Window6*
- National Institute of Standards and Technology (NIST)
  – *View3D*
- ZGF Architects Portland
  – funding for the case study

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Typical clothing ensemble insulation levels for 16 body parts

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Abstract

In order to accurately simulate skin and core temperatures and thermal comfort, some human physiology and comfort models now divide the human body into multiple body parts (such as head, hand, chest etc). Most of these parts are normally covered with clothing insulation, which must be quantified in the simulation. Unfortunately, existing clothing insulation databases only characterize clothing insulation for the whole body, not for individual body parts. That means every body part has the same clothing insulation level, even over the head and hands. In this study, we measured clothing insulation for 40 typical clothing ensembles using a 16-segment thermal manikin, and present here the insulation values for each body part, as well as for the whole body.

Keywords - Clothing insulation, thermal manikin, thermal physiology model, thermal comfort model

1. Introduction

In the past, clothing insulation data have been available only for the body as a whole [1-3]. This is because early human physiology and thermal comfort models treat the whole-body surface as one segment [4]. As models become more advanced and divide the body into individual parts [5, 6], whole-body clothing insulation values become inadequate. In this study, we obtained body-part-specific insulations using a 16-part thermal manikin, testing 40 typical clothing ensembles for summer, spring/fall, and winter. The clothing insulations are provided for each of the 16 body parts and for the whole-body (because left and right extremities can be considered identical, 10 parts are reported). A new database is provided for use by multiple-segmented models of physiology and comfort.

2. Method

The experiments were carried out in a climate chamber at the Center for the Built Environment (CBE) at University of California, Berkeley, between September and December 2012. The manikin used in the experiment is an average-sized Danish adult female as shown in Fig. 1. Its 16 segments are independently controlled and measured. Table 1 shows the 16 segments and

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their respective surface areas. Table 2 shows the experimental conditions. Air temperature was kept at 20ºC, and was measured continuously at 0.1m, 0.6m and 1.1 m height. All skin temperatures were set at 34ºC. The testing followed the recommended ASTM and ISO procedures [7, 8]. Each test continued for 2 hours to ensure that the manikin reached a stable condition. The manikin was seated in a mesh arm chair whose insulation level was measured.

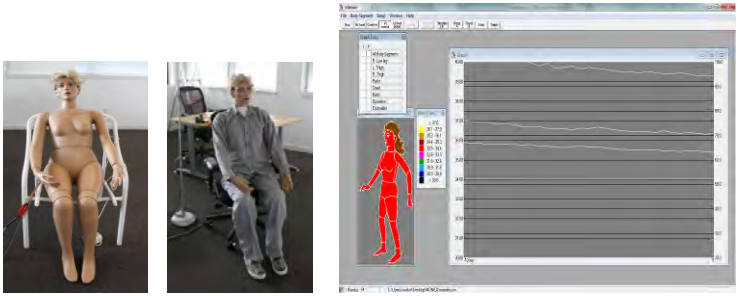

Fig. 1 Thermal manikin control screen(left) , climate chamber and thermal manikin_ 'Monica' (right)

Table 1 Body segments and respective areas of the manikin

	Name of Part	Area(m^2)
1	Left Foot	0.043
2	Right Foot	0.041
3	Left Leg	0.089
4	Right Leg	0.089
5	Left Thigh	0.16
6	Right Thigh	0.165
7	Pelvis	0.182
8	Head	0.1
9	Left Hand	0.038
10	Right Hand	0.037
11	Left Arm	0.052
12	Right Arm	0.052
13	Left Shoulder	0.073
14	Right Shoulder	0.073
15	Chest	0.144
16	Back	0.133
Total		1.471

Table 2. Experimental Conditions

Ambient temp (ºC).	Skin temp.	RH (%)	Air_ velocity	Posture	Chair
20.22 ±0.13	34	50	0.1	Sitting on a chair	Mesh arm chair

3. Clothing Ensemble Conditions and Clothing insulation results

We calculated the total insulation for 16 body parts for 40 typical ensembles including the nude condition, using Equation 1 and the manikin skin temperatures $T_{s,i}$ and heat fluxes $Q_{t,,i}$ measured for each body part:

$$I_{t,i} = (T_{s,i}-T_a)/(0.155*Q_{t,i}) \qquad [1]$$

where Ta is the ambient air temperature, and 1 clo = 0.155 m^2 ℃/W. The intrinsic insulation of the clothing itself was calculated by Equation 2:

$$I_{cl,i} = I_{t,i} - I_a/f_{cl} = I_{t,i} - I_a/(1+0.3\ I_{cl,i}) \qquad [2]$$

The thermal resistance of the nude body sitting on the mesh chair was measured as 0.8 clo during this experiment.

For whole-body clothing insulation, the summer ensembles tested range from 0.22 ~ 0. 57 clo. The spring/fall ensembles range from 0.53~0.80 clo. The winter clothing ranges from 0.8 clo to 1.4clo.

Table 3 shows all clothing ensembles and clothing insulation results. Values for left and right extremities are averaged and combined.

4. Conclusion

We measured and calculated the clothing insulation for 16 body parts for 40 typical ensembles. The values are useful for multi-segmented models of thermophysiology and comfort. They will be available in a public database maintained at the Center for the Built Environment.

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Table 3. Intrinsic clothing insulation for 40 ensembles

Clothing ensemble	*description*	*Whole-body*	*head*	*chest*	*back*	*Upper arm*	*Lower arm*	*hand*	*pelvis*	*thigh*	*Lower leg*	*foot*
	Nude (Mesh Chair)	0.01	0.13	0.01	0.01	0.02	0.00	0.01	0.04	0.01	0.03	0.05
	Nude (Nude Chair)	-0.02	0.13	0.05	-0.14	-0.01	-0.01	-0.02	-0.01	-0.10	0.00	0.00
	panty	0.03	0.00	0.00	0.00	0.00	0.00	0.00	0.24	0.05	0.00	0.05
	Bra+panty	0.05	0.00	0.22	0.00	0.00	0.00	0.00	0.18	0.03	0.03	0.08
	Bra+panty, tank top, shorts sandals	0.22	0.00	0.57	0.27	0.04	0.02	0.02	0.92	0.51	0.01	0.38
	Bra+panty long-sleeve shirt, shorts sandals	0.43	0.00	1.43	1.02	0.29	0.22	0.01	1.45	0.57	0.01	0.40
	Bra+panty sleeveless dress, sandals	0.29	0.00	0.85	0.48	0.00	0.00	0.00	0.94	0.72	0.00	0.41
	Bra+panty T-shirt, long pants,socks, sneakers	0.52	0.00	1.14	0.84	0.42	0.00	0.00	1.04	0.58	0.62	0.82

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Clothing ensemble	*description*	*Whole-body*	*head*	*chest*	*back*	*Upper arm*	*Lower arm*	*hand*	*pelvis*	*thigh*	*Lower leg*	*foot*
	Bra+panty sleeveless dress, cardigan, sandals	0.53	0.00	1.78	1.42	0.65	0.41	0.05	1.19	0.77	0.00	0.39
	Bra+panty song-sleeve dress, socks,sneakers	0.54	0.00	1.49	1.10	0.72	0.58	0.03	0.91	0.73	0.07	0.77
	Bra+panty long-sleeve dress, cardigan socks,sneakers	0.67	0.00	2.05	1.32	1.14	0.63	0.04	1.39	0.84	0.05	0.78
	Bra+panty tank top, skirt,sandals	0.31	0.00	0.83	0.22	0.00	0.00	0.03	0.99	0.88	0.05	0.44
	Bra+panty long sleeve shirts, Skirt,Sandals	0.52	0.00	1.62	0.99	0.31	0.28	0.03	1.41	0.82	0.04	0.41
	Bra+panty dress shirts, skirt,stocking, formal shoes	0.62	0.00	1.58	0.99	0.91	0.64	0.04	1.31	0.87	0.05	0.81
	Bra+panty dress shirts, skirt, leggings, sandals	0.65	0.13	1.59	1.04	0.91	0.67	0.07	1.36	1.26	0.12	0.43
	Bra+panty Thin dress shirts,long pants, socks, sneakers	0.69	0.10	1.33	0.93	0.79	0.66	0.13	1.39	0.60	0.57	0.76

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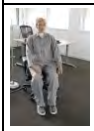
Clothing ensemble	*description*	*Whole-body*	*head*	*chest*	*back*	*Upper arm*	*Lower arm*	*hand*	*pelvis*	*thigh*	*Lower leg*	*foot*
	Bra+panty long sleeve shirts,long pants,socks, sneakers	0.80	0.00	2.47	1.48	0.98	0.58	0.04	1.58	0.69	0.65	0.89
	Bra+panty T-shirt, long sleeve shirts, long pants, socks,sneakers	0.83	0.25	3.88	2.28	1.89	1.41	0.16	2.07	0.83	0.66	0.86
	Bra+panty T-shirt, jeans,socks, sneakers	0.57	0.00	1.29	0.93	0.68	0.00	0.00	1.30	0.65	0.47	0.73
	Bra+panty long sleeve shirts, jeans,socks, sneakers	0.74	0.00	1.58	0.98	0.86	0.71	0.07	1.35	0.74	0.48	0.74
	Bra+panty Oxford shirts, Long thin pants, Socks,sneakers	0.83	0.16	1.39	1.02	0.83	0.69	0.22	1.34	1.02	0.68	0.80
	Bra+panty Thin dress shirts,Long pants, socks, sneakers	0.82	0.00	3.35	1.73	1.99	1.49	0.11	1.63	0.60	0.43	0.68
	Bra+panty Thin dress shirts(roll-up), Long pants, socks,sneakers	0.81	0.00	3.60	1.83	2.16	1.49	0.13	1.71	0.64	0.43	0.69
	Bra+panty Thin dress shirts(roll-up), Long pants, socks,sneakers	0.81	0.00	3.60	1.83	2.16	1.49	0.13	1.71	0.64	0.43	0.69

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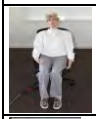
Clothing ensemble	*description*	*Whole-body*	*head*	*chest*	*back*	*Upper arm*	*Lower arm*	*hand*	*pelvis*	*thigh*	*Lower leg*	*foot*
	Bra+panty T-shirts, Short sleeve shirt, long pants, socks,sneakers	0.71	0.12	2.15	1.40	1.22	0.02	0.05	1.71	0.79	0.48	0.67
	Bra+panty, Sports shirts, long pants, socks,sneakers	0.80	0.05	1.92	1.31	1.14	0.86	0.18	1.41	0.59	0.49	0.75
	Bra+panty sports shirts, sports pants, sports socks, sports shoes	0.87	0.07	1.87	1.17	1.20	1.07	0.09	1.26	0.62	0.77	1.58
	Bra+panty sports shirts, sports pants, sports socks, sports shoes	0.87	0.07	1.87	1.17	1.20	1.07	0.09	1.26	0.62	0.77	1.58
	Bra+panty, Thin-dress shirts, Long pants, Wool sweater, Socks,sneakers	0.92	0.09	2.39	1.64	1.36	1.29	0.21	1.71	0.70	0.52	0.77
	Bra+panty, Thin dress shirts,Long pants,Cashmere sweater, Socks,sneakers	0.87	0.10	2.40	1.72	1.33	1.23	0.08	1.67	0.61	0.47	0.77
	Bra+panty T-shirt,Long sleeve shirts, Long pants, Winter Jacket Socks,sneakers	1.15	0.12	4.37	2.56	2.72	2.23	0.19	2.14	0.82	0.57	0.70
	Bra+panty T-shirts, long sleeve shirts, jeans, sports jumper, socks,sneakers	1.07	0.28	3.99	2.12	1.70	1.36	0.10	2.00	0.92	0.48	1.07

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Clothing ensemble	*description*	*Whole-body*	*head*	*chest*	*back*	*Upper arm*	*Lower arm*	*hand*	*pelvis*	*thigh*	*Lower leg*	*foot*
	Bra+panty T-shirt,long sleeve shirts, long pants, ventura jacket, socks,sneakers	0.90	0.09	2.66	1.42	1.32	0.99	0.14	1.57	0.73	0.66	0.85
	Bra+panty turtle neck, long pants, short trench coat, socks, sneakers	1.24	0.06	3.22	1.99	1.62	1.50	0.37	2.03	1.51	0.65	0.80
	Bra+panty T-shirt, long sleeve shirts, long pants, winter jacket socks,sneakers	1.08	0.31	3.76	2.00	2.00	1.37	0.25	1.92	0.83	0.61	1.08
	Bra+panty tank top, long sleeve shirts, blazer, skirt, sandals	0.86	0.00	3.24	1.81	1.98	1.13	0.07	2.06	1.19	0.04	0.44
	Bra+panty long sleeve shirts, wool skirt, socks, formal shoes	0.59	0.00	1.21	0.74	0.44	0.24	0.17	1.56	1.52	0.09	0.74
	Bra+panty turtleneck, wool skirt, socks, formal shoes	0.70	0.00	1.11	0.94	0.73	0.62	0.14	1.52	1.53	0.09	0.85
	Bra+panty long sleeve shirt, wool skirt, sweater, socks, formal shoes	0.91	0.14	2.82	1.53	1.22	0.97	0.08	1.79	1.53	0.11	0.83

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Clothing ensemble	*description*	*Whole-body*	*head*	*chest*	*back*	*Upper arm*	*Lower arm*	*hand*	*pelvis*	*thigh*	*Lower leg*	*foot*
	Bra+panty Thin dress shirts,slacks, tie, socks, sneakers	0.57	0.00	1.69	0.80	0.67	0.58	0.07	1.08	0.36	0.39	0.74
	Bra+panty Thin dress shirts,slacks, blazer,tie, belt, socks, formal shoes	0.93	0.00	3.60	1.83	2.16	1.49	0.13	1.71	0.64	0.43	0.69
	Bra+panty Long sleeve shirts,Long pants, Blazer, socks,sneakers	0.96	0.04	3.30	1.67	2.10	1.43	0.09	2.20	0.72	0.42	0.67
	Bra+panty T-shirts, Long sleeve shirts,Long Pants, Winter Jacket(Notica), socks,sneakers	1.05	0.04	3.88	2.26	1.82	1.46	0.17	1.97	0.81	0.57	0.78
	Bra+panty T-shirts,Long sleeve shirts, Long Pants, Winter Jacket, socks,sneakers	1.18	0.65	5.26	3.07	3.14	2.07	0.08	2.20	0.67	0.54	0.77
	Bra+panty Turtle neck, Ski-Jumper, Skin pants sports socks, sports shoes	1.84	0.89	5.24	2.87	2.55	2.16	0.46	2.64	1.49	1.82	1.56
	Bra+panty Turtle neck, Ski-Jumper and hood, Skin pants sports socks, sports shoes	1.87	1.63	5.12	2.70	2.58	2.16	0.49	2.57	1.44	1.76	1.54

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Clothing ensemble	*description*	*Whole-body*	*head*	*chest*	*back*	*Upper arm*	*Lower arm*	*hand*	*pelvis*	*thigh*	*Lower leg*	*foot*
	Bra+panty, Turtle neck, Goose down, ski pants, sports socks, sports shoes	2.53	1.17	15.44	5.50	6.55	5.58	0.35	5.20	2.12	1.70	1.54
	Bra+panty, Turtle neck, Goose down-with hood, ski pants, sports socks, sports shoes	2.75	3.52	12.62	3.99	6.20	5.73	0.53	5.05	2.11	1.81	1.58
	Bra+panty, Turtle neck, Goose down-with hood and gloves, ski pants, sports socks, sports shoes	3.27	3.92	16.13	4.47	7.12	5.37	2.54	5.71	2.14	1.82	1.61

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Moving Air For Comfort

By Edward Arens, Ph.D., Member ASHRAE; **Stephen Turner, P.E.,** Member ASHRAE; **Hui Zhang, Ph.D.;** And **Gwelen Paliaga,** Associate Member ASHRAE

Air movement can be an energy-efficient alternative to air cooling. However, in recent years it has been regarded more as a possible source of undesirable draft. Thermal comfort standards set room air speed limits low, even for temperatures as warm as 26°C (79°F). An exception was granted if the air speed source was under individual control, such as a window in a private office, or a desk fan, but only above 26°C (79°F).

Recent studies of occupied buildings provide consistent evidence that large percentages of occupants prefer more air movement than what they currently have, and small percentages prefer less. This is true in all conditions that occupants perceived as warm, thermally neutral, and even slightly cool. In terms of temperature, above 22.5°C (72.5°F) there is a small risk of draft and a strong preference for more air movement. This article describes these field study findings.

Responding to such findings, ANSI/ASHRAE Standard 55-2004, *Thermal Environmental Conditions for Human Occupancy*, is being updated with new provisions[1] that allow elevated air speed to broadly offset the need to cool the air in warm conditions,

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Thermal Sensation	Air Speed Range (m/s)	Percentage of Occupants Who Prefer			(N)	T_{op} (Standard Deviation) (°C)
		Less Air	No Change	More Air		
Cold (< −2.5)	0 to 0.2	33.33	46.85	19.82	111	22.66 (0.91)
	≥0.2	50.00	42.30	7.69	26	23.50 (1.45)
Cool (−2.5 to −1.5)	0 to 0.2	13.07	60.47	26.47	597	22.92 (1.08)
	≥0.2	11.55	72.51	15.94	251	24.28 (2.0)
Slightly Cool (−1.5 to −0.5)	0 to 0.2	10.75	53.08	36.17	1153	23.05 (1.23)
	≥0.2	11.35	62.23	26.42	458	24.59 (2.16)
Neutral (±0.5)	0 to 0.2	2.62	51.46	45.92	1407	23.30 (1.23)
	≥0.2	4.62	57.26	38.12	585	24.86 (2.03)
Slightly Warm (0.5 to 1.5)	0 to 0.2	2.31	27.73	69.95	822	23.65 (1.41)
	≥0.2	3.36	30.87	65.77	298	25.46 (1.85)
Warm (1.5 to 2.5)	0 to 0.2	4.24	18.37	77.39	283	23.75 (1.58)
	≥0.2	4.96	28.93	66.12	121	25.79 (2.08)
Hot (>2.5)	0 to 0.2	4.55	0	95.45	22	24.96 (1.28)
	≥0.2	7.14	14.29	78.57	14	26.23 (2.04)

Table 1: Air movement preferences by thermal sensation and for two ranges of air speed, n = 6,148.

replacing provisions that originated primarily from climate chamber studies.

Building Occupants' Air Movement Preferences

Under ASHRAE sponsorship, field studies of occupant comfort in buildings worldwide have been assembled in a database.[2] Eleven studies in the database (comprising 53 buildings and 6,148 sets of data) included direct questions about the occupants' air movement preference. These studies (from Montreal and Honolulu, Sydney, Kalgoorlie, and Townsville in Australia) are used for the following analyses. The six Honolulu buildings are schools, and the others are office buildings. Except for two naturally ventilated schools and one mixed-mode office in Sydney, all the buildings are fully air conditioned.

Table 1 lists occupants' stated air movement preference by the thermal sensation they reported at the same time. They are also arranged into two speed ranges: V<0.2 m/s (39 fpm; 0.44 mph) representing conditions below the draft limit in Standard 55-2004, and V≥0.2 m/s (39 fpm), representing potentially drafty conditions (for the latter, the mean air speed was 0.32 m/s [63 fpm]). It is obvious that when people felt "neutral" or "warm" ("slightly warm," "warm," or "hot"), a small percentage of them (7% or less) wanted less air speed. This is true even for the higher speed range (V≥0.2 m/s [39 fpm]) in which at least 93% of people accepted the higher air speed, or wanted even more. For both speed ranges, it is only under "cold" sensation that more people "want less" air speed than "want more." Even under "cool" and "slightly cool" sensations, substantially more people preferred more air speed than less.

The associated operative temperatures are also shown.* Under the same thermal sensation category, the operative temperature for the higher air speed is between 1.5 K to 2 K (3°F to 4°F) higher than the temperature for lower speed. This difference illustrates the trade-off between air speed and temperature in producing equivalent levels of comfort.

Air Movement for People Feeling Neutral to Slightly Warm

The following figures summarize air movement preferences in the thermal sensation range of –0.7 to 1.5. A sensation of –0.7 is a little cooler than the usual definition of neutral, but it matches the cool end of the Category III range (for acceptable existing buildings) in the European standard CEN 15251.[3] The figures exclude sensations above 1.5 because, although *Table 1* shows that air movement is clearly welcome when sensation is above 1.5, this sensation is too warm for normal office environments and ideally would not occur often. The –0.7 to 1.5 range should give a conservative estimate of air movement preference for neutral and slightly warm people.

Figure 1 shows that under the sensation range (–0.7 to 1.5), far more people (52%) wanted more air movement than less air movement (3%). The percentage wanting more was greater than the percentage preferring "no change."

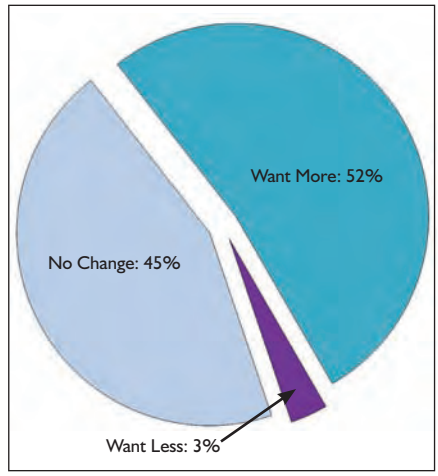
Figure 2 shows that when their air speed was at the higher range (≥0.2 m/s [39 fpm], *Figure 2*), a much higher percentage of people (47%) wanted more, or accepted the higher air speed (no change = 49%), than wanted less (4%).

Figure 3 shows that in the surveys, occupants also were asked whether the air movement was "acceptable." Twenty-nine percent of all occupants said it was not. Looking at these

* The operative temperature is a measure combining the air temperature (affecting the body by convection) and the surface temperatures of the surroundings (affecting the body by radiant exchange). It is defined as the uniform temperature of an imaginary black enclosure in which an occupant would exchange the same amount of heat by radiation plus convection as in the actual nonuniform environment.

respondent's preference votes, one can see that the unacceptable air movement is due to too little air movement, not too much. 84% wanted more, 7% wanted less (*Figure 3*).

When people felt the air speed was not acceptable, *and* when they were experiencing the higher range (≥0.2 m/s [39 fpm]), did they feel the speed was too much and, therefore, not acceptable? *Figure 4* shows that for the higher speed range, when people said that they felt the speed was not acceptable, the unacceptability was still due to insufficient air movement. The percentage wanting more (73%) is still far more than the percentage wanting less (17%). The difference is now smaller, but it is still clear that the majority found the speed unacceptable because there wasn't enough, even with the speed in the higher range.



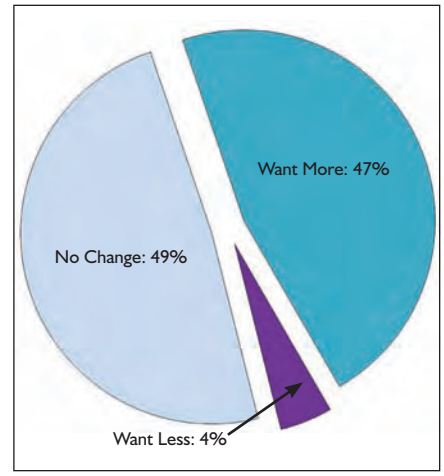


Figure 1 (left): Air movement preference (sensation –0.7 to 1.5), all air speeds (n = 3,230). Figure 2 (right): Air movement preference (sensation –0.7 to 1.5), air speed ≥ 0.2 m/s (n = 924).

One may also calculate the draft risk percentage (DR) for all the database's occupants based on the equations provided in the previous ASHRAE standard, using air speed, temperature, and turbulence intensity as inputs. Looking at the air movement preference for those people when their DR exceeded 20% (thermal sensation again within –0.7 and 1.5, n=172), the percentage wanting less air speed was 8%; 59% wanted no change, and 33% wanted more. The result is that 92% of a population predicted to be at an unacceptable risk of draft accepted their air speed or actually wanted it higher.

Summarizing these ASHRAE field study results, it is clear that for sensations –0.7 to 1.5, air movement should be encouraged. The air movement should not be made so great that it leaves people feeling cold, but a certain amount of it does answer a basic need found in the surveys, and can offset an increase in temperature in the space. Similar results have been found for a building in which occupants have personal or group control over window ventilation.[4]

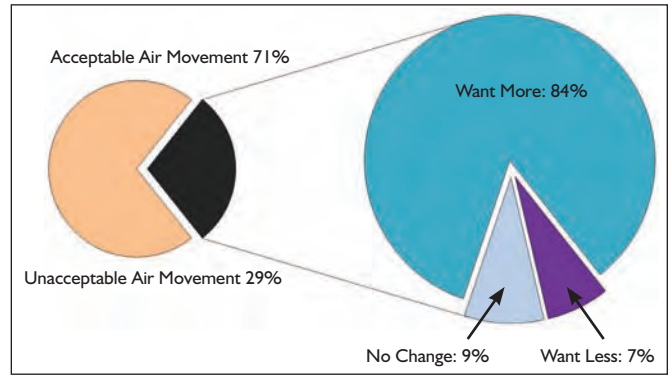


Figure 3: Air movement preference for people (sensation –0.7 to 1.5) who said the air speed was unacceptable. (n = 2,091).

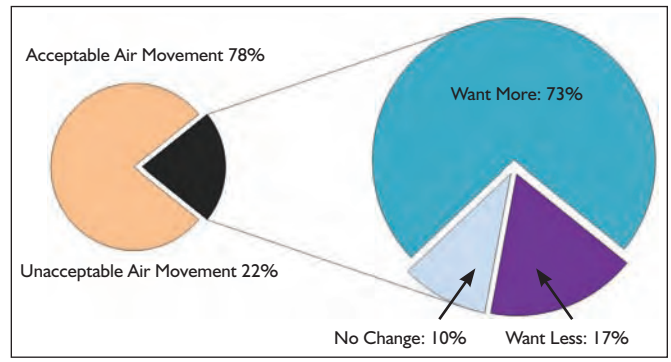


Figure 4: Air movement preference for people (sensation –0.7 to 1.5) who said the air speed was unacceptable, V≥ 0.2 m/s (n = 324).

Provisions for Elevated Air Speed in Standard 55

Standard 55-2004 will soon provide a two-step procedure for setting a comfort zone for temperature, radiation, humidity, and air movement. Both steps of this process can be carried out using the ASHRAE Thermal Comfort Tool,[5] or can be interpolated from figures in the standard. The first step, unchanged from the previous standard, combines air temperature and radiant temperature in the index "operative temperature" as defined earlier, which is then combined with humidity to produce comfort zones for still-air conditions displayed on a psychrometric chart. This step uses the predicted mean vote (PMV) human heat balance model, combining these environmental variables with the occupant's clothing level (expressed in the insulation unit, clo) and activity level (expressed in the metabolic rate unit, met).[6,7]

The second step evaluates elevated air speeds. It uses a different model that better accounts for convective cooling of the body. The standard effective temperature (SET) thermo-physiological model is based on long-standing ASHRAE research and practice.[6,8] Equal heat balance and skin-wettedness for different air speeds can be plotted in

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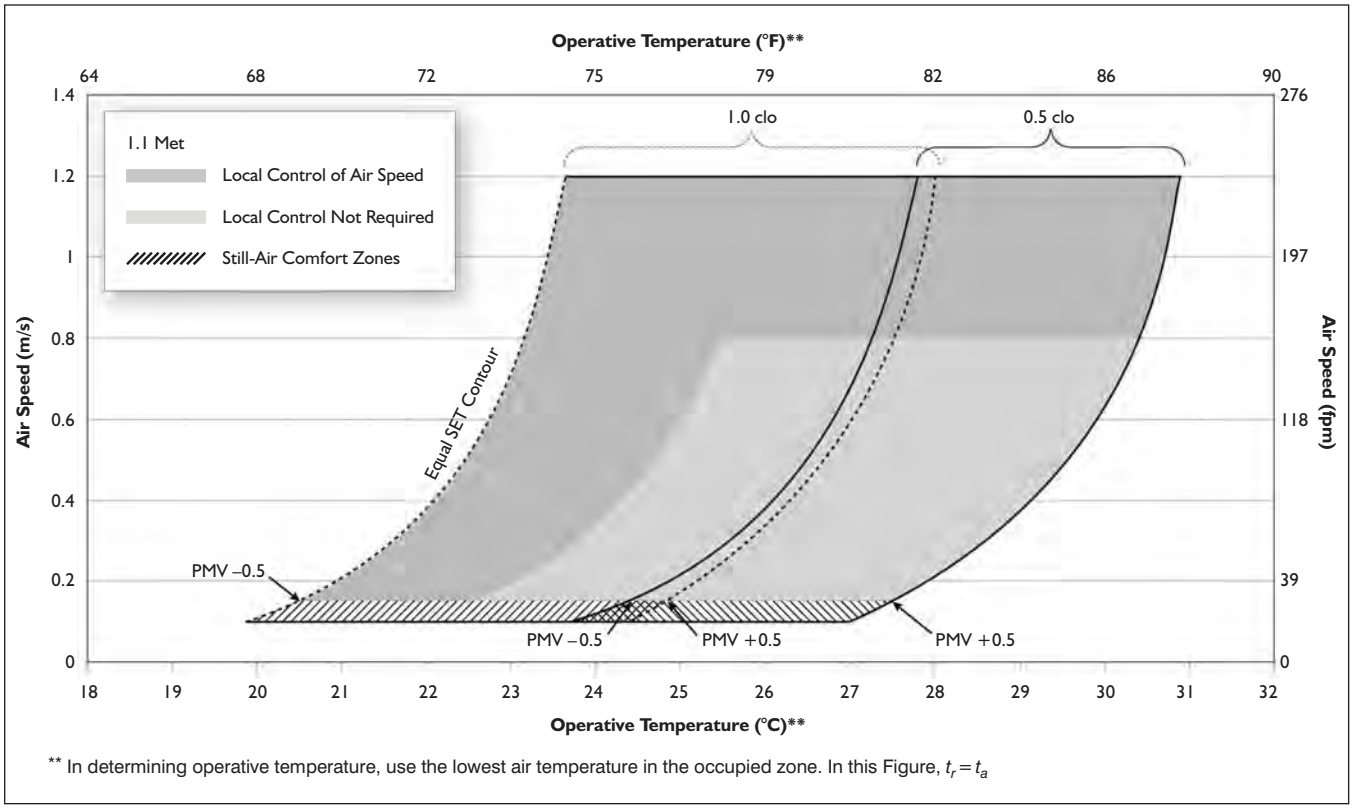

Figure 5: Elevated air speed for warm air temperatures.

terms of SET† contours. If one starts with the underlying PMV comfort zone, these SET contours form the boundaries of an air-movement comfort zone in which air speed is plotted against operative temperature.

These steps are shown in *Figure 5*, taken from the new standard. *Figure 5* shows two example comfort zones, for the clothing insulation levels: 0.5 clo (e.g., summer short-sleeved shirt and slacks) and 1.0 clo (winter heavy business suit), and the sedentary activity level of 1.1 met. These clo and met levels have been traditionally used in the standard's first figure, which plots PMV-based still-air comfort zones on a psychrometric chart. In *Figure 5*, those zones are transposed to the bottom of the y-axis, in the region defined as "still air" (0.1 to 0.15 m/s [20 fpm to 30 fpm]), for 50% relative humidity. The comfort zones each extend from –0.5 PMV (the thermal sensation midway between "neutral" and "slightly cool") and +0.5 (between "neutral" and "slightly warm"). One can see some temperature dependence on air speed within this still-air region, as predicted by the PMV model.

Comfort envelope. Starting with these still-air zones, 0.5 and 1.0 clo comfort envelopes for air speeds above 0.15 m/s (30 fpm) are generated using the SET model. SET contours of equal heat-loss relate a range of air speeds and temperatures to the equivalent PMV value in the still-air zones. These two comfort envelopes can be interpolated to roughly account for other clothing levels, but in practice, envelopes for any clothing, activity, and humidity level would be determined more accurately using the ASHRAE Thermal Comfort Tool.

Limits to air speed. *Figure 5* provides additional limits not based on SET. These are divided into two categories, with and without local control.

With local control. The full equal heat-loss envelope for a given clothing level in *Figure 5* applies when control of local air speed is provided to occupants. For control over their local air speed, control directly accessible to occupants must be provided for every six occupants (or less) or for every 84 m^2 (900 ft^2) (or less). The range of control shall encompass air speeds suitable to maintain comfort for sedentary occupants. The air speed should be adjustable continuously or in maximum steps of 0.25 m/s (50 fpm) as measured at the occupant's location.

Without local control. Within the equal heat-loss envelope, if occupants do not have control over the local air speed in their space, limits apply as shown by the light gray area in *Figure 5*. They apply for light, primarily sedentary office activities. For other types of activities or occupancies, such as retail or warehousing, these limits may differ or not be necessary. There is little quantitative information available for other types of occupancies.

For operative temperatures *above 25.5°C* (77.9°F), the upper limit to air speed is 0.8 m/s (160 fpm).

† **temperature, standard effective (SET):** the temperature of an imaginary environment at 50% RH, <0.1 m/s air speed, and with *mean radiant temperature equal to air temperature*, in which the total heat loss from the skin of an imaginary occupant with an activity level of 1.0 met and a clothing level of 0.6 clo is the same as that from a person in the actual environment, with actual clothing and activity level.[1,6,8]

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Below 22.5°C (72.5°F), the limit is 0.15 m/s (30 fpm) to avoid cold discomfort due to draft. 0.15 m/s (30 fpm) is the upper limit of the still-air zone, and is coincidentally equal to the air speed self-generated by an office worker at 1.2 met.

Between 22.5°C and 25.5°C (72.5°F and 77.9°F), the allowable speed follows an equal SET curve dividing the light and dark gray areas. This local-control boundary between 22.5°C and 25.5°C (72.5°F and 77.9°F) is not linked to any PMV comfort zone, but is based on temperatures that have been observed in office field studies to cause virtually no risk of draft.[9]

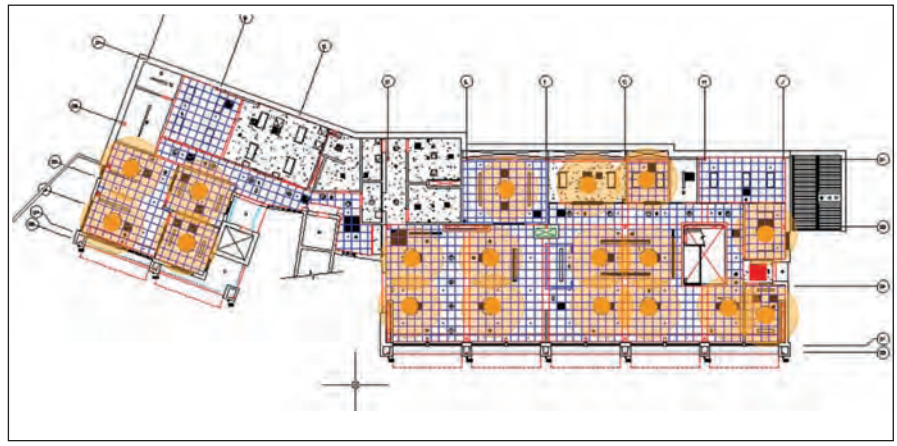

Figure 6: Orinda City Hall plan view. Orange circles are the ceiling fans, and the light orange circles are the areas in which 0.75 m/s (150 ft/min) air velocity is supplied in the occupied zone.

Using a temperature-based local-control boundary provides a substantial practical advantage in environmental control: decentralized air movement sources (such as ceiling fans) can be manufactured or retrofitted with temperature-based speed controllers for automatic control of common spaces.

The 1 clo zone in *Figure 5* represents a higher level of clothing than is typical for situations in which air-movement cooling is used. For more typical summer clothing levels such as 0.5 and 0.6 clo, the limits to air speed and temperature will be determined by the cool-side SET boundary of the comfort zone, not by this occupant-control boundary.

Air speed measurement. Above 22.5°C (72.5°F), the overall heat balance of the body determines comfort. For this, the standard uses the average air speed of the three measurement heights that represent the seated occupied zone—0.1, 0.6, and 1.1 m (4, 24, and 43 in.). Measurements should be taken in the occupants' estimated vicinity. To prevent anyone from attempting to use the standard to justify cold-air dumping from poorly designed HVAC supply air outlets, the coldest temperature in the occupied zone must be used to represent the entire zone. This also renders it improbable that cold air-distribution systems can be used as the source of elevated air movement because they are likely to produce cold air jets, as opposed to using fans, stack effect, or window ventilation as a means to elevate air speed.

Below 22.5°C (72.5°F), the problem is avoiding thermal discomfort on a local, usually unclothed, part of the body. The SET and PMV models do not distinguish between clothed and unclothed portions of the body, so the following conservative approach is adopted: the *maximum* mean air speed of the three measurement heights, and the *lowest* air temperature is used for the SET calculations, thereby over-predicting the whole-body cooling to a level that more closely approximates the cooling of the most affected local part.

There is no longer a requirement to measure turbulence intensity for determining draft risk.

Validation. In lab studies where air speed has been imposed on people, conditions on the cool side of the without-local-control (light gray zone) boundary have been found to be comfortable, even recommended as optimal.[10,11,12] Imposed air movement does not appear to be a problem at these temperatures. This is also supported in the ASHRAE field database described previously. For air speeds above 0.4 m/s (79 fpm) and air temperatures between 22.5 and 25.5°C (73°F and 78°F), 32% wanted more air movement, 59% wanted no change, and only 9% wanted less.

On the warm side of 25.5°C (77.9°F), ceiling fan studies (with no occupant control) have found 1 m/s (197 fpm, 2.2 mph) acceptable to 77% of subjects[13] and above (95%,[14] 80% to 100%[15]). A frontal desktop breeze of 0.8 m/s (157 fpm) air speed produced an acceptability level of 80%.[12] Subjects in preferred air speed studies[16,17,18] often chose air speeds well above 1 m/s (197 fpm). Fountain[19] summarized many of these studies, from which one can see that the SET-based curves in Standard 55 represent the laboratory results well.

Air speed requirements for industrial occupancies. Standard 55 has also added an informative appendix (Addendum *f*, forthcoming) to address more strenuous indoor activities than office work, such as industrial work. For such work in hot environments, sweating may be acceptable to the occupants. Elevated air speeds provide high rates of cooling by sweat evaporation. Using SET as described above, hot conditions with high air speeds may be equated to their temperature equivalents at lower air speeds. There is no specified limit to air speed in this range.

Impact in Practice

The new provisions allow designers to use fans, stack effect, or window ventilation to offset mechanical cooling, or in some climates to supplant it entirely. The following two examples are of offices using ceiling fans.

City Hall, Orinda, Calif., (*Figure 6*). Ceiling fans are used to enable a mixed-mode system that combines passive cooling through operable windows and a stack vent with indirect-direct evaporative cooling, instead of compressor-based cooling. The building is 1300 m^2 (14,000 ft^2) with 40 full-time occupants along with a community meeting room and conference rooms.

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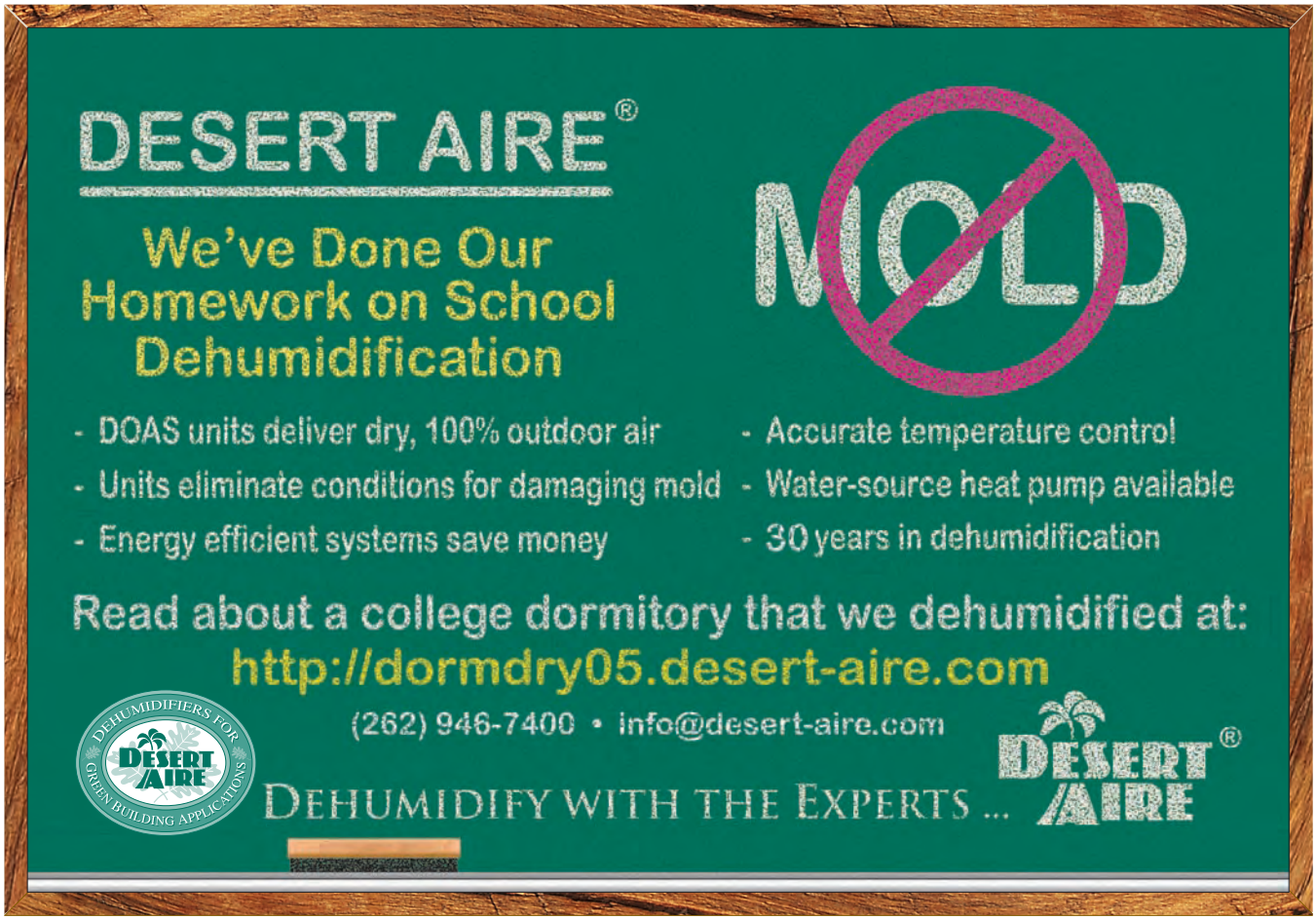
Figure 7: Orinda City Hall with ceiling fans. The left photo shows open space, and the right photo shows cubicles and hallway.

The building management system controls clerestory window opening based on indoor-outdoor temperature differential and illuminates signs that instruct occupants to open/close their windows. A total of 36 fans with 1321 mm (52 in.) blade diameter are provided in office, conference, and circulation spaces. *Figure 7* shows two spaces, one with four fans arrayed in a 100 m^2 (1,076 ft^2) open space (left image), another with 10 fans arrayed for 273 m^2 (2,938 ft^2) of cubicles and hallways (right image). Fans are controlled individually or in groups of two or three from wall-mounted switches and have three speed levels. The fans use approximately 30 to 70 W per fan. In the design, they were assumed to allow the air temperature



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to increase 2.6 K (4.7°F) at 0.75 m/s (148 fpm). Simulations of indoor temperature and comfort predict that the fans are critical to maintain comfort for 100 to 200 hours per year when the evaporative cooler has limited capacity and space temperatures rise.

The building has been occupied for a year and a half, and the users are happy with the fans.

Loisos+Ubbelohde Office, Alameda, Calif., (*Figure 8*). This 223 m^2 (2,400 ft^2) design office is cooled only by a system of operable windows, automated shading, and ceiling and desk fans. The occupancy is 18, for an occupant density of 12 m^2 (133 ft^2) per person. Ceiling fans are individually controllable from one control panel in the entry area. They are controlled in groups of one or two. Fans have high-efficiency twisted airfoil blades designed by Florida Solar Energy Center/AeroVironment, generating approximately 40% higher cfm/W than paddle-blade fans, drawing between 9 W to 50 W from low to full speed (0.4 to 1.6 m/s [79 fpm to 315 fpm]).[20]

The system has been in place through one summer in its present configuration, with owner and occupants expressing satisfaction. The owner feels that the fan density shown could be increased for more even coverage. He suggests that air motion sectional diagrams, similar to photometric curves, be published to facilitate the design layout of fans in accordance with work stations.

Conclusions

Forthcoming addenda to Standard 55-2004 include new provisions for using air movement to offset warm air temperatures. They are based on field studies that consistently demonstrate that in neutral and warm environments, people want more air speed. The SET model is used to generate comfort zones in which elevated air speed offsets warm air temperature. New criteria for group local control are specified, making it possible to use air movement in open-plan offices as shown in the two examples. The new air movement provisions should assist the design of both passive and mechanical systems such as natural ventilation, thermal mass cooling, evaporative cooling, and mixed-mode HVAC.

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Figure 8: Design office in Alameda, Calif. Left photo shows ceiling fans in open office space, and the right photo shows ceiling fans.

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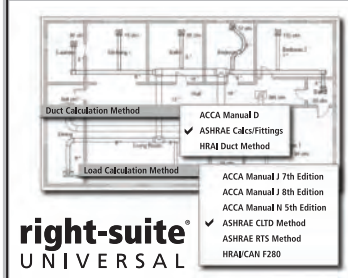

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Are 'class A' temperature requirements realistic or desirable?

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ABSTRACT

It requires more energy to maintain a narrow indoor temperature range than a broader range, in which the building may be allowed to float with reduced conditioning for longer periods of time. A narrow range should presumably be preferable to the building occupants to justify its increased energy cost. At what widths are temperature ranges detected, preferred, or judged unacceptable? Three databases of occupant satisfaction in buildings are used to examine the acceptability of three classes of temperature range currently employed in the ISO and European standards, and proposed for the ASHRAE standard. These are alternatively identified as class A, B, and C, or category I, II, and III, but their specifications are identical. The A class (I category) is found to confer no relative satisfaction benefit to individuals or to realistic building occupancies. In addition, the differences in B and C class satisfaction are small.

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1. Introduction

We are facing the need to reduce building energy use dramatically. Reductions of 50 to 90% are being called for in both existing and new buildings over the next 30 years. This will not be technologically or financially easy to accomplish. From the climate perspective, early reductions are more valuable than later ones, so there is a particular need for measures with immediate impact.

Since over 15% of US energy use goes to heating and cooling interior space, it is prudent to examine our requirements for interior space conditioning to make sure that truly unnecessary conditioning is minimized, and that potentially energy-efficient space-conditioning technologies are not penalized in present and future buildings.

2. Background

ISO standard 7730 [1] and CEN 15251 [2] include three categories of environmental quality for mechanically cooled buildings: I, II, and III, in which category I requires the tightest control of interior conditions. This schema has been proposed for ASHRAE Standard 55 as well [3], using the terms class A, B, and C to represent the three categories (see Table 1 for their thermal specifications). In this paper we will use the class terminology. Class A if adopted would require tighter control than the existing standard 55, whose specifications are now at the B level.

Class categories have been created for several aspects of indoor comfort. One addresses the overall thermal environment surrounding the occupant, as it affects the average heat balance of the occupant. Several others address localized environmental variables that might cause local discomfort: draught, vertical air temperature difference, floor temperature, and radiant temperature asymmetry. A 'predicted percent dissatisfied' (PPD) is associated with each class level for each variable, ranging in all from 3 to 35%.

In this paper, we address occupants' comfort and acceptance of the overall thermal environment surrounding them. In all three standards, this is quantified in terms of the variable 'predicted mean vote' (PMV), the outcome of a human heat balance model. The PMV model [4] incorporates the four basic environmental variables that affect human heat balance: temperature, humidity, thermal radiation, and air movement. It predicts the mean thermal sensation of a group of people exposed to that environment, assuming they all have the same clothing insulation and activity level. The mean thermal sensation values range from negative (cold) through neutral to positive (hot).

Thermal acceptability (or its complement, PPD) is typically indirectly inferred from PMV. The three classes in ISO 7730 and CEN 15251 each have a PMV range whose center is zero, with class A

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Table 1
Thermal requirements of the three classes of indoor environment

Class (category)	PMV	Temperature range for typical clo and met (K)	PPD (%)
A (I)	$-0.2 < \text{PMV} < +0.2$	2	<6
B (II)	$-0.5 < \text{PMV} < +0.5$	4	<10
C (III)	$-0.7 < \text{PMV} < +0.7$	6	<15

being defined by the tightest band (Table 1) [5,6]. The PPDs associated with the three PMV ranges vary from 6 to 15%.

It should be noted that the PMV/PPD relationship has been derived from climate chamber studies. There have been few rigorous tests of PPD or thermal acceptability for actual occupants in office buildings (e.g. [7–10]). In the case of draught, the PPD from climate chamber experiments has been found to differ substantially from dissatisfaction data gathered in actual buildings [11,12].

Table 1 also gives temperature ranges for a typical office occupancy. In practice it is possible to represent PMV ranges in terms of temperature by assuming the occupants' clothing insulation and metabolic activity levels. In still-air internal environments with moderate humidity, the thermal environment can be adequately characterized by operative temperature or globe temperature, combining the air and surface radiant temperatures. When air and interior surface temperatures are roughly the same, it is sufficient to measure air temperature alone. Building thermal control is ultimately implemented in terms of the temperature measured in a thermostat, a combination of air and radiant temperatures.

Maintaining a narrow PMV or temperature range requires more HVAC energy than maintaining a broad range. In commercial buildings, heating/cooling energy-use changes roughly 7% for each degree K change in interior temperature range (space–temperature–setpoint deadband), starting from a class B baseline [13]. In residences, the change per degree is conservatively 10%, exceeding 20%/K in the vicinity of the baseline [14]. Therefore, the difference between adjacent classes translates to about 12% of annual heating + cooling energy use for commercial buildings. For residential and small commercial, it is closer to 20–30%.

This invites the questions: is the so-called class A thermal environment worth its inherent energy penalty? Is it even more comfortable than class B (or C) in a realistic environment?

3. Statement of the problem

To date there has been very little critical analysis of how the three classes' PMV ranges are accepted by occupants in real buildings. There are a number of considerations: (1) individuals have their own acceptable temperature ranges. These differ from those of other individuals based on personal characteristics, clothing and activity levels. (2) Building occupancies consist of collections of such differing individuals who often have to share common thermal environments. How do real occupancy acceptances relate to the standard's three PMV ranges, bearing in mind that the PMV and PPD values are based on uniform occupancies (i.e. sedentary activity and uniform clothing in laboratory studies)?

Until recently, few laboratory or field studies included the direct question about whether an environment was acceptable or not. Dissatisfaction, acceptability, and general thermal comfort have all been inferred from whole-body thermal sensation votes. The relationship between these different metrics of subjective comfort has yet to be successfully established.

There may well be other factors determining thermal acceptance that are outside the parameters of a heat balance model, such as perceived access to environmental controls, or freedom of dress code.

3.1. Individuals' acceptance range

If occupants were in individual rooms with individual thermostats which they could adjust according to their clothing, met, and personal preferences, would they control their temperature around a narrow band as in class A, or would they control to a band more like class B or C? If, rather than controlling the temperature themselves, it were controlled for them, would they really prefer class A control to class B control? Would they notice the difference?

These simple but important questions can be tested using observations of acceptance and satisfaction from field studies in which comprehensive indoor environmental measurements were recorded, as well as occupants' clothing and met rate. From these, an occupant's PMV values can be calculated, allowing individual observations to be placed in class A, B, and C relative to the occupant's personal ideal (neutral) temperature. The occupants' acceptance of their state would indicate the extent of their *individual* need for tight environmental control as imposed by the HVAC system.

3.2. Variability in building occupant populations

In many cases we would expect multiple occupants to be in zones controlled by a single thermostat. The thermostat's temperature setpoint will not be ideal for each of the occupants because in addition to normal person-to-person variability people vary in their clothing and metabolic rate. These variations cause each individual's acceptable temperature range to differ from those of the other occupants with whom they share the space. Is it then possible to find any narrow band of control setpoints (e.g. class A, with a width of 2 K) that a substantial majority of an occupant population finds acceptable?

Here again, we can test these questions directly from field studies in which building *populations* have been directly asked about thermal acceptability (or in its absence, thermal sensation). Are there temperature bands that are acceptable to a substantial percentage of occupants? How wide are these temperature bands? Do they differ with seasonal variations of the outdoor temperature? The portions of the acceptability variation caused by *clothing* and *metabolic rate* could be obtained from the same field study databases. It would be useful to know how much these clothing- and

Table 2
Results of the ASHRAE RP-884 database enquiry.

Office rating	PMV range	Townsville summer wet season (% accept)	Townsville summer dry season (% accept)	Kalgoorlie–Boulder summer season (% accept)	Kalgoorlie–Boulder winter season (% accept)	Montreal summer season (% accept)	Montreal winter season (% accept)
Class A	±0.2	74.4 (n = 160)	84.2 (n = 203)	88.9 (n = 163)	86.7 (n = 166)	81.2 (n = 129)	86.3 (n = 102)
Class B	±0.5	77.5 (n = 346)	81.0 (n = 394)	87.8 (n = 320)	84.5 (n = 373)	84.2 (n = 272)	86.0 (n = 250)
Class C	±0.7	77.2 (n = 425)	79.2 (n = 476)	88.3 (n = 393)	84.3 (n = 452)	84.4 (n = 333)	86.0 (n = 321)

Observed acceptability percentages (sample sizes indicated in parentheses) within the three office quality ratings defined in terms of PMV.

Table 3
Results of the SCATs database enquiry (the percentages voting comfortable, and voting within the three central categories of the ASHRAE thermal sensation scale).

PMV range	*N*	% voting in central three categories of ASHRAE scale (±SE)	% comfortable (overall comfort ≥ 4)
$-0.2 < \text{PMV} < 0.2$	966	87.2 ± 1.1	80.0 ± 1.3
$-0.5 < \text{PMV} < 0.5$	2210	87.9 ± 0.7	78.6 ± 0.9
$-0.7 < \text{PMV} < 0.7$	2902	87.3 ± 0.6	78.2 ± 0.7

'Overall comfort' (in column 5) is based on a six-point scale: 1, very uncomfortable; 2, moderately uncomfortable; 3, slightly uncomfortable; 4, slightly comfortable; 5, moderately comfortable; 6, very comfortable. The numbers are the percentages on the comfortable portion of the scale.

metabolic-rate-related acceptable temperature variations compare to the 2, 4, and 6 K sizes of the class A, B, and C bands of setpoint control.

4. Method

We used three databases of field studies in which detailed occupant surveys are matched with coincident measurements made with high-grade instrumentation. The field studies use a variety of question formats, so that the analyses used subsets of the available data that have high compatibility.

(1) The ASHRAE RP 884 database [7]. This public-domain database contains quality-controlled data from thermal comfort field studies conducted in various countries and climate zones around the world. We excerpted three field studies from the database; ASHRAE RP-702 in Townsville, Australia, ASHRAE RP-921 in Kalgoorlie–Boulder, Australia, and ASHRAE RP-821 in Montreal, Canada. All 45 buildings in these three field surveys were centrally air-conditioned office buildings. The three studies represent three distinct climate zones, and both summer and winter surveys were conducted in each location. The 3350 questionnaires were accompanied by simultaneous and local indoor climate measurements (air temperature, mean radiant temperature, air speed and humidity) made in accordance with ASHRAE Std 55 [3] and ISO 7726 (1985) [15] requirements. Each questionnaire included estimates of the subject's clothing ensemble insulation and metabolic rate.
(2) The Smart Controls And Thermal Comfort database (SCATs) [16]. This database includes subjective survey and occupants' environmental data on 26 office buildings in five European countries, France, Greece, Portugal, Sweden, and UK, and different building types. The total number of environmental questionnaires, accompanied by simultaneous indoor climate measurements and estimates of clothing insulation and metabolic rate, exceeds 4600, and covers the entire year. Experimental protocols were uniform across all surveys, apart from the language of presentation of the questionnaire.
(3) Berkeley City Center (BCC) Project database [17]. This is a modern naturally ventilated building that was monitored and surveyed in great detail for 3-week periods during both summer and winter. The total number of surveys was 2075, with 38 subjects in each season repeating the survey throughout the day.

5. Individual occupants' acceptance of the three classes

The three databases were examined to compare individuals' ratings of acceptability against which PMV ranges they were in.

5.1. ASHRAE database: PMV-based ranges

The PMV comfort index calculations in the ASHRAE RP-884 database were performed with a standardized software package distributed by ASHRAE [18]. The PMV calculations were based on 5-min averages across all three levels (0.1, 0.6 and 1.1 m) of the measured environmental parameters (ta, tr, rh and v). Occupant acceptance of the measured thermal environments was assessed directly with this questionnaire item:

Is the thermal environment acceptable to you? ACCEPTABLE/UNACCEPTABLE

In the discussion below, we equate the terms 'accept' and 'acceptable' with 'being satisfied with' and 'satisfactory' (conversely 'unacceptable' is equated to 'dissatisfied with'). We do not know whether there should be a distinction made here. It seems that for practical purposes there is no difference. The term 'satisfied' is rarely used in questionnaires, even though 'predicted percent dissatisfied' (PPD) is a commonly invoked metric. The acceptability percentages for the three classes in the three studies are presented in Table 2.

The results suggest that it does not matter to the occupants of these buildings whether the physical environmental conditions inside their air-conditioned office building comply with class A, B or C specifications; there is a fairly constant level of dissatisfaction across all grades of about 20%. The small differences between the three classes are not statistically significant. There is some variance between studies, with the general level of acceptability lower in Townsville's wet season and higher in Kalgoorlie's summer, but there is no consistent pattern of satisfaction between classes A, B and C.

One might conclude that occupants' sense of acceptability is unrelated to the narrow temperature ranges defined by the classes, or that PMV lacks the discriminatory power to predict acceptability within these ranges.

5.2. SCATs database

We did a similar study for the 26 buildings in the SCATs database (Table 3). There are several differences, however. Here the acceptability metric is replaced by a right-now measure of 'comfort', which the subjects could interpret to include thermal, acoustical, lighting, and air quality effects. In addition, percentage of occupants voting within the central three categories of the ASHRAE thermal sensation scale is also shown, since this is the operational definition of comfort within PPD, and integral to the way PMV has been proposed as the basis for grading office environments. The three central categories have often been interpreted as equivalent to acceptability although the linkage is tenuous and has not been well established in the published field studies.

Table 4
Results of the BCC database enquiry (observed acceptability percentages within the three office quality ratings defined in terms of PMV ranges).

PMV range	Sample size (inclusive)	Thermal acceptability (% ± SE)	Want warmer (%)	No change (%)	Want cooler (%)
±0.2	721	89.0 ± 1.2	9.9	62.9	27.2
±0.5	1427	87.3 ± 0.9	10.4	61.7	28.5
±0.7	1686	86.2 ± 0.8	10.7	59.3	29.9

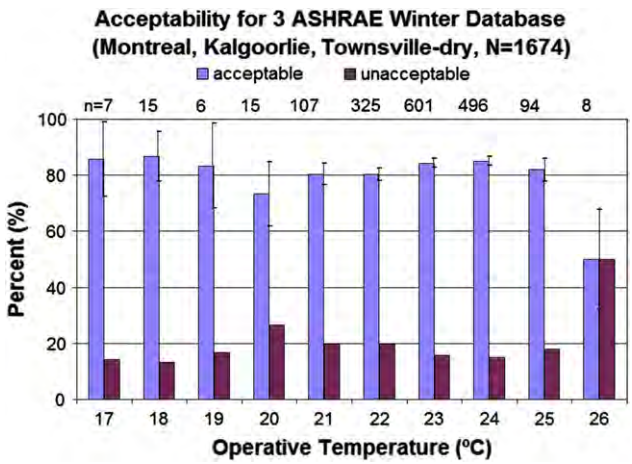

Fig. 1. Acceptability against temperature at the workstation, winter season (brackets represent 95% confidence intervals).

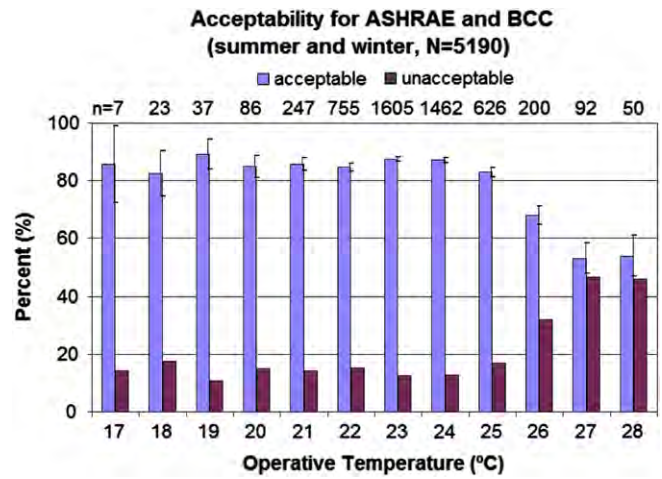

Fig. 3. Acceptability against temperature at the workstation, annual, pooled ASHRAE and BCC data.

Again the SCATs results show that there is little if any advantage in restricting the PMV range to ±0.2. The differences between the comfortable percentages within the PMV bands ±0.2, ±0.5, and ±0.7 are not significant on either measure ('comfort' or 'central three categories of the ASHRAE sensation scale').

As with the ASHRAE database, the conclusion must be that either PMV does not have the necessary discriminatory power, or that it does not matter to the occupants which of the three classes the environment belongs to.

5.3. BCC database

We did a similar analysis for the BCC database, shown in Table 4. Here thermal acceptability was directly assessed, as with the ASHRAE RP-884 analysis in Table 2. There is a small increase in acceptability with tightening range, but the differences between the three classes are not statistically significant. The warmer/cooler preferences are also shown, showing that the building conditions were not equally distributed around neutral. Almost 30% of the occupants perceived the building as being warm—three times as many people would have preferred cooler conditions than warmer. Nonetheless at least 86% regarded the environment as being acceptable.

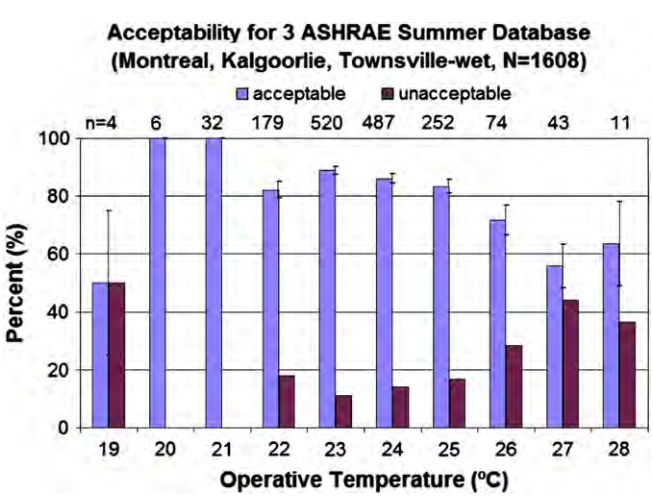

Fig. 2. Acceptability against temperature at the workstation, summer season.

5.4. Summary: individual acceptability

The PMV estimates underlying the tables are necessarily imprecise, since they entail estimates of the occupants' clothing insulation and metabolic rate at the time of the survey measurements. This will result in some 'blurring' of the class boundaries. However, such imprecision is intrinsic to any practical measurement of PMV – and a drawback to its use as a control variable.

Occupants' ranges of acceptance appear to be broader than the narrow range of PMV that defines class A, and perhaps also that of class B.

6. Acceptable temperature ranges for occupant populations

The ASHRAE (and class B) definition of an acceptable environment is one in which 80% of the occupants find it acceptable. For populations of individuals whose PMVs differ, the acceptable temperature range is best examined in terms of actual temperatures – operative, or globe – since temperature is the variable by which environments are usually controlled. The temperature range may be examined for various time periods, e.g. annual, seasonal,

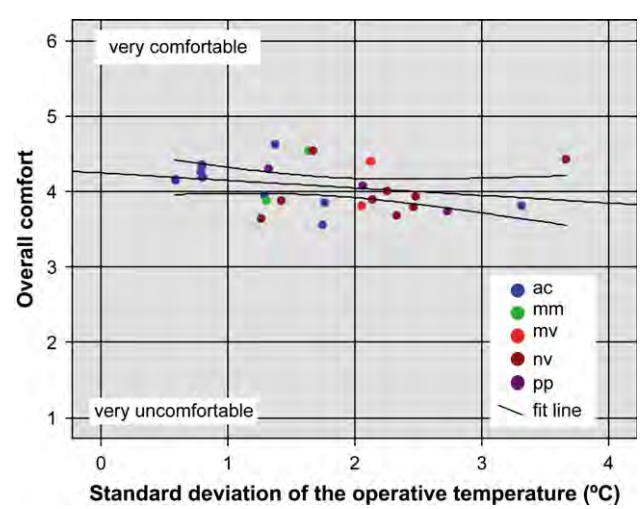

Fig. 4. Annual room temperature variability versus overall comfort (entire SCATs database).

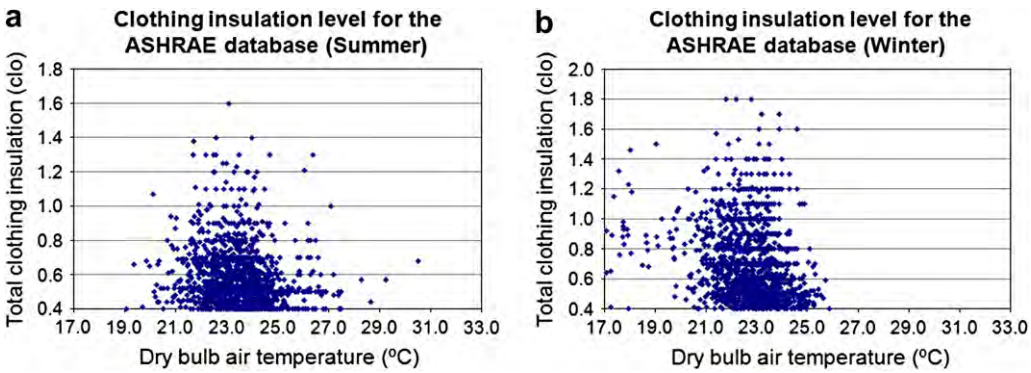

Fig. 5. (a) ASHRAE database clothing insulation for summer. (b) ASHRAE database clothing insulation for winter.

daily. The ASHRAE and BCC databases contain largely season-specific data. The SCATs data is analyzed in annual terms.

6.1. ASHRAE and BCC databases

Over the winter and summer test periods, the selected studies taken from the ASHRAE database showed a wide spread of operative temperatures acceptable to over 80% of the population, from 17 through 25 °C for winter (Fig. 1), and from 20 through 25 °C for summer (Fig. 2), although the fewness of the data makes for great uncertainty at the cold extremes. At 26 °C operative temperature, the percentage of the population voting acceptable dropped below 80% in both seasons, 72% in summer and 50% in winter.

A very similar 80% acceptability range is also found in the BCC database (not shown here). In winter the range was 19 through 25 °C, and in summer from 21 though 25 °C. In Fig. 3, all the ASHRAE and BCC data are pooled to give an acceptable range for the entire year, with a greater than 7 K range of acceptable temperature, from below 19 through 25 °C.

6.2. SCATs database

Fig. 4, from the SCATs database, shows the relationship of comfort to annual indoor temperature variation. The horizontal axis (the standard deviation) is a measure of indoor temperature variability over the whole year. The vertical axis is the mean assessment of overall comfort. Each point represents one building. (mm, mixed mode (few of these); mv, mechanical ventilation (a requirement for offices in some European lands; pp is a building in which some spaces were air-conditioned while others were not). The error lines are 95% confidence limits for the position of the regression line.

There is little relation between tight control (low standard deviation) and overall comfort. The relation, while in the expected direction, does not reach statistical significance even over the wide (~7 K) range of variability encountered.

6.3. Summary: population acceptability

The ASHRAE data indicate that acceptability began to fall at temperatures above 25 °C, but there was no clear lower temperature limit within the range of the data. The pooled (ASHRAE + BCC) data suggest a range from 19 to 25 °C. The SCATs data suggest that the overall variation of temperature in a building over a period of about a year makes little difference to the overall assessment of environmental comfort in that building. The generally null results of these analyses do nothing to encourage the belief that close control of the operative temperature will much improve comfort and acceptability. However, caution is necessary, because Figs. 1–4 include the effects of adaptation during the course of the surveys, whether seasonally as in the ASHRAE data, or year-round in the (combined ASHRAE + BCC) and SCATs data. Thus the temperature for thermal neutrality is continually varying. If it occurred in the course of a single day, one would not expect this full range of temperatures to be acceptable.

7. Sources of variation in occupants' thermal requirements: clothing level

Some of the variation in an occupant's acceptable range of temperature is due to clothing. The clothing levels observed in the three databases are presented in Figs. 5–7. Figs. 5 and 6 give the summer and winter clothing versus interior air temperature for the ASHRAE database and the Berkeley Civic Center. Fig. 7 gives clothing versus indoor operative temperature from the SCATs database. From all three databases, it is very clear that at the same air or operative temperature, the clothing insulation varies by about 0.8 clo unit, from 0.4 to 1.2 clo.

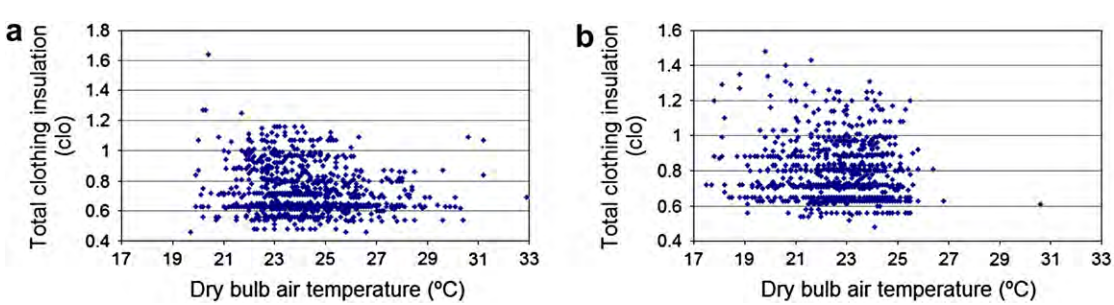

Fig. 6. (a) Berkeley Civic Center (a naturally ventilated building) clothing insulation for summer. (b) Berkeley Civic Center (a naturally ventilated building) clothing insulation for winter.

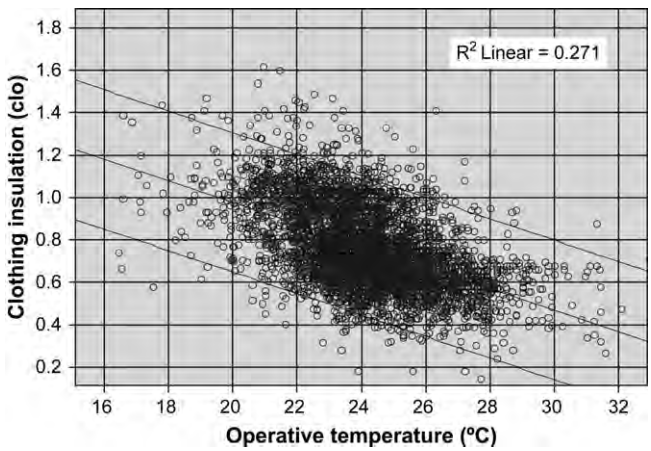

Fig. 7. SCATs: clothing versus indoor operative temperature. The bands enclose 95% of the observations and show a variation of about 0.6 clo at any given temperature.

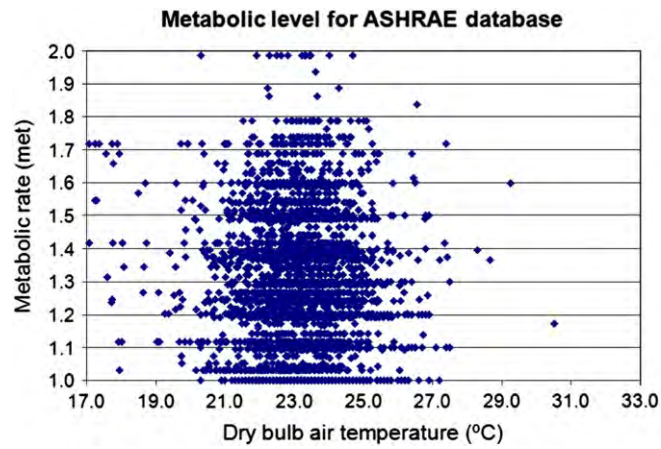

Fig. 9. ASHRAE database: metabolic level varies from 1.0 to 1.8 met.

Fig. 8 shows PMV simulation results for a range of clothing insulation values. It can be seen that a clothing change from 0.5 to 1 clo corresponds to a 3 K air temperature change, a change that exceeds the 2 K width of the proposed class A environment. Similarly, PMV varies approximately one scale unit for this spread of clothing at a given temperature near neutrality.

8. Sources of variation: activity level

Some of the variation in acceptable temperatures is due to differences in metabolic rate caused by occupants' varied activities. The results from the ASHRAE data, the Berkeley Civic Center, and the SCATs are presented in Figs. 9–11. These field data indicate large met-rate variability in realistic commercial building occupancies. The ASHRAE data cover metabolic levels from 1 to 1.8 met, the BCC metabolic range is from 1.1 to 1.4 met, and the SCATs data cover a much larger range, from 1 to 2.8 met.

How much does variation in the met level affect predicted PMV values? Fig. 12 shows the PMV variation over the met level from 1 (quietly seated) to 1.3 met (periodic standing and walking). The influence of met on PMV is larger when the environment is cool. At 21 °C, PMV changes about one scale unit when met changes from 1 to 1.3 met. The figure also shows that a metabolic change from 1 to 1.3 met corresponds to about a 3 K air temperature change.

Typical metabolic rate differences can be seen to cause as much change in PMV and air temperature as is caused by typical clothing variability (Fig. 8).

8.1. Summary: sources of variation in predicted acceptability

The observed clothing and metabolic variation in actual occupancies (Figs. 5–7, and 9–11) does not allow a single PMV or PPD value to differentiate between classes, for either design or operation of buildings. Properly, PMV should be calculated for a collection of metabolic rates and clothing levels and combined in some way to account for their distributions in the population. This would require empirical evidence that does not currently exist.

9. Discussion: shorter time periods

This paper does not address the acceptable ranges of temperature variation within shorter timeframes, such as a week, day, hour, or minute. Acceptable range is affected by how quickly people can adapt to changing conditions. Adaptation can be due to people's clothing behavior and to changes in their physiology. In most cases, clothing insulation is fixed on a daily basis by what people choose to wear that day (unless they can add or shed a layer at work), but it can vary from day to day, and it is usual for clothing insulation to differ between the seasons. Similarly, physiological adaptation to

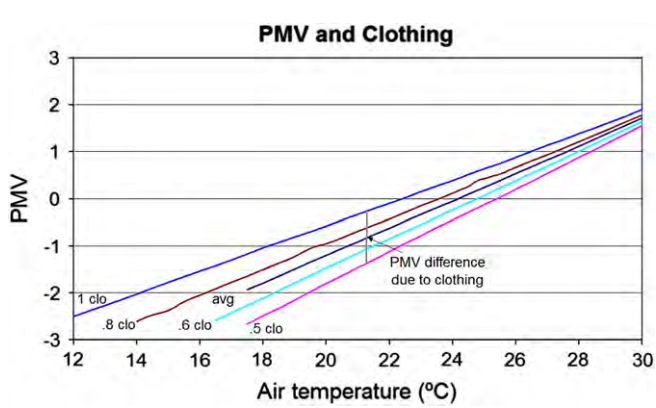

Fig. 8. Clothing insulation from 0.5 to 1 corresponds to PMV variation of about one scale unit or a 3 K air temperature difference, assuming the air and radiant temperatures are equal.

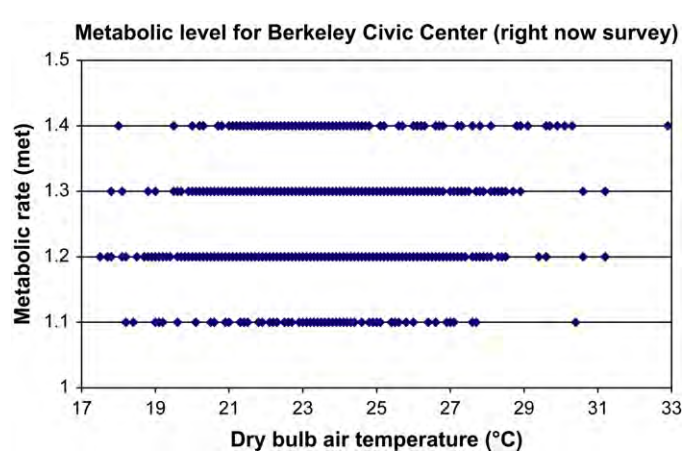

Fig. 10. Berkeley Civic Center: metabolic level varies from 1.1 to 1.4 met.

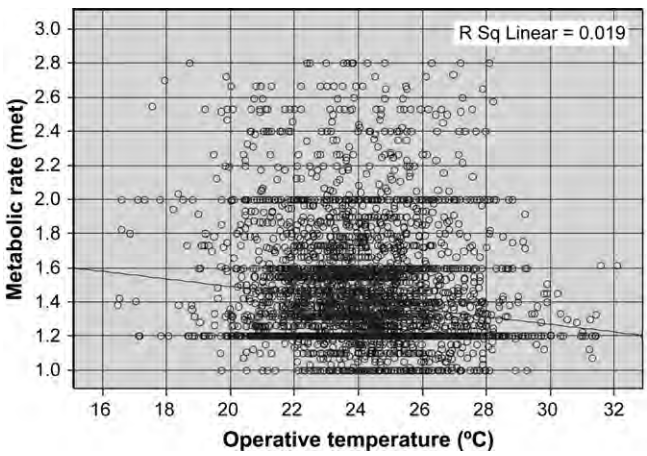

Fig. 11. SCATs data: metabolic rate varies from 1.0 to 2.8 met.

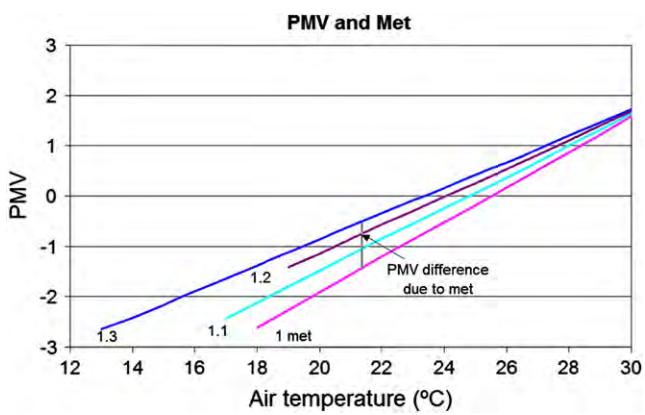

Fig. 12. Metabolic level from 1.1 to 1.3 corresponds to PMV variation of about one scale unit, or a temperature variation of 3 K (the figure assumes the air and radiant temperatures are equal).

changed thermal environments is a process requiring a few days, up to about a week. Physiology therefore also requires a more limited range over the course of a day, or from one day to the next, than over periods of a week or longer, when people may have adapted and can accept warmer or cooler conditions. Our data addresses periods of a week and longer.

10. Conclusions

In an analysis of high-quality field studies, the three classes do not exhibit different comfort/acceptability outcomes. The tightly air-temperature-controlled space (class A) does not provide higher acceptability for occupants than non-tightly air-temperature-controlled spaces (class B and C).

The theoretical basis of tight PMV/PPD building control is flawed. Real populations are diverse in clothing insulation and metabolic rate, causing variation in their acceptable temperature that is wider than the class deadband. PMV itself may lack the precision needed to handle the fine distinctions needed for the three-class system of control (see [19] for a more detailed discussion of this issue).

Class A as a category is unsupportable as a basis for environmental control in office buildings, given the energy costs of designing and controlling to its specifications. The most restrictive situation for a space temperature setpoint deadband involves within-day variation, where the neutral point has been reset daily. Even for this, the class B temperature range is the narrowest justifiable limit for control.

Building temperature ranges should be based instead on real-time empirical feedback about their occupants' requirements. In the future, one can envision measures that enhance occupant feedback capability being incorporated in normal building control and operation, and being specified in building designs.

If we do not see advantages in classifying buildings in terms of their temperature or PMV ranges, we could instead consider classifying them in terms of access to personal control, or their energy use in providing occupant comfort [20].

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Air temperature thresholds for indoor comfort and perceived air quality

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RESEARCH PAPER

Air temperature thresholds for indoor comfort and perceived air quality

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Air temperature thresholds are investigated and proposed for acceptable comfort in air-conditioned buildings. Using the ASHRAE database of field studies in which acceptability votes were obtained from occupants, it is shown that within the thresholds the acceptability is indistinguishable. Therefore, there is little gain from conditioning spaces to an 'optimum' air temperature, a practice that involves significant energy cost. However, beyond the thresholds there is a significant drop-off in acceptability. Ideally, air-conditioning would be used only when the environmental conditions are beyond the thresholds. The use of ceiling fans or personal environmental control systems broadens the threshold range. Thresholds are determined for both air-conditioned and ventilation-cooled buildings in the database. The equally acceptable range between the thresholds is 8–10 K in both types of buildings. It is possible that a perception of reduced air quality in warm environments could impose an upper temperature threshold. Perceived air quality is examined in two laboratory studies with air temperatures ranging from 18 to 30°C. Perceived air quality is seen to be closely correlated to thermal comfort rather than to temperature; as long as thermal comfort is maintained by air movement, perceived air quality will be acceptable. Relationships between temperature thresholds and productivity, operating set-points, and energy use are also discussed.

Keywords: adaptive comfort, air movement, air-conditioned buildings, free running, personal environmental control (PEC), thermal acceptability, thermal comfort

Les seuils de température de l'air sont étudiés et proposés pour un confort acceptable dans les immeubles climatisés. En utilisant les études de terrain figurant dans la base de données de la Société américaine des Ingénieurs en Chauffage, Réfrigération et Climatisation (ASHRAE), dans lesquelles il a été obtenu de la part des occupants des réponses relatives à l'acceptabilité, il est montré que, dans la limite des seuils, l'acceptabilité n'est pas discernable. Il y a par conséquent peu de bénéfice à climatiser des espaces à une température « optimale » de l'air ce qui êntraine des frais significatifs. Cependant, au-delà des seuils, l'on constate une baisse sensible de l'acceptabilité. Dans l'idéal, la climatisation ne serait utilisée que lorsque les conditions environnementales dépassent les seuils. L'utilisation de ventilateurs de plafond ou de systèmes de contrôle individuel de l'environnement élargit la plage des seuils. Les seuils sont définis à la fois pour les immeubles climatisés et les immeubles refroidis par ventilation figurant dans la base de données. La plage offrant une égale acceptabilité entre les seuils se situe à 8-10 K dans les deux types d'immeubles. Il est possible que la perception d'une réduction de la qualité de l'air dans des environnements chauds puisse imposer un seuil de température supérieur. La qualité de l'air perçue est examinée dans deux études de laboratoire pour des températures de l'air allant de 18 °C à 30 °C. Il apparaît que la qualité de l'air perçue est étroitement corrélée au confort thermique plutôt qu'à la température; tant que le confort thermique est maintenu par la circulation d'air, la qualité de l'air perçue sera acceptable. Il est également discuté des relations entre les seuils de température et la productivité, les points de consigne et l'utilisation de l'énergie.

Mots clés: confort adaptatif, circulation d'air, immeubles climatisés, fonctionnement libre, contrôle individuel de l'environnement (CIE), acceptabilité thermique, confort thermique

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Introduction

A suite of physiological and behavioural adjustments allows one to adapt unconsciously to one's environments. As a result, one is able to experience a range of environmental conditions as equally comfortable, or acceptable. For thermal environments in buildings, this acceptable range roughly centres on the 'neutral' or 'optimal' temperature for the particular activity or occupancy in the building. Since individual occupants will differ physiologically, and may have somewhat different requirements for their behaviour, their optimal temperatures (and acceptable comfort ranges) will be diverse. A change from a given temperature can increase the acceptability for some people while decreasing it for others. This effect lowers the maximum percentage of the population that finds any given temperature acceptable, and broadens the zone that any given percentage of the population finds acceptable. Beyond the zone boundaries, acceptability will drop off, both for a single person and for a given majority of the population. Although the comfort literature (and the air-conditioning industry) tends to focus on neutral and optimal temperatures, it is perhaps more important to examine these zone boundaries and find ways to expand them because they are the 'thresholds' beyond which energy is needed to condition interior space.

Historically, mechanical air-conditioning (heating, ventilation and air-conditioning systems, HVAC) has enabled designers to engineer buildings to previously unimaginable levels of certainty, guaranteeing standards' comfort criteria for specific fractions of the year (*e.g.* 99%). This type of engineering had not been done for the naturally ventilated (NV) or free-running designs of the past.

The shift toward air-conditioning over NV in design practice was accompanied by more restrictive comfort requirements. If one could do it, why not? From pre-HVAC to now, the 'comfort zone' has shrunk by at least 1 K at each end. The comfort of a building's entire occupancy is now extrapolated from the predicted thermal sensation of a single representative occupant (using predicted mean votes (PMV) calculated for one clothing and activity level); the extrapolation to the borders where unacceptability or dissatisfaction begins (using predicted percentage of dissatisfied (PPD)) is entirely based on laboratory studies (Standard 55 of the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), 2010). This approach is not based on real buildings with real populations of occupants, nor on direct measurement of comfort or acceptability. It is based on an ideal of what HVAC could make possible.

The need to address energy efficiency has forced the authors to look at space-conditioning requirements more directly to understand how they affect the occupants of actual functioning buildings. For this one must use the results of field studies in which occupants were surveyed and concurrent physical measurements taken. A number of such studies have been made. A database of their results has been accumulated under the auspices of ASHRAE (de Dear, 1998), allowing the analysis of real occupant populations in a number of counties worldwide. Another database of European results (SCATs) has been assembled for European countries (McCartney and Nicol, 2002).

Initial examination of these databases found that space-conditioning requirements in NV buildings differ from those in HVAC buildings. Analyses (de Dear and Brager, 1998) supported earlier suggestions (Nicol and Humphreys, 1972) that human adaptation was more active in NV buildings, and that zones of equal thermal sensation for occupants of NV buildings were generally broader than for those in HVAC buildings. The resulting 'adaptive' comfort zones have obvious energy-efficiency implications, and adaptive models of comfort have now been adopted into building codes for buildings with operable windows (ASHRAE, 2010) and for free-running buildings (Standard 15251 of Comité Européen de Normalisation (CEN), 2007).

The adaptive comfort zones, like earlier comfort zones, were developed from thermal sensation votes in occupant surveys, employing an assumed relationship between a person's warm/cool sensations and his/her thermal satisfaction. Satisfaction in field studies is assumed when sensation votes are between −1 (slightly cool) and + 1 (slightly warm) on a seven-point scale ranging from cold to hot. A more direct approach to determining comfort zones is to obtain the occupants' assessment of whether they find their thermal environment acceptable. (The question is usually a binary 'yes' or 'no' on the survey, but it can also be measured in a continuous scale from 'very unacceptable' to 'very acceptable'.) With this question the occupants are judging their thermal condition in the context of their expectations for the type of environment or work setting in which they find themselves. It should be the 'bottom-line' question for determining a building's comfort zone.

Not all field studies have asked an acceptability question, but a substantial number in the ASHRAE database have done so. A previous examination of the 'acceptability' votes in a subset of ASHRAE field studies showed a broader and flatter acceptable temperature range than when the range was extrapolated from thermal sensation votes (Arens *et al.*, 2010). The zone boundaries, or thresholds beyond which space conditioning is needed, tended to suggest 'thresholds', since the drop-off beyond them is rapid. In this paper the threshold concept is examined further using field survey data from the entire ASHRAE database of acceptability votes. Of particular interest are additional expanded threshold values that might be possible with energy-efficient cooling

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systems, such as ceiling fans and personal fans that offset warm temperatures, and personal heating systems that offset cool ambient temperatures.

The paper also examines whether temperature thresholds might be imposed by occupants' perception of air quality. Perceived air quality (PAQ) is a temperature-related issue. If perceptions of bad air quality inevitably occurred at warm temperatures (Fang *et al.*, 1998), the upper threshold of the comfort zone might be ultimately dictated by PAQ acceptability rather than by thermal acceptability. Recent laboratory test results are used to analyse this fact (Zhang *et al.*, 2010).

The paper briefly discusses relationships between temperature thresholds and occupant productivity, HVAC operating set-points and building energy use.

Method

Acceptability votes are used for determining temperature thresholds. Within the ASHRAE field study database, 16 studies of 71 buildings in seven locations include acceptability votes. These are assembled for analysis into Dataset 1. Dataset 2 supplements the ASHRAE database with acceptability votes from an additional building and location, studied in two seasons. It is an NV building: the Berkeley Civic Center (BCC) (Brager *et al.*, 2004).

Table 1 shows the study locations for the 72 buildings in Datasets 1 and 2. HVAC buildings are distinguished from NV plus mixed-mode (MM) buildings. MM buildings contain air-conditioning but operate at times in NV mode.

In these field studies, the acceptability scale is a binary scale. The occupants chose either 'yes' or 'no' to decide whether their thermal environments were acceptable. From these data, the percentage of occupants voting acceptable or unacceptable can be calculated for a range of temperature bins. The criterion for a successful design in comfort standards (ASHRAE, 2010) is 80% of the occupants satisfied. If one can equate 'satisfaction' with 'acceptability' (which seems reasonable), it can be said the warm and cold thresholds occur when acceptable votes drop below 80%.

The analysis of PAQ is performed using the BCC results (Dataset 2), since Database 1 surveys did not include a PAQ question. In the BCC study PAQ was measured with a seven-point scale ranging from 'very dissatisfied' to 'very satisfied'. The velocities measured in BCC were in general quite low. In order to find thresholds for warm conditions with higher levels of air movement, the results from two human subject studies of local cooling/heating devices (Datasets 3 and 4) were used. These studies were performed in an environmental chamber at the University of California at Berkeley at 18, 20, 25, 28 and 30°C, and the PAQ effect of temperature and air movement was tested, so the results of Datasets 3 and 4 are used together with Dataset 2 to define the PAQ thresholds. The PAQ scale used in the human subject tests is a continuous scale (Figure 1). It has a break in the middle to force a clear decision in the neutral region. In both environmental chamber studies, 18 college students experienced each test condition.

The results are presented below in two parts, 'I' for thermal acceptability and 'II' for PAQ.

Table 1 ASHRAE database locations analysed in this paper by building type

Location	Season	Number of buildings	T_{max} (°C)	T_{min} (°C)	T_{avg} (°C)
HVAC buildings *(n = 4730)*					
Kalgoorlie	Summer	21	31.7	19.8	23.7
	Winter	22	24.5	16.6	22.1
Townsville	Summer	12	27.7	21.2	23.8
	Winter	12	25.7	19.8	23.4
Montreal	Summer	12	26.9	21.0	23.6
	Winter	11	25.0	19.9	22.6
Sydney	Winter	2	23.8	20.9	22.3
Honolulu	Hot season	2	26.9	19.6	23.3
	Cool season	2	23.5	21.0	22.5
NV and MM buildings *(summer, n = 2512)*					
Berkeley	NV	1	30.3	19.7	21.4
Sydney	MM	1	27.3	20.8	24.0
Merseyside	NV	3	25.9	16.6	21.9
Athens	NV	6	36.4	17.7	30.1
NV and MM buildings *(winter, n = 2632)*					
Berkeley	NV	1	27.9	17.6	22.9
Sydney	MM	1	27.3	16.8	23.2
Merseyside	MM	1	25.9	18.7	23.4
Merseyside	NV	8	25.9	18.6	21.9
Honolulu	NV	4	27.6	23.1	26.1

Notes: n = number of votes in each category. Also shown are the number of buildings in each field study, and their mean measured indoor temperature maxima, minima and averages.
HVAC = heating, mechanical ventilation and air-conditioning
MM = mixed mode
NV = naturally ventilated.

Results I: temperature thresholds for thermal acceptability

Previous work

Arens *et al.* (2010) analysed occupants' operative temperature acceptability votes in the ASHRAE

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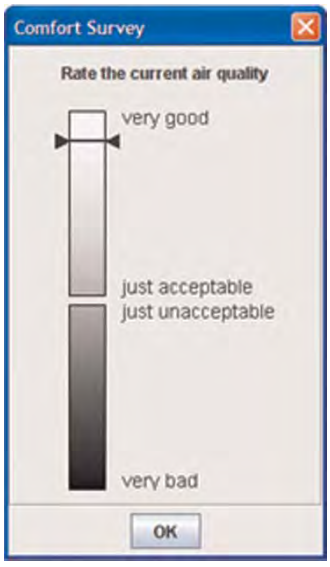

Figure 1 Perceived air quality scale used in the laboratory studies in Datasets 3 and 4

database of 45 central-HVAC office buildings in Kalgoorlie and Townsville (both in Australia), Montreal in Canada, and the NV BCC building in Berkeley (US). The airspeeds in the buildings were generally very low. The results showed very similar acceptability levels over large temperature ranges. A figure combining all four locations and both summer and winter is shown in Figure 2. The data were binned at the middle of each degree, and the brackets indicate ± 1 standard deviation.

Figure 2 shows a roughly flat top in acceptability over a 9 K temperature range (16.5–25.5°C). Between 25.5 and 28.5°C the acceptability reduced significantly and consistently, well below the 80% acceptability. Because of low vote numbers in cool conditions, it is unclear whether 17 or 18°C is acceptable.

Acceptability thresholds for air-conditioned versus naturally ventilated buildings

The authors have here enlarged the above-mentioned dataset by including all ASHRAE database studies in which thermal acceptability was measured. Three HVAC building studies (Honolulu in hot and cool seasons, and Sydney in winter) are added to the locations included in Figure 2 (Kalgoorlie, Townsville, Montreal). Now HVAC buildings are treated separately from NV and MM buildings. Because in the HVAC buildings the indoor temperatures are not very different between summer and winter, the two seasons are combined in Figure 3. Figure 4 and 5 show results for NV and MM buildings, separating winter and summer. The building locations for each figure are listed in Table 1.

Figure 3 shows a roughly similar acceptability profile as Figure 2. Again, it shows a drop below 80% acceptability above 25.5°C. Within 16.5–25.5°C the acceptability does not show a significant and consistent peak value over a specific range, but is mostly above 80% as presented by the flat line. From both figures it can be observed that the temperatures 16.5–25.5°C are likely to be the thresholds for the HVAC buildings when putting both summer and winter together. Again, there are only 32 votes at temperatures below 19.5°C, so the location of the lower threshold remains unclear.

In NV and MM buildings, the indoor operative temperatures are quite different for winter and summer. In winter, the upper threshold at which acceptability

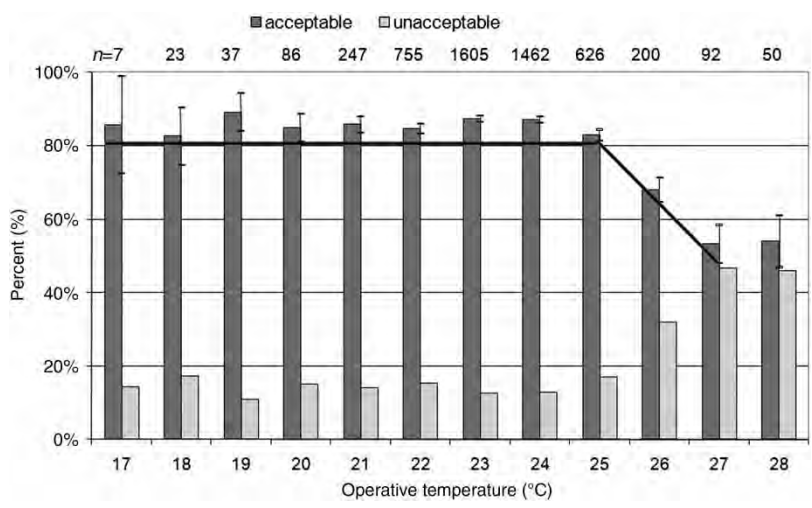

Figure 2 Acceptability against temperature at the workstation, annual, pooled three locations of ASHRAE and Berkeley Civic Center (BCC) data (summer and winter, $n = 5190$)

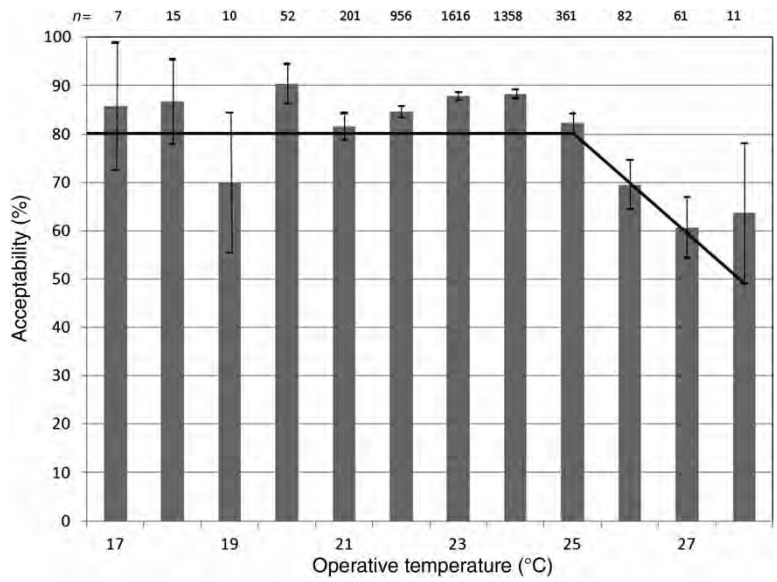

Figure 3 Acceptability against temperature at the workstation; winter and summer, HVAC buildings in the ASHRAE database ($n = 4730$); five locations shown in Table 1

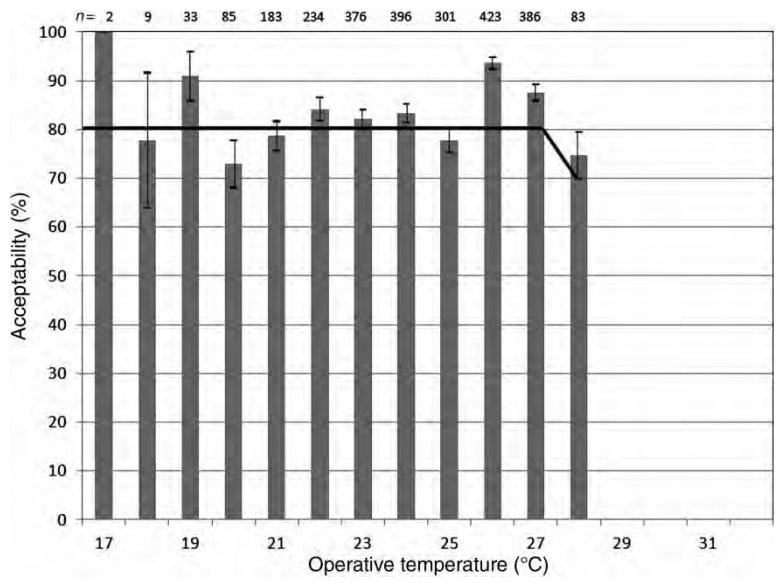

Figure 4 Acceptability against temperature at the workstation; winter; naturally ventilated and mixed-mode buildings ($n = 2512$) in the ASHRAE database; five locations shown in Table 1

drops below 80% occurs at 27.5°C, 2 K higher than the threshold for the HVAC buildings. Between 16.5 and 27.5°C, there is no specific range that the acceptability is significantly better than the others, although the number of votes at 17 and 18°C are very small (Figure 4).

In summer, the NV and MM thresholds move towards the warm side (Figure 5). The significant drop-off below 80% acceptability occurs at 30°C, 2 K above the threshold in winter. On the cool side, the drop happens at 21.5°C, clearly at a warmer temperature than in winter. Most acceptability below this temperature is less than 80%.

There are other field studies in the ASHRAE database that do not have the acceptability question but which have been examined for comfort limits based on sensation and comfort questions (*e.g.* Busch, 1992; Nicol *et al.*, 1999). These studies have found comfortable ranges that are somewhat greater than those shown here.

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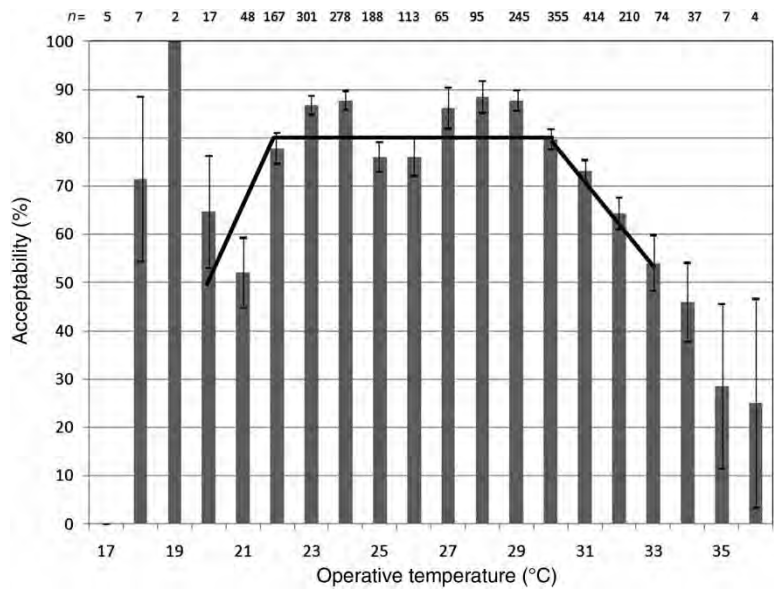

Figure 5 Acceptability against temperature at the workstation; summer; naturally ventilated and mixed-mode buildings ($n = 2632$) in the ASHRAE database; four locations shown in Table 1

Example of difference between air-conditioned and NV buildings

If the ASHRAE database studies of HVAC and NV in the same climate (Singapore) are compared, it is evident the acceptability of the two thermal environments (in this case approximated by sensations between −1 and 1 inclusive because the acceptability votes are not available) was very similar (78% and 76%), although the indoor thermal environmental conditions were very different (Figure 6). The higher thermal thresholds in the NV building than in the HVAC building are presumably due to expectation, physical and behaviour adaptation (de Dear and Brager, 1998), and the somewhat stronger air movement in the NV building (average 0.22 m/s versus 0.11 m/s for the HVAC building).

Threshold concept and values for HVAC buildings

Figure 7 uses the above field results to suggest thresholds for air-conditioned buildings. The thresholds provide guidance for operating HVAC buildings in 'free-running' mode as much as possible, with HVAC heating and cooling applied only after more energy-efficient alternatives such as fans and local heat sources and sinks have reached their outer

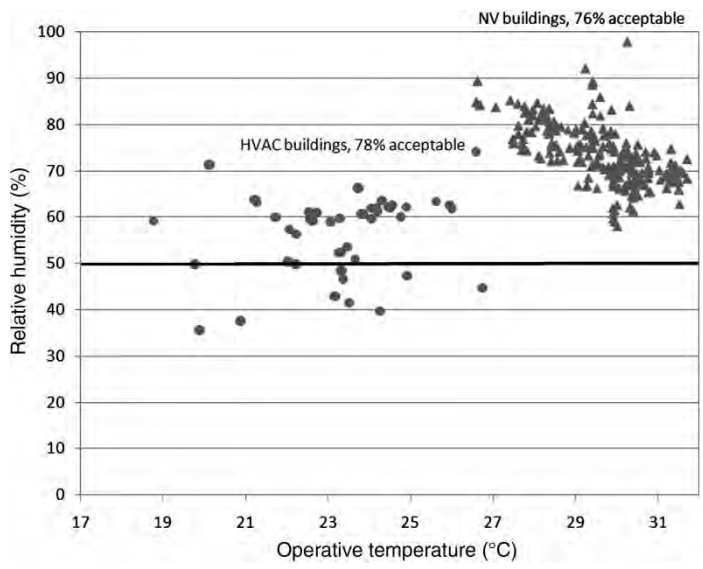

Figure 6 Similar acceptable rate for very different indoor thermal environments in HVAC and naturally ventilated buildings in Singapore

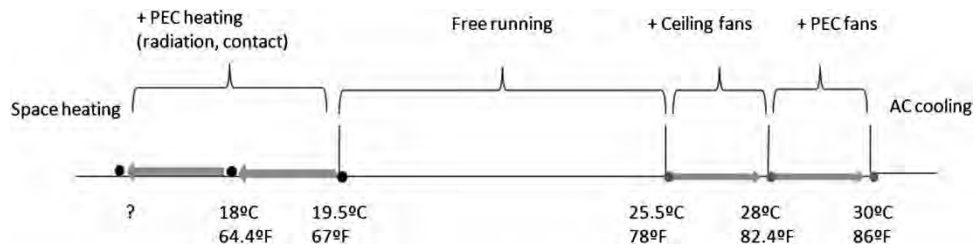

Figure 7 Thermal comfort temperature ranges for HVAC buildings with fans and radiant sources in buildings

temperature limits. The data for the alternative limits are gathered from laboratory studies.

The energy-efficient alternatives in Figure 7 are divided into two categories: (1) sources that affect communal space conditions such as ceiling fans; and (2) sources that affect the occupant directly and are under their personal control (PEC) fans and heaters.

Figure 3 reveals that acceptability in HVAC buildings in the ASHRAE database dropped below 80% between 25 and 26°C in warm conditions, and between 19 and 20°C in cool conditions. Therefore, the Figure 7 temperature range for free-running mode is defined as being between 19.5 and 25.5°C.

On the warm side, area sources like ceiling fans provide comfort, with or without group control, from 25.5 to 28°C (McIntyre, 1978; Fountain and Arens, 1993; Rohles *et al.*, 1983; Scheatzle *et al.*, 1989; Arens *et al.*, 2009). PEC fans provide individual-level comfort from 25.5 to 30°C. Recent PEC studies at 28 and 30°C show that comfort is well maintained with breathing-zone air movements of 0.6 and 1.0 m/s (Zhang and Zhao, 2009; Zhang *et al.*, 2010).

On the cool side, the authors' laboratory study (Zhang *et al.*, 2010) showed that local radiation sources provide comfort for people at ambient condition 18°C. Below that, a study for the automobile industry by Zhang and Wyon (2007) showed that a warmed contact seat can make people comfortable at an ambient air temperature of 15°C. Recent tests at University of California at Berkeley found comfort with a heated seat to 12°C ambient. Because it may be too much to extrapolate these tests to building environments, the authors have left the lower limit unknown. Below the unknown mark space heating is needed.

Results II: thresholds for perceived air quality (PAQ)

If PAQ were reduced in warm temperatures, it might impose a practical limit on the warmer comfort zone boundaries. The ASHRAE database does not include PAQ questions. This is investigated using the PEC studies conducted in the University of California at Berkeley environmental chamber (Datasets 3 and 4) and the field study at the Berkeley Civic Center (Dataset 2).

PAQ related to air temperature and air movement

Figure 8 shows that at air temperatures ranging from 18 to 25°C PAQ does not vary much (Dataset 3). It drops significantly at an air temperature of 28°C, and is further reduced at 30°C. The drop happens between 25 and 28°C, although the particular test

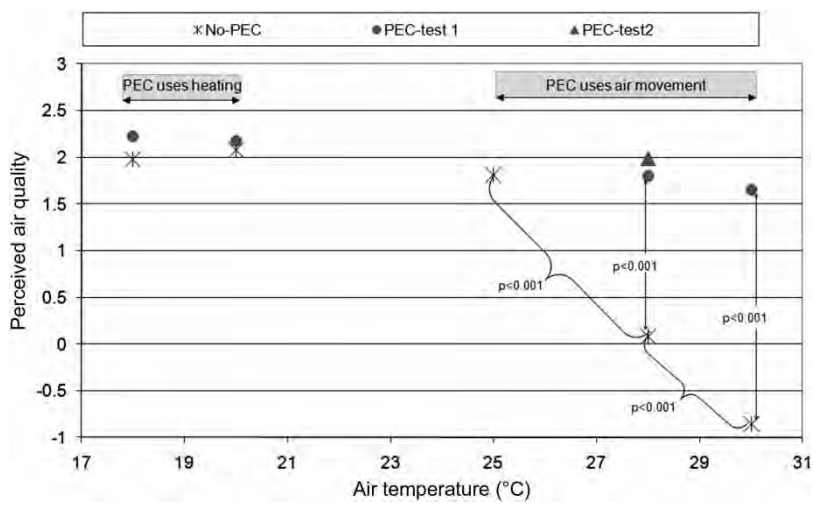

Figure 8 Perceived air quality versus air temperature and air movement

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conditions do not allow the exact temperature to be determined. Air movement is seen to bring PAQ at 28 and 30°C back to the level found under neutral conditions (see the circles and triangles in Figure 8; Arens *et al.*, 2008). Therefore, with air movement the PAQ threshold is beyond 30°C.

PAQ as measured in the Berkeley Civic Center (BCC)

The threshold for PAQ without air movement is not clear from Figure 8 – it is somewhere between 25 and 28°C. The authors looked for the threshold in the BCC field data (Dataset 2), the only study in which PAQ and concurrent air temperatures were measured. The questions and answers are presented in Figure 9a and b for winter and summer studies, respectively. It shows that for the air temperature ranges measured (up to 26°C in winter and 28°C in summer), no clear threshold was reached. It should be noted that although the BCC is an NV building, measured velocities were low, averaging 0.04 m/s in summer and 0.05 m/s in winter.

PAQ related to thermal comfort

In the absence of a clear temperature threshold for PAQ, the authors looked for other means of assuring good PAQ in practice. Humphreys *et al.* (2002) found that PAQ is mostly related to thermal comfort as opposed to air temperature. Figure 10 confirms this, showing PAQ to be closely correlated with thermal comfort at a mixture of temperatures and air

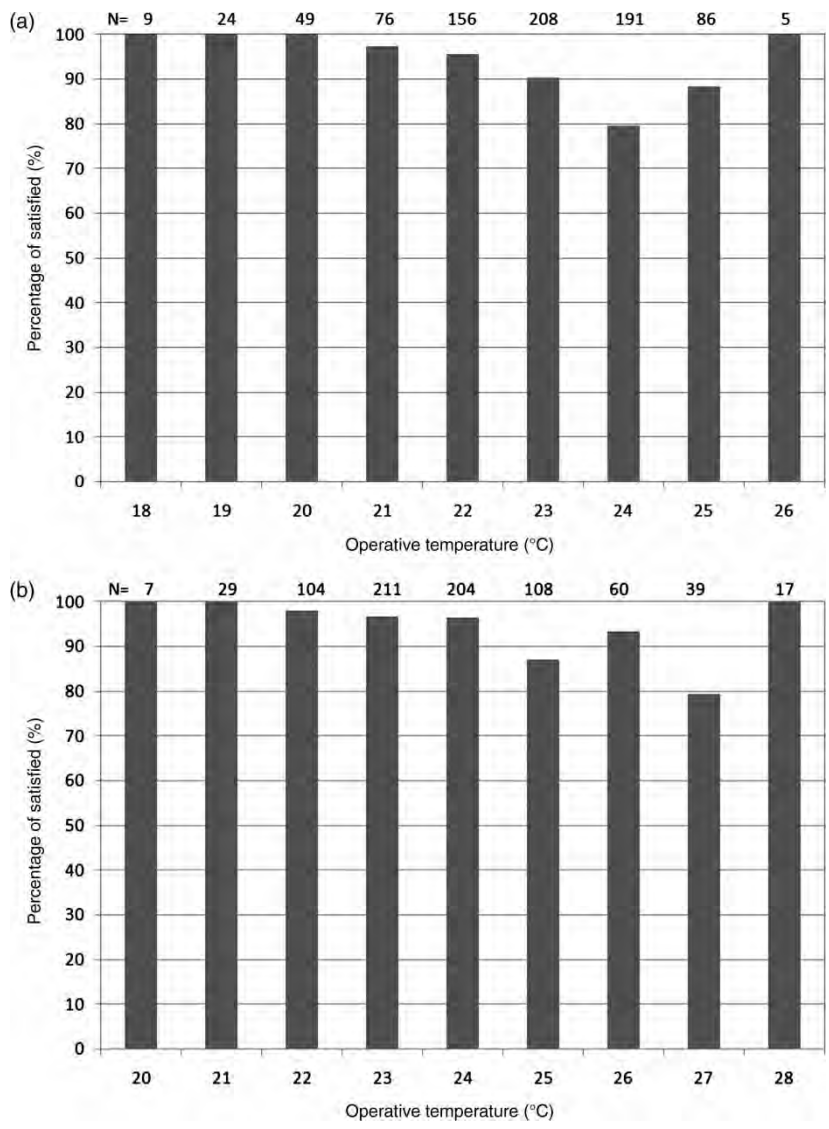

Figure 9 Perceived air quality versus temperature for the BCC database: 'How satisfied are you with the air quality in your workspace?' (a): winter BCC; $n = 804$, clear threshold not reached within 18–26°C, and (b): summer BCC; $n = 779$: clear threshold not visible within 20–28°C.

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movements. Figure 10 is from the two laboratory studies (Datasets 3 and 4) where ambient air temperatures were at 28 and 30°C, and PEC fans were used to provide thermal comfort (the comfort scale is presented on the *x*-axis). If this strong correlation holds in real buildings, it might be assumed that when comfort is maintained PAQ is maintained as well, and that air movement can be used to provide the necessary thermal comfort at high operative temperatures.

Discussion

Thresholds for productivity

Productivity might also follow thresholds in temperature or comfort. Figure 11 (adapted from Seppanen *et al.*, 2004, by adding results from Tawada *et al.*, 2010) shows that within air temperatures from 21 to 27°C there is no obvious best temperature for productivity. Beyond this range, productivity declines in most of the studies. However, it should be noted that these tests did not have elevated air movement under warm conditions. If productivity were actually a function of occupant comfort instead of temperature, the temperatures shown might not represent productivity in buildings where air movement is present.

Uchida *et al.* (2009) showed that self-estimated performance is strongly related to thermal comfort satisfaction (Figure 12, $r = 0.97$). When correlating the self-estimated performance with air temperature, the correlation is very poor, $r = -0.21$. The result shown in Figure 12 is strongly similar to that shown for PAQ in Figure 10. This might lead to the hypothesis that making people thermally comfortable is the key factor in maintaining PAQ and productivity.

Overcooling and overheating in HVAC buildings

Recent field studies in a large number of US office buildings (Mendell and Mirer, 2009, for 95 buildings; and Choi *et al.*, 2010, for 20 buildings) show that the

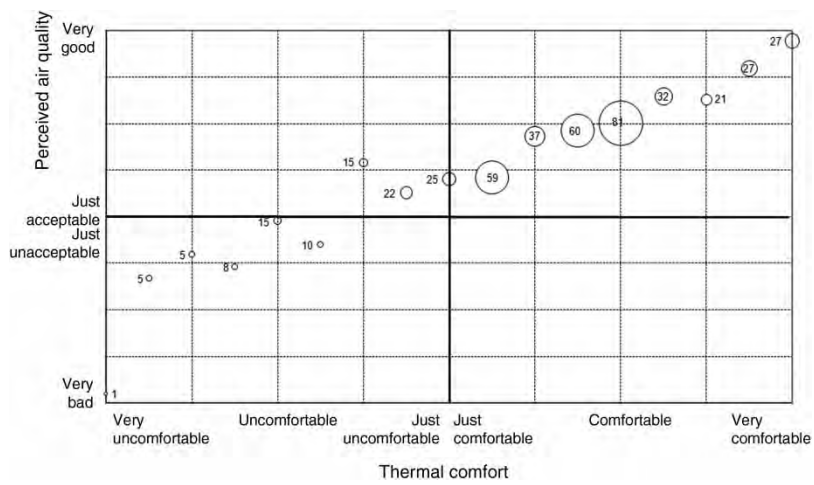

Figure 10 Perceived air quality versus thermal comfort, binned data. Circle diameters represent the number of votes shown nearby ($n = 450$, Datasets 3 and 4)

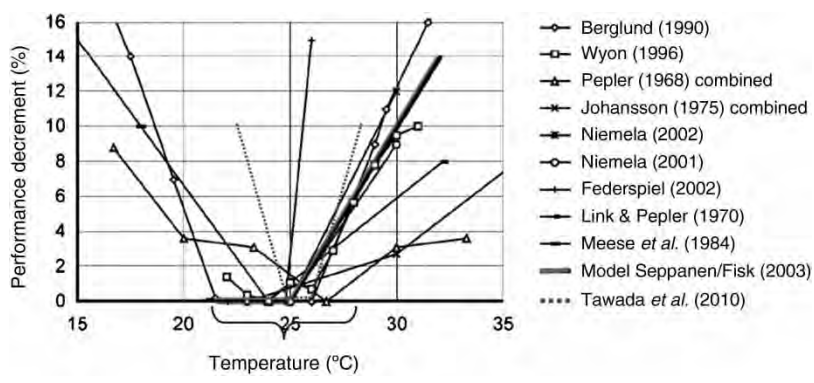

Figure 11 Summary of the studies on the effect of room temperature on the decrement of performance and productivity. Sources: adapted from Seppanen *et al.* (2004), with the addition of Tawada *et al.* (2010)

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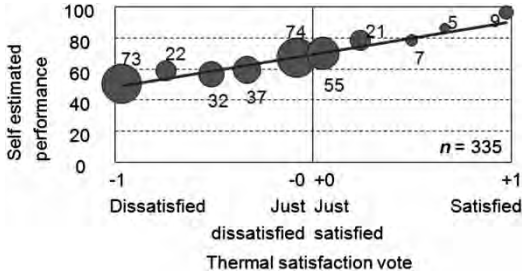

Figure 12 Self-estimated performance versus thermal satisfaction – 335 observations from workers. $R = 0.97$. Source: Uchida *et al.* (2009)

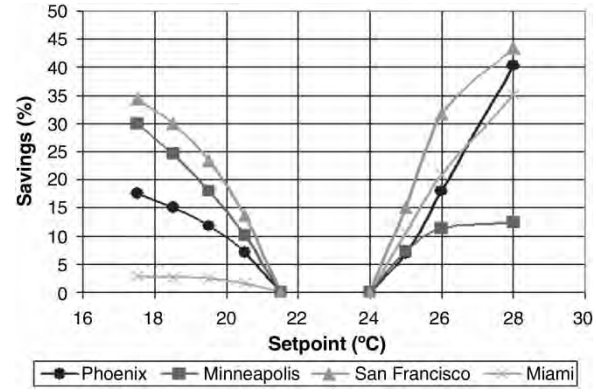

Figure 13 HVAC energy savings for widened air temperature set-points relative to conventional set-point range in San Francisco, Miami, Phoenix and Minneapolis. Source: Hoyt *et al.* (2009)

average indoor air temperature is being maintained cooler in summer than in winter (22.9°C in summer and in 23.4°C winter in Mendell and Mirer; and 23.3°C and 23.5°C in Choi *et al.*). This summer over-cooling works against human adaptation; the reasons for it are unclear since the practice increased both discomfort (Mendell and Choi *et al.*) and sickness symptoms (Mendell) in each season. In addition, health surveys in a large number of buildings (Burge *et al.*, 1987; Zweers *et al.*, 1992; Fisk *et al.*, 1993) found that sick building syndrome is significantly more prevalent in air-conditioned buildings than in NV buildings. It is necessary to understand why this obviously non-adaptive operation of HVAC buildings is so widespread, and what role it might be playing in the worldwide conversion of NV buildings to HVAC.

Energy impacts of thresholds in HVAC buildings

By focusing on the environmental conditions outside the thresholds and not tightly controlling within them, the threshold concept encourages the designs of free-running MM or NV buildings. It also encourages the use within HVAC buildings of energy-efficient technologies with limited cooling capacity (such as evaporative coolers); and it increases the effectiveness of other energy-efficient measures that may be inherently slow acting or unpredictable (such as radiant ceilings/floors) and that inherently cause fluctuation in space air temperature.

Figure 13 shows that by broadening the interior temperature thresholds in HVAC buildings, each 1 K broadening corresponds to about 7–15% in energy saving (Hoyt *et al.*, 2009).

In addition to the operational savings seen in Figure 13, savings may be obtained by reducing the required sizes of HVAC equipment.

Threshold applicability

The thresholds described in this paper are based on occupant surveys taken at random times. The rate of change that may have been occurring in the occupants' environments was not measured. If the temperature is changing rapidly, people may not adapt to the full threshold range observed. The thresholds do not shed light on the extent of adaptation within short time ranges, such as one hour or a day, but they do reflect typical changes occurring in real buildings.

Conclusions

The threshold concept presented in this paper suggests that when indoor air temperature is within defined thresholds, there is little advantage from fine-tuning the air temperature to an optimum. The air-conditioning system should focus on bringing environmental conditions outside the thresholds within them. In NV buildings the thresholds are broad due to occupant-adaptive behaviour in the presence of outdoor climate. The acceptability-based thresholds presented in Figure 7 are independent of seasonal climate, and also of operable windows. The thresholds may be broadened in both HVAC and NV buildings by adding air movement and radiation to the occupied space, or by the occupants directly through PEC systems. PAQ in warm conditions does not appear to be a problem as long as occupants are kept thermally comfortable. The energy impacts of broadened thresholds are very substantial. The threshold concept makes the design of free-running mode and NV buildings more feasible, and reduces the need for energy-intensive air-conditioning in buildings.

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