Title
How the number and placement of sensors controlling room air distribution systems affect energy use and comfort

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HOW THE NUMBER AND PLACEMENT OF SENSORS CONTROLLING ROOM AIR DISTRIBUTION SYSTEMS AFFECT ENERGY USE AND COMFORT

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ABSTRACT

This study assesses the impact of sensor number and placement on the energy needed to condition a typical office using several likely variants of an underfloor air distribution system (UFAD). The study uses an empirical-based room stratification model developed from full-scale tests of UFAD systems. Annual energy consumption is calculated for an interior zone using outside air temperature bin data. The comfort criteria are taken from ASHRAE standard 55-92. The simulations indicate that there are benefits derived from using more than one temperature sensor to control conditions in the occupied zone of a room. Among these are: 1. By adjusting both supply air temperature and volume to maintain the maximum allowable thermal gradient in the occupied (lower) part of the room, an optimal supply air condition can reduce energy use (relative to the best arrangement of a single sensor) while maintaining comfort; 2. Discomfort caused by stratification can be detected by having one of the sensors located at foot level; 3. For the simulated UFAD interior zone of a typical office building in Sacramento, an overall energy saving of 8%/24% (VAV/CAV respectively) can be achieved when two sensors as opposed to one are used to control room conditions.

INTRODUCTION

Buildings use a considerable amount of energy to condition the interiors for thermal and lighting comfort. Yet a substantial proportion of occupants are uncomfortable because typical building systems cannot accommodate the different needs of individuals within office spaces shared by more than one person. Much of this can be attributed to the following factors:

1. Ineffective operation of environmental control systems because they are unresponsive to important environmental conditions. The lack of response may be due to:

- Insufficient sensors to detect the variables of interest,
- sub-optimal placement of sensors due to physical and economic constraints imposed by wiring,
- inability to detect sensors that are faulty or out of calibration, since the opportunities for comparative checking are limited.

2. Poor integration of mechanical systems with the building itself. There are promising integrated systems that can save energy and produce individualized microclimates within the occupied zone, but their adoption is being hampered partly because they tend to require more intensive control. These include:

- Systems like underfloor air distribution (UFAD) that take advantage of thermal stratification to promote thermal comfort and ventilation effectiveness,
- ‘task-ambient’ air conditioning systems that allow occupant control over the local microclimate while the ambient system is controlled to loose, energy-conserving tolerances,
- daylight-responsive light-dimming systems,
- blinds/solar controls responsive to solar heat gain,
- ‘mixed-mode’ designs with operable fenestration in the facades that interacts with the heating, ventilating, and air conditioning (HVAC) system.

Since all these systems produce variable conditions within the occupied zone they could benefit from sensor densities greater than is currently typical.

In this study we use a simulated UFAD system, one of the newer HVAC systems that is capable of more refined control of the interior thermal environment, to investigate the energy saving potential of using more room temperature sensors for environmental control. Underfloor air distribution is a more efficient way to condition occupied spaces than conventional overhead (OH) systems. In overhead systems, cool (usually 55°F) air is injected into the space from above with sufficient momentum to have it uniformly mix within the space below, so that it will at best provide a uniform temperature and pollutant distribution within the space. In UFAD systems, somewhat less cool air (usually 63-65°F) is supplied from diffusers installed in a raised floor, so that it interacts with thermal plumes generated by occupants and other heat (and contamination) sources. As the fresh supply air absorbs the internal heat (and contaminants), it naturally rises by buoyancy to the ceiling area where the exhaust registers remove air that is warmer and more polluted than the exhaust air from an OH system. Efficient UFAD operation inherently involves a temperature gradient in the occupied zone of the room. The increased supply and exhaust air temperatures in UFAD systems save cooling energy by allowing an increase in the amount of heat removed by outside air alone (through what is termed the 'outside air economizer'). Fan power can also be saved if systems are designed and operated to exploit the potential of the stratification by minimizing supply airflow rate at different loads while maintaining comfort conditions in the occupied zone.[1]

OBJECTIVE

This study is intended to demonstrate the impact of sensors on the energy needed to condition a typical office using several likely variants of UFAD. It also examines how these UFAD cases perform relative to an OH system conditioning the same space to the same level of comfort.

APROACHES

<u>Room air stratification model</u>

To perform this study, it is necessary to have a model to define the coupling relationship between the supply air condition and room air distribution for different cooling loads.

Previous work on stratified temperature gradients in rooms include nodal models (Li (1992) and Mundt (1996)) and computational fluid dynamics (CFD) (Chen (1990), Gan (1995), Yuan (1999)), each for modeling displacement ventilation systems that are similar to underfloor systems. Ito (1993) and Fujita (1996) built empirical room-stratification models for underfloor systems based on full-scale tests. Linden (1990, 1996) proposed analytical models of stack-driven natural ventilation, and Lin (2002) built a scaled physical model to simulate underfloor systems.

To assure that our room stratification model embodied the characteristics of a typical current underfloor installation, we developed an empirical model from a series of full-scale tests done by the authors. The tests quantified temperature gradients for several underfloor technologies installed in realistic arrangements of air supply vents, heat loads, and office furniture.

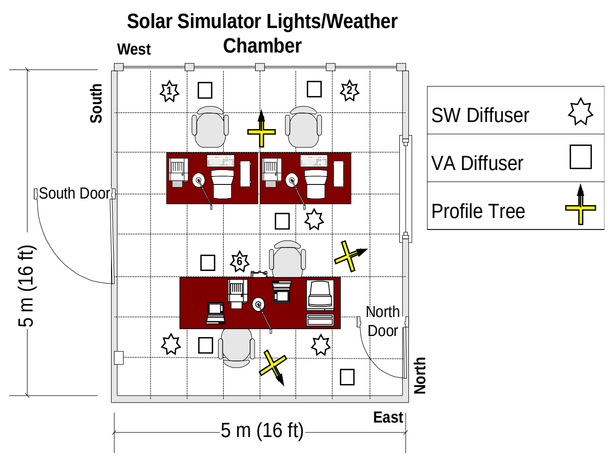


Figure 1 Test chamber layout[2]

Experiments were performed at McGrath Laboratories (St. Louis, MO) beginning in 1999. The test chamber (see Figure 1) was configured like a regular office space including overhead and task lighting fixtures, computers, printers, and occupants. The chamber included a 20 in. high raised floor and a suspended acoustical ceiling 10 ft above the raised floor. A weather chamber adjacent to a curtain wall

[1] Fan energy is also saved for UFAD systems because the static pressure in the supply ducts is lower than that required for traditional overhead systems.

[2] VA diffuser refers to a variable area diffuser where airflow rate is controlled by a damper that restricts the discharge area. The profile tree is a vertical array of temperature sensors used to measure room air stratification profiles.

on one side of the chamber allowed ambient conditions typical of perimeter office zones to be created. (For details of the test chamber and test results, please refer to Webster (2002)). This weather chamber can also be operated to minimize heat transfer allowing interior rooms to be simulated.

A multivariate regression model was developed from 23 tests for simulated interior rooms. Tests were conducted for different supply air conditions, and diffuser operating conditions for two diffuser types, *swirl* (SW) and *variable area* (VA). Independent variables for the regression model are diffuser design ratio (DDR) and load volume ratio (LVR); dependent variables are temperature gradient in the occupied zone (the region from 4 to 67 in. from the floor), and the difference between the average occupied zone and supply air temperatures. For swirl diffusers, the regression model can be written as follows:

$$\Delta T_1 = 4.41 + 0.03 e^{LVR} - 3.23 DDR \qquad (1)$$

$$\Delta T_2 = 2.13 LVR + 1.76 DDR \qquad (2)$$

where

$$\Delta T_1 = T_{5.5\,ft} - T_{0.5\,ft}$$

$$\Delta T_2 = T_{avg\,0.5-5.5\,ft} - T_{sa}$$

and

$LVR \propto Q/V$, *LVR* is proportional to the ratio of load and zone supply airflow rate.

$DDR \propto V/n$, DDR is ratio of diffuser flow rate to the manufacturer's nominal design flow rate.

The UFAD room model consists of a typical occupied office space with a raised floor that covers a supply plenum bounded on the lower side by a concrete slab that separates building floors; the underside of the slab is exposed to a return air plenum from the floor below. The overhead model assumes this same slab is the floor surface of the room. In both cases the return path for the conditioned room air is assumed to be a return plenum bounded on the bottom by an acoustical ceiling and on the top by the floor slab. Heat transfer to the supply plenum will reduce the cooling load to some extent for UFAD systems while heat transfer from the return plenum will increase cooling load. For these systems the floor and ceiling conduction is calculated by:

$$q = U_{f,c}(TD) \qquad (3)$$

where

q is the total heat transfer through the floor by conduction Btuh/ft^2.

$U_{f,c}$ = 0.33 and 0.35 Btuh/ft^2/°F, is the air-to-air U-factor for the floor and ceiling, respectively.

$TD = T_{air\_plenum} - T_{air\_room@\ floor}$ is the temperature difference.

For the overhead simulation the loads were modified to reflect the differences in floor heat transfer since overhead systems typically do not have a cool plenum under a raised floor. The heat transfer through the slab from an assumed return plenum below was assumed to be zero for both UFAD[3] and OH systems. Heat transfer through the ceiling was assumed to be constant for full load and part load cases, respectively.

Both systems were sized (i.e., the supply air flow rate was determined) at full load. For UFAD, the supply air temperature is constrained to be above 63°F because lower temperatures have been shown to cause discomfort at the feet. For OH, a supply air temperature of 55°F was assumed. At partial load conditions, system control parameters were derived individually for VAV and CAV configurations.

For the UFAD system (see Strategies 1-3 below), the temperature stratification is calculated interactively by equations (1) and (2). For overhead, the room air is assumed fully mixed so that all room temperatures including that of the air exhausted from the room is equal to the control temperature of 75°F.

Energy consumption

Annual energy consumption was calculated using the outside air temperature bin-method. The bin-method is a simplified steady state analysis method that allows outside-air-driven loads to be modeled and the effects of outside air temperature

[3] CBE experience with testing UFAD systems indicates that heat transfer from the floor slab into the supply plenum is small. However, this may not be true for all cases.

variations to be included in central plant energy calculations. For this study, room-cooling loads were not affected by outside environmental conditions since the simulated room was an interior zone with no interaction with outside walls. The outside air variation allows economizer usage to be captured for the differences in supply and return air temperatures caused by the various control strategies. All load components were assumed to be instantaneous without time lag effects.

Meteorology data was obtained from the National Renewable Energy Laboratory (NREL) weather database for Sacramento, CA. The heat gain profile shown in Figure 2 was assumed. The full load condition from 9 AM to 4 PM, consisted of the components listed in Table 1. At partial load conditions from 6-9 AM and 4-6 PM equipment and people heat gains are 60% of full load values.

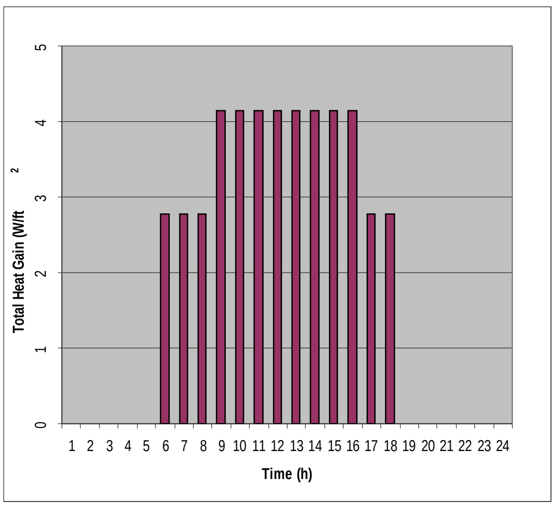


Figure 2: Heat gain profile for simulations

Economizer performance depends on the supply and return temperatures relative to outdoor air temperature. In this study a "return air temperature" economizer was assumed that uses the following operating strategy:[4]

If $T_{oa} > T_{ra}$
Use minimum outside air (20 cfm/person)
If $T_{sa} < T_{oa} < T_{ra}$,
Use 100% outside air;
If $T_{oa} < T_{sa}$,
Proportion T_{ra} *and* T_{ma} *to satsify* T_{sa}
Else
use minimum outside air + reheat
where,
T_{oa} = *Outside air temperature*
T_{ra} = *Return air temperature*

Table 1: Zone load component assumptions

Load component	**Remarks**
Small equipment	Includes task lights
People	110/ft^2 per person
Overhead lights[5]	1.8 W/ft^2 total input; 40% to room, 60% to return plenum
Raised floor conduction	U = 0.33 Btuh/ft^2/°F at 7°F TD
Ceiling conduction	U = 0.35 Btuh/ft^2/°F, at ~ 2°F TD
Total cooling load, UFAD/OH	3.5 W/ft^2 / 4.3 W/ft^2

Fan performance was simulated using the model from Webster (2000). For the chilled water plant a constant COP = 4 was assumed for all situations.[6]

[4] This strategy is appropriate for climates like California's but would not be appropriate for humid climates.

[5] Light heat to space is assumed to be 40% of total lighting power input; 60% of lighting input was assumed to increase plenum return (i.e. return to conditioning equipment) temperatures by 2-4°F above the space temperature at the ceiling (i.e., ceiling return temperature).

[6] This is a simplifying assumption that is appropriate for this study only; the comparisons between energy use for the cases shown were found to be insensitive to variations in COP over ranges normally encountered for typical chilled water systems.

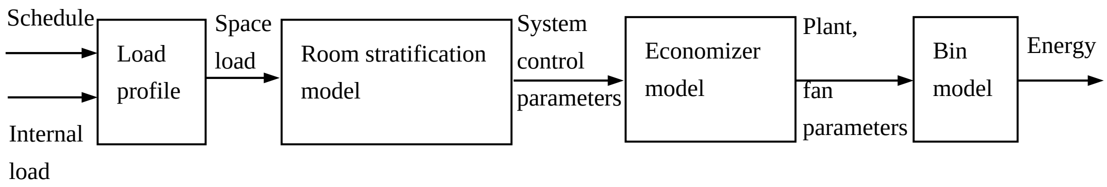


Figure 3: Diagram of the modeling process

Modeling process

The modeling process is illustrated in Figure 3. This modeling process was used to simulate both traditional overhead (OH) and UFAD variable air volume (VAV) and constant air volume (CAV) systems including three different temperature-sensing scenarios for room temperature control of a UFAD system:

Strategy 0: Traditional overhead VAV and CAV systems (Simulations VAV-OH and CAV-OH) with a sensor located at 5.5 ft (typical thermostat height, near the top of the occupied zone).

Strategy 1: UFAD VAV and CAV systems (Simulations VAV1, CAV1) with a single sensor located at 5.5 ft.

Strategy 2: UFAD VAV and CAV systems (Simulations VAV2, CAV2) with a single sensor at 3.75 ft (average occupied zone height).

Strategy 3: UFAD variable temperature and volume (Simulation VTV) system with two sensors located at 0.3 ft and 5.5 ft, respectively to control average temperature and gradient in the occupied zone.

For each of these strategies, the annual energy consumption was calculated for both VAV and CAV system types (the last strategy which is a combination of VAV and CAV), the two predominate forms of all-air systems used in commercial buildings. Control of VAV systems is accomplished by varying the supply flow rate at constant supply air temperature while CAV systems are controlled in the opposite manner, i.e., fixed supply air flow rate but varying supply air temperature. For Strategy 3, variable temperature and volume (VTV), both the supply airflow rate and temperature are controlled to achieve the control objective.

RESULTS AND DISCUSSION

Strategy 0: Overhead CAV and VAV systems

This scenario provides a baseline for comparing the energy use of overhead and UFAD systems under comparable heat gain conditions. Overhead CAV and overhead VAV systems were both modeled, assuming fully-mixed room air with a single control sensor located at 5.5 ft with a set-point of 75°F. The results are discussed in the following sections.

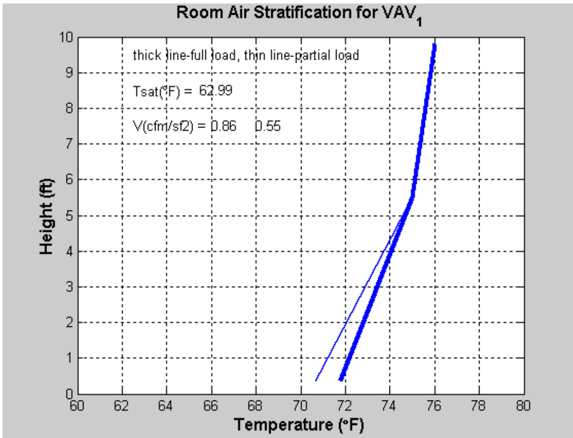
Strategy 1: UFAD with single temperature sensor at 5.5 ft

Figure 4 shows the predicted room air temperature profile at two load conditions for a UFAD VAV system.[7] For this strategy a control setpoint of 75°F was used. This figure shows that as load decreases, less air is supplied to the room and more stratification is generated in the occupied zone (lower region of the room). The temperatures in the upper region of the room are very close for the two load conditions. Thermal comfort studies show that 5% or more of occupants feel uncomfortable when the foot to head vertical temperature difference is 5.4°F or more. [ASHRAE 1992, ASHRAE In Press]. As shown by the partial load line, the gradient from foot to head is very close to this limit. For larger load variations it is likely that this gradient will exceed the limit and cause stratification discomfort.

For a CAV system, as load decreases, supply air temperature is increased and the temperature gradient in the occupied zone will decrease. For this reason it is less possible to cause stratification discomfort at partial load conditions. However, the CAV system will consume more energy at partial load than VAV systems since the room airflow is not decreased.

[7] For the UFAD VAV cases the LVR was assumed to be constant between full and partial load. As shown by equation (1) the gradient in the occupied zone is increased as the diffuser flow rate is decreased when room airflow is decreased since the number of diffusers is fixed.

Figure 4: Simulated stratification for VA



V1

The results for these two cases suggest that one sensor is not sufficient to achieve energy savings and comfort simultaneously for either VAV or CAV systems. In practice, there will be variations in the occupied zone comfort conditions because the temperature profile in the occupied zone will depend on how the profile pivots around the sensor location (the control point). One sensor will not assure a consistent average room temperature around the occupant. Also, for VAV systems, there are times when the stratification in the occupant zone is less than the complaint limit, thus by reducing the air volume until the stratification is closer to the complaint limit, more energy can be saved.

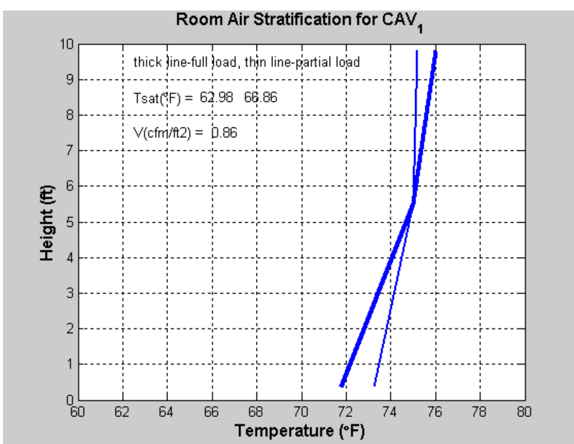


Figure 5: Simulated stratification for CAV1

A comparison of energy consumption between Strategies 0 and 1 (i.e., OH vs UFAD) is shown in Figure 6. This figure shows that UFAD uses about 31% and 29% less cooling energy than OH in VAV and CAV configurations, respectively. Fan energy of UFAD is greater than OH by 5% for VAV and 5% less for CAV, so overall consumption for UFAD is 16% less than OH for both configurations.

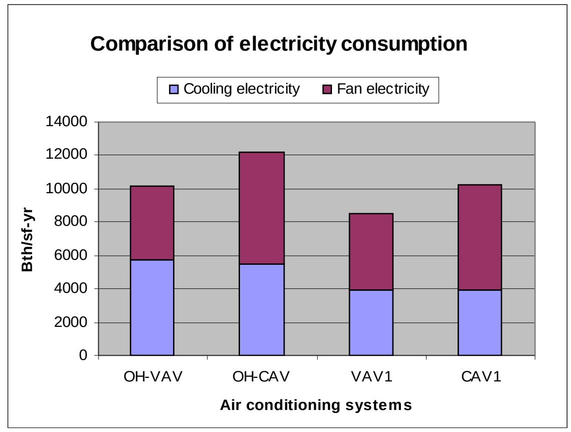


Figure 6: Comparison of OH and UFAD

<u>Strategy 2: UFAD with single temperature sensor at 3.75 ft</u>

This scenario explores the opportunity to improve on the performance of using one sensor located at typical thermostat heights by adjusting the height of the room sensor. With the current room air model, two straight lines are used to represent the room air profile. In this case, the average of the occupied zone occurs at the mid-point of the occupied zone; controlling the temperature at this point will achieve consistent thermal comfort as long as the stratification limit is not exceeded. The results of the simulations for VAV and CAV systems are shown in Figure 7 and 8, respectively.

A comparison of energy consumption between Strategies 1 and 2 in Figure 9 shows that energy consumption of Strategy 2 is less than Strategy 1 by about 3%. Although the control setpoint is the same (75°F), comfort conditions (average occupied zone temperature and gradient) for the two strategies are not the same, but they are both within the ASHRAE comfort range. [ASHRAE 1992] Strategy 2 has the same set-point temperature as Strategy 1 but at a lower height. This suggests that by simply lowering the room sensor, comfort could be improved somewhat while saving some energy.

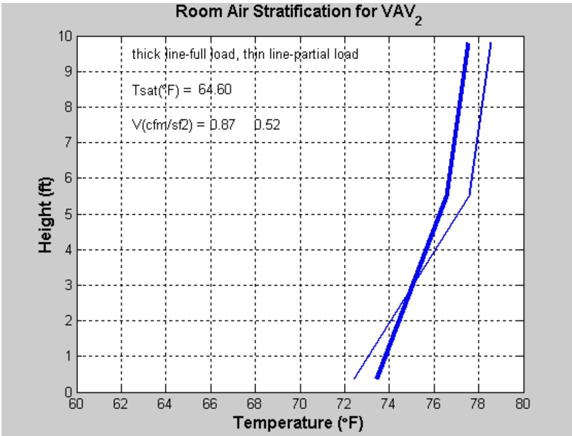


Figure 7: Simulated stratification for VAV2

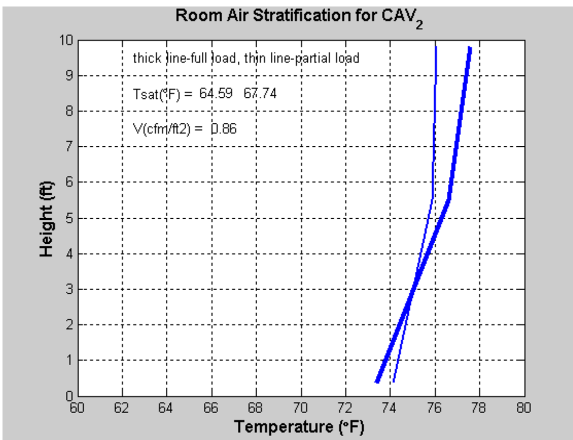


Figure 8: Simulated stratification for CAV2

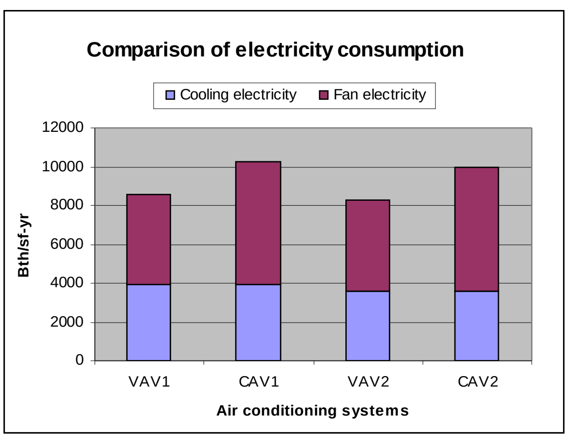


Figure 9: Fan and plant electricity consumption for VAV1, CAV1 vs. VAV2, CAV2

Strategy 3: UFAD with two temperature sensors located at head and foot levels

To evaluate the thermal comfort in a non-uniform environment, stratification must be considered. As shown in Table 2 [ASHRAE In press], the vertical stratification between foot and head has an inverse proportional relationship with comfort.

Table 2: Thermal comfort parameters for ASHRAE Standard 55-2000R

Allowable vertical air temperature difference between head and ankles for the three classes of thermal environment.		
Class	Vertical air temperature difference °C (°F)	Predicted percentage of discomfort (PPD)
A	< 2 (< 3.6)	PPD < 3
B	< 3 (< 5.4)	PPD < 5
C	< 4 (< 7.2)	PPD < 10

To achieve energy savings and thermal comfort simultaneously, an optimal supply air condition is needed that generates an appropriate average temperature and degree of stratification in the occupied zone but is sufficient to remove the heat load from the room. A hybrid system, which controls both supply air volume/airflow and temperature, labeled a VTV system, could provide this capability.[8] The temperature profile for a VTV system derived from the simplified model by adjusting both supply air temperature and airflow rate is displayed in Figure 10. As shown, the temperature gradient between foot and head are maintained at 5.4°F, which corresponds to a PPD of less than 5%.

[8] This is a conceptual solution; many practical issues would have to be addressed to actually implement such a scheme.

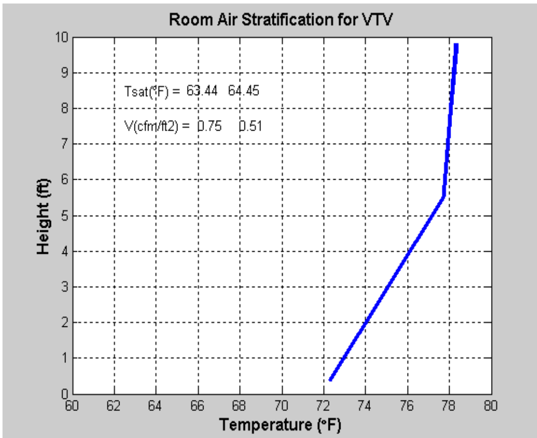


Figure10: Simulated stratification for VTV

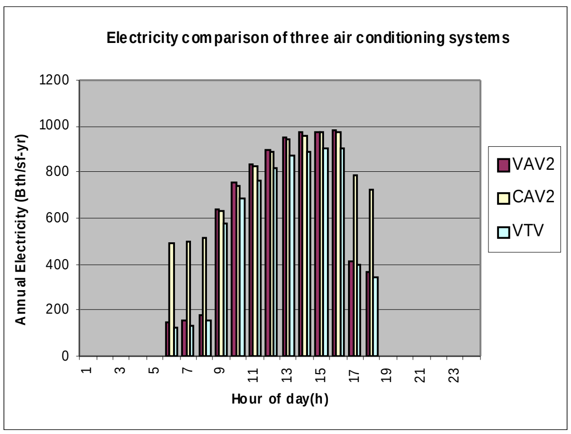


Figure 11: Electricity consumption for VAV2, CAV2 and VTV

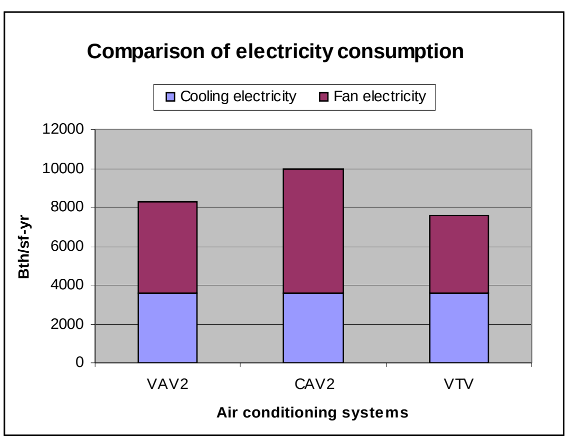


Figure 12: Fan and plant electricity for VAV2, CAV2, and VTV

Figure 11 shows the daily profile for Strategies 2 and 3 and Figure 12 compares energy use. The three cases shown have the same average comfort temperature in the occupant zone and vertical temperature gradients are all within acceptable limits. Figure 12 shows that VTV uses the least energy among the three cases with savings of 8% compared with VAV2. It saves up to 24% when compared with CAV2. It also shows that the major energy difference is from fan energy consumption. VTV consumes 14% less fan energy compared to that of VAV2 and 37% to that of CAV2.

UFAD with three or more temperature sensors located at various heights in the occupied zone.

Because the empirically based UFAD model used in this paper assumes a linear temperature profile in the occupied zone, it cannot be used to study more than two sensors arranged vertically. Experience has shown that the temperature profile in the occupied zone sometimes may not be linear, with a higher and/or reversed order curvature caused by complex heat transfer processes. In such cases, additional temperature sensors would provide the control system more accurate temperature profiles for evaluating and controlling occupant thermal comfort.

CONCLUSIONS

1. Simplified models used to simulate annual energy consumption for UFAD systems in interior zones indicate that there are benefits associated with using more than one temperature sensor to control conditions in the occupied zone of a room. Among these are: (1) by adjusting both supply air temperature and volume, an optimal supply air condition can be achieved such that energy use can be reduced (relative to a single sensor) while maintaining comfort; (2) discomfort caused by stratification could be detected by having one of the sensors located at foot level; and (3) multiple sensors provides the kind of information that can be used for sophisticated control strategies; i.e., control of both average temperature and gradient in occupied zones of stratified rooms facilitates optimization of comfort, energy or both. For the simulated UFAD interior zone of a typical office building in Sacramento, an overall energy saving of 8% was shown when two sensors were used instead of one to more optimally control room conditions using a VAV control strategy.

2. The results indicate that using a single sensor at a non-standard height can improve thermal comfort by reducing temperature variability inherent with typical UFAD control methods. The simulations

showed a slight saving in energy consumption as well. These results underscore the benefits of having sensors that can be located in a flexible manner so various applications can be accommodated easily.

3. The methodology employed for this analysis provides insight into the benefits of using UFAD compared to traditional overhead systems. The simulation corroborates other research showing overall energy benefits to UFAD compared to traditional overhead systems. Although UFAD fan energy may be greater because higher supply air temperatures are used to accommodate a given heat gain, the increased supply air temperature always extends the period when the economizer can be used, reducing the energy needed for mechanical cooling.

RECOMMENDATIONS FOR THE FUTURE

The positive results from these simplified simulations suggest that further work is warranted to examine more complex and realistic situations in buildings. This is especially so because the case for multiple sensors will probably be stronger as the environmental conditions in the building zones become more variable and asymmetric than those in the interior zone described here. Also, it is probable that the interaction between a building's multiple zones and its HVAC system will improve with more new sensor input about the detailed conditions in the zones:

1. Perimeter zones should be included in future simulations. In a perimeter zone, thermal comfort is more complicated than in an interior zone because the radiant environment is often asymmetric. And the shape of the room stratification profile predicted for the interior zone is different than for perimeter zones because of plumes generated next to heated or cooled window surfaces. It is challenging to model these thermal behaviors, but it is important because multiple sensors may offer even greater opportunities to improve energy and comfort performance of perimeter zones.

2. A detailed human thermophysiological comfort model such as UCBMultiNode might be used in determining the comfort effects of non-uniform thermal environments. Up to now we have used the temperature gradient limits specified by the comfort standards as the criteria for comfort, but in complex environments it would be better to evaluate comfort effects directly on simulated occupants.

3. CBE is now working on a project to develop a detailed model of UFAD in interior and perimeter zones, to be incorporated into the building energy simulation program EnergyPlus. This will add physical room air distribution to the conventional zone-by-zone simulation of building energy use. It will allow project-specific factors to be considered beyond what is possible in this paper: the distribution of zones and occupancies in the building, the detailed characteristics of the mechanical system, and how these characteristics interact in using energy and delivering comfort. In addition, this simulation program will allow the impact of various climates to be more fully explored.

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