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Thermal and behavioral modeling of occupant-controlled heating, ventilating and air conditioning systems

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Abstract

Occupant-controlled heating, ventilating and air conditioning (HVAC) systems allow inhabitants of open-plan spaces some degree of control over their immediate microclimate. Typically, cooled air is supplied at floor or desktop levels. The amount and direction of air flow is under occupant control. Productivity increases have been attributed to this form of control. This paper proposes a simplified model of the thermal environment created by an occupant-controlled HVAC system and the behavior of the occupants within it. The thermal environment is characterized by individual nodes representing sub-areas of the conditioned space and a single well-mixed ceiling space above the occupied zone. Random processes are used to simulate the comings and goings of individual occupants and their HVAC control behavior. The model is used to identify the parameters which have the largest influence on the energy efficiency. Energy use of task HVAC with occupant sensors is found to be less than that of a conventional HVAC system by 13%. Individual HVAC control requires about 10% more energy than uniform temperature conditions.

Keywords: Energy modeling; Underfloor HVAC; Thermal environment; Occupant-controlled HVAC; Task conditioning system

1. Introduction

Occupant-controlled heating, ventilating and air conditioning (HVAC) reflects a change in the underlying philosophy of the maintenance of thermal environments in office buildings. Common HVAC design practice in office buildings today is to provide a steady environment which will be acceptable to 80% of the occupants [1]. While adequate, this philosophy is far from ideal. Nearly half of the office building occupants surveyed by Schiller et al. [2] expressed a desire for a different thermal environment than the one they had, even though the majority of offices surveyed fell within the standard ASHRAE comfort guidelines [1].

Occupant-controlled HVAC attempts to address this situation. Rather than a central system providing a single, uniform thermal habitat which is supposed to be adequate for most people, each occupant controls his or her own microclimate. Typically, cooled air is supplied at floor or desktop levels. The amount and direction of air flow is under occupant control. Worker productivity increases have been attributed to this form of control in studies by Kroner et al. [3] and Paciuk [4]. Hedge et al. [5] found that two-thirds of the workers they surveyed felt that the underfloor HVAC system provided increased comfort compared to conventional systems in previous office buildings. There are a number of other reasons to look at occupant-controlled HVAC systems. Such a system can also decrease energy consumption, increase the flexibility of building interior layout, and eliminate regions of inadequate ventilation.

Previous studies of task conditioning systems have described planned or existing installations or laboratory and field studies. Key physical parameters and comfort control performance have been studied in the laboratory [6–8] and in the field [9]. Researchers have constructed physical and behavioral models of occupant-controlled HVAC using conventional HVAC modeling software in order to estimate energy consumption: Heinemeier et al. [10] modeled energy use using TrakLoad, while Braun and Seem [11] used TRNSYS. Both studies modeled characteristics of task conditioning systems such as elevated supply and return temperatures, local fan heat and power loads, and floating temperatures in unoccupied areas. Conventional HVAC energy modeling systems such as TrakLoad and TRNSYS assume that the temperature of the conditioned space is uni-

[1] Schiller et al. [2] found that 78.2% of buildings surveyed in winter and 52.8% of those surveyed in summer fell within the ASHRAE Standard 55-81 comfort zone. In these buildings, 46.7% of the occupants surveyed in winter and 47.9% of 1034 surveyed in summer expressed a desire to be either warmer or cooler.

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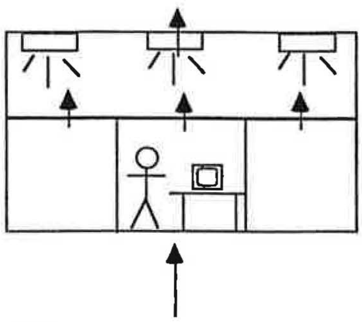

Fig. 1. Schematic of office space model.

form, so neither Heinemeier et al. nor Braun and Seem considered the details of the thermal environment created by the task conditioning system. Occupant behavior and HVAC control strategies are modeled deterministically by Heinemeier et al. and Braun and Seem, for example, assuming that local fan use would be a function of space temperature [11].

This paper presents a simple model of the thermal environment created by a task conditioning system and the results of an analysis of HVAC energy consumption using the model. In particular, the temperature differences and heat transfer between adjacent work stations and common areas and the resulting effects on energy consumption have been modeled. A simple model of the behavior of individual occupants, based on random processes, is included.

1.1. Physical model

The basic model consists of one interior zone on a floor of a multiple-story office building. The space is subdivided into a $m \times n$ rectangular grid of well-mixed square cells ('cells' hereafter) extending from floor level to the tops of the partitions, a height of about two meters. These cells represent the occupied portion of the space. Each cell has a single temperature T_i. Above these cells is an open area between the partitions and the ceiling with a limited number of exhausts. It is reasonable to expect the air in this area to approach a uniform temperature. This zone will be represented by a single temperature T_c (Fig. 1).

Space cooling loads are a function of the internal heat gains and heat exchanges with the outdoors. They are modeled in the steady state. An interior space does not directly exchange heat with the outdoors, so that there are no solar or infiltration effects. Except for perimeter zones extending 12 to 18 ft from the outer wall, office buildings are dominated by internal heat loads and have fairly constant cooling loads throughout the year [12]. This justifies the ideal assumption for a typical open-plan office space [2].

[2] Perimeter zones are often reserved for private offices with their own thermostats. Thermal discomfort occurs in the perimeter zones of some open-plan offices which do not have such private offices. Although not studied here, occupant-controlled HVAC could also address this problem.

1.2. Probabilistic occupant behavior model

Since the occupants have control over the operation of the HVAC system, it is necessary to make some assumptions about their actions. Two factors affect the calculations: the number and location of the people; and their control strategy, including preferred climates or comfort conditions. These are best represented in a probabilistic manner.

Every cell in the conditioned space is assumed to fall into one of four categories: (i) unoccupied areas such as corridors, (ii) unoccupied work stations with equipment turned off, (iii) unoccupied work stations with equipment left on, and (iv) occupied work stations and equipment turned on. The locations of cells in category 1 are known a priori. As a simple approximation, all persons are considered identical. Two parameters control a random process that assigns type 2, 3 or 4 to the remaining cells. The 'occupancy rate', x, is the probability that a particular occupant is present at any given time. The 'equipment leave-on rate', y, describes the tendency of occupants to shut off task lights and personal computers when they depart. It is straightforward to construct a tree, as shown in Fig. 2. This categorization reduces the description of any single cell to two random Bernoulli-type trials, with probabilities of success x and y. Bias can be eliminated by assigning cell types randomly. Occupant sensors are easily modeled as forcing the number of type 3 cells to zero by simply setting $y = 0\%$.

These numbers allow us to answer two basic questions: which cells are being conditioned, and what is the internal heat gain in each cell? HVAC controls are presumed to be deactivated when a person shuts off other equipment. Cells of types 1 and 2 (see Fig. 2) do not have active HVAC control. Their temperatures 'float' up to a maximum value at the extreme of the comfort range, maintained by the HVAC equipment, if necessary. Types 3 and 4 have active HVAC systems keeping them at a specified temperature determined by the occupant.

One of the chief shortcomings in this model of occupant behavior is the lack of hard data for occupancy rate and equipment leave-on rate, parameters x and y. Library research has failed to find even a single field study of these phenomena. Furthermore, x and y can be expected to vary widely depending on the type of business conducted in the office, and even on day-to-day or hour-by-hour basis. A survey published in 1984 [13] included data that can be used to derive a value of 60 to 70% for x (the occupancy fraction), but these results

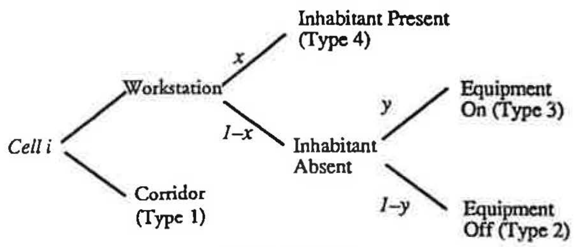

Fig. 2. Cell tree.

are based only on subjective assessments. Norford et al. [14] use daily electric load profiles from a large and small building to estimate that 30–40% of electronic equipment is left on over nights and weekends. In the absence of better data, a value of 40% was used to reflect the higher leave-on rate for shorter absences.

1.3. HVAC control behavior

In addition to the comings and goings of the occupants, their climate preferences must be modeled. One model of occupant behavior allows the occupant of each work station to select a single temperature for his or her microclimate. The HVAC equipment will then maintain that temperature by adjusting the cooling air volume flow at a fixed supply temperature to compensate for the space load and the heat exchange with other cells and the ceiling area. Alternatively, occupants could be given control of the temperature of a constant flow of cooling air. In order to approximate the behavior of an unknown population of individuals, the ASHRAE standard comfort zone was used [12]. This zone, based on extensive laboratory studies of thermal comfort conditions, defines a temperature and humidity range which is expected to be acceptable to 80% of the occupants. Assuming individual temperature preferences to be normally distributed, the comfort zone can be interpreted as a confidence interval where 90% of occupants prefer a temperature below the upper (summer) temperature limit of the comfort zone and 10% prefer a temperature below the lower (winter) temperature limit (Fig. 3). This model results in a mean preferred temperature of 23°C and a standard deviation of 1.5° at 50% humidity, the middle of the range.

Random numbers sampled from a normal distribution can be used to assign the temperature preference of the occupant of each cell. In this way, the temperature differences between adjacent cells are also randomized. The ability of occupant-controlled HVAC systems to sustain temperature differences in adjacent work stations is supported by Bauman et al. [6] who observed differences of up to 2.5°C. Limits placed at ± 2 standard deviations (20 and 26°C) reflect the limitations of the equipment, i.e., the two 'tails' of the normal distribution are omitted above and below these limits. This is important because excessively cold cells require very large amounts of cooling air, while excessively hot ones may not have enough

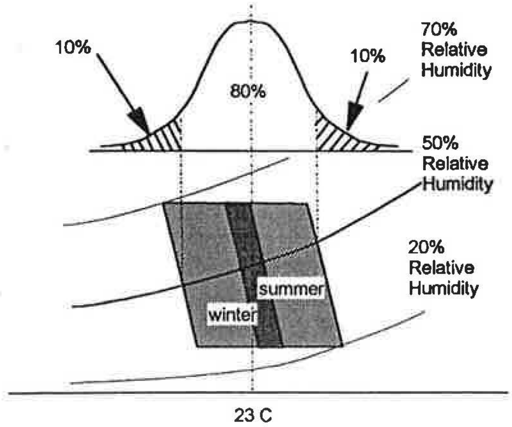

Fig. 3. ASHRAE comfort zone basis for temperature preferences.

internal heat generation to produce the desired temperature. Note that no heating function is modeled; only cooling equipment is considered.

Control behavior of the occupants is more than choosing a temperature. Few people consistently pursue a thermostatic control strategy [15,16]. Some occupants will adjust the HVAC controls frequently, while others will almost never adjust them, introducing thermal transients. As an alternative control model, occupants can be thought to randomly select the amount of cooling air supplied to them. Lower limits can be set according to the ASHRAE ventilation standard of 10 l/s per person of fresh air [17]. Upper limits simply reflect the capacity of the local fan unit. A limit of 200 l/s gives adequate cooling capacity for the loads described herein. It is conceivable that individuals would have multiple units under their control, especially in high heat load areas. This could easily be modeled, although the authors have not done so.

1.4. Heat balance calculations

The space heat gain for each cell is calculated separately as described in Table 1. The lighting heat gain formulae allow for varying amounts of task lighting where each watt per square meter of task lighting is assumed to replace 2 W/m^2 of overhead lighting. One-third of the power consumption of the overhead lights is modeled as radiant heating of the cells, while the remaining two-thirds is the space heat gain of the ceiling zone. Heat transfer among cells and the ceiling zone is modeled separately.

Table 1
Cell heat loads

Heat source	Value	Notes
People	75 W sensible 45 W latent [15]	Assumed constant. Humidity control for comfort neglected. Absent persons assumed to leave the conditioned space.
Equipment	17 W/m^2 [11]	Assumed to be uniformly distributed.
Localcontrols	5 W/unit [7]	Local controls may be lower.
Lighting	19 W/m^2 [14]	1/3 of power consumption modeled as radiant heating of cells; remaining 2/3 of lighting power consumption becomes heat gain in the ceiling zone. One watt of task lighting (in cell) replaces 2 W of overhead.
Local fans	0.43 W per l/s	Power assumed to be proportional to flow rate [10]. System may not require local fans.

Heat balance equations are used to calculate the temperature or cooling air flow of a given cell—one of the two will always be unknown. Each cell is assumed to exchange heat only with its four neighbors and the ceiling zone. Air is assumed to flow unidirectionally from the floor-level supply vents through the cells, into the ceiling zone and out the exhausts.

Cooling air flow is governed by the simplified equations (neglecting latent effects):

$$Q_{\text{cooling}} = 1.232V(T_s - T_i) \tag{1}$$

where Q is in watts, V in l/s, T in °C and 1.232 is used as an average heat capacity per unit volume of moist air [18]. T_s is the temperature of the cooling air supplied to the cell and T_i is the cell temperature. A heat balance can be performed on a given cell which yields the unknown temperature (if cooling air flow is known) or cooling air flow (if temperature is unknown). Assuming a uniform temperature in cell i:

$$m_i c_i \frac{dT_i}{dt} = Q_{\text{internal}} + Q_{\text{exchange}} + Q_{\text{cooling}} \tag{2}$$

Consider a general cell, as shown in Fig. 4. The cell exchanges heat with the surrounding cells, numbered 1–4, and the ceiling cell, indicated by subscript c. Cooling air flows are denoted by the letter V while temperatures are denoted by T. Surfaces between neighboring cells are characterized by area A_v and heat transfer coefficient h_v. The top surface of the cell is similarly described by A_h and h_h. (The v denotes a vertical surface, the h a horizontal one.) The power usage of the local fan is denoted by Q_{fan}. In the steady state, the energy balance for one cell can be written as:

$$\begin{aligned} Q_{\text{person}} &+ Q_{\text{equip}} + Q_{\text{fan}} + h_{v_1}A_{v_1}(T_1 - T_i) + h_{v_2}A_{v_2}(T_2 \\ &- T_i) + h_{v_3}A_{v_3}(T_3 - T_i) + h_{v_4}A_{v_4}(T_4 - T_i) \\ &+ h_h A_h(T_c - T_i) + 1.232V(T_s - T_i) \\ &= 0 \end{aligned} \tag{3}$$

For cells where temperature is known, the equation can be solved for the unknown cooling air flow. For cells where cooling air flow is known, the unknown cell temperature must be found.

The ceiling area is handled somewhat differently. Consider an $m \times n$ block of floor-level cells. Some may have known temperature T_i, and some known cooling air flow V_i. Internal heat gains for the ceiling zone are calculated as described in

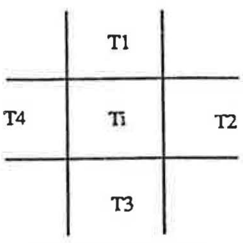

Fig. 4. General cell.

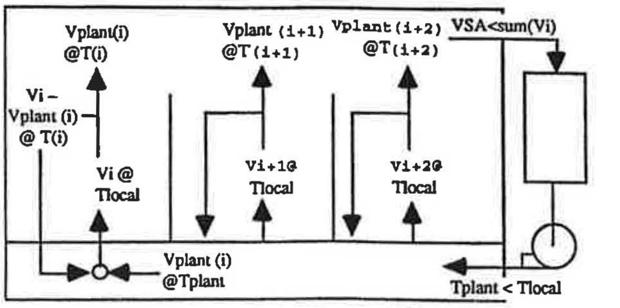

Fig. 5. Plant supply temperature < local supply temperature.

Table 1. It is assumed that the ceiling is cooled by air which flows up from the cells—air is not supplied directly to the area near the ceiling, consistent with the concept and applications of floor-based HVAC. The heat balance equation for a well-mixed ceiling cell at unknown temperature T_c can then be written as:

$$Q_{\text{internal}} + h_h A_h \sum_{i=1}^{mn} (T_i - T_c) + 1.232 \sum_{i=1}^{mn} V_i(T_i - T_c) = 0 \tag{4}$$

In the more general case, local supply conditions are achieved by mixing plant air with cell air at the local fan. The air flow of the mixture of plant and cell air is designated as the cell air flow. Plant air supply can be cooler than the local supply. By mixing cold supply air with warmer room air using local fans or ejectors, the plant air flow may be reduced below the total of the individual air flows (Fig. 5). The flow through the ceiling zone can be less than the sum of the individual cell air flows, although it is still the sum of the individual cells' plant air flows. Reduced plant air flow means that the return temperature will also rise. In general, one can imagine a system where the plant supply temperature determines the plant flow rate and the return temperature. The plant air volume is calculated by

$$V_{\text{ceiling}} = \sum_{i=1}^{mn} V_{\text{plant}_i} = \sum_{i=1}^{mn} V_i \frac{T_{\text{s local}_i} - T_i}{T_{\text{s plant}_i} - T_i} \tag{5}$$

This reduces to $V_{\text{ceiling}} = \sum V_i$ when $T_{\text{s plant}_i} = T_{\text{s local}_i}$.

These equations completely describe the model. There are a total of $(mn+1)$ unknowns in this system of equations: one T or V for each cell plus one for the ceiling zone. The equation for the cooling air flow or temperature of each cell is always linear. If any of the cell cooling air flows is unknown, the equation for the ceiling zone temperature is non-linear since all of the unknown V_i appear in the denominator. The V_iT_i product does not introduce non-linearities because only one of the terms can be unknown for a given cell i.

1.5. Heat transfer coefficients

Estimates for h_h and h_v are needed to simulate offices equipped with personal comfort control systems. To estimate h_v, the cell-to-cell heat transfer coefficient, we must consider two cases: partitions and doorways. Using standard correlations for natural convection on a vertical surface, and assuming an air exchange of 5 l/s between cells, the heat transfer coefficient across partitions is calculated to be about 3 W/m^2 °C. The same value is used for doorways for the sake of simplicity. Although heat transfer is expected to be greater through open doorways than partitions, the authors' simulations have revealed the cell-to-cell heat transfer coefficient to have a relatively weak effect on HVAC energy use, as can be seen in the results which follow.

Heat transfer between the ceiling zone and cells is assumed to consist of natural convection from the ceiling zone air to the ceiling surface in series with blackbody radiation from the ceiling surface to the cells. The air is assumed to be a non-participating medium. An overall heat transfer coefficient of 1.5 W/m^2 °C between cells and the ceiling has been calculated using standard correlations for natural convection on the underside of a cooled plate with the ceiling air at 26°C, the ceiling surface at 24.5°C and an average cell temperature of 23°C.

2. Results

We have obtained a few interesting results using the models described in this paper, summarized here. The parameters controlling annual HVAC energy consumption were evaluated using a number of parametric runs. Results indicate that the most important parameters are: the occupancy rate, x; the use of occupant sensors to control HVAC use and heat loads; and the amount of task lighting and the floor-to-ceiling heat transfer coefficient, h_h, both of which determine the thermal stratification of the room. Figs. 6–9 are the results of parametric runs using a randomly generated population of temperature preferences on an 11×14 grid of cells representing a 1000 m^2 space containing 100 work stations configured so that every work station is adjacent to a corridor (Fig. 10). Washington, DC, weather conditions are used [19].

HVAC energy use in Figs. 6–9 is normalized by dividing the occupant-controlled HVAC system energy use by the conventional system HVAC energy use. Total energy use is

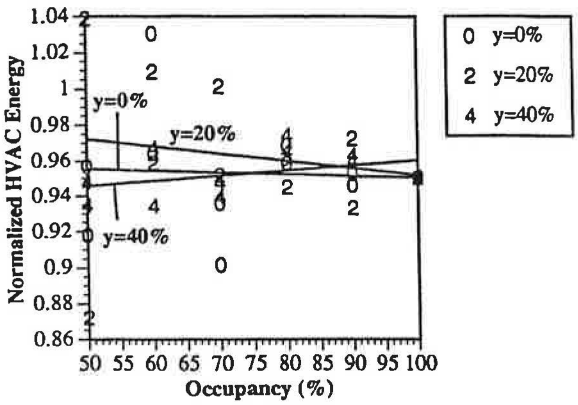

Fig. 6. Effect of occupant behavior on HVAC energy use for a task conditioning system relative to a conventional system, both without occupant sensors.

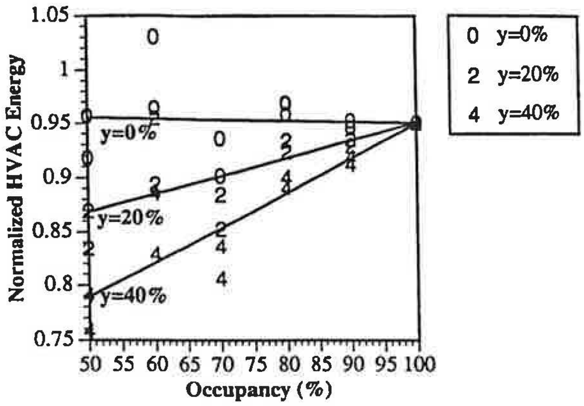

Fig. 7. Effect of occupant behavior on HVAC energy use for a task conditioning system with occupant sensors relative to a conventional system without them.

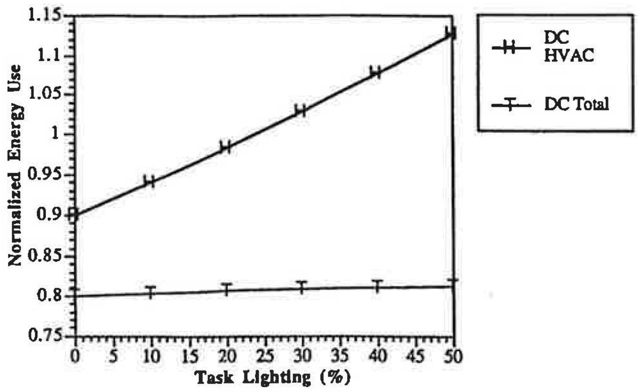

Fig. 8. Task lighting effect on energy use—HVAC energy and total energy (HVAC + lights + plug loads).

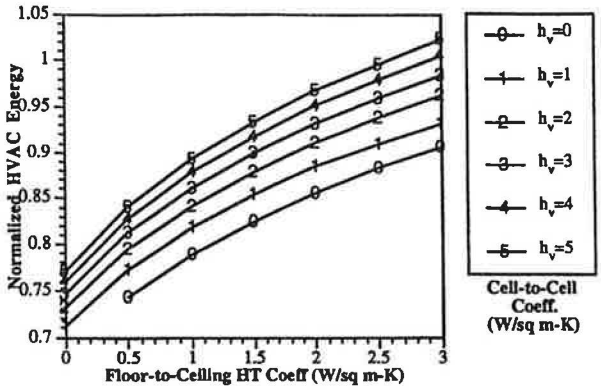

Fig. 9. Effects of heat transfer coefficients on energy use relative to a conventional system.

similarly normalized by dividing the total energy use of the building with occupant-controlled HVAC by the total energy use of the same building with conventional HVAC.

Energy use was simulated for a variable air volume HVAC plant in the steady state. A schematic of the plant simulated is shown in Fig. 10. Except as noted later, the model for the comparative conventional HVAC system is identical to that of the occupant-controlled system.

Central fan. Pressure drops are 5.5 in. of water for the supply fan and 1.5 in. of water for the return fan. For com-

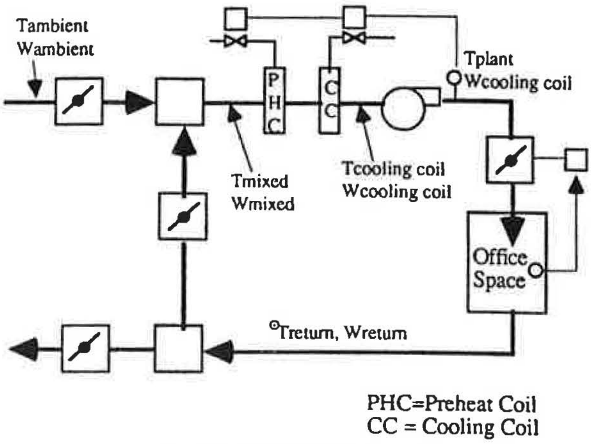

Fig. 10. HVAC system diagram.

parison, a conventional system with higher supply fan pressure drop, 6 in. of water, was used to reflect the resistance of additional supply duct work. The fan efficiency was assumed to be 50% with an ideal cubic part-load characteristic. The central fan was sized for 120% of the maximum expected sensible heat load as described by ASHRAE [18].

Chiller. The chiller is modeled as a steady-state vapor compression refrigeration cycle having a compressor efficiency of 60% and heat exchanger effectiveness of 90%. The chiller was sized to accommodate 120% of the maximum expected cooling load as described by ASHRAE [18]. The working fluid is R-12. Supply temperature is 13°C for a comparative conventional system. For an occupant-controlled system, 10°C was selected to minimize total central plant energy use (fans plus chiller). Plant supply air is mixed with room air at each local fan unit to provide a load supply temperature of 18°C. A return temperature of 24°C is used for the comparative conventional system. An enthalpy-controlled outside air economizer is modeled in all cases.

Schedule. Occupied hours are 7:00 a.m. through 7:00 p.m., Monday through Friday, for a total of 4383 occupied hours per year. Overhead lights are assumed to be shut off over nights and weekends, while task lights and electronic equipment are only shut off by the occupants.

2.1. *Energy effect of non-uniform temperatures*

Non-uniform temperatures are the hallmark of floor-based HVAC systems. In order to estimate their effect on energy use, we compared an energy use estimate of a room having a uniform temperature preference of 23°C with that of ten randomly generated sets of temperature preferences using a mean of 23°C. The non-uniform temperature energy use estimates were, on average, 8% more than the energy estimate for the uniform temperature case. This happens because the energy penalty from the cooler-than-average cells is larger than the savings from the warmer-than-average cells. Local fan energy use is assumed to be proportional to flow rate, which is inversely proportional to the difference between the local air supply temperature and the cell temperature. Thus, local fan energy is non-linearly related to cell temperature. Since the power consumption of the local fans contributes to the cooling load, the central plant chiller and fan energy increase as well. Compounding this effect is the fact that the warmer cells shift some of their heat load to their cooler neighbors.

Both estimates were substantially less than the result for a conventional system generated using the same HVAC plant model. The conventional system used an estimated 13% more energy over the course of a year than the occupant-controlled system with non-uniform temperatures and 21% more than the uniform temperature case. This is mainly a result of stratification within the room and reduced conditioning of corridor areas. Occupant sensors also reduce the heat load in the room by shutting off equipment and task lights. Simulations of various climates (Albuquerque, NM, Houston, TX, and Oakland, CA) show HVAC savings ranging from 5% to 16%. Energy savings rise to 17–20% when lighting and plug loads are taken into account.

2.2. *Influence of occupant behavior on energy use*

The influence of occupant behavior on HVAC energy use is illustrated in Figs. 6 and 7. In both cases, the scatter in the results is due to the probabilistic models of the presence or absence and HVAC control behavior of the occupants. Without occupant sensors, the energy advantage of occupant-controlled HVAC is small, and occupant behavior has little discernible effect on energy use. When occupant sensors are used to reduce conditioning in unoccupied areas, energy consumption falls dramatically, particularly when occupants are often absent and tend to leave office electronic and HVAC systems running when they are absent.

2.3. *Task lighting, heat transfer coefficients and stratification*

Energy savings from occupant-controlled HVAC are mainly due to stratification and reduced conditioning in unoccupied areas. Task lighting affects thermal stratification by shifting the space heat gain away from the ceiling area towards the occupied areas. Reduced lighting levels compensate for the detrimental effect of task lighting on HVAC energy use (Fig. 8).

The heat transfer coefficient between the ceiling zone and the cells also affects stratification (Fig. 9). Decreasing the heat transfer coefficient between the cells saves energy for the same reasons (described above) that uniform temperatures save energy.

3. Conclusions

These results serve to demonstrate that a properly designed occupant-controlled HVAC system can save a substantial amount of energy (13% for the Washington DC climate), while providing increased comfort. These energy savings are due to stratification and reduced conditioning in unoccupied areas. Simulations of various climates (Albuquerque, Houston and Oakland) show savings ranging from 5% to 16%. The energy penalty associated with non-uniform temperatures is real, on the order of 10%, but not enough to negate the energy advantages of the system. For more extensive discussions of the material presented in this paper, see Taub [20]. For more discussion of simulations using the models presented here, see Glicksman and Taub [21].

Further research could be directed towards several ends. Investigations of the behavior of occupants who control their own HVAC systems—their control strategies, their comfort conditions, their comings and goings and how often they remember to turn out the lights—would fill a large gap in the model presented in this paper. Another fruitful avenue of inquiry would be an improved model of heat transfer and air flow in office spaces equipped with floor-based HVAC.

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