Energy Efficiency of Occupant Controlled Heating, Ventilating and Air Conditioning Systems for Office Buildings

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Occupant controlled HVAC systems offer inhabitants of open office spaces some degree of control over their immediate microclimate typically by control of air supplied at floor or desk top level. Productivity gains have been attributed to these systems but it is unclear whether these systems will use less energy than conventional HVAC systems. It is also not clear what the controlling parameters will be.

To study energy consumption, a simplified model of the thermal environment was created for an occupant controlled system. This model was combined with a model of the central HVAC plant and ambient weather conditions to simulate the annual energy usage for several climates. The HVAC control behavior of the occupants (e.g. comfort preferences) and the occupancy of each work station in the space were modeled as random processes. Factors affecting energy use are identified with this model.

Typical occupant controlled systems are shown to offer HVAC savings of 5-16% depending on climate. Savings are achieved by occupancy sensors, properly selected plant and local supply temperatures, reduced cool air supply requirements due to thermal stratification, and reduced conditioning in areas which do not contain a workstation. The influence of occupant behavior, minimum temperature limits, local fan design and task lighting on energy savings is presented.

INTRODUCTION

Occupant controlled HVAC systems offer inhabitants of open office spaces some degree of control over their immediate microclimate typically by control of air supplied at floor or desk top level. Personalized thermal comfort control is the defining feature of occupant controlled HVAC. Worker productivity gains have been attributed to these systems by Kroner et al. (1992) and Paciuk (1989). Hedge et al. (1995) 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. It is unclear whether these systems will use less energy than conventional HVAC systems. It is also not clear what the controlling parameters for energy use will be.

Background

Previous studies of occupant controlled HVAC used conventional HVAC modeling software to estimate energy consumption: Heinemeier et al. (1991) modeled energy use using TrakLoad, while Braun and Seem (1992) used TRNSYS. Both studies considered key physical characteristics of task conditioning systems such as elevated supply and return temperatures, heat and power loads from local fans, and floating temperatures in unoccupied areas. However, conventional HVAC energy modeling systems such as TrakLoad and TRNSYS assume that the temperature of the conditioned space is uniform. Nonuniform temperatures within the conditioned space, which are a key characteristic of occupant controlled HVAC, were not modeled by these authors. As a result their effect on energy consumption is not known. The lack of a detailed thermal model of the conditioned space also hampers the investigation of the parameters that influence energy consumption.

Scope

To study energy consumption, a simplified model of the thermal environment was created for an occupant controlled system. This model was combined with a model of the central HVAC plant and ambient weather conditions to simulate the annual energy usage for several climates. The behavior of the occupants (e.g. comfort preferences) and the occupancy of each work station in the space were modeled as random processes. Mathematical details of the modeling strategy are described in Glicksman and Taub (1996). Factors affecting energy use are identified with this model.

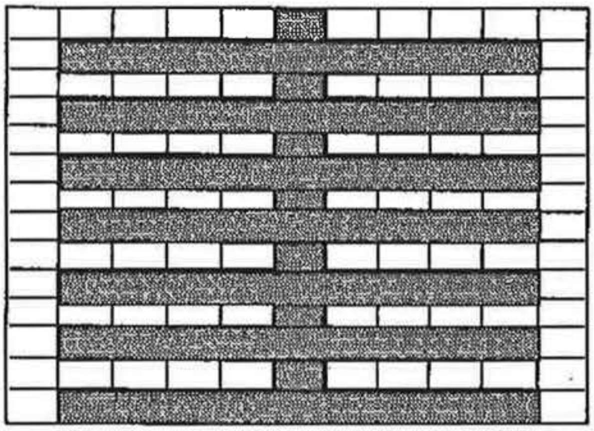
METHODOLOGY

A single interior zone on a floor of a multi-story office building was modeled as a rectangular grid of well-mixed square cells extending from floor level to a height of 2 meters (6.6 ft). These cells represent the occupied portion of the space, including work stations and corridors. The cells are laid out in an 11 by 14 grid to represent a typical office space of 1000 m^2 (10,800 ft^2). Each of the 100 occupied cells is adjacent to a corridor, as shown in the floor plan in Figure 1. Above the array of cells is a single well-mixed ceiling zone. Cool air is supplied through a floor plenum, and exhausted through the ceiling zone, see Figure 2. Recirculation of air from the ceiling zone back to the cells is not included. Since the office space is an interior zone, no heating was modeled.

Space Heat Gains

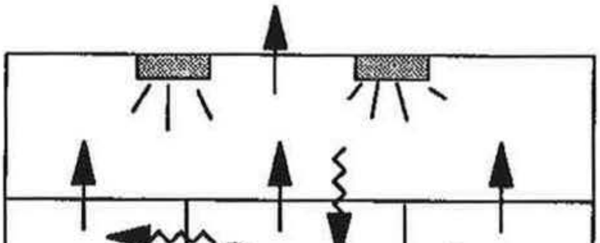
Work station cells are assumed to contain a single person, generating 75 Watts of sensible heat and 45 Watts of latent heat (ASHRAE, 1985). The occupant-controlled HVAC unit is assumed to consume approximately 0.4 Watts per L/s of airflow (0.2 W per cfm), plus 5 Watts for the control unit (Heinemeier et al., 1991). A minimum of 10 L/s (20 cfm) of air flow per unit is maintained according to ASHRAE Standard 62-1989 (ASHRAE, 1989). Each workstation contains 110 Watts of heat-generating electronic equipment, i.e. personal computers[1] (Norford et al., 1988). The ceiling zone contains 19 Watts per square meter of overhead lighting. One third of the overhead lighting heat load is arbitrarily modeled as radiant heating of the cells, and two thirds as heating in the ceiling zone. Task lighting was varied from none up to 50%. Each Watt of task lighting is assumed to supplant 2 Watts of overhead lighting, so that the maximum task lighting case represents 4.8 W/m^2 (0.4 W/ft^2) of task lighting and 9.5 W/m^2 (0.9 W/ft^2) of overhead lighting.

Figure 1. *Floor Plan of Simulated Office Space*



Note: Shaded areas represent corridors.

Figure 2. *Schematic of Office Space Model*

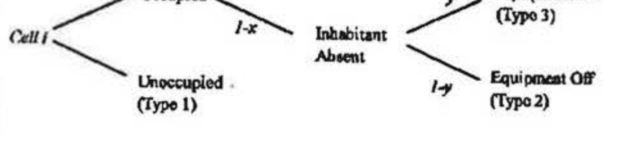


A simple steady-state heat balance for the cells and the ceiling zone accounted for heat exchange between neighboring cells, heat exchange with the ceiling zone, internal heat loads and cooling air flow. Heat transfer coefficients were 3 W/ m^2-°C (0.53 Btu/h-ft^2-°F) between cells and 1.5 W/ m^2-°C (0.26 Btu/h-ft^2-°F) between cells and the ceiling zone. For those cells whose temperature was controlled by the occupant, the model was solved for cooling air flow. For areas such as corridors with "floating" temperatures, the model was solved for temperature. The return temperature was an output of the model since the air flow was determined by the comfort preferences of the occupants. Return temperatures varied from 24.5°C (76°F) and 29°C (84°F) depending on the amount of overhead lighting, the floor-to-ceiling heat transfer coefficient and occupant behavior. See Glicksman and Taub (1996) for a summary of the equations used.

Occupant Behavior Model

Occupant behavior determines the heat and power loads in the conditioned space, affecting energy consumption. Heat loads are determined by a random process using two parameters, see Figure 3. HVAC control behavior is assumed to consist of a selection of a preferred temperature. HVAC system power consumption is a function of these tempera-

Figure 3. *Cell Tree for Determining Heat Loads in Cells*



tures. These temperatures are selected according to a normal distribution having a mean of 23°C (73°F) and a standard deviation of 1.5°C (2.7°F) reflecting individual comfort preferences within the ASHRAE standard comfort limits. Temperature limits were placed at ±2 standard deviations (20°C and 26°C, 68°F and 79°F) to reflect the limitations of the HVAC equipment. Lutzenheiser (1992) and Kempton et al. (1992) offer some insight into occupant's control strategies. The temperature of unoccupied areas was allowed to float up to 26°C (79°F). Relative humidity of 50% was assumed, but humidity was not directly modeled. Temperature was the only comfort criterion used.

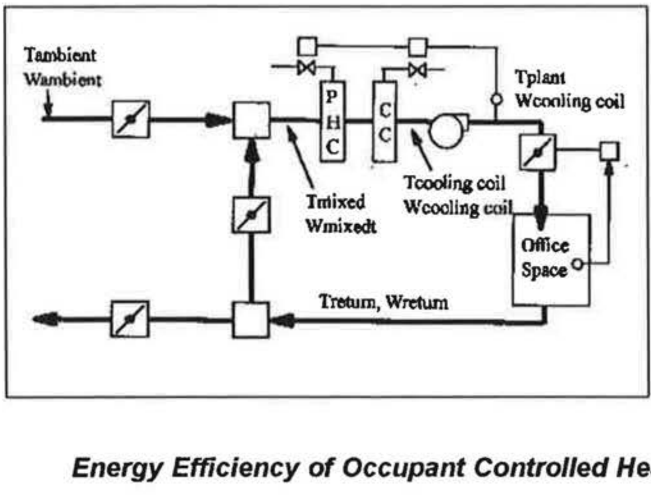
Random trials determined if an individual workstation was occupied, using an 80% probability of an occupant being present as a base case. A second random trial determined whether an absent occupant switched off the control task lighting, electronic equipment, and local HVAC controls in their workstation. Occupant sensors, which are assumed to control task lighting, electronic equipment, and local HVAC controls, are modeled by setting the probability of switching these items off at 100%. In cases where occupant sensors are not modeled, it is assumed that the occupant either shuts off all of these items or none of them. A 40% probability of leaving these items on is used as a base case.

HVAC Plant and Ambient Climate Simulation

Energy use was simulated for a variable air volume HVAC plant in steady state. A schematic of the plant simulated is shown in Figure 4. Energy consumption was modeled in steady state using the "bin method" described in the ASHRAE Handbook (ASHRAE, 1985). Only cooling equipment was modeled, since the interior zone is heated by internal heat gains.

Fan. Fan power is calculated as the product of flow rate and pressure drop divided by efficiency, with an ideal cubic part-load characteristic. Total fan and motor system efficiency was assumed to be 50%. Static pressure drops are 14 cm (5.5 in) of water for the supply fan and 3.8 cm (1.5 in) of water for the return fan. For comparison, a conventional system with a supply fan static pressure drop of 15.3 cm (6 in) of water was used to reflect the resistance of additional supply duct work. The central fan was sized for 120% of the maximum expected sensible heat load as described in the ASHRAE Handbook (ASHRAE, 1985).

Figure 4. Schematic of Simulated Central HVAC System



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 coefficient of performance calculated by the thermodynamic model was supplemented with a quadratic part-load curve taken from the ASHRAE Handbook (ASHRAE, 1985). The chiller was sized to accommodate 120% of the maximum expected cooling load as described in the ASHRAE Handbook (ASHRAE, 1985). The working fluid is R-12. Supply temperature is 13°C (55°F) for a comparative conventional system and 10°C (50°F) for an occupant controlled system. This plant supply temperature is 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 local supply temperature of 18°C (64°F). A return temperature of 24°C (75°F) is used for the conventional system comparison. An enthalpy-controlled outside air economizer is modeled.

Climate. Bin weather data from the U.S. Air Force (1978) for Washington, DC, Albuquerque, NM, Houston, TX, and Oakland, CA are used.

Schedule. Occupied hours are 7:00 AM through 7:00 PM, Monday through Friday, for a total of 4,383 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 or by occupant sensors if they are installed.

RESULTS

Energy use of the occupant controlled system for the Washington, DC climate is presented in Table 1 as the average of the results of ten trials using different sets of randomly-generated temperature preferences and occupant behavior parameters. A conventional HVAC system simulated using a similar model consumed 65,000 kilowatt hours per year. The average of 10 trials using Washington, DC climate data is 56,100 kilowatt hours per year, for an average energy savings 13% of HVAC energy use, 9% when compared to a conventional system with occupant sensors. Standard deviations are included in Table 1 to illustrate the amount of variation due to occupant behavior. These variations are not great enough to eliminate the energy savings relative to conventional HVAC. When lighting and plug loads are taken

Table 1. Energy Use of Occupant Controlled HVAC for Washington, DC Climate—Ten Trials assuming use of occupant sensors and 80% of occupants present

	Mean (kWh/year	Standard Deviation
Chiller	27,045	377
Central Fan	20,871	1,564
Local Fans	8,162	529
TOTAL HVAC	56,076	2,377
Overhead Lights	47,587	
Plug Loads	38,445	
TOTAL	142,108	

into account, energy use of the occupant controlled HVAC system is approximately 142,100 kWh per year against 181,000 kWh per year for the conventional system, a savings of 22%. These additional savings occur because the occupant sensors included in the local HVAC system shut off task lights and plug loads during unoccupied hours.

Typical occupant controlled systems are estimated to offer HVAC savings of 5–16% depending on climate. When lighting and plug loads are taken into account, savings are 17–22%. Table 2. Savings are achieved by occupancy sensors, properly selected plant and local supply temperatures, reduced cool air supply requirements due to thermal stratification, and reduced conditioning in areas which do not contain a workstation. Occupant sensors offer the largest portion of the energy savings. Compared to a conventional HVAC system equipped with occupant sensors, the occupant controlled system savings are 1–12% of HVAC energy consumption and 0.6–4.4% of total energy consumption. Figure 5 illustrates the reduced conditioning in the corridors.

Factors Influencing Energy Use of Occupant Controlled HVAC Systems

Using the model described in this paper, the most significant factors influencing energy use were found to be occupant behavior, minimum temperature limits, local fan design and task lighting. Local and plant supply temperatures and heat transfer coefficients within the room were found to have less significant effects.

The Energy "Cost" of Individual Temperature Preferences. When all occupant temperature preferences are set at 23°C (73°F), energy consumption of the occupant controlled HVAC system is estimated to be 51,600 kWh/year for Washington, DC. This value is 8% less than the average result using normally-distributed individual temperature preferences. This is the energy "cost" of allowing occupants to control the temperature of their individual microclimates. The effect is caused by the local fans, which increase the cooling load in addition to their direct power consumption. Cooling air flow requirements and the corresponding local fan energy consumption grow rapidly (proportional to the inverse of the difference between the local supply temperature and the cell temperature) as the temperature of the cell approaches the local supply temperature. When cell temperatures are nonuniform the local fans in the warmer-than-average cells use less energy. However, this savings is overwhelmed by the penalty from local fans in the cooler-than-average cells. Occupant comfort can be individualized by allowing occupants to control the airflow direction but not control the ambient temperatures if additional energy savings are desired.

Effect of Occupant Behavior on Energy Use. Since little information was available in the literature on occupant behavior, the two behavior parameters were varied to gage their influence on energy consumption. In the process, individual temperature (comfort) preferences were the same for each trial, though different for each occupant. The results for occupant controlled HVAC energy use are presented in Figure 6 and Figure 7 for HVAC energy use plus lights and plug loads. These results are normalized against energy consumption for the same climate and behavior parameters for a conventional HVAC system. The range of results is due to the use of probabilistic models to determine which specific cells are occupied and the resulting temperature preferences for each trial. The importance of the occupant sensor is apparent. The more likely occupants are to be absent or leave their electronic equipment on, the greater the energy savings. These savings are mostly a feature of the occupant sensor, however, occupant controlled HVAC systems offer an opportunity to incorporate these sensors into the work space that is absent in offices equipped with conventional HVAC systems. When compared to a conventional HVAC system that incorporates occupant sensors, the occupant controlled HVAC system exhibited HVAC energy savings of 6% and total energy use savings of 2% independent of occupant behavior parameters.

Effect of Task Lighting on Energy Use. Task lighting is an increasingly popular way to increase productivity and decrease energy consumption because it offers more efficient lighting along with reduced lighting in non-work space areas. However, since task lights can be left on during unoccupied hours while overhead lights are shut off, they could increase

__Table 2.__ Typical Energy Use of Occupant Controlled HVAC Using Occupant Sensors Compared to Conventional HVAC without occupant sensors—80% occupancy, 40% of occupants do not switch off plug loads when absent.

	CONVENTIONAL HVAC (kWh/yr)		OCCUPANT CONTROLLED (kWh/yr)
	No Sensor	Occupant Sensor	Occupant Sensor
Washington, DC			
HVAC Only	65,000	62,058	58,526
Total Energy Use	180,711	148,090	144,558
Albuquerque, NM			
HVAC Only	57,500	54,630	48,474
Total Energy Use	173,211	140,662	134,506
Oakland, CA			
HVAC Only	55,727	52,908	50,969
Total Energy Use	171,438	138,940	137,001
Houston, TX			
HVAC Only	90,164	86,964	85,888
Total Energy Use	205,875	172,996	171,920

NOTE: Results in this table are for a single trial for each climate using identical temperature preferences and occupant locations. As a result the Washington, DC data differ somewhat from the value in Table 1, which is based on the average of 10 trials.

__Figure 5.__ Distribution of Cell Temperatures

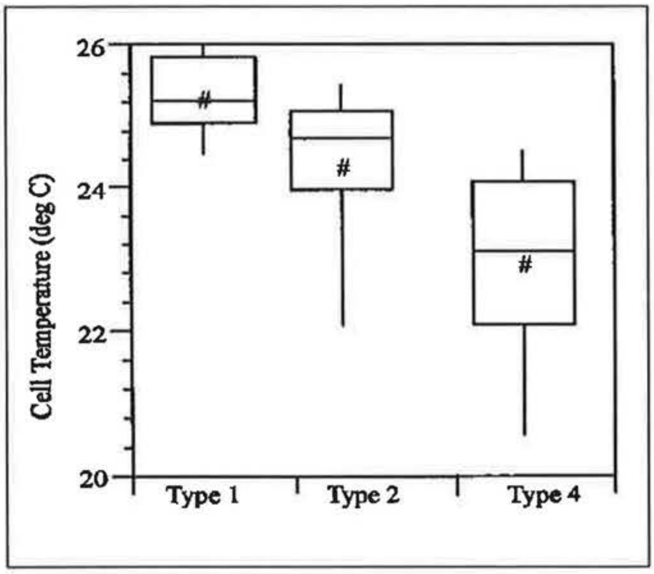
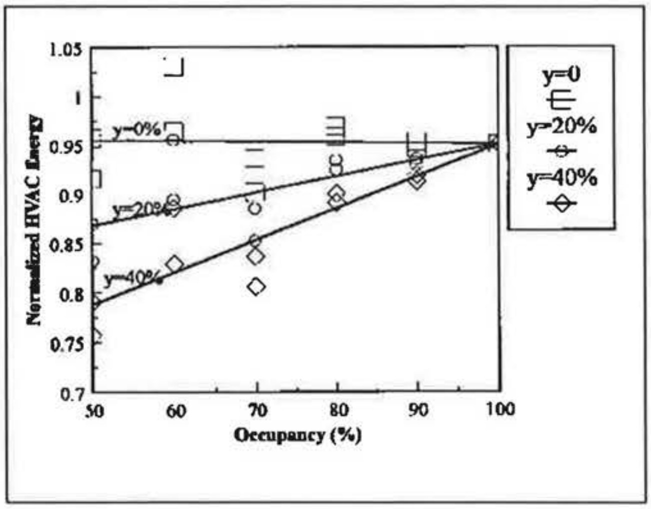


__Figure 6.__ HVAC Energy Use of Occupant Controlled System Relative to a Conventional HVAC System as a Function of Occupant Behavior



rather than decrease energy use. In these simulations, occupant sensors are assumed to shut off task lights after hours. Task lighting reduces the overall cooling load, and shifts it away from the ceiling area and into the work space, so it cannot be examined independently of its effect on HVAC energy use. It is found that while task lighting increases occupant controlled HVAC energy use, the decreased energy use for lighting balances the effect so that the net effect on energy use should be negligible. See Figure 8.

Figure 7. Total Energy Use as a Function of Occupant Behavior for an Office with Occupant Controlled HVAC Relative to an office with a Conventional HVAC System

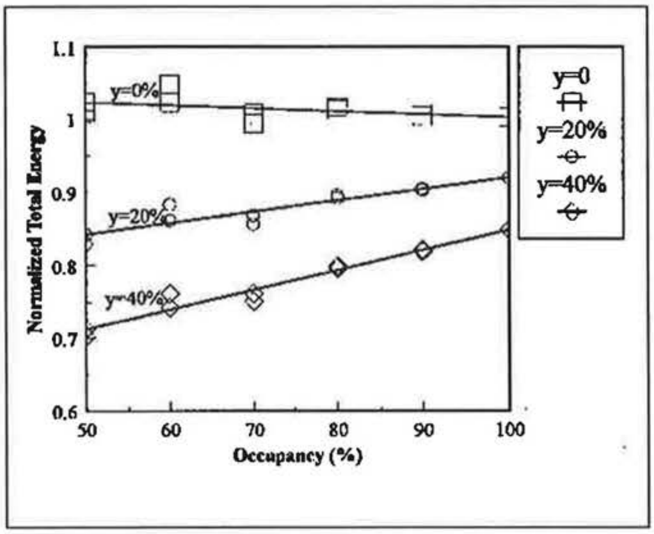
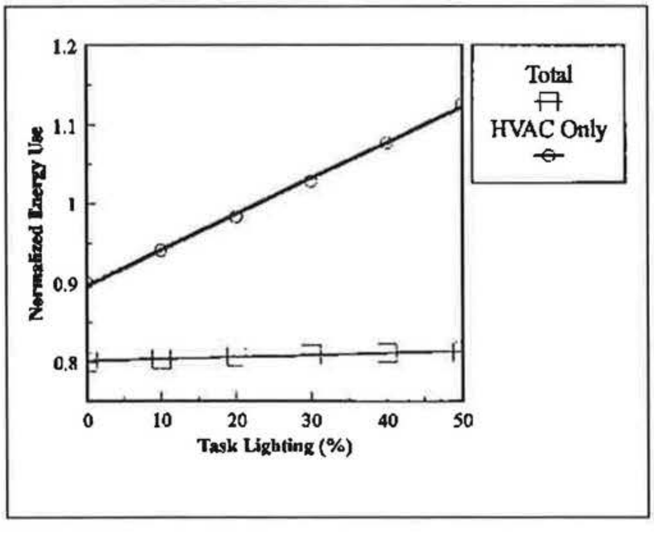


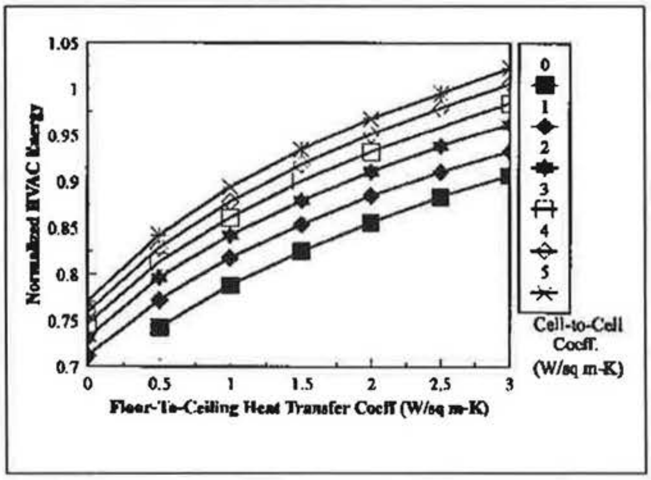
Figure 8. Effect of Task Lighting on the Energy Use of an Office with an Occupant Controlled HVAC System Relative to a Conventional HVAC System



Effect of Heat Transfer Coefficients on Energy Use. The estimates of heat transfer coefficients used in these simulations cannot be expected to be very accurate due to the complex geometry and the interaction of forced and natural convection. Figure 9 presents the results of parametric runs for various values of the floor-to-ceiling and cell-to-cell heat transfer coefficients. The floor-to-ceiling heat transfer coefficient dominates because of its effect on thermal stratification.

Effect of Local Fan Design on Energy Use. As illustrated by Table 1, local fans account for approximately 15%

Figure 9. Sensitivity of Energy Use of Occupant Controlled HVAC to Heat Transfer Coefficient Values



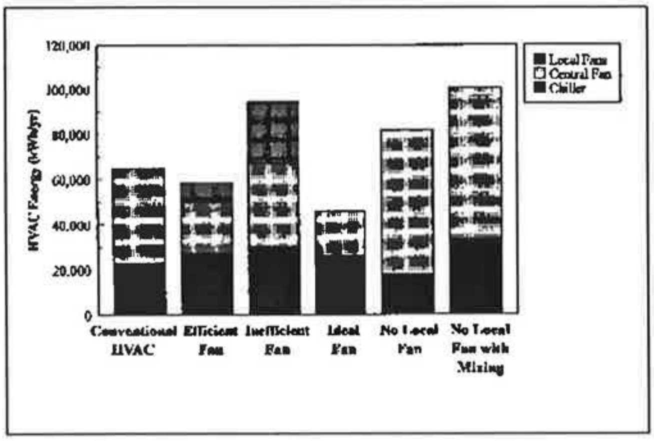
of the energy consumption of occupant controlled HVAC. The fan simulated in this paper has a part load characteristic of 0.43 W per L/s (0.2 W per cfm) of air flow based on measurements of a commercial system by Heinemeier et al. (1991). Another commercial system tested by the same authors consumed 82 Watts at 150 cubic feet per minute of airflow. These systems both include filters. Based on kinetic energy, an ideal fan would consume approximately 0.01 W per L/s (0.005 W per cfm).

Increasing the supply plenum pressure could allow occupants to control cooling air flow using variable dampers. However, without local fans to mix plant supply air with room air, plant supply must be warm enough to supply directly to the occupants. The increased plant supply temperature offers increased chiller efficiency and economizer savings, but this is outweighed by the increased energy consumption due to the higher central fan pressure.

Results using an ideal local fan, the base case local fan and the less efficient local fan are shown in Figure 10. This figure also presents the results of simulations without local fans, one where plant and local supply temperature is 18°C (64°F) and another where plant supply is 10°C (50°F) and local supply is 18°C (64°F), so that room air is mixed into the local supply. Although the ideal fan case shows significant energy savings (22% less than the standard fan), the inefficient fan and no fan cases both show significant penalties.

Effect of Minimum Temperature Limits on Energy Use. Occupants who chose low temperatures have a disproportionate effect on the energy use of the HVAC system. Cooling air flow rises rapidly as the temperature of the conditioned space approaches that of the cooling air. The heat output of the local fan compounds this effect, as does the increasing heat load from warmer neighbors. The

Figure 10. Effect of Local Fan Design on Energy Use of an Occupant Controlled HVAC System



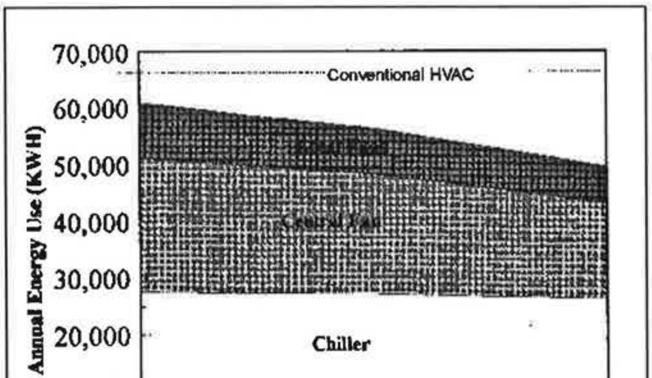
increased air flow reduces thermal stratification in the room, causing greater central fan power consumption. The lower plant return temperature also decreases the available hours of economizer operation.

A minimum temperature of 20°C (68°F) has been chosen for this simulation. The results of varying this parameter are found in Figure 11. As expected, energy use falls as the minimum temperature is raised.

CONCLUSION

Simulations demonstrate that occupant controlled HVAC systems can offer significant energy savings in addition to increased occupant comfort. However, improperly designed systems can result in an even larger energy penalty. The greatest influence on energy consumption, occupant behavior, is also the greatest unknown.

Figure 11. Effect of Minimum Allowed Workstation Temperature on HVAC Energy Use of an Occupant Controlled HVAC System Relative to a Conventional HVAC System



Occupant sensors offer considerable HVAC energy savings. Energy efficient systems will also incorporate efficient local fans that mix cold air with room air to provide warmer air at the outlet vents. Substantial energy savings can be realized by restricting the minimum temperature that occupants can select. Occupants can compensate by directing airflow toward themselves. Offices incorporating occupant controlled HVAC should be designed to reduce heat transfer between workstations and to thermally isolate the ceiling area from the occupied levels to promote thermal stratification.

ENDNOTES

1. Larger office equipment generates considerably more heat. For example, a typical mid-sized photocopier generates 335 Watts of heat in standby mode and 2.2 kilowatts when copying (Xerox, 1993).

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