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THE IMPACT OF COMFORT CONTROL ON AIR CONDITIONER ENERGY USE IN HUMID CLIMATES

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ABSTRACT

This paper assesses the impact of controlling an air-conditioning (AC) system to maintain constant comfort instead of constant temperature. Three different indices of thermal comfort were used in the analysis: new effective temperature (ET^*) from Gagge et al. (1971), predicted mean vote (PMV) from Fanger (1970), and a modified PMV (PMV^*) from Gagge et al. (1986). Maintaining constant comfort on hourly and daily bases was simulated. Hourly comfort control represented the performance of an ideal "comfort" thermostat, while daily comfort control was more typical of human interactions with a conventional thermostat. Daily and hourly comfort control resulted in similar impacts on energy use.

A building simulation model (Kerestecioglu et al. 1989) was used to simulate the impact of comfort control on energy use in a typical residence in Miami and Atlanta using three different AC systems: a high sensible heat ratio (SHR) AC, a medium-SHR AC, and a heat-pipe-assisted AC. A more generalized analysis of how comfort control would impact other advanced low-SHR AC systems is also presented.

Under conventional temperature control, adding heat pipes lowered the space relative humidity (RH) but increased annual energy costs in Miami by 15%. Under comfort control (PMV^*), the energy cost of adding heat pipes dropped threefold to 5%. Conversely, comfort control penalized the high-SHR AC system because a lower temperature was required to compensate for higher space RH.

A general analysis of the building load at a moderate infiltration rate (0.5 air changes per hour [ach]) showed that lowering space RH from 50% to 40% while holding temperature constant increased the total load by 11%. When the RH was reduced while maintaining constant comfort (PMV^*), total load actually decreased by 3%. This implies that advanced high-efficiency, low-SHR systems may be able to reduce space RH with little or no increase in energy costs when comfort is considered.

INTRODUCTION

Several advanced air-conditioning (AC) systems are now available that offer improved dehumidification performance. Examples of these systems include heat-pipe-assisted ACs, heat-actuated desiccant cooling machines (both liquid and solid), and desiccant-enhanced air conditioners (Khattar 1985; Marsala et al. 1989; Cromer 1988). All of these systems offer improved dehumidification by providing a larger fraction of their total cooling output as latent capacity (moisture removal) rather than sensible capacity (temperature reduction). Conventional AC systems typically have sensible heat ratios (SHRs) in the range of 0.7 to 0.85 (70% to 85% of their cooling capacity is sensible). The advanced AC systems have SHRs much lower than conventional AC systems, with only slightly lower efficiencies. Therefore, they offer the potential to cost-effectively maintain lower space humidity than is presently possible with conventional AC systems.

Maintaining lower humidities is especially important in hot, humid climates such as the southeastern United States. Using conventional AC systems in humid climates, indoor relative humidity (RH) is typically in the 55% to 60% range (Cummings 1990); this is at the upper limit of the 40% to 60% RH range generally recommended for human health (Sterling et al. 1985). The advanced AC systems have been proposed as a means to lower indoor RH in residences.

While it is clear that these advanced AC systems can lower RH, there is controversy over the energy consumption and operating costs of these systems. Often, the efficiency of these advanced systems is slightly lower than that of conventional AC systems. Still, several proponents cite the potential for lower energy costs. The rationale given for potential savings is that by lowering RH in the space, a higher dry-bulb temperature is permissible while maintaining equivalent comfort. As a result of the higher temperature setpoint, the energy costs of the system are reduced compared to conventional AC systems.

Purpose of the Study

The purpose of the study was to determine the impact of comfort on AC energy use. Specifically, the effect of maintaining constant comfort (instead of constant temperature) on energy use was studied for three systems: a high-SHR AC, a medium-SHR AC, and a heat-pipe-assisted (i.e., low-SHR) AC. The building simulation model FSEC 2.1 (Kerestecioglu et al. 1989) was used to determine the

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impact of comfort control on these systems. This paper also presents a general analysis of how comfort control affects the energy consumption of other advanced dehumidification systems.

APPROACH

Comfort Indices: ET*, PMV, and PMV*

Thermal comfort depends on several factors, including temperature, humidity, air velocity, clothing level, and metabolic rate. Several indices have been developed to quantify comfort. Two of the most popular are the new effective temperature (ET*) from ASHRAE (1981) and the predicted mean vote (PMV) (Fanger 1970; ISO 1984). Recently, Gagge et al. (1986) proposed a modified version of PMV (PMV*) to better account for the effect of humidity.

ET* is a physically based rationalization of the original ASHVE ET comfort chart developed by Houghten and Yaglou (1923). The physical basis for ET* was developed and presented by Gagge et al. (1971). ET* is defined as the condition that imposes thermal strain equivalent to dry-bulb (DB) temperature at 50% RH. ET* equals DB at 50% RH. Lines of constant ET* on the psychrometric chart represent lines of constant comfort. Constant ET* lines define the left- and right-hand borders of the ASHRAE comfort zone (ASHRAE 1981) on the psychrometric chart.

Unlike ET*, which explicitly accounts only for temperature and humidity, PMV accounts for additional factors including clothing, metabolic rate, and air velocity. PMV has been widely accepted as an overall index of comfort. However, Gagge et al. (1986) stated that PMV only accounts for total body load and does not correctly consider the impact of humidity on comfort. Therefore, they developed a new index, PMV*, to better account for humidity effects. PMV* has the same functional form as PMV (ISO 1984), except ET* is substituted into the comfort equations in place of the operative DB temperature.

A requirement for the current study was to develop contours of iso-comfort as a function of temperature and humidity. Figure 1 shows iso-comfort lines for PMV, ET*, and PMV* on the psychrometric chart. The lines (PMV = −0.01, ET* = 78°F, and PMV* = −0.01) all intersect at 78°F, 50% RH, and all apply for a normally clothed person at rest (met = 1.2, clo = 0.5, v = 40 fpm [0.2 m/s]). The humidity dependence of these iso-comfort lines is lowest for PMV, followed by ET* and PMV*. The intersecting wet-bulb (WB) line has also been included in the figure as a point of reference. The humidity dependence, or slope, of the lines was of particular interest in this study. The slope indicates the rate at which dry-bulb temperature can be increased with decreasing RH while maintaining constant comfort. All three of these indices were used in the analysis.

AC Models

To simulate AC performance in humid climates, it is important to accurately predict the SHR and energy efficiency ratio (EER) at off-design conditions. Accurately modeling AC performance is especially important for a heat-pipe-assisted AC, where the inlet air temperature can

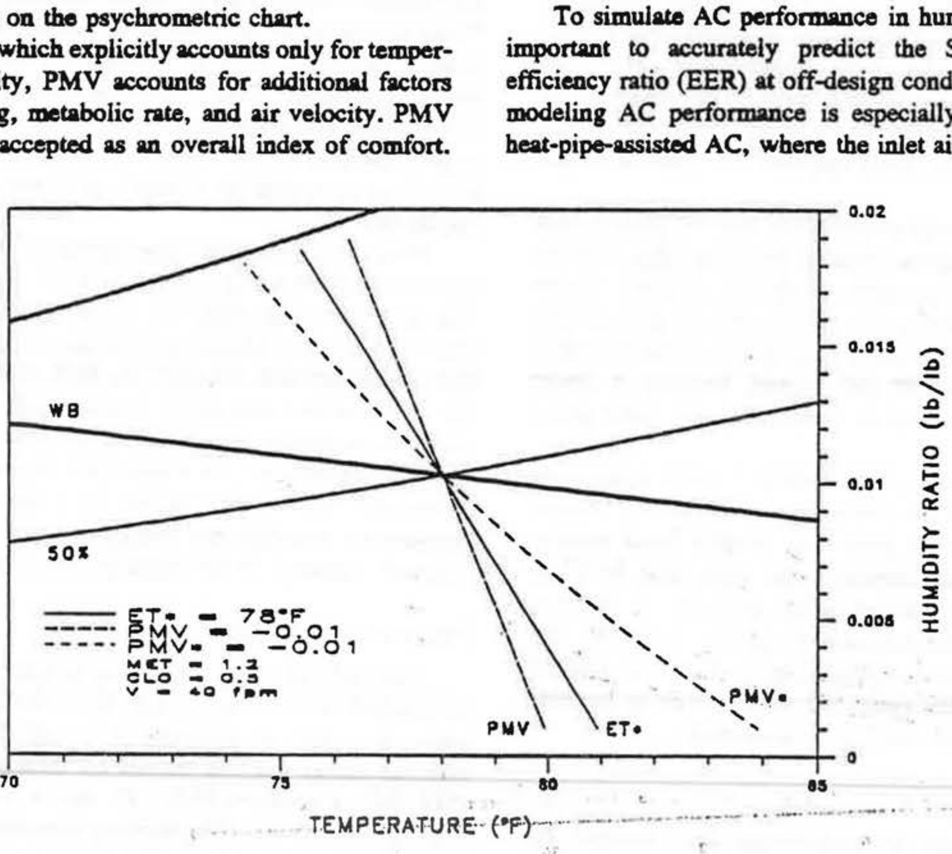


Figure 1 Iso-comfort lines for ET, PMV, and PMV* on the Psychrometric Chart.*

be much lower than rated (or design) conditions (80°F and 50% RH).

The AC model developed for this study combined the DOE-2 default performance curves (DOE 1982) with the apparatus dew point (ADP)/bypass factor (BF) approach (Carrier et al. 1959) to predict off-design sensible/latent performance. This method is described in detail in the appendix. The resulting performance map for the medium-SHR AC unit is shown in Figure 2. Contours of constant SHR and EER for the AC unit are plotted on the psychrometric chart for evaporator inlet temperature and humidity. Several real-world aspects of AC unit performance are reflected in Figure 2, including the variation of evaporator ADP with inlet WB and evaporator dryout at low humidities.

The main advantage of using this method for modeling AC performance is that any system can be rationally simulated with minimal information; only total capacity, EER, and SHR at rated conditions (i.e., 95°F [35°C] outdoors, 80°F [26.7°C], and 50% RH indoors, and 400 cfm/ton [0.05 m^3/kW·s]) are required to generate the entire performance map.

Three AC systems were simulated for this study: a high-SHR AC, a medium-SHR AC, and a heat-pipe-assisted AC. Table 1 lists the characteristics of each system.

Figure 3 shows the psychrometric process and the schematic for the heat-pipe-assisted AC system. This system uses the medium-SHR AC with a heat pipe air-to-air heat exchanger added to enhance dehumidification performance. Sensible heat exchange between the inlet and outlet of the AC evaporator results in both lower SHR and EER (see Figure 2).

Building Model

The computer model FSEC 2.1 (Kerestecioglu et al. 1989) was used to simulate building performance. This public domain program simulates thermal and moisture transport and storage in building components. Proper treatment of thermal and moisture transport and storage is important when simulating air conditioner/building interactions in humid climates. FSEC 2.1 was selected for this study to accurately model time-varying indoor humidity levels.

In this study, the thermal performance of each building component (walls, floor, etc.) was modeled using the conduction transfer function (CTF) method. The CTF method requires the boundary condition for each component to be a convective link to either a zone or to ambient conditions. Solar flux and interzonal radiation are treated as heat flux boundary conditions.

Moisture storage inside the building was modeled with the effective moisture penetration depth (EMPD) method described by Kerestecioglu et al. (1990). This simplified method assumes moisture is lumped at the surface of interior components and is in equilibrium with the material at the surface temperature. This approach is computationally more efficient than detailed numerical methods and is more rational than the lumped-air methods that are commonly used.

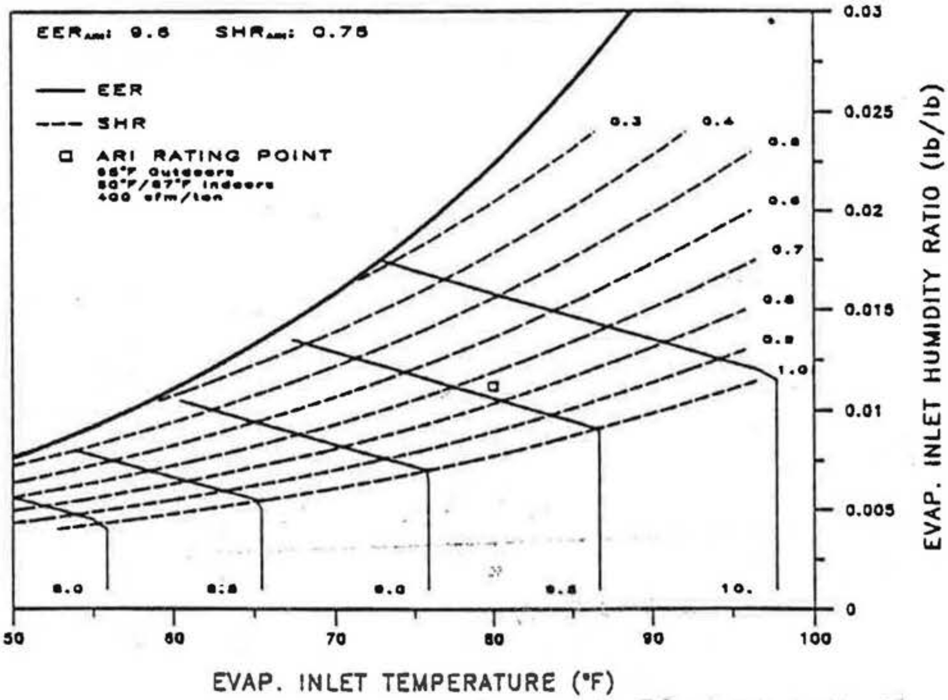
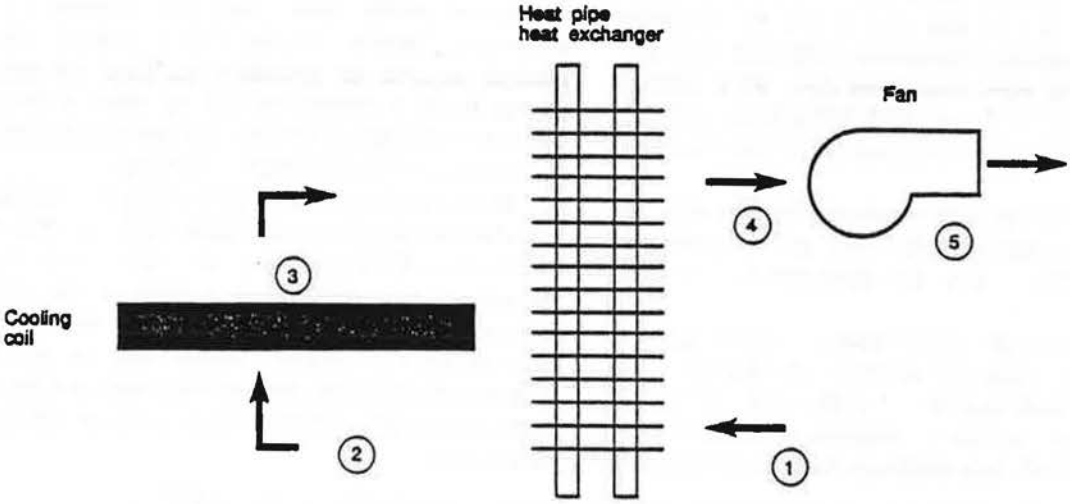


Figure 2 Contours of constant EER and SHR on the Psychrometric Chart of evaporator inlet conditions.

TABLE 1
Description of AC Systems

System Type	Rated Capacity kBtu/h (kW)	Efficiency Btu/Wh (W/W)		Rated SHR
		Seasonal	Rated	
High-SHR	22.5 (6.6)	10.1 (3.0)	9.6 (2.8)	0.83
Medium-SHR	22.5 (6.6)	10.1 (3.0)	9.6 (2.8)	0.75
Heat-Pipe	21.1 (6.2)	9.4 (2.8)	8.9 (2.6)	0.60

Notes: All systems include indoor fans.
Rated Conditions: 80°F (26.7°C) and 50% RH indoors, 95°F (35°C) outdoors, 400 cfm/ton (0.05 m^3/kW-s).
Seasonal efficiencies calculated assuming C_D=0.2.
Heat Pipe: 2 ft^2 (0.2 m^2) per side, two-row plate fin, 14 fins/inch (0.5 fins/mm), heat transfer effectiveness: 0.35.



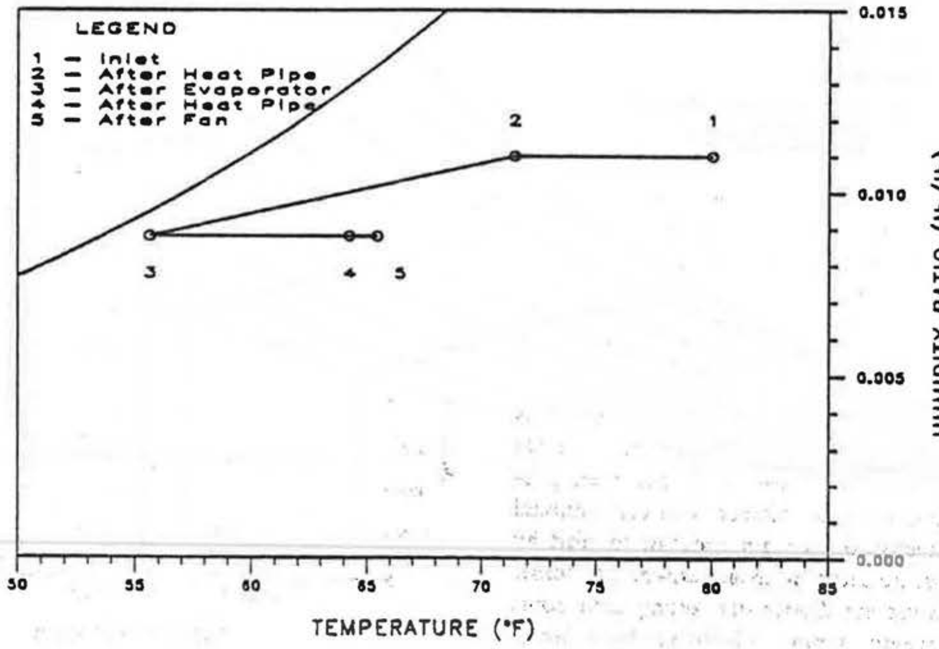


Figure 3 Schematic and psychrometric process of a heat-pipe-assisted AC.

TABLE 2
Description of House

Overall	
Construction:	Slab-on-grade, single story
Floor Area:	1,500 ft² (139.4 m²)
Glazing:	18% of wall area, equally distributed
Insulation	
Walls:	R-11 (drywall, fiberglass, siding)
Ceiling:	R-24 (drywall, fiberglass)
Infiltration and Internal Gains	
Air Changes Per Hour (ach):	0.54 average, 0.72 peak
Internal Gains:	2.8 kBtu/h (0.8 kW) average, 6.1 kBtu/h (1.8 kW) peak, (62% sensible, 38% latent)

Typical meteorological year (TMY) data were used by the model to run hour-by-hour annual simulations. The residence simulated in this study is described in Table 2.

Temperature and Comfort Control

The AC equipment model was integrated into the building simulation model such that either constant temperature or constant comfort could be maintained. In the building simulation model, the conventional method of AC system control was to calculate the sensible cooling required to maintain the temperature setpoint for each hour. The required sensible cooling was then used to determine the run-time fraction of the AC unit based on available sensible capacity. The AC run-time and latent capacity were used to determine the moisture removed by the AC unit, and then the resulting average space RH was calculated for each hour.

For the comfort control mode, the comfort index was specified as the "setpoint." Then the temperature setpoint was calculated for each hour based on the required comfort setpoint and the average RH. Iterations were required since the average RH was a function of the temperature setpoint. The net result was that the temperature setpoint moved up and down the comfort contour (see Figure 1) as the humidity level in the space varied. Any of the comfort contours in Figure 1 (ET*, PMV, or PMV*) could be used to adjust the space temperature.

While the comfort control mode described above is technically possible, the authors know of no actual device that performs this function. However, it has been postulated that building occupants interact with conventional thermostats in an attempt to maintain constant comfort by adjusting the temperature setpoint up and down. Of course, occupants do not adjust the thermostat setting each hour; the adjustment frequency is more likely on the order of once per day.

To simulate daily thermostat adjustment, logic was included in the computer algorithm to adjust the temperature setpoint only once a day at a specified time. At the specified time, the temperature was adjusted to match the comfort setpoint for that hour only. The temperature setpoint then remained fixed until the specified hour of the following day. This daily comfort control mode was included to emulate occupant interaction with the thermostat.

RESULTS

Conventional Temperature Control

The three AC systems described in Table 1 were simulated for the entire year using TMY data for Miami and Atlanta. Miami is representative of a humid coastal climate, while Atlanta is typical of a warmer, humid inland climate.

Figure 4 compares the annual energy use for all three machines in the two climates. For all the AC machines under temperature control, Miami used more than twice as

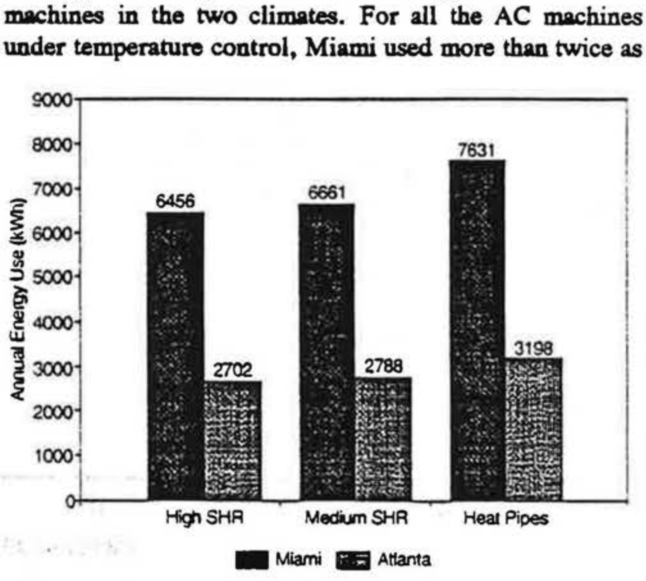


Figure 4 - Annual AC energy use in Miami and Atlanta: temperature control.

much electricity as Atlanta for air conditioning. For both climates, the high-SHR AC used the least energy, followed by the medium-SHR AC and the heat-pipe-assisted AC. As expected, the AC unit with the higher SHR, and therefore more sensible capacity, maintained the temperature setpoint while consuming less energy.

Energy use, however, is not the only consideration. Figure 5 shows the average annual RH for the three AC systems in both climates when under temperature control. The RH was highest for the high-SHR AC unit in both climates. Consequently, the lower energy use of the high-SHR AC was achieved at the cost of higher humidity in the space. Conversely, the low-SHR system maintained lower humidity but at the cost of increased energy use.

Figure 6 compares the rated SHR from Table 1 to the annual operating SHR (delivered) for all three AC systems in both climates. It is interesting to note that the rated SHR of the heat-pipe AC system is 0.23 lower than the high-SHR AC, yet the resulting difference between the annual operating SHRs for these machines was only 0.08. This is a result of the self-compensating nature of AC systems. If an AC system has a high rated SHR, the RH in the space will rise, which, in turn, tends to lower the operating SHR (see Figure 2). By definition, the rated SHR of an AC system only applies at the rating point of 50% RH. When the RH is lower than 50%, the operating SHR will be higher than the rated SHR. When the RH is higher than 50%, the operating SHR will be lower. Figure 6 demonstrates the difficulty of predicting the actual operating SHR from the rated SHR alone.

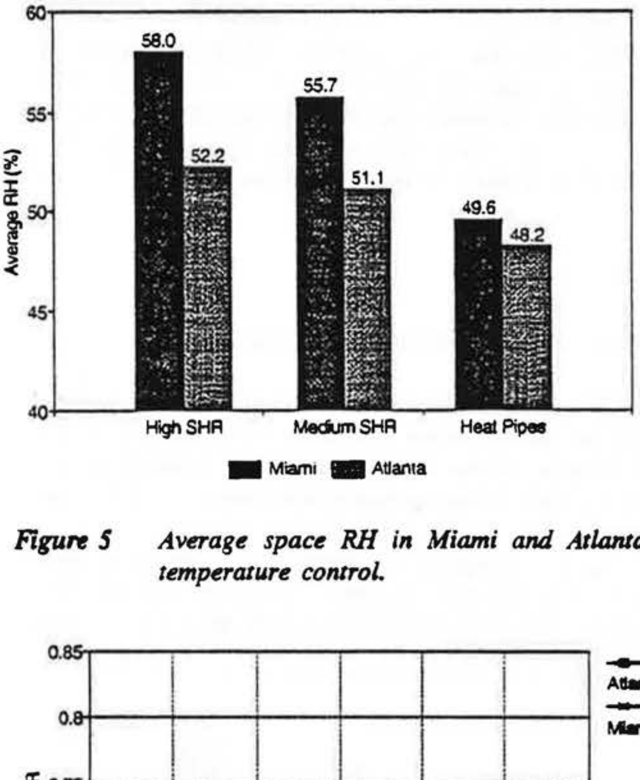


Figure 5 *Average space RH in Miami and Atlanta: temperature control.*

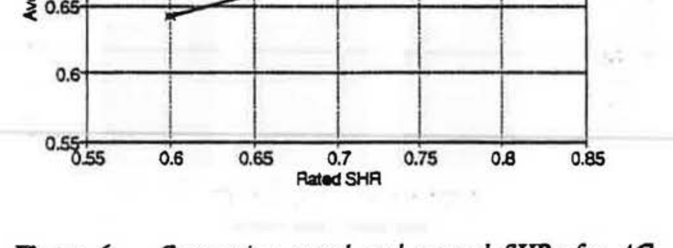


Figure 6 *Comparing rated and annual SHRs for AC systems in Miami and Atlanta.*

Hourly Comfort Control: ET*, PMV, and PMV*

To this point, all results have been presented for temperature control. Figure 7 compares the annual energy use of the systems when maintaining constant "comfort" in addition to constant temperature in Miami. The results are shown for temperature control (T) as well as for three types of comfort control (ET*, PMV, and PMV*). Comfort control increased the energy use (kWh) of the high- and medium-SHR AC units while only slightly reducing energy use for the low-SHR, heat-pipe AC unit. Since temperature and comfort control are equivalent at 50% RH, the impact of comfort control depends on how far the space RH ranged from 50%. The heat-pipe AC system operated close to 50% RH in Miami; therefore, the differences between comfort and temperature control were small. However, the space humidity was higher than 50% RH for the medium- and high-SHR AC systems in Miami; therefore, comfort control increased energy use (compared to temperature control) for these systems by decreasing the average space temperature.

Under comfort control, the differences in energy use between the AC systems decreased compared to tempera-

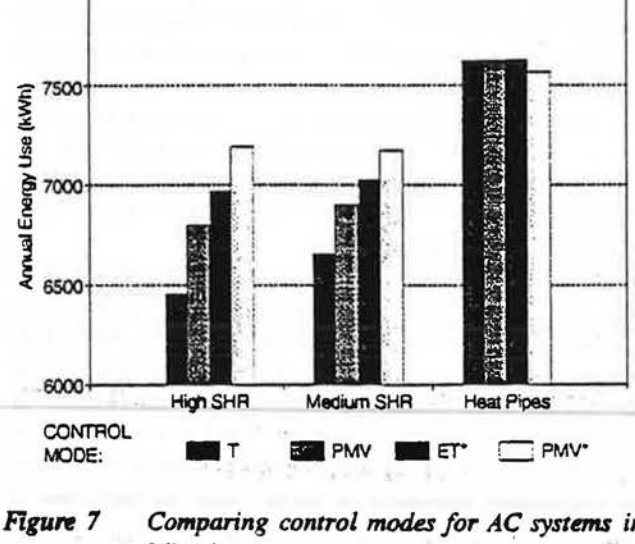


Figure 7 *Comparing control modes for AC systems in Miami.*

ture control. The decreased difference in energy use was a function of the humidity dependence of the comfort index used (see Figure 1). PMV* has the strongest humidity dependence, followed by ET* and PMV.

Figure 8 compares the relative energy use of the heat-pipe and high-SHR ACs to the medium-SHR AC in Miami under each control mode. Under temperature control, adding heat pipes to the medium-SHR AC resulted in a 14.6% increase in annual energy use. Under comfort control, the increase in energy use due to adding heat pipes was smaller, only 5% to 11%, depending on the comfort index.

For the high-SHR AC, energy use was 3.1% lower than for the medium-SHR AC under temperature control. Comfort control reduced the difference in energy use between the high- and medium-SHR ACs; in fact, the trend reversed slightly for PMV* control. Accounting for the effect of comfort control tends to reduce the energy cost of maintaining lower humidity in the space and penalizes high-SHR systems, which allow the RH to increase.

Similar results were also found for Atlanta. Figure 9 compares the increased energy use due to adding heat pipes in both Miami and Atlanta when using temperature (T) and comfort (ET*) control. The increase in energy consumption was nearly identical for both climates under the two control modes; this indicates that the impact of comfort control on *relative* energy use is not strongly dependent on climate.

Comparing Daily and Hourly Comfort Control

As noted previously, it is likely that building occupants interact with conventional temperature thermostats such that they mimic comfort control. To simulate this scenario, the temperature setpoint was adjusted once a day, at a specified hour, to satisfy the comfort requirements for that hour. The temperature setpoint then remained fixed until the specified hour occurred the following day.

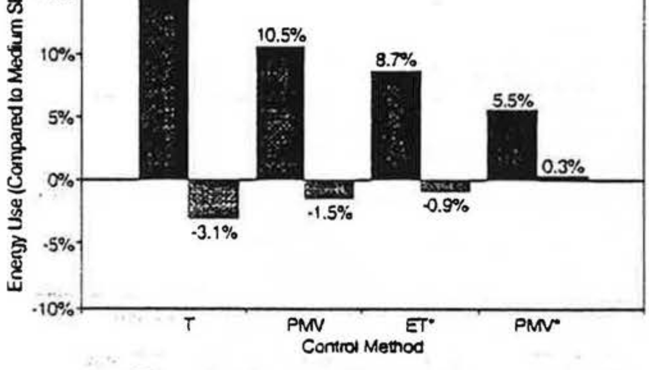


Figure 8 *High-SHR and heat-pipe AC compared to medium-SHR AC*

Figure 10 compares increased energy use due to heat pipes using daily and hourly comfort (ET*) control. The temperature setpoint is adjusted each day at 6:00 p.m. for daily comfort control. Daily comfort control resulted in lower energy use for both systems; this occurred because the setpoint was adjusted at 6:00 p.m., the hour of the day when the space RH was lowest. The lower RH allowed the temperature setpoint to be raised and maintained at the higher setting for the remainder of the day. If the adjustment period had been 6:00 a.m., the opposite trend would have been observed.

The relative changes in energy use between the medium-SHR and heat-pipe AC systems were similar; adding heat pipes to the AC unit increased energy use by 8.7% under hourly ET* control and 7.5% under daily ET* control. This indicates that daily comfort control, which attempts to mimic occupant behavior, is similar to hourly comfort control.

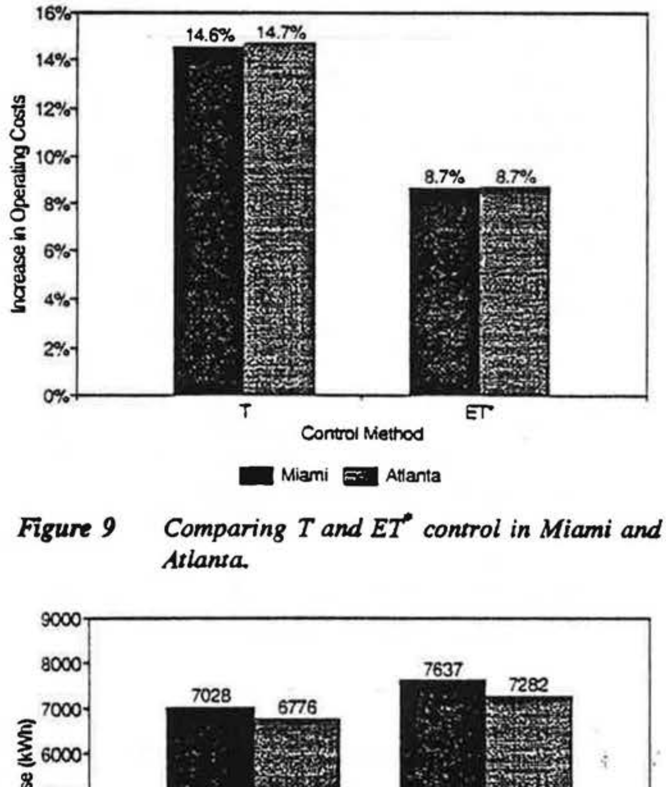


Figure 9 *Comparing T and ET* control in Miami and Atlanta.*

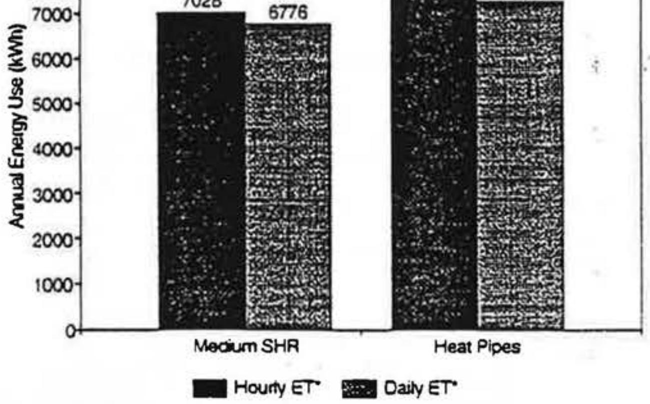


Figure 10 *Comparing daily and hourly ET* control.*

The General Impact of Constant Comfort Control

This analysis has examined the impact of comfort control on three specific types of AC systems. It is also of interest to determine how comfort control affects the energy use of other types of AC machines (e.g., gas-fired desiccant systems). Figure 11 shows how the total ideal building load[1] changes with RH while maintaining constant temperature or comfort. The relative load (normalized to 50% RH) is shown for constant temperature (78°F), constant ET* (78°F), and constant PMV* (−0.01). The temperature setpoints used for each case are listed in Table 3.

For constant temperature, decreasing the space RH from 50% to 40% RH increased the total load by 11%. For constant ET*, the total load increased by only 3% when the humidity was reduced to 40% RH. When constant PMV* was maintained, reducing the RH from 50% to 40% actually decreased the total load by 3%.

The results shown in Figure 11 are for an average infiltration rate of 0.54 air changes per hour (ach). Figure 12 demonstrates the impact higher infiltration rates have on the results in Figure 11. At higher values of ach, maintaining the RH at 40% has a greater impact on the load. At a constant temperature (78°F), the load increase jumps from 11% at 0.54 ach to 17% at 1.08 ach. Likewise, at a constant value of PMV*, the load decrease of 3% at 0.54 ach changes to a load increase of 5% at 1.08 ach.

When accounting for comfort control with PMV* (the comfort index with the strongest humidity dependence), maintaining 40% RH actually lowers the total load (compared to 50% RH) for the house simulated in this study. This implies that, at low to moderate infiltration rates, lowering indoor RH has little or no impact on total load if comfort is considered. Therefore, advanced AC systems that maintain lower humidity than conventional AC systems (and have the same efficiency) may actually cost the same or less to operate.

[1] *Ideal* refers to the load required to perfectly maintain the required temperature and humidity setpoints for each hour.

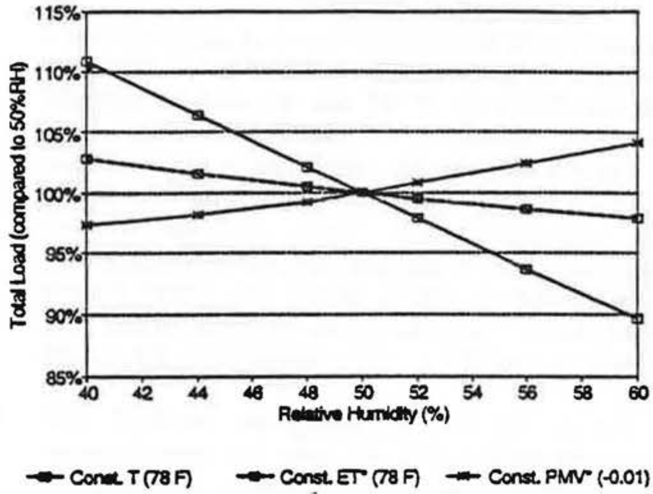


Figure 11 *Total load vs. RH (normalized to 50% RH, with 0.54 ACH).*

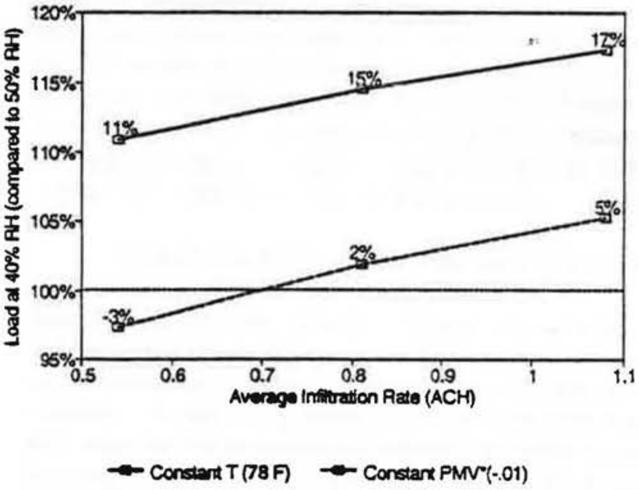


Figure 12 *Normalized load at 40% RH vs. infiltration rate.*

TABLE 3
Temperature Setpoints Used in Figure 11

Relative Humidity %	Temperature Setpoint °F (°C)		
	Constant T	Constant ET**	Constant PMV**
40	78.0 (25.6)	78.6 (25.9)	79.1 (26.2)
44	78.0 (25.6)	78.4 (25.8)	78.7 (25.9)
50	78.0 (25.6)	78.0 (25.6)	78.0 (25.6)
56	78.0 (25.6)	77.7 (25.4)	77.3 (25.2)
60	78.0 (25.6)	77.4 (25.2)	76.9 (24.9)
ET and PMV* calculated using (met=1.2, clo=0.5, v=0.2 m/s or 40 fpm)			

CONCLUSIONS

This paper has assessed the impact of controlling an AC system to maintain constant comfort instead of constant temperature. Comfort and temperature control have been compared by simulating a typical house in Miami and Atlanta. Three AC systems were analyzed: a high-SHR AC, a medium-SHR AC, and a heat-pipe-assisted AC.

Three indices of thermal comfort (ET^*, PMV, and PMV^*) were used to determine the impact of comfort control on the AC system energy use. While no commercially available comfort control "thermostat" presently exists, the purpose of this study was to determine the potential impact such a device would have, especially on AC systems with high or low SHR.

When under normal temperature control, the heat-pipe AC (with low SHR) used 14.6% more energy, while the high-SHR AC used 3.1% less than the conventional, medium-SHR AC. The low- and high-SHR systems used more and less energy because they maintained different space humidities (49.6% and 58.0% RH, respectively) and therefore met different loads. When humidity is not a concern, high-SHR AC systems always have the lowest operating costs. In other words, it is always cheaper not to meet the latent load.

When the impact of comfort control is considered, the energy cost of adding heat pipes is reduced threefold from 14.6% to 5.5% (using PMV^*). Conversely, the energy savings of the high-SHR system are reduced from 3.1% to slightly less than zero. Comfort control tends to decrease the energy cost of systems that maintain low humidity and penalize systems that maintain high humidity. In general, considering comfort tends to reduce the energy differences between high- and low-SHR systems.

In addition to hour-by-hour comfort control, an algorithm for daily comfort control was also analyzed. Daily comfort control was included to emulate how building occupants might interact with a conventional temperature thermostat. The temperature setpoint was adjusted each day at 6:00 p.m., as a homeowner might do, to achieve comfort for that hour. On an annual basis, daily comfort control had an impact on energy use similar to hourly comfort control.

A general analysis was also conducted to determine the impact comfort control has on the total building load. When maintaining constant temperature, reducing the space RH from 50% to 40% increased the load by 11%. When maintaining constant comfort (PMV^*), reducing the space RH actually *decreased* the load by 3%. This indicates that, at moderate infiltration rates (i.e., 0.5 ach), high-efficiency, low-SHR AC equipment may cost the same or less to operate as conventional systems while maintaining lower space RH.

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APPENDIX

AC Model

The AC model described here uses empirical functions for capacity and efficiency in conjunction with the apparatus dew point (ADP)/bypass factor (BF) relations to determine off-design performance. Sensible heat ratio (SHR) and efficiency are predicted as a function of evaporator inlet temperature and humidity.

The default performance functions for total capacity and efficiency from DOE-2 (DOE 1982) are used in this routine, though any set of empirical relations could be used. The default performance functions are of the following form:

$$C = C_{rated} \cdot F_C(iwb, odb, Q) \qquad \text{(A1)}$$

$$e = e_{rated} \cdot F_e(iwb, odb, Q) \qquad \text{(A2)}$$

where

iwb = indoor wet-bulb (°F or °C),
odb = outdoor dry-bulb (°F or °C),
Q = indoor airflow rate (ft^3/min or m^3/s).

C_{rated} and e_{rated} are the capacity and efficiency at the rated conditions (80°F [26.7°C] and 50% RH indoors; 95°F [35°C] outdoors; 450 cfm/ton [0.06 $m^3/s{\cdot}kW$]). The functions F_C and F_e adjust capacity and efficiency for off-design conditions (i.e., F_C and F_e equal one at rated conditions).

The ADP/BF method (Carrier et al. 1959) of determining the sensible and latent fractions of total capacity is used:

$$T_{exit} = \mathrm{BF} \cdot T_{inlet} + (1 - \mathrm{BF}) \cdot T_{\mathrm{ADP}} \qquad \text{(A3)}$$

$$w_{exit} = \mathrm{BF} \cdot w_{inlet} + (1 - \mathrm{BF}) \cdot w_{\mathrm{ADP}} \qquad \text{(A4)}$$

where

w = absolute humidity (lb/lb or kg/kg),
T = dry-bulb temperature (°F or °C),
$inlet$ = entering the evaporator,
$exit$ = leaving the evaporator,
T_{ADP}, w_{ADP} = average saturated conditions at the evaporator surface,

and where

$$\mathrm{BF} = e^{-\mathrm{NTU}}, \ \mathrm{NTU} = \frac{a_o}{Q}. \qquad \text{(A5)}$$

The constant a_o is determined from the rated conditions.

Procedure

The calculation procedure is described below.

The first time the routine is called,

(1) Specify the desired C_{rated}, e_{rated}, and SHR_{rated}.
(2) At the rated conditions, calculate $T_{exit,rated}$ and $w_{exit,rated}$.
(3) Iteratively determine BF_{rated}, $T_{\mathrm{ADP},rated}$, and $w_{\mathrm{ADP},rated}$ using Equations A3 and A4; then find a_o by rearranging Equation A5 to

$$a_o = -\log_e(\mathrm{BF}_{\mathrm{rated}}) \cdot Q_{rated}. \qquad \text{(A6)}$$

For subsequent calls,

(4) At the operating conditions, use Equations A1 and A2 to calculate C and e. From C and the airflow, Q, calculate the enthalpy change of the air to determine the exiting enthalpy, h_{exit} using Equation A7:

$$h_{exit} = h_{inlet} - \frac{C}{\rho\, Q}. \qquad \text{(A7)}$$

T_{exit} and w_{exit} must lie on this constant enthalpy line.

(5) Use the airflow rate, Q, and Equation A5 to calculate BF.
(6) Iterate on ADP, keeping T_{exit} and w_{exit} on the constant enthalpy line, h_{exit}. Calculate BF and adjust ADP until BF converges to the required value of BF (from step 5). Only one value of ADP will satisfy the h_{exit} and BF conditions.
(7) If the evaporator is dry ($w_{inlet} < w_{\mathrm{ADP}}$), then Equations A1 and A2 are not valid. Increase w_{inlet}' while maintaining T_{inlet} constant, until the dryout point is reached ($w_{\mathrm{ADP}} = w_{inlet}'$). Use the modified inlet humidity (w_{inlet}') to calculate capacity and power.

Features

This AC model has several features:

- Single values of capacity, efficiency, and SHR are used to develop the entire performance map.
- At wet-coil conditions, SHR and ADP are predicted for any evaporator inlet temperature and humidity.
- At dry-coil conditions, efficiency and capacity depend on DB only.