Thermal Comfort Controller for HVAC Systems With Mechanical Ventilators

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ABSTRACT

The operation and performance of a thermal comfort controller for HVAC systems with forced-air ventilators were analyzed with the aid of a dynamic computer program. The functions and features of GEMS (Generalized Engineering Modeling and Simulation), a dynamic modeling and simulation software tool, are briefly described. Using GEMS, the effects of different ventilation airflow rates and sensible and latent efficiencies on the thermal comfort environment within the conditioned space were analyzed for Miami, Florida. The simulations indicate that when using a conventional thermostat, any form of mechanical ventilation will increase the moisture content of the air within a conditioned space in a cooling-dominated hot and humid climate. Therefore, while providing "fresh" outdoor air, mechanical ventilators in HVAC systems with conventional controllers may be a contributing factor to indoor air quality problems associated with high levels of moisture within a conditioned space. With conventional thermostats, occupants typically resort to undesirable and inefficient interventions (e.g., manually lowering the space dry-bulb temperature set point to "sweat out" the water from the air). Thus, there is a need for a method to control both the dry-bulb temperature and the moisture content within the conditioned space. The Auctioneering Temperature and Humidity Controller (ATHC) fulfills this need. These new algorithms were exercised for an HVAC system with mechanical ventilation. Simulation results from typical-day analysis and results from annual cooling operation are presented for Miami, Florida. When compared to results for systems with a conventional dry-bulb temperature controller, these results, consisting of annual cooling energy requirements, predictive mean vote (PMV) in the conditioned space, and so on, validate the superior thermal comfort provided by the auctioneering controller.

INTRODUCTION

The advent of "tighter, less leaky" residential construction has resulted in poor indoor air quality (IAQ) within conditioned spaces. One major reason for deteriorating IAQ is the reduced level of infiltration/exfiltration for ventilating the space. Increasingly, forced-air mechanical ventilation systems, such as heat or energy recovery ventilators (HRVs or ERVs), are being used to improve the IAQ by introducing preconditioned "fresh" outdoor air into an occupied space. Although these systems aid in reducing irritants and pollutants such as odors, volatile organic compounds (VOCs), and the like, they impose additional sensible and latent loads on the building's primary heating, ventilation, and air-conditioning (HVAC) system. The objective of this study was to use dynamic simulation models to analyze the impact of residential mechanical ventilation system parameters and HVAC system controllers on the thermal comfort conditions within a conditioned space and on the annual heating and cooling energy requirements. The GEMS program was used to study the effect of ventilator airflow rates and sensible and moisture removal efficiencies (η_s and η_m, respectively) on the indoor air moisture levels and annual energy requirements for two U.S. cities—Miami, Florida (a cooling-dominated climate), and Minneapolis/St. Paul, Minnesota (a significant heating climate with substantial cooling loads). Also analyzed were the impact of two distinctly different system controllers—a

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conventional thermostat (dry-bulb temperature controller) and an auctioneering temperature and humidity controller. Due to space limitations, only the results for Miami are presented.

First, a brief description of GEMS and the simulation models is presented. Next, the auctioneering controller, and technical approach used in this study are discussed. Then, the simulation results for Miami are presented. Finally, the analysis effort is summarized and conclusions are drawn.

GENERALIZED ENGINEERING MODELING AND SIMULATION (GEMS)

GEMS is structured, generalized modeling and simulation software that enables development and analysis of control algorithms and systems and prediction of thermal performance and energy use in buildings (Benton et al. 1982). The basis for this tool is the state-space technique, which casts differential and algebraic equations describing the system into a vector-matrix form. In doing so, several powerful tools, such as linear algebra, modern control theory, and vector-matrix numerical methods, can be used to study the system. In modeling even a moderately detailed system, the state-space form presents an enormous bookkeeping and coupling problem. Moreover, each time the system is modified, the coupling among the equations necessitates a derivation of all the state-space vector-matrix equations. Herein lies the power of GEMS, which contains the tools to automatically generate the state-space representation of the entire system from user-input descriptions of the basic subsystem block dynamics and the interconnections of these blocks. Detailed and simplified building models can be generated directly from plans by using a library of construction element models (walls, floors, etc.). A closed-loop system simulation can be generated by interconnecting the building model with other component models selected from a library of HVAC equipment, controls, internal loads and schedules, and weather data.

Over the years, GEMS has been used extensively in the development and analysis of control algorithms for a large number of existing and future products. With the proliferation of inexpensive microprocessors, control algorithms developed with the aid of GEMS have been directly embedded into microprocessor-based controllers. Furthermore, GEMS has been used in emulators (closed-loop or real-time simulations) for testing and analyzing product prototypes.

SIMULATION MODELS

Layout of the simulated house, construction, and HVAC system was based on an actual two-story single-family house with full basement located in Eden Prairie, Minnesota, which has been used in the past for field testing prototype controllers. The house is divided into four thermal zones and has a single HVAC system consisting of a central forced-air heating (furnace) and cooling (air conditioner [a/c]) plant whose operation is controlled by a proportional-plus-integral thermostat located within one of the conditioned zones. Time-scheduled internal heat and moisture gains, such as from occupants, appliances, household activities (e.g., cooking, showers, laundry), are included in the model. The only paths for air to flow between the inside and the outside of the house are the furnace stack, the make-up duct, and cracks in the building's structure. Stack flow is calculated as a function of the stack temperature, and make-up air is considered to be a constant value whenever the central fan is on and zero when the fan is off. In the simulated system, the make-up flow equals the nominal stack flow of 84 cfm (39.65 L/s), and the central fan is on only when required. Infiltration was modeled as a function of both wind speed and indoor-to-outdoor temperature difference (ASHRAE 1997). The baseline infiltration rate, using blower door testing, was determined to be 0.281 ach in the heating mode. The average annual infiltration rate was estimated to be 0.33 ach. No mechanical ventilation is designated as the reference or baseline case.

A simple steady-state model of a mechanical ventilator with constant η_s and η_m and fixed airflow rates was used in this study. This single model was used for analyzing the following three different ventilation systems: (a) direct mechanical ventilation to introduce unconditioned outside air (η_s = 0% and η_m = 0%), (b) an HRV (η_s > 0% and η_m = 0%), and (c) an ERV (η_s > 0% and η_m > 0%). The ventilation systems were assumed to be on all the time, supplying outside ("fresh") air into the supply side of the central duct distribution system. Simulations were performed with ventilation airflow rates of 100 and 200 cfm (47.2 and 94.4 L/s), with corresponding fan power consumption of 82 W and 164 W.

THE AUCTIONEERING CONTROLLER

The Auctioneering Temperature and Humidity Controller (ATHC) shown in Figure 1 consists of a dry-bulb temperature sensor and a relative humidity sensor. The sensed dry-bulb temperature and the sensed relative humidity are used to determine the "sensed" dew-point temperature of the air in the conditioned space. Note that the dew-point temperature can be directly available if an appropriately reliable low-cost sensor were used in place of the relative humidity sensor. The user is required to specify two setpoints—the dry-bulb temperature setpoint and either the relative humidity setpoint or the dew-point temperature setpoint. If provisions are made for the user to input the relative humidity setpoint, then the dew-point temperature setpoint can be determined from the specified dry-bulb temperature and relative humidity setpoints. The goal is to obtain two error signals—a dry-bulb temperature error and a dew-point temperature error. These errors are then compared, and the numerically larger of the two is used in the classical PID control block, with anticipation and a hysteretic switch, to generate the ON/OFF control signal for the cooling plant. Note that it is not mandatory to use the maximum of the two errors as inputs to the PID block; rather, any desired functional relationship can be used to deliver an appropriate error to the PID block. One advantage of using the numerically larger value is that it is simple, and another is that any step

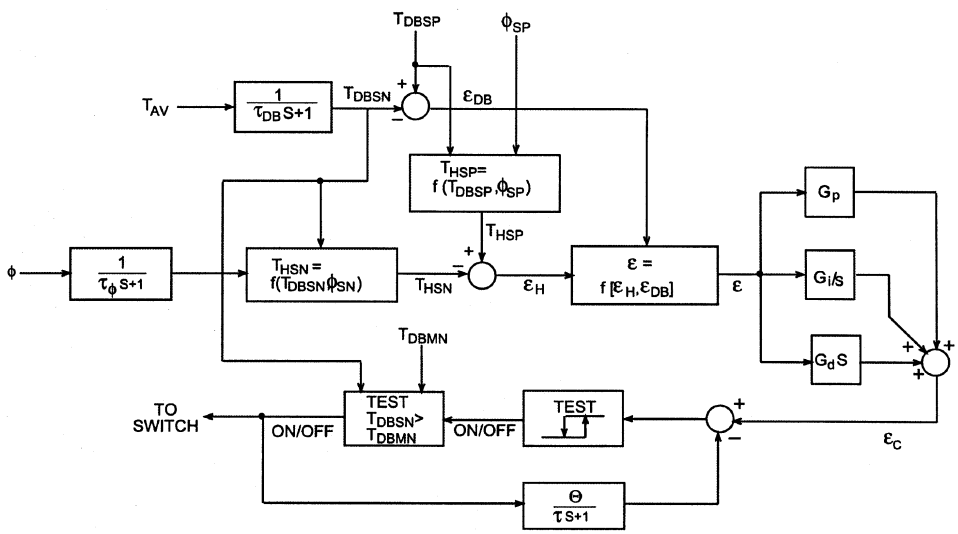

Figure 1 *Block diagram for the auctioneering temperature and humidity controller.*

change(s) in the error, for example, due to the step change(s) in the control setpoint(s), will be easily and quickly relayed to the PID block for a fast and robust response by the controller. Provisions have been made to ensure that the space dry-bulb temperature does not fall below a prespecified value during cooling plant operation. Additional details on the operation and performance of this controller compared to a conventional dry-bulb temperature controller (thermostat) are published elsewhere (Shah 2000a; Shah et al. 1994).

TECHNICAL APPROACH

Using the previously described GEMS model of a two-story single-family house with full basement and its associated HVAC components and system, "typical" day and seasonal (heating and cooling) simulations were conducted with "typical" year weather data for Minneapolis, Minnesota (significant heating climate with substantial cooling loads). A slab-on-grade version of this Minnesota house, reflecting Florida building construction practices (no basements), was also developed for simulations in a cooling-dominated hot and humid climate such as Miami, Florida, which was assumed to have a year-round cooling mode of operation.

Several simulations were conducted with different combinations of the following ventilator parameters for studying their impact on energy consumption and indoor conditions:

- Ventilator airflow rates of 100 and 200 cfm (47.2 and 94.4 L/s) with fan power consumption of 82 W and 164 W, respectively
- η_m = 0%, 30%, 50%, and 80%
- η_s = 0%, and 70%

Note that $\eta_s = \eta_m = 0\%$ reflects direct introduction of outside air into the conditioned space without any heat and/or moisture recovery for preconditioning this airstream. Also, $\eta_s > 0\%$ and $\eta_m = 0\%$ represents an HRV, and $\eta_s > 0\%$ and $\eta_m > 0\%$ implies an ERV.

Identical sets of simulations were performed using models for both a conventional thermostat (dry-bulb temperature controller) and the auctioneering temperature and humidity controller. A year-round cooling requirement with no heating was assumed for the Miami simulations, and the controller setpoints were assumed to be constant: 78°F (25.6°C) for the dry-bulb temperature (T_{dbsp})and 60°F (15.6°C) for the dew-point temperature (T_{dpsp}).

SIMULATION RESULTS

As previously mentioned, "typical" day and seasonal/annual simulations were performed with "typical" year weather data for Miami, Florida, and Minneapolis/St. Paul, Minnesota, with appropriate two-story single-family house and HVAC system simulation models representative of construction practices in the two geographic areas. Results for Miami, assumed to have a year-round cooling mode of operation (no heating), will be presented; however, data for Minneapolis/St. Paul, which has seasonal heating and seasonal cooling requirements, will not be presented due to space limitations.

"Typical" day simulations for Miami consisted of outdoor conditions that were (a) mild and humid (day 121), (b) hot and humid (day 169), and (c) cold and humid (day 304). Scenarios considered included a baseline case of no mechanical ventilation, systems with direct mechanical ventilation introducing unconditioned outside air into the conditioned space, and recovery ventilators with different sensible and moisture removal efficiencies and two airflow rates.

Dry-bulb temperatures of the outdoor air and within the conditioned space are shown in Figure 2 for day 169 for the baseline and different ventilation schemes using a conven-

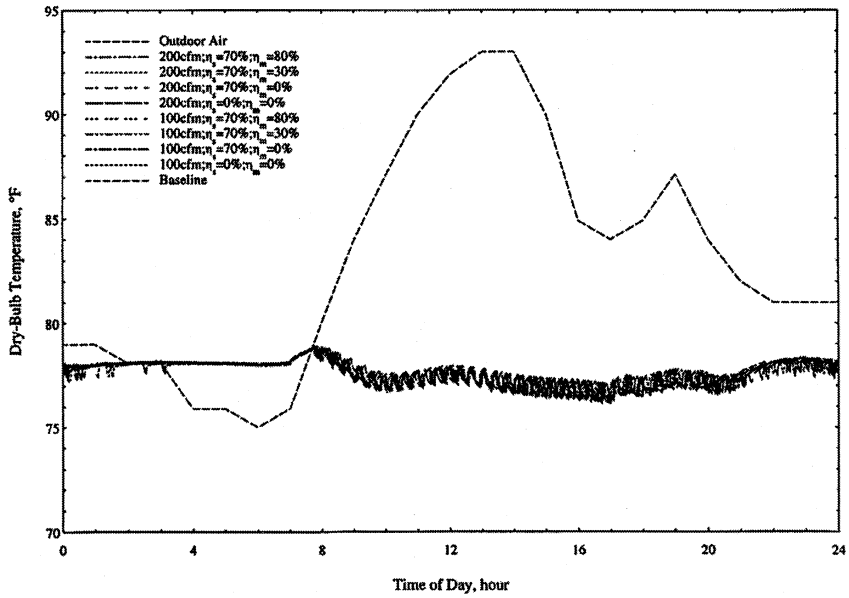

Figure 2 *Space dry-bulb temperatures on a hot and humid Miami day under conventional thermostat control (day 169;* T_{dbsp} *= 78ºF [25.6ºC]).*

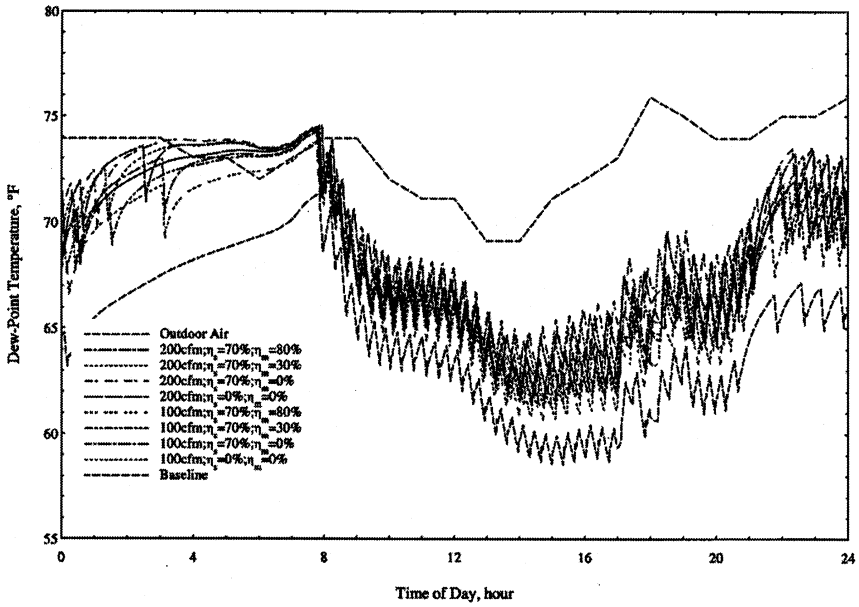

Figure 3 *Effect of ventilation schemes on space dew-point temperatures on a hot and humid Miami day under conventional thermostat control (day 169;* T_{dbsp} *= 78ºF [25.6ºC]).*

tional controller. Note that the dry-bulb temperature within the conditioned space is maintained close to the 78ºF (25.6°C) set point by the conventional thermostat, which is basically a dry-bulb temperature controller. The "sawtooth" variation in the space air temperature is due to the a/c cycling on and off. For this hot and humid day, note that between 8:30 p.m. and 11:00 p.m., the air dry-bulb temperature does not oscillate and increases slightly for the baseline case and some of the ventilation schemes, indicating that the a/c is locked on due to the cooling demand exceeding equipment capacity. This indicates that any form of mechanical ventilation, with or without preconditioning the outside air, will increase the sensible cooling load on the house/HVAC system. For some of the ventilation cases, sensible heat recovery appears to aid in cooling the outdoor air being brought into the space.

Dew-point temperatures of the outdoor air and within the conditioned space are shown in Figure 3 for day 169 for the baseline and different ventilation schemes using a conven-

tional controller. The bottommost curve in these figures is for the baseline case of no mechanical ventilation, and the topmost curve shows the outdoor dew-point temperature. It is obvious that any form of mechanical ventilation tends to raise the dew-point temperature within the conditioned space relative to the baseline. Again, the sawtooth variation in the temperature is due to the a/c cycling on and off—decreasing when the a/c is on and extracting moisture from the space air and increasing when the a/c is off due to internal gains and moisture brought in by the ventilation system. Ventilation airflow rates of 200 cfm (94.4 L/s) introduce more water and heat into the space than 100 cfm (47.2 L/s) and hence have higher space dew-point temperatures. Also, direct mechanical ventilation systems ($\eta_s = \eta_m = 0\%$) that introduce unconditioned outside air into the space cause the space dew-point temperature to be higher than systems with moisture recovery (i.e., ERVs, $\eta_m > 0\%$). Additionally, systems with higher η_m have lower indoor dew-point temperatures than those with low η_m. Figure 3 shows that between 8:30 p.m. and 10:00 p.m., when the a/c equipment does not have sufficient capacity to satisfy the sensible load and is locked on (Figure 2), the space dew-point temperature continues to increase, despite the removal of water by the a/c coil. This implies that the a/c does not have sufficient latent removal capacity either. The sharp rises in the dew-point temperatures between 5:00 p.m. and 7:00 p.m. are due to prescheduled internal moisture gain within the conditioned space. During the evening hours, the space dew-point temperatures tend to increase due to decreased a/c cycling because of reduced sensible loads.

Additional simulation results for "typical" day operations and annual energy requirements for Miami with conventional thermostat control are available (Shah 2000b).

Next, the module representing the conventional thermostat was replaced with the ATHC model and all simulations were repeated. Dry-bulb temperatures of the outdoor air and within the conditioned space are shown in Figure 4 for day 169 for the baseline and different ventilation schemes. The "sawtooth" variation in the space air temperature is due to the a/c cycling on and off. Note that for the conventional thermostat case ("DBTC"), the dry-bulb temperature within the conditioned space is maintained close to the 78ºF (25.6°C) setpoint during the day, and the a/c does not operate between midnight and 7:00 a.m. and after 10:00 p.m. due to no sensible load. When using the ATHC (T_{dpsp} = 60ºF (15.6°C)) without any mechanical ventilation ("ATHC Baseline"), the space dry-bulb temperature is maintained about 2°F to 3°F (1.1°C to 1.7°C) below the set point temperature for most of the day because the a/c is being operated under dew-point temperature control for a portion of that period. As previously mentioned, any form of mechanical ventilation tends to increase the moisture content within the conditioned space, which results in more frequent cycling of the a/c when under ATHC control. This then causes the controller to operate under dew-point control mode and the space being maintained at dry-bulb temperatures in the high 60s and low 70s °F (high 10s and low 20s °C), depending on the type of ventilation scheme being used. Generally, 100 cfm (47.2 L/s) of ventilation imposes less sensible and latent load than 200 cfm (94.4 L/s) ventilation, resulting in lower a/c run times and warmer space dry-bulb temperatures. In each flow category, HRV systems are the

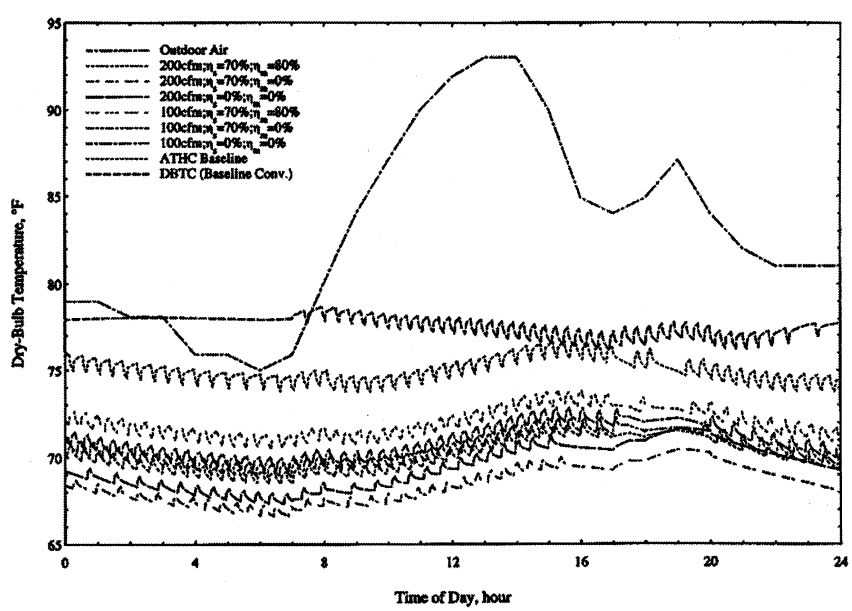

Figure 4 *Effect of ATHC and ventilation schemes on space dry-bulb temperatures on a hot and humid Miami day (day 169;* T_{dbsp} *= 78ºF [25.6ºC[,* T_{dpsp} *= 60ºF [15.6ºC[).*

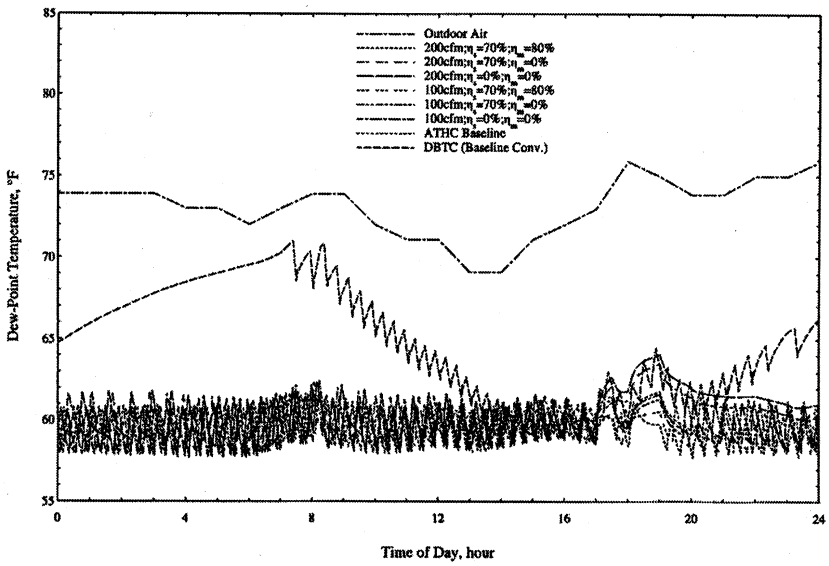

Figure 5 *Space dew-point temperatures on a hot and humid Miami day for various ventilation schemes under auctioneering control (day 169;* T_{dbsp} *= 78ºF [25.6ºC],* T_{dpsp} *= 60ºF [15.6ºC]).*

worst, followed by direct ventilation (i.e., without any form of recovery). For this hot and humid day, an ERV with the highest possible η_s and η_m is highly desirable. The figure also shows that after 3:00 p.m., the a/c operated by the ATHC gets locked on (dry-bulb temperature does not oscillate in sawtooth pattern) for various lengths of time depending on the type of ventilation scheme used because the space moisture content levels are sufficiently high to initiate dew-point temperature control mode. In some cases, the dry-bulb temperature increases slightly, indicating that the cooling demand has exceeded equipment capacity.

The impact of the auctioneering controller on the space dew-point temperatures for the various ventilation schemes is shown in Figure 5. The topmost curve represents the outdoor air, and the one immediately below that, ranging from about 58ºF to 71ºF (14.4°C to 21.7°C), is for the baseline case of conventional control without any ventilation ("DBTC"). All the other curves represent ATHC operation. Even without any mechanical ventilation introducing moisture into the conditioned enclosure, the space dew-point temperatures can reach uncomfortable levels when using a conventional thermostat. This is particularly true at night and in the morning when there is insufficient sensible load to operate the a/c to cool and dehumidify the air. For the most part, the auctioneering controller helps maintain the 60ºF (15.6°C) dew-point temperature setpoint, albeit at the expense of somewhat lower dry-bulb temperatures (Figure 4). An exception to this is after 5:00 p.m. Moisture is added to the space from prescheduled internal loads between 5:00 p.m. and 7:00 p.m., which, when combined with ventilation-induced latent load, exceeds the moisture removal capacity of the a/c and leads to higher than "desired" dew-point temperatures. After 8:00 p.m., all but two dew-point temperatures display the sawtooth pattern, indicating on/off operation of the a/c. However, for the 200 cfm (94.4 L/s) cases of direct mechanical ventilation (no recovery) and HRV, the space dew-point temperatures do not oscillate but exhibit a steady decline. This is because significant amounts of moisture got added into the conditioned space, causing the a/c to lock on under the dew-point temperature control mode due to insufficient latent removal capacity.

When using the ATHC, space dry-bulb temperatures below the desired 78ºF (25.6°C) dry-bulb temperature setpoint (Figure 4), resulting from trying to maintain the space dew-point temperature at (or below) the 60ºF (15.6°C) comfort value (Figure 5), can be very misleading at first glance. In fact, such low dry-bulb temperatures help reduce the mean radiant temperature within the conditioned space, which, in combination, leads to improved predictive mean vote (PMV) as discussed below. Furthermore, it should be realized that whenever occupants feel uncomfortable due to elevated space dew-point temperatures (generally when greater than 60ºF (15.6°C)), they will approach their conventional thermostat to manually reduce the space dry-bulb temperature below the normal setpoint in order to force the a/c to operate and remove moisture. Typically, the occupants will then forget to reset the dry-bulb setpoint to the normal value until after they start feeling uncomfortably cold, which could be much later than necessary. The auctioneering controller automates and eliminates the need for manual intervention and thus prevents energy waste by reverting back to dry-bulb temperature control as soon as practical.

"Typical" day simulations, such as those discussed above, provide invaluable insight into the operation and performance of control algorithms, as well as the impact and interactions

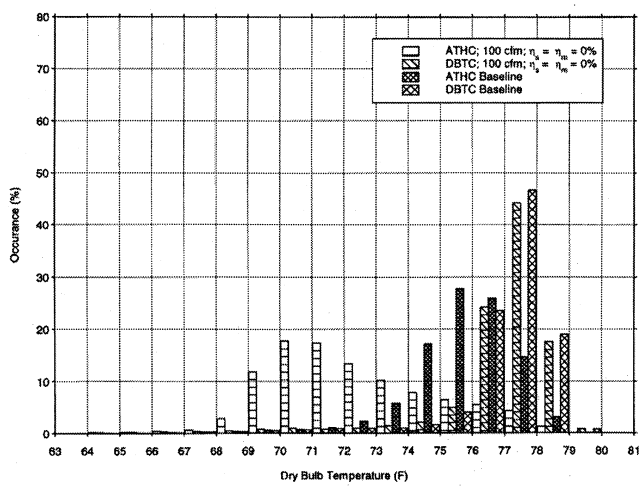

Figure 6 *Annual dry-bulb temperature distribution for 100 cfm (47.2 L/s) direct mechanical ventilation (no recovery) (T_{dbsp} = 78ºF [25.6ºC], T_{dpsp} = 60ºF [15.6ºC]).*

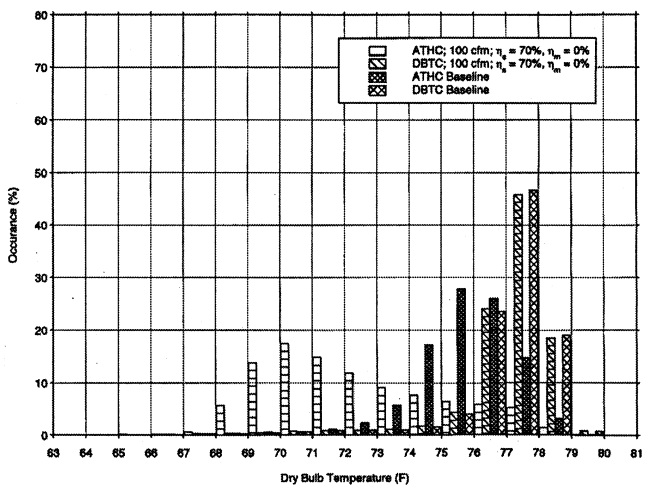

Figure 7 *Annual dry-bulb temperature distribution for 100 cfm (47.2 L/s) heat recovery ventilation (T_{dbsp} = 78ºF [25.6ºC], T_{dpsp} = 60ºF [15.6ºC]).*

between the various components and subsystems and any local and global (supervisory) controllers. This is made possible by the GEMS software tool, which can also be used to study the impact of various control options on annual energy consumption, thermal comfort, and so on.

Annual simulations were conducted assuming a year-round cooling season for Miami, Florida. The annual distribution of the space dry-bulb temperature for a ventilation rate of 100 cfm (47.2 L/s) is shown in Figures 6, 7, and 8, respectively, for direct mechanical ventilation (no recovery), HRV, and ERV. For comparison purposes, annual dry-bulb temperature distributions for the baseline cases of no mechanical ventilation are also included on all these figures for both

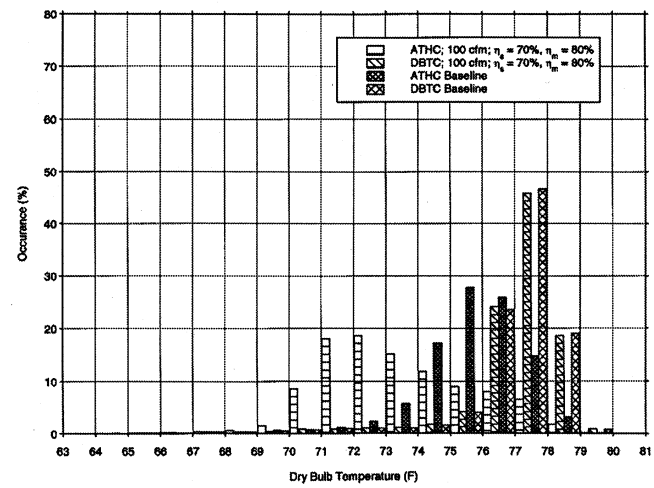

Figure 8 *Annual dry-bulb temperature distribution for 100 cfm (47.2 L/s) energy recovery ventilation (T_{dbsp} = 78ºF [25.6ºC], T_{dpsp} = 60ºF [15.6ºC]).*

conventional thermostat ("DBTC Baseline") and the auctioneering controller ("ATHC Baseline"). These figures show that any form of mechanical ventilation, when installed in an HVAC system with conventional thermostat, will not significantly impact the average dry-bulb temperature and its annual distribution within the space because there is not any substantial change in the space sensible load. Also, an ATHC-operated HVAC system without any mechanical ventilation will be maintained at a space dry-bulb temperature somewhat lower than the system with a conventional thermostat because the system gets operated under the dew-point temperature control mode for some period due to the addition of moisture into the occupied areas. The data indicate that any form of mechanical ventilation, when installed in an HVAC system with the auctioneering controller, further reduces the space dry-bulb temperatures due to increasingly frequent operations under dew-point temperature control. The ATHC drives the average dry-bulb temperature toward the low 70s °F, and the distribution appears to become a bit more "spread out." In and of themselves, these lower dry-bulb temperatures might be of concern; however, the space thermal comfort conditions, as indicated by the PMV, are markedly improved and, hence, such low temperatures may not be of significance because the occupants will feel more comfortable relative to systems with conventional thermostats.

The annual distribution of the PMV for a ventilation rate of 100 cfm (47.2 L/s) is shown in Figures 9, 10, and 11, respectively, for direct mechanical ventilation (no recovery), HRV, and ERV. For comparison purposes, annual PMV distributions for the baseline cases of no mechanical ventilation are also included on all these figures for both conventional thermostat ("DBTC Baseline") and the auctioneering controller ("ATHC Baseline"). The annual average PMVs for each

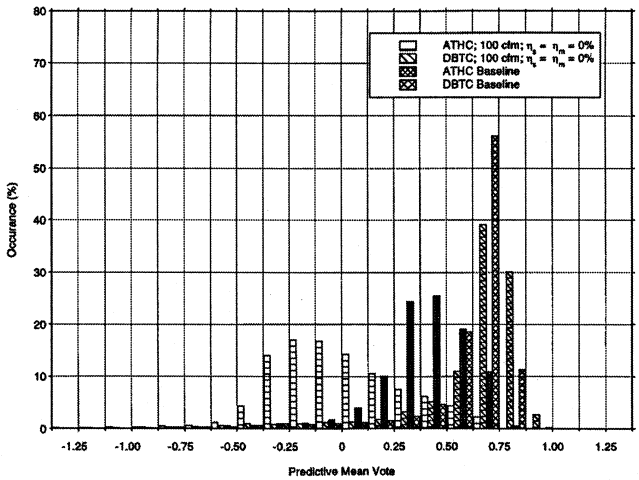

Figure 9 *Annual PMV distribution for 100 cfm (47.2 L/s) direct mechanical ventilation (no recovery) (T_{dbsp} = 78ºF [25.6ºC], T_{dpsp} = 60ºF [15.6ºC]).*

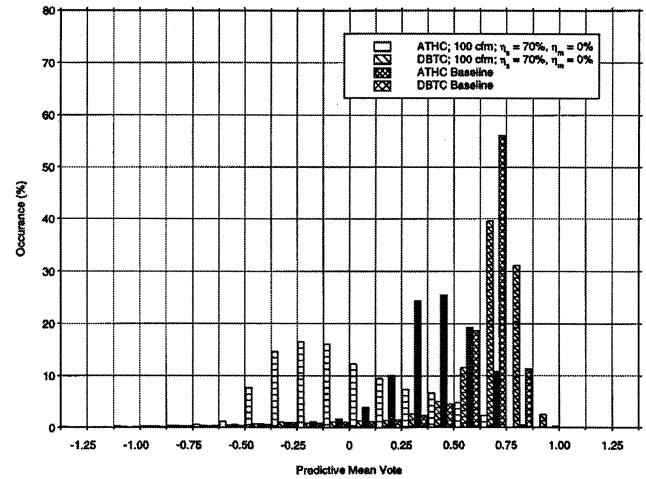

Figure 10 *Annual PMV distribution for 100 cfm (47.2 L/s) heat recovery ventilation (T_{dbsp} = 78ºF [25.6ºC], T_{dpsp} = 60ºF [15.6ºC]).*

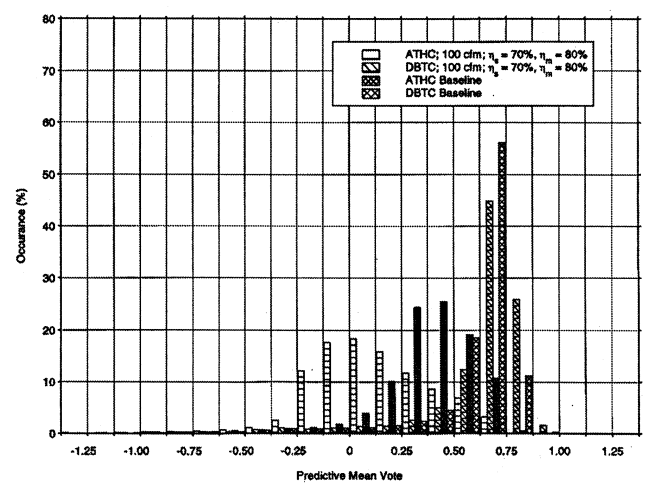

Figure 11 *Annual PMV distribution for 100 cfm (47.2 L/s) energy recovery ventilation (T_{dbsp} = 78ºF [25.6ºC], T_{dpsp} = 60ºF [15.6ºC]).*

ventilation strategy are listed in Table 1 for both control modes.

These figures and the table indicate that a system without any mechanical ventilation (baseline) will be maintained at a space PMV level of about 0.600 for approximately 85% of the time by a conventional thermostat and about 0.375 for approximately 90% of the year by the auctioneering controller. The data also show that any form of mechanical ventilation, when installed in an HVAC system with conventional thermostat, will not significantly impact the average PMV within the space and will only slightly affect the overall distribution of the annual PMV. For example, Figure 9 shows that direct mechanical ventilation with a conventional thermostat will reduce the occurrence of 0.75 PMV from about 55% to roughly 40% and decrease the occurrence of 0.625 PMV from about 20% to approximately 10%; however, the occurrence of 0.825 PMV increases by about 20% (from 10% to 30%). All other areas of PMV distribution with the conventional thermostat are unaffected. Similar trends are observed in Figures 10 and 11, respectively, for HRVs and ERVs in HVAC systems with a conventional thermostat.

The simulation results presented in Figures 9, 10, and 11 and Table 1 indicate that an ATHC-operated HVAC system without any mechanical ventilation will be maintained at a space PMV level of about 0.375 for approximately 90% of the year. This in itself shows a marked improvement over the 0.600 PMV level possible with a conventional thermostat. The data also show that any form of mechanical ventilation, when installed in an HVAC system with the auctioneering controller, further significantly impacts the average PMV within the space and also beneficially affects the overall distribution of the annual PMV. The ATHC drives the average PMV toward the neutral level of zero (0), and the distribution becomes more "spread out," appearing slightly biased toward values just above the neutral point.

Table 1 also shows that the overall ventilation rate increases from 0.235 ach for the baseline case of no mechanical ventilation to 0.5 ach and 0.76 ach, respectively, for ventilation rates of 100 and 200 cfm (47.2 and 94.4 L/s). Also, any form of mechanical ventilation helps reduce the CO_2 levels within the conditioned space.

The overall trends and distributions for the 200 cfm (94.4 L/s) ventilation airflow rates in HVAC systems with both conventional thermostat and ATHC are similar to those shown in Figures 6 through 11 for the 100 cfm (47.2 L/s) ventilation rates, and, hence, they are not presented here.

TABLE 1
Annually Averaged Conditions for the Miami, Florida, House
(Setpoints: 78°F [25.6°C]Dry-Bulb Temperature; 60ºF [15.6ºC] Dew-point Temperature)

System Configuration	PMV		Ventilation Rate (ach)	Carbon Dioxide Concentration (ppm)	
	DBTC	ATHC		Outdoor	Indoor
Baseline Conventional	0.5991	n/a	0.235	400	663
Baseline ATHC	n/a	0.3757	0.236	400	651
100 cfm $\eta_s = \eta_m = 0\%$	0.6109	-0.0126	0.500	400	506
100 cfm $\eta_s = 70\%$ $\eta_m = 0\%$	0.6293	-0.0189	0.501	400	506
100 cfm $\eta_s = 70\%$ $\eta_m = 30\%$	0.6264	0.0281	0.499	400	506
100 cfm $\eta_s = 70\%$ $\eta_m = 80\%$	0.6202	0.1157	0.497	400	507
200cfm $\eta_s = \eta_m = 0\%$	0.6008	-0.1976	0.760	400	468
200 cfm $\eta_s = 70\%$ $\eta_m = 0\%$	0.6375	-0.2287	0.762	400	468
200 cfm $\eta_s = 70\%$ $\eta_m = 30\%$	0.6344	-0.1732	0.760	400	468
200 cfm $\eta_s = 70\%$ $\eta_m = 80\%$	0.6278	-0.0555	0.755	400	468

TABLE 2
Annual Energy Operating Cost for Miami, Florida
(Setpoints: 78°F [25.6°C] Dry-Bulb Temperature; 60°F [15.6°C] Dew-Point Temperature)

System Configuration	Increased Operating Cost ($/year @$0.08/kwh)	
	DBTC	ATHC
Baseline Conventional	-	n/a
Baseline ATHC	84	-
100 cfm $\eta_s = \eta_m = 0\%$	128	438
100 cfm $\eta_s = 70\%$ $\eta_m = 0\%$	130	426
100 cfm $\eta_s = 70\%$ $\eta_m = 30\%$	123	377
100 cfm $\eta_s = 70\%$ $\eta_m = 80\%$	111	292
200cfm $\eta_s = \eta_m = 0\%$	237	803
200 cfm $\eta_s = 70\%$ $\eta_m = 0\%$	241	777
200 cfm $\eta_s = 70\%$ $\eta_m = 30\%$	231	701
200 cfm $\eta_s = 70\%$ $\eta_m = 80\%$	212	558

Increased annual energy cost for operating the HVAC system for the baseline ATHC and various ventilation strategies with both conventional thermostat and the auctioneering controller are listed in Table 2 for the Miami house. These costs are relative to the baseline systems (i.e., without mechanical ventilation). As expected, any mechanical ventilation increases the energy consumption (cost) for operating the HVAC system. The "DBTC" column shows that the auctioneering controller baseline system (without mechanical ventilation) will cost $84/year more to operate than the conventional thermostat system. It is interesting to note that when using a conventional thermostat, the ventilation-induced increased annual operating cost is more a function of the airflow rate and not a function of ventilation scheme (direct, HRV, or ERV). This indicates that the space most likely benefits from some "free cooling" when using direct ventilation and that the system does not benefit from the use of recovery ventilators with η_s = 70%. It appears that the recovery ventilators merely shift the sensible load to a different time (e.g., warming the relatively colder incoming outdoor air that could have provided "free cooling" during some periods and cooling the relatively warmer incoming outdoor air during other periods). For HVAC systems with a conventional thermostat, this offsetting of the sensible load appears to negate the need for relatively expensive recovery ventilators in favor of direct ventilation. Recall that Figures 9, 10, and 11 and Table 1 also did not reveal any significant thermal comfort benefits that could be attributed to recovery ventilation systems, and any improvement in indoor air quality could easily be provided by a direct ventilator. The annual energy cost with an ERV (η_s = 70% and η_m > 0%) is somewhat less than that for both direct ventilation and HRV and decreases with increasing η_m because moisture removal by the relatively drier exhaust air provides some cooling of the incoming outside airstream. It should also be noted that any recovery ventilation system (HRV or ERV) will have an additional first cost and annual maintenance costs associated with it. The direct mechanical ventilation system will be the least expensive to install with negligible maintenance requirements and will provide the same benefits as a recovery ventilation system.

The "ATHC" column shows that when using the auctioneering controller, the increased annual operating cost is a function of both the ventilator airflow rate and the ventilation scheme. Direct mechanical ventilation systems without recovery ($\eta_s = \eta_m$ = 0%) have lower first costs but are more expensive to operate because they introduce unconditioned

outside air into the space. The annual energy cost with HRVs (η_s = 70% and η_m = 0%) is somewhat less than that for direct mechanical ventilation because the outdoor air is sensibly cooled by the exhaust airstream. Both direct ventilation and HRVs introduce moisture into the space, causing the auctioneering controller to enter the dew-point control mode for operating the a/c and thus increasing the annual energy cost. The simulations indicate that the annual energy cost with ERVs (η_s = 70% and η_m > 0%) is substantially less than that for HRVs and direct ventilation because the relatively drier exhaust air removes moisture from the incoming outside air. This reduces the amount of moisture entering the conditioned space, resulting in the HVAC system spending less time under dew-point control mode. Also, the moisture removal process from the exhaust air further cools the incoming outside airstream. Obviously, higher ventilator airflow rates lead to higher energy consumption and cost.

As listed in Table 2, an ATHC-operated HVAC system without any mechanical ventilation will cost about $84/year more to operate than a baseline system with a conventional thermostat. This increase in annual operating cost is insignificant when compared to the aggregated costs for installing and operating mechanical ventilators (first cost, maintenance costs, and operating costs) on baseline conventional HVAC systems. Furthermore, the baseline ATHC system provides marked improvement in the space thermal comfort (average PMV = 0.375) compared to the conventional thermostat-controlled HVAC systems with or without mechanical ventilation (average PMV ~ 0.600). Additionally, all manual interactions by the occupant raising and/or lowering the dry-bulb temperature setpoint will be eliminated or significantly reduced when using the auctioneering controller.

SUMMARY AND CONCLUSIONS

Assuming year-round cooling operation for Miami, Florida, "typical" day and annual simulations were conducted for a single-family, two-story, slab-on-grade house with different mechanical ventilation parameters such as airflow rates and sensible and moisture removal efficiencies (η_s and η_m). No mechanical ventilation is designated as the reference or baseline case. Ventilation strategies analyzed included (a) direct introduction of unconditioned outside air into the house (η_s = η_m = 0%), (b) heat recovery ventilator (HRV, η_s = 70%, η_m = 0%), and (c) energy recovery ventilator (ERV, η_s = 70%, η_m = 30%, 50%, and 80%). Two distinctly different control strategies for operating the HVAC system were analyzed—a conventional thermostat (dry-bulb temperature controller) and the auctioneering temperature and humidity controller.

"Typical" day simulations for Miami consisted of outdoor conditions that were (a) mild and humid (day 121), (b) hot and humid (day 169), and (c) cold and humid (day 304). These results indicate the following.

1. Relative to the baseline case of no mechanical ventilation, any form of ventilation (i.e., with or without recovery) will increase the moisture content, and hence the dew-point temperature, within the conditioned space

   - when using a conventional thermostat
     - for a given η_m, indoor moisture content increases with ventilation airflow rate;
     - for a given ventilation airflow rate, indoor moisture content decreases with increasing η_m;
   - the auctioneering controller maintains the space dew-point temperature at or below the specified set point.

2. With a conventional thermostat (dry-bulb temperature controller) operating the HVAC system, the space dry-bulb temperature is maintained near the specified set point.
3. The ATHC causes the space dry-bulb temperature to often fall below the setpoint because the cooling plant is operated under dew-point control mode due to excessive ventilation-induced moisture within the conditioned space.

The annual simulations for Miami show the following.

1. With any form of mechanical ventilation,

   - whole-house ventilation rate increases with "fresh" (outside) airflow rate;
   - carbon dioxide concentration levels within the occupied conditioned space decrease with increasing ventilation airflow rates;
   - when using a conventional thermostat, moisture content levels (or dew-point temperatures) within the conditioned space are higher than for the baseline case of no mechanical ventilation.

2. Total power consumption and energy cost are higher than the baseline cases and increase with ventilation airflow rate. Ventilation with the auctioneering controller results in significantly higher energy costs than with the conventional thermostat because the a/c plant is quite frequently operated under the dew-point control mode. The ATHC-induced energy costs decrease with increasing η_m; however, the energy costs associated with a conventional thermostat are not a significant function of the ventilation method.
3. Thermal comfort (PMV) within the conditioned space is not significantly affected by the ventilation airflow rates, η_s and η_m; however, the ATHC provides substantial improvement over the conventional thermostat because it maintains lower dry-bulb and dew-point temperatures within the conditioned space. When combined with any ventilation method, the HVAC system operated by the auctioneering controller will provide almost neutral thermal comfort conditions (PMV ~ 0) within the occupied space.
4. In a warm and humid climate, such as that of Miami, Florida, it is more advantageous to use an ERV than an HRV because the increase in indoor moisture content will be less. This is particularly advantageous when using the ATHC to operate the HVAC system.

In today's "tighter, less leaky" residential construction, the primary purpose of any form of mechanical ventilation is to improve the indoor air quality. Consequently, simulations have shown that for HVAC systems operated by a conventional thermostat, the indoor moisture content levels are higher than those for the baseline case of no mechanical ventilation. Elevated levels of indoor moisture content can result in fungal and bacterial growth, leading to more adverse health problems and damage to property. One potential solution is to lower the thermostat (dry-bulb temperature) setpoint; however, this results in increased energy cost, a "chilly" space, and low temperatures being maintained, even when not necessary. Alternatively, the robust auctioneering temperature and humidity controller can be used for maintaining desired levels of dry-bulb and dew-point temperatures within the conditioned space without installing any additional equipment, such as recovery ventilators (HRV, ERV), desiccant systems, or heat pipe units. Such controllers are beneficially attractive in that there is no capital and/or annual maintenance cost for additional equipment. These simulations have shown that for a small increase in the annual energy cost, the ATHC provides marked improvement in the PMV over the conventional thermostat. Another potential solution might be to use an ERV with the lowest possible airflow rate and the highest possible η_s and η_m. It has been shown that HVAC systems with a ventilator and operated by the auctioneering controller can provide almost neutral comfort conditions within the occupied space. The optimal ventilation airflow rate will be that needed to satisfy code requirements and will have to be determined on a case-by-case basis.

Given the current state of the art wherein HVAC systems are operated by conventional thermostats (dry-bulb temperature controllers), a recommended solution is to use a direct mechanical ventilator (i.e., without recovery). Alternatively, an energy recovery ventilator with the lowest possible ventilation airflow rate and the highest possible sensible and moisture removal efficiencies can be used in these systems with conventional controllers.

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