SIMULATION OF DEHUMIDIFICATION CHARACTERISTICS OF HIGH EFFICIENCY RESIDENTIAL CENTRAL AIR-CONDITIONERS IN HOT AND HUMID CLIMATES

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ABSTRACT

This study assesses the dehumidifying performance of the high efficiency residential central air-conditioners (CAC) in hot/humid climates typified by that of Houston and Galveston. The performance study is based on such factors as: (i) weather (ii) thermostat set point and dead band, and (iii) sizing of unit relative to the design load of the residence. The units are evaluated on their ability to maintain conditions in the ASHRAE comfort zone in a typical residence in Houston area. The units, the thermostat, and the residence are simulated on a minute-by-minute basis using a commercial software (TRNSYS) after making certain modifications to it.

INTRODUCTION

The primary purpose of air conditioning equipment is to provide a comfortable environment for the occupants in a conditioned space. What is perceived as a comfortable environment by an individual will depend on such variables as the air temperature, environmental radiation, humidity, air movement, clothing, and activity level [ASHRAE 1981]. A well-designed air-conditioning and ventilation system can directly affect the temperature, humidity, and air movement in a conditioned space. The other variables either depend on the occupant (clothing and activity level) or other factors (environmental radiation).

The conventional residential central air conditioner (CAC) is designed to both cool and dehumidify. However, most of these units are controlled by a single thermostat, which only senses the indoor dry-bulb temperature. Because the thermostat does not sense humidity, the air conditioner does not directly respond to changes in humidity within the conditioned space. Instead, air-conditioning systems have traditionally been designed with large latent capacities (30% to 40% of total at 95 F design conditions) to adequately dehumidify while satisfying the sensible cooling load.

In recent years, manufacturers have been under both competitive and regulatory pressures to increase the efficiency of residential CACs. One of the methods that has been used to improve CAC efficiency has been through the use of increased heat exchanger surface areas. Such a strategy allows the CACs to run at higher refrigerant temperatures in the evaporator and lower refrigerant temperatures in the condenser. With a higher evaporator temperature, the latent capacity of a unit would drop. This could potentially lead to a situation where a CAC would provide the desired temperature control but not the desired humidity control.

Another method of improving the efficiency is by varying the speed of the compressor. Several manufacturers offer two-speed residential CAC units with Seasonal Energy Efficiency Ratios (SEERs) of above 10.

This study assesses the dehumidifying performance of the high efficiency single and two-speed residential CACs in hot/humid climates typified by that of Houston. The major criteria for determining dehumidification performance is the ability of a unit to maintain a "typical" residence within the comfort zone defined by ASHRAE Standard 55-1981. The performance is based on such factors as (1) weather, (2) thermostat set point and dead band, and (3) sizing of the unit relative to the design load on the residence. The units, the thermostat, and the residence were simulated on a minute-by-minute basis using TRaNsient SYstem Simulation model (TRNSYS) [Klein et al. 1983].

METHODOLOGY

This study attempts to quantify the dehumidification performance of residential CACs by simulating the actual performance in a residence. Before this analysis could be performed, a methodology had to be developed to provide a systematic approach to defining how the performance would be measured.

Many equipment manufacturers publish steady-state dehumidification data on CACs in their engineering data sheets (Table 1). Typically, the measure of performance is the Sensible Heat Ratio (SHR), which is provided as a function of outdoor temperature and indoor dry-bulb and wet-bulb temperatures. However, the SHR, by itself does not provide any indication of whether an individual unit will maintain comfort conditions within a given residence. Because comfort is the primary aim of an air-conditioning system, using some measure of comfort would be more informative than the SHR data provided by the manufacturer.

The basic methodology included:

1. Defining the comfort criteria
2. Developing a model for the dynamic performance of an air-conditioning/control/residential system
3. Defining a "typical" residence

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4. Obtaining detailed performance information on a selected number of CACs
5. Defining a "base" case, and
6. Testing the sensitivity of the results of the "base" case to several key variables

DEFINITION OF COMFORT ZONE

Significant research has already been done on defining what combination of temperature, humidity, airflow, etc., are needed to provide a comfortable environment. ASHRAE has defined an acceptable comfort zone for both summer and winter conditions (Figure 1) [ASHRAE 1981]. The comfort zone specifies conditions in which 80% or more of the occupants will find the environment thermally acceptable. Humidity is described in terms of the dew-point temperature. The acceptable ranges of operative temperature and humidity for persons clothed in typical summer clothing, at light, mainly sedentary, activity are: 72.7 – 78.8 F (22.6 – 26.1°C) at 62 F (16.7 °C) dew point and 74 – 80.9 F (23.3 – 27.2 °C) at 35 F (1.7°C) dew point (Table 2).

The comfort zone is used in this study as the basis for estimating whether an air conditioner is providing acceptable comfort conditions or not. The length of time that an air conditioner maintains conditions within a comfort zone is estimated and then compared to other air conditioners. A modification had to be made in the comfort zone for this study. The comfort zone is basically a "steady–state" zone, and if conditions are maintained within the comfort zone for a long enough period of time, the occupants should be comfortable. However, conditions in a room may deviate slightly (for a short period of time) outside the comfort zone before the occupants would sense any discomfort. While some recent studies [Ohno et al. 1987] have begun to address the length of this period of time, its value is still uncertain.

Thermal comfort greatly depends upon the occupants' clothing and activity level. In the present study, it is assumed that the occupants will be at rest or slightly active and wearing typical indoor clothing ($\leq$ 1.2 met).

SIMULATION MODEL

Two options were considered for estimating the dehumidification performance of a CAC: (1) hour–by–hour and (2) minute–by–minute simulation. While the hour–by–hour is a more common simulation methodology, it has several severe restrictions for this study. First, many of the ON/OFF cycles for CACs occur over a few minutes. An hour–by–hour program could not adequately capture the instantaneous dehumidification during most ON/OFF cycles and average hourly values would have to be used. Second, an hour–by–hour program would not allow us to look at the effect of changes of the thermostat. Thermostat interactions with the CAC occur on the order of minutes. An hour–by–hour program would not allow us to adequately model that interaction.

A decision, therefore, was made to use a minute–by–minute modeling methodology. Because the TRNSYS model was readily available and was capable of simulating a residence and HVAC equipment on a minute–by–minute basis, it was selected for this study. Previous work with TRNSYS by Hackner [1984 1985] indicates that the program could simulate the process control dynamics of a building and HVAC systems.

TRNSYS is modular in structure and an individual user can develop new modules and add them to the standard TRNSYS library. This feature enables the users to create modules for different types of equipment and control functions. In addition to the standard TRNSYS library, two more subroutines were developed for this study to (1) control the HVAC equipment and (2) define the comfort zone and analyze the minute-by-minute output from TRNSYS.

The first function of the HVAC control module was to prevent short–cycling of the compressor. This was accomplished by specifying a fixed time delay between the compressor ON and OFF periods. The qualitative comparisons between two CAC units can only be made if their rated capacities are equal. Therefore, a scaling factor was introduced to match the rated capacity of the CAC unit to the design cooling load of the test residence. The scaling factor scales both the capacity and the compressor power. Due to wide variations of thermostat dead bands, a variable dead band parameter was introduced. The dead band forces the equipment ON until the room's dry–bulb temperature reaches a value equal to the set point minus half the specified dead band. Similarly, the equipment remains OFF until the room's dry–bulb temperature reaches a temperature equal to the set point plus half the specified dead band. The module makes use of manufacturers' data to simulate the cooling equipment. These data relate input power and sensible and latent heat removal rates to the dry–bulb and wet–bulb temperatures of the air entering the coil and the outdoor temperature. A typical example of the manufacturers' data used in this simulation is shown in Table 1.

A program was written to analyze the minute-by-minute output from the TRNSYS. First, the time spent outside the comfort zone in each hour was evaluated by checking the temperature and the specific humidity. The condition would be outside the comfort zone if the humidity ratio was greater than 0.0012 lb/lba for indoor dry–bulb temperatures between 70 F and 80 F from 6 a.m. to 11 p.m. The limit on the humidity ratio was increased to 0.00135 lb/lba from 12 a.m. to 5 a.m. Second, the equipment ON time and the number of ON/OFF cycles in each hour were evaluated. Finally, the integrated hourly values for indoor relative humidity, supply dry–bulb temperature, relative humidity, capacity, SHR, power consumed, and energy efficiency ratio (EER) were evaluated.

DESCRIPTION OF THE HOUSE AND ITS LOADS

Residential buildings vary widely in the United States. A typical house was defined for estimating the loads and simulating the system's performance.

Test House

The house used for the simulation is a single-story residence. The schematic of the building is shown in Figure 2. The building has a conditioned floor area of 1672 ft^2. The total exterior surface is 1280 ft^2 of which 265 ft^2 is glazed surface (glass). The percentage of glass area to the floor area is 15%. A brief description of the test house is given in Table 3.

Thermal Loads

The external loads (weather conditions) used in this study represents a TMY (Typical Meteorological Year) based on the period from 1953 to 1975 for Houston, Texas. The ambient conditions include continuous hourly weather data (dry-bulb temperature, humidity ratio, wind speed, radiation, etc.), which were interpolated between hours to get weather data at each minute.

The internal loads consist of sensible heat gains from lights and equipment, sensible and latent heat gains from the people, and latent heat gains from cooking, showers, dishwashing, etc. The operating schedules and their profiles are required to estimate the thermal loads and the system performance. The operating schedules include (1) occupancy, (2) lighting, (3) equipment, and (4) internal latent load (other than people).

The number of occupants of the house is assumed to be three. The occupancy schedule of the people is shown in Figure 3. It is assumed that all the occupants stay home between 6 p.m. and 8 a.m. The instantaneous latent heat gains (other than people) include cooking, washing dishes, and showers. Cooking, dishwashing, and showers contribute 4.9 lbs, 3.6 lbs, and 2 lbs of moisture, respectively, into the conditioned space [Olivieri 1980]. Most of this contribution occurs from 6 to 8 a.m. and 6 to 8 p.m. because of the moisture gain from the showers and cooking. The schedule of the latent loads is shown in Figure 3.

The maximum amount of lighting used in the house is 660 watts (2250 Btu/h). The schedule of the lighting is shown in Figure 4. Since the occupants are asleep between 12 and 5 a.m., only 132 watts (450 Btu/h) of the total lights are assumed to be ON during that period. Between 5 and 6 a.m., when the occupants wake up, 410 watts (1400 Btu/h) of the lights are assumed to be ON. Between 7 a.m. and 5 p.m., when the house is not fully occupied, 191 watts (650 Btu/h) of the lights are assumed to be ON. Since all the occupants are assumed to be back by 6 p.m., 660 watts (2250 Btu/h) of the lights are assumed to be ON between 6 and 11 p.m.

The peak equipment wattage is assumed to be 1400 watts (4775 Btu/h). The schedule of the equipment is shown in Figure 4. Some equipment, such as refrigerators, run 24 hours; therefore, 418 watts (1400 Btu/h) of the equipment is assumed to be ON even when the occupants are asleep between 12 and 5 a.m. The peaks (4775 Btu/h) occur at 6 a.m., 12 noon and between 6 and 9 p.m., during the breakfast, lunch and dinner times, respectively. Between 8 a.m. and 5 p.m., when the house is not fully occupied, 704 watts (2400 Btu/h) of the equipment is assumed to be ON.

SELECTION OF THE TEST DAYS

The main objective of this study is to evaluate the dehumidifying performance of the CAC units as a function of SHR. Although steady-state dehumidifying performance is well defined in terms of the sensible heat ratio, there are acceptable criteria for either daily or seasonal performances of the residential CACs. In this study, dehumidification performance will be based on "typical" summer days.

A set of criteria was developed to select test days from the weather files. Three days, (i) a hot day, (ii) a hot/humid day and (iii) a mild/humid day. The selection criterion for the hot day was to pick the day with the maximum dry-bulb temperature (at or above design conditions), from the entire weather file, to typify the peak load conditions.

The selection criteria for the mild/humid day were (1) to select all the days with maximum dry-bulb temperatures in the range of 80 F to 86 F and (2) then select a day (among them) with the least difference between the maximum and the minimum dry-bulb temperatures. When the difference between the maximum and the minimum dry-bulb temperatures is small, it is a good indication of a humid day. The day was to represent a high humidity when the latent loads would be high even though the dry-bulb temperature is comparatively mild. The maximum dry-bulb temperature, solar radiation and wind speed, average dry-bulb temperature, and relative humidity for all the test days are shown in Table 4.

BASE CASE

Estimating the design cooling load (maximum cooling load) for the test house is essential in the selection of the CAC units. Before the design load was estimated, certain variables that influence the dehumidifying performance of the CACs had to be identified and base values for some of them had to be assigned.

Due to wide variations in the values of these variables, a base case was established. Five variables were identified as the most important for estimating dehumidification performance: room temperature set point, dead band on the thermostat, external load (weather), infiltration, and sizing of the CAC unit. The base values for the five variables are: (1) temperature set point of 78 F, (2) dead band of 3 F (± 1.5 F) for single-speed units and 6 F (± 3 F) for two-speed units on the thermostat, (3) design day with the maximum dry-bulb temperature for the season, (4) infiltration rate at 0.5 air-changes per hour, and (5) sizing the CAC unit to the design load.

Subsequent to the estimation of the design cooling load, six single- and six two-speed CAC units from several manufacturers were selected for evaluation of their ability to maintain conditions within the specified comfort zone. The data provided by the manufacturers for each unit were essentially steady-state capacity, power, and SHR as a function of outdoor temperature and indoor temperature and humidity. We assumed that when a unit comes on, it reaches its steady-state conditions instantaneously. This assumption probably overstates the dehumidification capacity of the units.

Evaluation of the Design Load

Two possible control modes for modeling the building energy use with TRNSYS are (1) energy rate and (2) temperature level. In the energy rate control mode, the model calculates energy loads based only upon the net gains or losses from the conditioned space. The loads are considered to be independent of the HVAC equipment. The model determines the energy necessary to keep the room at a specified dry-bulb temperature and specific humidity.

In the temperature level control mode, the room state reflects both the ambient conditions and the HVAC equipment inputs. The advantage of the temperature rate control mode is that detailed switching dynamics are not lost; however, the energy rate control mode gives the total load on the building, which is not possible with temperature level control mode.

The design load for Houston was estimated for the test house by energy rate control method. The set point temperature and relative humidity were set at 78 F and 50%, respectively.

Base Case Simulation

The dehumidification provided by the 12 CAC units selected was simulated using the design performance data. The units have SHRs between 0.65 and 0.96, SEERs between 8 and 15, and indoor flow rates of 1200 to 1400 cfm. The SEERs and the SHRs of these units at ARI Standard 210 conditions (95 F outdoor dry-bulb, 67 F indoor wet-bulb, and 80 F entering dry-bulb temperature) are shown in Table 5. The rated capacity of the units was adjusted to match the design cooling load of the test house by scaling down the rated capacity and compressor power. This was done to make qualitative comparisons between different units with different rated capacities.

Dehumidifying performance for all the units was studied with temperature rate control for the base case. The test day (weather) was the hottest day in the weather files. The minute-by-minute output generated by TRNSYS included room temperature and relative humidity, equipment ON/OFF status, supply temperature, relative humidity, capacity, SHR, and compressor power. The equipment ON/OFF status, room temperature, and relative humidity were further processed to obtain integrated hourly information. The time spent outside the comfort zone, equipment ON time, power consumed, and ON/OFF cycles for each of the 12 units integrated over the entire day are shown in Tables 6 and 7. The time spent outside the comfort zone is the ratio of the number of minutes the room conditions were outside the comfort zone to the total number of minutes in a day (1440).

Column A is the time spent outside the comfort zone between 6 a.m. and 11 p.m. whereas column B is the time spent outside the comfort zone between 12 a.m. and 11 p.m. The time spent outside the comfort zone increased with SHR for all the units. It increased from 9% for the unit with SHR of 0.67 to 23% for the unit with a SHR of 0.85. As the SHR increases, the capacity to dehumidify reduces. Therefore, the units with high SHRs spent considerable time outside the comfort zone. Although most of the single speed units selected had reasonably good SHRs (SHRs of less than 0.8) these units have spent an average of 12% of the time outside the comfort zone. The units with SHRs of greater than 0.8 have spent almost 20% of the time outside the comfort zone in a day (6 a.m. to 11 p.m.). Most of the time spent outside the comfort zone for all the units was between 6 and 8 a.m., when the sensible loads are low while the outdoor and indoor relative humidities are high.

The time spent outside the comfort zone increased with SHR even for the two-speed units. The units with lower SHR at low speed spent significantly less time outside the comfort zone than the units with high SHR at low speed. The average ON-time for the units TA – TC was about 84%, and for the units TD – TG it was about 67%. The energy consumption, in general, decreased with the increasing SHR at low speed. The units TA – TC had only one ON/OFF cycle, while the units TD – TG had four.

All the units operated at low speed for approximately 90% of the ON time; therefore, the correlation between the SHR at low speed and time spent outside the comfort zone is significant. Since the SHR at low speed for the units TA – TC was better and the ON time of the units was higher than the units TD – TG, the units TA – TC removed more moisture from the zone. The flow rate for the units TD – TG remains constant, while the flow rate for the units TA – TC at low speed is half that at high speed; therefore, the total capacities of the units TA – TC were lower than the units TD – TG at low speed; however, the latent capacity of these units was better. Since the total capacities of the units TA – TC were lower at low speed, these units had higher ON time and fewer ON/OFF cycles.

Compared to the single speed unit SJ, the two–speed units TD – TG spent between 90 and 200% more time outside the comfort zone (between 6 a.m. and 11 p.m.) for the same conditions. However, the units TA – TC spent about 25% less time than the unit SJ.

SENSITIVITY ANALYSIS

A sensitivity analysis was carried out to determine the effect of a change in the values of the major parameters on the dehumidifying performance of a CAC unit. The major parameters were:

1. The weather (design, hot/humid and mild/humid days)
2. Thermostat set point (78 & 76 F)
3. Thermostat dead band (3 & 6 F)
4. Infiltration (0.25 to 2.0 air–changes/hr)
4. Sizing of the CAC unit (70% to 130% rated capacity)

The simulation was carried out to determine the time spent outside the comfort zone, equipment ON time, energy consumed, and the number of ON/OFF cycles as a function of each of the parameters considered above. The results are given in Tables 8 – 12. All other parameters are the same as for the base case and the time interval is 6 a.m. to 11 p.m. The unit used for sensitivity analysis was the single–speed unit SJ.

The unit spent 14%, 34%, and 58% of the time outside the comfort zone for the design, hot/humid and mild/humid days, respectively. The dehumidifying performance of the CAC depends on the ambient dry-bulb temperature and relative humidity. The difference in the time spent outside the comfort zone is due to two reasons: (i) the average outdoor dry-bulb temperature on the mild/humid day (between 6 a.m. and 11 p.m.) is 82.9 F which is 2.3 F lower than the design day and (ii) the average outdoor humidity ratio on the mild/humid day is much higher than on the the design day. At higher ambient temperatures on the design day the unit operates longer than on the hot/humid or mild/humid day. Because the unit operates longer, it is able to extract more moisture from the air on the design day than on the other two days. When the ambient relative humidity is high as on the hot/humid and mild/humid days, latent gains from infiltration are much higher than on a hot day with a lower ambient relative humidity. Therefore, the time spent outside the comfort zone is higher for hot/humid and mild/humid days as compared to the design day. Low ambient dry-bulb temperatures also mean lower ON times for the unit. Hence, unit ON time and number of ON/OFF cycles increase with average dry-bulb temperature. The time spent outside the comfort zone increases with decreasing average dry-bulb temperature and increasing humidity ratio.

Decreasing the set point temperature from 78 F to 76 F reduced the time spent outside the comfort zone but increased the equipment ON time and the energy consumption. There is a significant reduction in time spent outside the comfort zone, for example, on the mild/humid day. The time spent outside comfort zone reduced from 58% to 34%. These results can be explained on the basis of the units running longer when the set point is reduced. There is no significant change in the number of ON/OFF cycles.

Increasing the dead band from 3 F (± 1.5 F) to 6 F (± 3 F) with a set point temperature of 78 F, decreased the time spent outside the comfort zone for all the three days. The decrease in the time spent outside the comfort zone occurred between 8 and 11 a.m. and 4 and 7 p.m. Since the ambient dry-bulb temperatures during these hours are higher than the set point, a unit with 6 F dead band will run longer. Hence, there is more time for the dehumidifying process.

Infiltration is another parameter which plays a major role by increasing the latent load on CACs. A series of simulations were run at infiltration rates between 0.25 and 2.0 air-changes/hr. The performance of the CAC was simulated for indoor set point temperature of 78 F and a 6 F dead band. The single–speed unit, SJ, was used for all the simulations without a change in the capacity, and the test day was the hot/humid day. The time spent outside comfort zone and the energy consumption increased with the rate of infiltration.

Finally, a series of simulations were performed by varying the rated capacity of the unit SJ from 70% to 130% of the design load for the test house. The time spent outside the comfort zone increased with capacity and corresponding to that increase there was a slight decrease in energy consumption. The units with a lower capacity run longer and hence the time spent outside the comfort zone was lower for the undersized units.

CONCLUSIONS

The base case analysis was carried out for the hottest day of the year (design day) for Houston weather. The simulation results were then compared with the performance of one of the units (SJ) on a hot/humid and mild/humid days.

Most of the units with rated SHRs of less than 0.75 performed adequately on design days; however, the units with rated SHRs between 0.75 and 0.80 would not adequately dehumidify on mild/humid days in Houston, Texas. For SHR values above 0.80, the units spent a considerable amount of time outside the comfort zone, even on the design day, which is hot and dry. Therefore, it appears that units with SHR of greater than 0.8 may not be able to provide adequate comfort for locations with hot and humid weather conditions, such as Houston. This is also borne out by the fact that there is a strong relationship between comfort and rated SHR. On the other hand, this study also seems to indicate a relatively insignificant relationship between comfort and SEER and between SEER and SHR.

The sensitivity analysis showed that decreasing the set point temperature (from 78 F to 76 F) and/or increasing the dead band on the thermostat (from 3 F to 6 F) would improve the comfort level; however, the energy consumption would increase in both cases. The results also indicate that undersizing the CAC units slightly (80% of the design load) would improve the comfort level.

The data used for the simulation of the units were steady–state values provided by the individual manufacturers. However, in order to get a clear picture of the transient dehumidification performance of the CAC units, it is recommended that actual ON/OFF cycling dehumidification performance data be used. Unfortunately, this type of data is not readily available at present.

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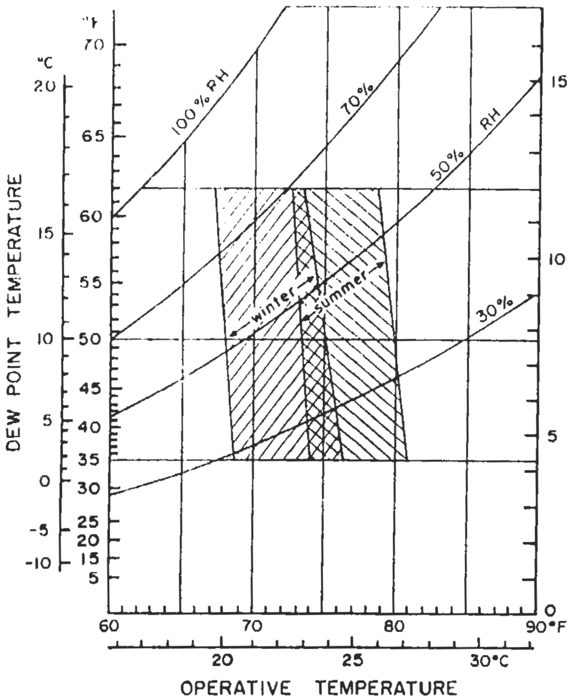

Figure 1 – ASHRAE Comfort Zone
Source: ASHRAE 55-81

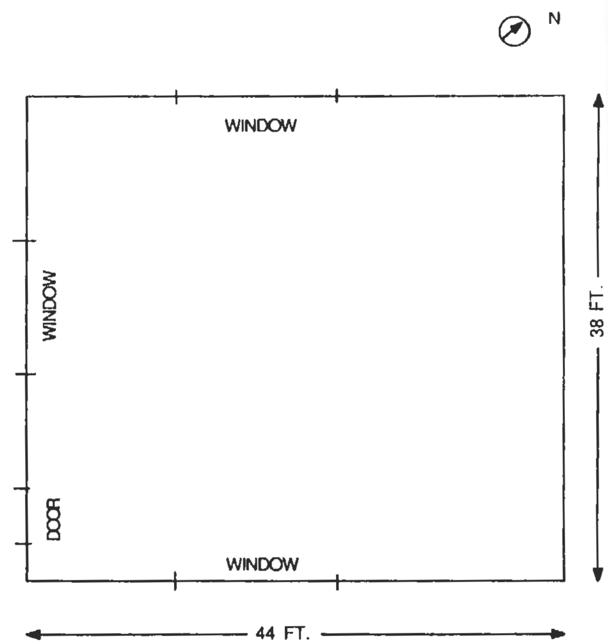

Figure 2 – Schematic of the Test House

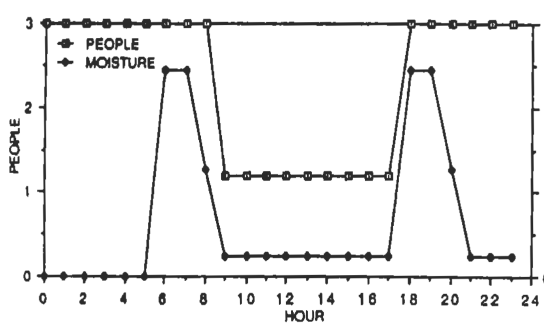

Figure 3 – People and Internal Latent Load Profiles for the Test House

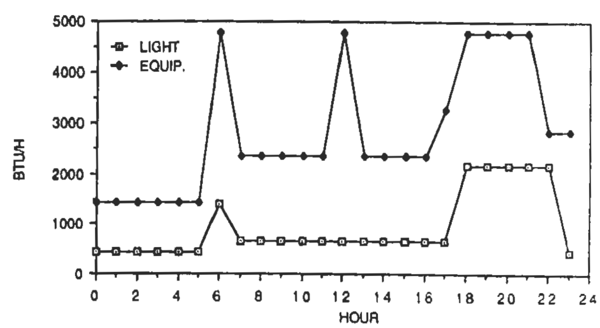

Figure 4 – Lighting and Equipment Profiles For the Test House

TABLE 1
Sample Manufacturers' Data

PERFORMANCE DATA COOLING
(CAPACITIES ARE NET IN BTUH/1000-INDOOR FAN HEAT DEDUCTED)

BTA036D WITH BXA736D AT 1200 CFM

O.D. D.B.	I.D. W.B.	TOTAL CAP.	SENS. CAP. AT ENTERING D.B. TEMP. 72	74	76	78	80	COMPR. KW	APP. DEW PT.
85	59	33.0	26.3	28.4	30.6	32.8	33.7*	3.35	45.7
	63	35.5	21.9	24.1	26.3	28.5	30.6	3.49	49.7
	67	38.1	17.2	19.4	21.6	23.8	25.9	3.63	53.9
	71	40.8	12.4	14.6	16.7	18.9	21.1	3.78	58.2
90	59	32.5	26.1	28.2	30.4	32.6*	33.4*	3.52	45.8
	63	35.0	21.7	23.9	26.1	28.3	30.4	3.66	49.8
	67	37.5	17.0	19.2	21.4	23.6	25.7	3.81	54.1
	71	40.2	12.2	14.4	16.5	18.7	20.9	3.95	58.4
95	59	32.1	25.9	28.0	30.2	32.2*	33.0*	3.70	46.1
	63	34.4	21.5	23.7	25.9	28.1	30.2	3.83	50.1
	67	37.0	16.8	19.0	21.2	23.3	25.5	3.98	54.3
	71	39.6	12.0	14.2	16.3	18.5	20.7	4.13	58.6
100	59	31.3	25.5	27.7	29.9	31.6*	32.4*	3.90	46.4
	63	33.7	21.2	23.4	25.6	27.7	29.9	4.04	50.4
	67	36.2	16.5	18.7	20.9	23.0	25.2	4.18	54.6
	71	38.7	11.7	13.9	16.0	18.2	20.4	4.33	58.9
105	59	30.8	25.2	27.4	29.6	31.0*	31.8*	4.11	46.8
	63	32.9	20.9	23.1	25.3	27.4	29.6	4.24	50.7
	67	35.3	16.2	18.4	20.6	22.7	24.9	4.38	54.9
	71	37.8	11.4	13.5	15.7	17.9	20.1	4.53	59.3
115	59	29.1	24.6	26.8	29.0	29.8*	30.5*	4.54	47.4
	63	31.3	20.3	22.5	24.6	26.8	29.0	4.65	51.3
	67	33.6	15.6	17.7	19.9	22.1	24.3	4.79	55.6
	71	36.0	10.7	12.9	15.1	17.3	19.5	4.92	59.9

CORRECTION FACTORS - OTHER AIRFLOWS
(multiply or add as indicated)

AIRFLOW	1050	1350
TOTAL CAP.	×0.98	×1.01
SENS. CAP.	×0.94	×1.06
COMPR. KW	×0.99	×1.01
A.D.P.	-1.4	+1.1

VALUES AT ARI RATING CONDITIONS

TOTAL NET CAPACITY = 37000 BTUH
AIRFLOW = 1200 CFM
APP. DEW PT. = 54.3 DEG. F
COMPR. POWER = 3960 WATTS
I.D. FAN POWER = 440 WATTS
O.D. FAN POWER = 280 WATTS
S.E.E.R. = 8.30 BTUH/WATT
E.E.R. = 7.80 BTUH/WATT

NOTE: RATED WITH 25 FEET OF 7/8 SUCT. AND 3/8 LIQUID LINES

* Dry coil condition (Tot. Cap. = Sens. Cap.) Tot. Cap., Comp. KW and App. Dew Pt. valid only for wet coil. All temperatures in degrees F.

TABLE 2
Operative Temperatures for Thermal Acceptability for Slightly Active Persons ($\le$ 1.2 mets[⊕]) at 50% Relative Humidity.

Description of typical clothing	clo *	Optimum operative temperature	Operative temperature range for 80% thermal acceptability
light slacks and short sleeve shirt	0.50	76.0 F 24.4°C	73.0 – 79.0 F 22.8 – 26.1°C
minimal	0.05	81.0 F 27.2°C	79.0 – 84.0 F 26.0 – 29.0°C

TABLE 3
Parameters of the Building

Type	Description	Area (ft^2)	U-value (Btu/h-F-ft^2)
Roof	Wood shingle + R-19; 45° slope	1672	—
Wall	Brick + R-11	1015	0.064
Glass	Single pane	265	0.610

TABLE 4
Summary of the Test Days

TEST DAY	MAXIMUM DRY-BULB TEMP. (F)	MAXIMUM SOLAR RADIATION (Btu/h/ft^2)	MAXIMUM WIND SPEED (mph)	AVERAGE DRY-BULB TEMP. (F)	AVERAGE RELATIVE HUMIDITY (%)
Design July 24	99	3000	16	83.1	69.8
Hot/Humid Aug. 14	92	2400	16	81.3	77.1
Mild/Humid Aug. 22	86	1500	15	76.7	85.6

⊕The activity level is expressed in terms of metabolic rate per unit area (W/m^2). One Met is the metabolic rate of a person at rest (58 W/m^2).

* Is a numerical representation of a clothing ensemble's thermal resistance, 1 Clo = 0.88 ft^2 h F/Btu (0.155 m^2 K/W).

TABLE 5
Design Specification for Selected CAC Units
at ARI Standard 210

Single–Speed

UNIT	SEER	SHR
SA	8.85	0.67
SB	8.40	0.70
SD	8.70	0.72
SF	11.85	0.75
SJ	8.75	0.79
SN	9.25	0.85

Two-Speed

UNIT	SEER	SHR (Low/High)
TA	10.00	0.65/0.68
TB	10.25	0.73/0.75
TC	10.50	0.72/0.77
TD	12.15	0.75/0.94
TF	15.00	0.74/0.95
TG	13.30	0.74/0.96

TABLE 6
Base Case Summary of Single–Speed Units.

UNIT	SHR	PERCENT OUTSIDE COMFORT ZONE		PERCENT EQUIP. ON TIME		ENERGY CONSUMED (kWh)		NUMBER ON/OFF CYCLES	
		A	B	A	B	A	B	A	B
SA	0.67	8.8	9.2	40.4	47.8	39.5	46.6	38	38
SB	0.69	9.8	11.2	40.3	47.2	37.5	43.8	38	38
SD	0.72	10.6	12.8	38.4	45.5	34.8	41.1	40	40
SF	0.75	11.1	13.8	38.6	44.9	26.9	31.2	40	40
SJ	0.79	14.3	19.2	34.3	43.1	33.5	39.0	44	44
SN	0.85	23.1	30.2	34.3	42.4	34.4	40.8	42	42

A: 6 a.m. to 11 p.m.
B: 12 a.m. to 11 p.m.

TABLE 7
Base Case Summary of Two–Speed Units.

UNIT	SHR (Low/High)	PERCENT OUTSIDE COMFORT ZONE		PERCENT EQUIP. ON TIME		ENERGY CONSUMED (kWh)		NUMBER ON/OFF CYCLES	
		A	B	A	B	A	B	A	B
TA	0.65/0.68	8.8	16.6	70.3	85.1	33.2	42.1	1	1
TB	0.73/0.75	9.8	19.0	70.4	84.4	29.9	37.5	1	1
TC	0.72/0.77	9.4	20.2	70.6	82.9	32.4	37.9	1	1
TD	0.75/0.94	25.5	50.3	61.9	65.3	25.3	26.9	4	4
TE	0.75/0.95	37.9	60.1	61.2	66.9	23.9	26.4	4	4
TF	0.74/0.95	36.6	61.6	63.4	67.0	25.8	27.4	4	4
TG	0.74/0.96	41.6	65.8	63.1	67.0	23.4	25.1	4	4

A: 6 a.m. to 11 p.m.
B: 12 a.m. to 11 p.m.

TABLE 8
Sensitivity of Results to Weather at a Set Point
Temperature of 78 F and 3 F Dead Band.

TEST DAY	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	ENERGY CONSUMED (kWh)	NUMBER ON/OFF CYCLES
DESIGN	14.3	34.3	33.5	44
HOT/HUMID	34.0	30.6	29.2	40
MILD/HUMID	58.4	25.1	24.0	36

TABLE 9
Sensitivity of Results to Set Point Temperature.

TEST DAY	SET POINT (F)	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	ENERGY CONSUMED (kWh)	NUMBER ON/OFF CYCLES
DESIGN	78	14.3	34.3	33.5	44
DESIGN	76	10.5	37.5	35.7	45
HOT/HUMID	78	34.0	30.6	29.2	40
HOT/HUMID	76	16.8	33.8	31.3	41
MILD/HUMID	78	58.4	25.1	24.0	36
MILD/HUMID	76	34.2	28.3	26.2	38

TABLE 10
Sensitivity of Results to Dead Band.

TEST DAY	DEAD BAND (F)	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	ENERGY CONSUMED (kWh)	NUMBER ON/OFF CYCLES
DESIGN	3	14.3	34.3	33.5	44
DESIGN	6	13.4	37.2	35.5	19
HOT/HUMID	3	34.0	30.6	29.2	40
HOT/HUMID	6	28.5	35.2	32.8	18
MILD/HUMID	3	58.4	25.1	24.0	36
MILD/HUMID	6	44.1	28.2	26.2	17

TABLE 11
Sensitivity of Results to Infiltration

INFILTRATION RATE AIR-CHANGES/HR	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	ENERGY CONSUMED (kWh)	NUMBER ON/OFF CYCLES
2.00	71.5	47.6	44.9	9
1.50	61.7	44.0	41.5	11
1.25	57.3	43.1	40.5	13
1.00	52.2	39.5	37.1	15
0.75	45.3	36.1	33.8	17
0.50	28.5	35.2	32.8	18
0.25	12.7	30.6	28.4	19

TABLE 12
Sensitivity of Results to the Capacity of the Unit
at the Set Point Temperature of 78 F and Dead Band of 6 F.

CAPACITY (%)	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	ENERGY CONSUMED (kWh)	NUMBER ON/OFF CYCLES
70	10.5	64.4	40.1	4
85	13.3	51.7	39.3	9
100	13.4	37.2	35.5	19
115	15.9	33.7	34.8	20
130	16.0	29.7	34.8	23