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DETERMINATION OF THE TRANSIENT DEHUMIDIFICATION CHARACTERISTICS OF HIGH EFFICIENFY CENTRAL AIR CONDITIONERS

FINAL REPORT

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Contract No. S-95013
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GLOSSARY OF TERMS

ARI	American Refrigeration Institute
ASHRAE	American Society of Heating Refrigeration and Air Conditioning Engineers
CAC	Central Air Conditioners
DOE	Department of Energy
EER	Energy Efficiency Ratio
EPRI	Electric and Power Research Institute
LOADSIM	LOAD SIMulation
SEER	Seasonal Energy Efficiency Ratio
SHR	Sensible Heat Ratio
TOCZ	Time Outside the Comfort Zone
TMY	Typical Meteorological Year
TRNSYS	TRaNsient SYstem Simulation

SUMMARY

A series of tests were performed to assesses the dehumidifying performance of residential central air conditioners (CACs). The performance studies were based on factors such as: (i) dynamic performance (ii) the ASHRAE comfort zone, (iii) control strategy and (iv) published performance characteristic of the units. The units were evaluated on their ability to maintain conditions in the ASHRAE comfort zone in a typical residence and typical summer days in Houston, Texas.

The minute-by-minute simulation was performed for three typical days. The primary concern was with dehumidifying performance of residential CACs rather than annual energy consumption. The design cooling load on the hottest day for the test house was estimated to be 32,500 Btu/hr. Based on the design cooling load a number of single speed and two-speed CAC units were selected for simulation. The single speed units had sensible heat ratios (SHRs) between 0.67 and 0.85, seasonal energy efficiency ratios (SEERs) between 8 and 12, and indoor flow rates between 1200 to 1400. The two-speed units had SHRs between 0.65 and 0.75 at high speed and 0.68 and 0.96 at low speed and SEERs between 10 and 15.

Before the design load was estimated, variables which influence the dehumidifying performance of residential CACs were identified and the base values were assigned to them. Five variables were identified as being the most important in determining the dehumidifying performance of residential CAC units: (i) type of day selected (design day, hot/humid day or mild/humid day) (ii) thermostat set point, (iii) thermostat dead band, (iv) infiltration and (v) sizing of the units. The base values for the five variables are: (i) temperature set point of 78 F, (ii) dead band of 3 F on the thermostat, (iii) design day with the maximum dry-bulb temperature for the season, (iv) infiltration rate of 0.5 air-changes/hr, and (v) sizing the CAC unit to the design load.

Fourteen single speed units were evaluated for base values. Although the units with less than 0.8 SHR at design conditions are considered reasonable by CAC designs, these units allowed conditioned space to drift outside the comfort zone for approximately 12% of the time during the day (6 a.m and 11 p.m). The unit with SHRs greater than 0.8 spent more than 20% of the time outside the comfort zone. Most of the time spent outside the comfort zone for all the units was between 6 a.m. and 8 a.m., when the sensible loads were low and the outdoor and indoor specific humidities high.

On a day with mild ambient dry-bulb temperatures and high humidity it appears that the units will spent considerable time outside the comfort zone. The

correlation between the time spent outside the comfort zone and SHR is quiet significant (.92) for the single speed units. In contrast, the correlation between the time spent outside the comfort zone and the correlation between the SHR and SEER are insignificant.

Typically there is a wide variation in the values of the variables, identified as being most important for the dehumidifying performance. Alternative values were assigned to these variables and the performance of the CAC units were evaluated to estimate the sensitivity of the base case results.

The influence of the weather on the dehumidifying performance was studied for the three typical days: base case day (design day), hot/humid day and mild/humid day. The CACs spent the most time outside the comfort zone on mild/humid day and the least time on the design day. The difference in the time spent outside the comfort zone was due to two reasons: (i) the average outdoor dry-bulb temperature on the mild/humid day (between 6 a.m. and 11 p.m.) was 82.9 F which was 2.3 F lower than the design day and (ii) the average outdoor specific humidity on the mild/humid day was much higher than on the design or the hot/humid day. The time spent outside the comfort zone increases with decreasing average dry-bulb temperature and increasing specific humidity.

Decreasing the set point temperature from 78 to 76 F reduced the time spent outside the comfort zone and increased the power consumption on all the three test days. The decrease was greater for days with low average ambient dry-bulb temperature and was primarily because of increase in the ON. Increasing the dead band from 3 F to 6 F also reduced the time spent outside the comfort zone. The unit with a 6 F dead band was operating longer and therefore extracting more moisture from the zone. There was significant reduction in the ON/OFF cycles with increase in the dead band (50%). The comparison of time outside the comfort zone for two set point temperatures and two dead bands are shown in Table 1.

Table 1 - Comparison of Time Outside Comfort Zone (TOCZ) at Two Set Point Temperatures and Two Dead Bands on Hot/Humid Day.

Set Point (F)	Dead Band (F)	TOCZ (%)	Reduction TOCZ (%)	Power Consumed (kWh)
78	3	34.0	-	29.2
78	6	28.5	17.4	32.8
76	3	16.8	50.6	31.3
76	6	16.9	50.3	33.9

Infiltration is another parameter which plays a major role by increasing the latent loads on the CACs. A series of simulations were run at infiltration rates between 0.25 and 2 air-changes/hr. A house with a properly sized unit will spent more than 60% of the time outside the comfort zone if the infiltration rate exceeds 1 air-change/hr. On a humid day the latent load will be significantly higher as the rate of infiltration increases.

The rated capacity of the unit was varied from 70 to 130% of the design load and a series of simulations were performed. There was no change in time outside the comfort zone with capacity on the design day for both 78 F and 76 F set point temperatures. The ON time increased with decreasing capacity. The time outside the comfort zone increased with capacity on hot/humid day for both set points (78 F and 76 F) and for both dead bands (3 and 6 F). The units with lower capacity were operating for longer periods than the units with higher capacity and also the outdoor specific humidity levels were higher on the hot/humid day. Therefore, the units with low capacity removed more moisture from the zone and spent less time outside the comfort zone. The comparison of the time outside the comfort zone and power consumption at various capacities on the hot/humid day are shown in the Table 2.

Table 2 – Comparison of Time Outside Comfort Zone (TOCZ) for 76 F Set Point Temperature and 6 F Dead Band on Hot/Humid Day.

Percent Rated Capacity	TOCZ (%)	Change (%)	Power Consumed (kWh)
70	12.3	–	40
85	11.9	-3	38
100	16.9	37	34
115	18.5	50	33
130	19.6	59	33

The performance of the seven two-speed units with SEERs ranging from 10 to 15 were also studied. Although the SHRs at the design conditions at high speed for all the units appear to be reasonable, these units spent more time outside the comfort zone than single speed units. The units were operating at low speed for 90% of the ON time. The units have a very high SHRs at low speed (0.95); therefore, these units spent more than 50% of the time outside the comfort zone even on the design day (at 78 F set point).

In almost all cases the two-speed units spent considerably more time outside the comfort zone as compared to the single speed unit. However, the power consumption of the two-speed units was lower than single speed units. Finally, the results indicate a strong correlation between the SHR at low speed and the time spent outside the comfort zone for all the seven units.

TABLE OF CONTENTS

CHAPTER 1

INTRODUCTION

The primary purpose of air conditioning equipment is to provide a comfortable environment for the occupants in a building. Comfort is affected by several variables: (i) temperature, (ii) environmental radiation, (iii) humidity (iv) air movement,(v) clothing and (vi) activity level [1]. A well designed air conditioning and ventilating system can directly affect temperature, humidity and air movement. However, most conventional residential central air conditioner (CAC) units are controlled by a single thermostat which only senses the indoor dry-bulb temperature. Thus a CAC may provide precise temperature control in a residence, but not adequate humidity control to maintain comfort.

In recent years, manufacturers have placed a high priority on improving the energy efficiency of residential CACs. One method of improving CAC efficiency has been to increase the surface area of the heat exchanger on both the evaporator and the condenser. Such a strategy allows the CAC to run at higher refrigerant temperatures in the evaporator and lower refrigerant temperatures in the condenser. As the temperature of the refrigerant in the evaporator is increased, the ability of the CAC to dehumidify is reduced. The ability of the CACs to dehumidify is measured by sensible heat ratio (SHR). It is defined as the ratio of the sensible capacity to the total capacity. Typically, residential CACs have SHRs ranging from 0.65 to 0.90.

Another method of improving the efficiency is by varying the speed of the compressor. Lennox and Trane offer residential CAC units with SEERs as high as 15 and 10.5, respectively. The major feature of these units is a two–speed compressor.

This study assesses the dehumidifying performance of the high efficiency two–speed residential CACs in hot/humid climates typified by that of Houston and Galveston. The performance study is based on such factors as: (i) dynamic performance of the CACs, (ii) the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) comfort zone (Standard 55-1981), (iii) control strategy of the CACs and (iv) published SHR data for CACs. The units are evaluated on their ability to maintain conditions in the ASHRAE comfort zone [2] in a typical residence in Houston, Texas. This is the final report for the Houston Lighting & Power Company (HL&P), purchase order number S-95013.

Chapter 2 of this report contains the definition of the comfort zone based on the ASHRAE Standard 55–1981, which specifies the combination of factors necessary for thermal comfort in the building environment. A commercially available software, TRNSYS (TRaNsient SYstem Simulation) [3], is used with minor changes to estimate the loads and simulate the equipment in the test house. TRNSYS is a transient system simulation program with a modular structure developed by the Solar Energy Laboratory, University of Wisconsin, Madison. The details of TRNSYS software, its merits and demerits, its relevance and suitability in the study of the dehumidifying performance of residential CACs are discussed in Chapter 3. The steps involved in the simulation are also discussed in Chapter 3. Due to a wide variation of residential buildings in the HL&P service area, a typical single-story residential building is assumed. The description of the test house, and the internal and external loads associated with it are discussed in Chapter 4.

The standard library provided by TRNSYS did not contain a HVAC control module. Therefore, a module was developed to control and simulate HVAC equipment. The functions of the control module are discussed in Chapter 5. The selection of suitable test days are essential in the performance studies of the residential CACs. Three typical test days were selected from the TMY (Typical Meteorological Year) Houston weather file. The three days selected are categorized as hot, hot/humid and mild/humid days, respectively. The details of the selection criterion are discussed in Chapter 6.

A number of variables influence the dehumidifying performance of residential CACs. A base case with major variables was formulated. The formulation of the base case, evaluation of design load for the test building and evaluation of the dehumidifying performance of the six CAC units are discussed in Chapter 7. Several variables were evaluated as to their influence on the dehumidifying performance: (i) thermostat set point and dead band, (ii) weather, (iii) infiltration and (iv) sizing of CAC units.

A special case of interest is the dehumidification performance of high efficiency two-speed CACs. Units from two manufacturers were studied and the results are presented in Chapter 9.

Finally, certain conclusions and recommendations will be drawn as to the overall capability of the TRNSYS program, and some of the results obtained from the present simulation study.

CHAPTER 2

BACKGROUND OF THE ASHRAE COMFORT STANDARD

The definition of the ASHRAE comfort zone was based on human physiological responses to temperature and humidity. Numerous studies have tried to measure the effect of temperature, and humidity on human comfort. Fanger at Kansas State University (KSU) [4], studied the thermal and physiological responses of human beings to the environment in buildings. His work is the oldest and most widely used in the comfort studies. Rholes and Nevins [5] and Nevins et al. [6], also at KSU, studied the affects of ambient temperature and humidity level on comfort and thermal sensation. Berglund and Gonzales studied the affect of temperature ramps, on human comfort, in the summer at low and high humidity conditions [7].

Numerous other studies have tried to measure the effect of one or two of the six variables (temperature, humidity, radiation, air movement, activity level and thermal resistance of the clothing) on human comfort. Some of those studies were extensive and have quantified the affects of some variables, but no attempt was made to generalize the results beyond the actual test conditions [8]. Based on these studies, ASHRAE has defined the comfort Standard 55-81 [1].

DEFINITION OF COMFORT ZONE

The ASHRAE standard specifies conditions in which 80% or more of the occupants will find the environment thermally acceptable. The perception of comfort, temperature and thermal acceptability are related to one's metabolic heat production, its transfer to the environment and the resulting physiological adjustments and body temperatures. The heat transfer is influenced by the environmental factors of air temperature, thermal radiation, air movement and humidity and by the personal factors of activity and clothing. Based on the experience gained over the years of how thermal sensations are related to environmental parameters acceptable range of operative temperatures and humidities for the winter and the summer seasons were defined. The operative temperatures for maximum relative humidity of 50% for the summer are shown in Table 2.1.

Table 2.1 – Operative Temperatures for Thermal Acceptability for Slightly Active Persons ($\leq$ 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

Humidity is described in terms of dew point temperature. In the zone occupied by people at rest, the dew point temperature is always excepted to be between 35 F and 62 F. The thermal effect of humidity on an inactive person is quite small, and the limits are based on respiratory health, mold growth and other moisture related phenomena. The acceptable ranges of operative temperature and humidity for persons clothed in typical summer and winter clothing, at light, mainly sedentary, activity ($\leq$ 1.2 met) are shown in Figure 2.1.

Thermal comfort greatly depends upon the occupant's clothing and activity level. In the present study, it is assumed that the occupant's will be at rest or slightly active and wearing typical indoor clothing ($\leq$ 1.2 met). Therefore, the upper limit on dew point temperature as shown in the Figure 2.1 is 62 F. Also, for this study the qualitative analysis of the performance of the residential CACs between 12 a.m. and 6 a.m. will be ignored, because of inactivity (sleeping) of the people at this time.

[@]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).

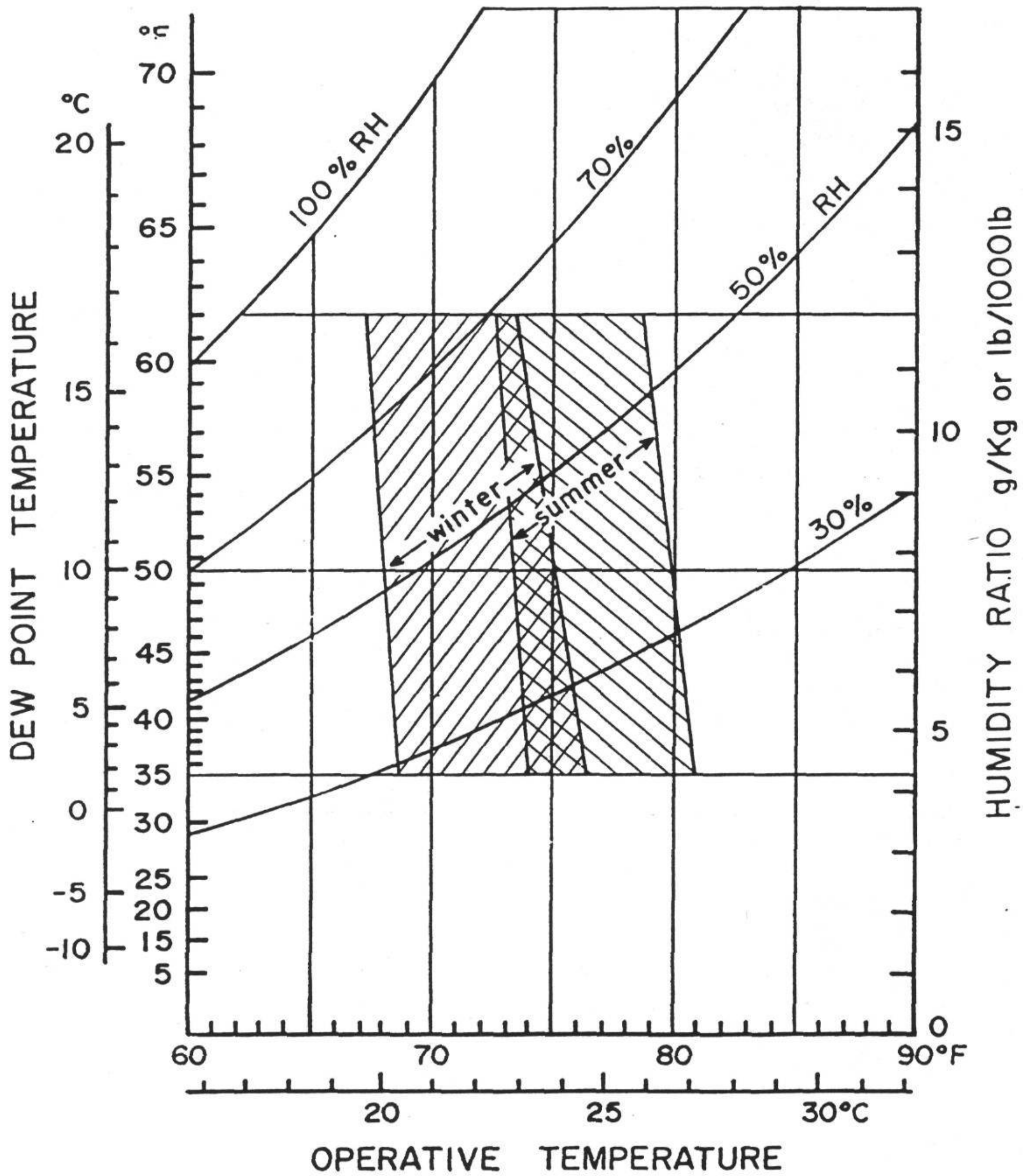

Figure 2.1 – ASHRAE Comfort Zone.

CHAPTER 3

SIMULATION MODEL & METHODOLOGY

To analyze the dehumidifying characteristics of the residential CACs for typical summer days a simulation model called TRNSYS was used. TRNSYS which is used by EPRI in LOADSIM to model the hourly variation of loads in buildings due to weather and internal loads was adopted for this study. In addition to the standard TRNSYS library, three more subroutines were developed to: (i) control the HVAC equipment, (ii) define the comfort zone and (iii) analyze the minute-by-minute output from TRNSYS. The details of the TRNSYS program are given in this chapter, and the details of the subroutines are given in Chapter 5.

SIMULATION MODEL

TRNSYS is a transient system simulation program with a modular structure developed by the Solar Energy Laboratory, University of Wisconsin, Madison. The program functions as a compiler for a high level computer language designed specifically to connect component models of transient systems, and solve the resulting algebraic and differential equations describing the system [9]. The system components are described by individual FORTRAN subroutines. To convey the information to TRNSYS, a simple language with essentially seven key-words (SIMULATION, UNIT, TYPE, PARAMETERS, INPUTS, DERIVATIVES and END) is used.

TRNSYS modules include a main module, utility modules, and component modules. Each module has a series of fixed parameters and inputs to perform the necessary calculations and obtain the outputs. Individual TRNSYS component modules are interconnected to form a simulation model for the system to be studied. Performance of the system depends on the values of the parameters, and operating schedules. Throughout the simulation, the values of the parameters are constant while inputs and outputs continually change according to the dynamic behavior of the components forming the system.

During each simulation time step, TRNSYS uses successive substitution to solve for output values. The solution is obtained by iteratively solving the set of simultaneous equations in the component modules until all changes in the values of inputs are less than the allowable tolerance values. When all inputs converge, the simulation proceeds to the next time step. In case the simulation does not converge within the specified number of iterations, the program will terminate with an error message. The tolerance values specified by the user affect the degree of accuracy of the simulation, and the time required for convergence.

A standard TRNSYS library consists of several component modules. The modules consist of a building zone load module, a pump and fan module, a cooling coil module, etc. The library also includes several utility routines for performing psychrometric analysis, processing solar radiation, solving differential equations, etc.

Previous work with TRNSYS by Hackner [10,11] indicates that the program was sufficient for simulating the process control dynamics of the building and HVAC systems. Nevertheless, two limitations of the program were mentioned. First, the program did not accurately model the components at a small time step (less than one minute). Control dynamics such as hysteresis were not modeled in TRNSYS. Second, the simulation with a time step greater than one hour did not accurately show the dynamics of components built into the models.

Despite the above limitations, TRNSYS has some advantages over other energy simulation programs because of its modular structure. Also, an individual user can develop new TRNSYS 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.

SIMULATION METHODOLOGY

To analyze the transient dehumidification performance of the residential CACs the following steps were taken.

(1) TRNSYS was made operational on a local computing system.

(2) A test house was described in a form compatible to TRNSYS

(3) A new TRNSYS module was developed to control the equipment ON/OFF and to force a desired dead band on the thermostat.

(4) The comfort zone was defined as specified by ASHRAE with minor changes.

(5) Suitable test days for simulation were selected from the TMY (Typical Meteorological Year) Houston weather file.

(6) Variables which influence the dehumidification performance of the CACs were identified.

(7) A base case was defined.

(8) Certain modifications to the base case were studied for single speed units.

(9) Finally, CACs with high SEERs (two-speed units) were selected and their performance was studied.

CHAPTER 4

DESCRIPTION OF THE HOUSE AND ITS LOADS

Residential buildings vary widely in the HL&P service area. A typical house is defined for estimating the loads and simulating the system's performance. The test house and associated thermal loads are described in this chapter.

TEST HOUSE

The house used for the simulation is a typical single-story residential building. The schematic of the building is shown in Figure 4.1. The building has a conditioned floor area of 1672 sf. The total exterior surface is 1280 sf of which 265 sf is glazed surface (glass). The percentage of glass area compared to the floor area is 15%. A brief description of the test house is shown in Table 4.1.

Table 4.1 - Physical Parameters of the Building

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

THERMAL LOADS

The external loads (weather conditions) used in this study represents a TMY 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.).

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, shower, dish washing, etc. The operating schedules and their profiles are required to estimate the thermal loads and the system performance. The operating schedules include: (i) occupancy, (ii) lighting, (iii) equipment and (iv) 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 4.2. It is assumed that all the occupants stay home between 12 a.m. and 8 a.m., and also between 6 p.m. and 11 p.m. The instantaneous latent heat gains (other than people) include cooking, washing dishes, and showers. Cooking, dish-washing and shower contribute 4.9 lbs, 3.6 lbs and 2 lbs of moisture respectively, into the conditioned space [12].

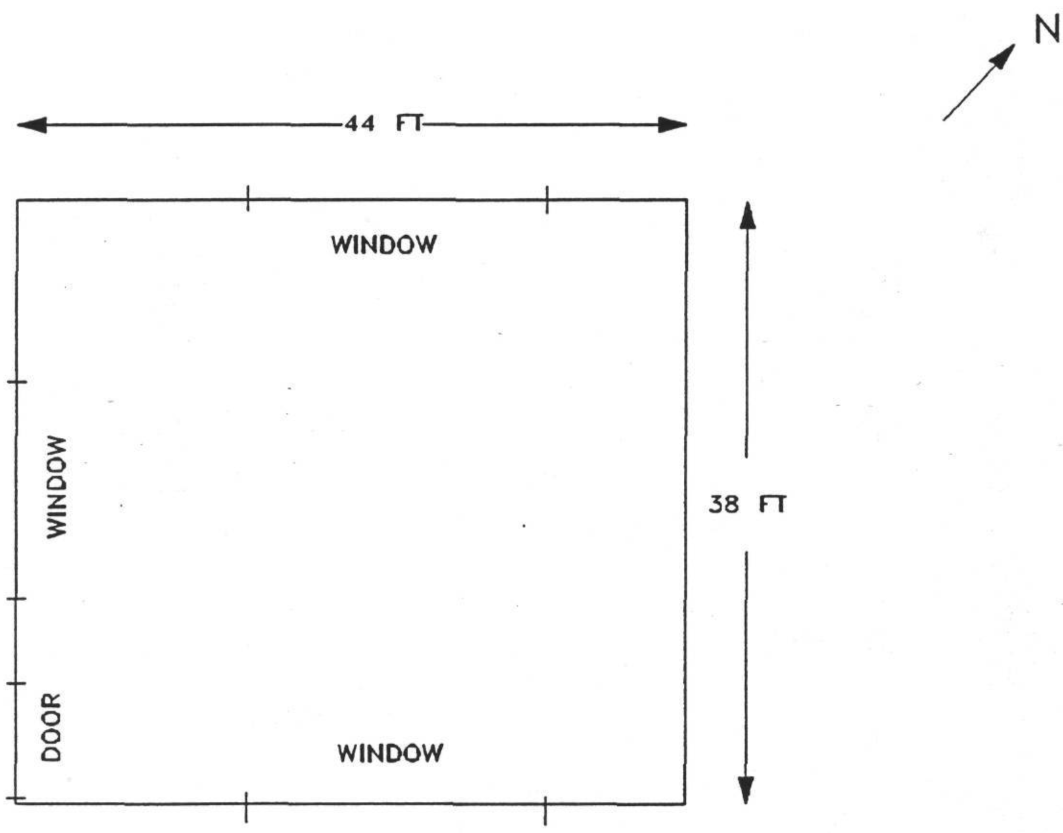

Figure 4.1 – Schematic of the Test House.

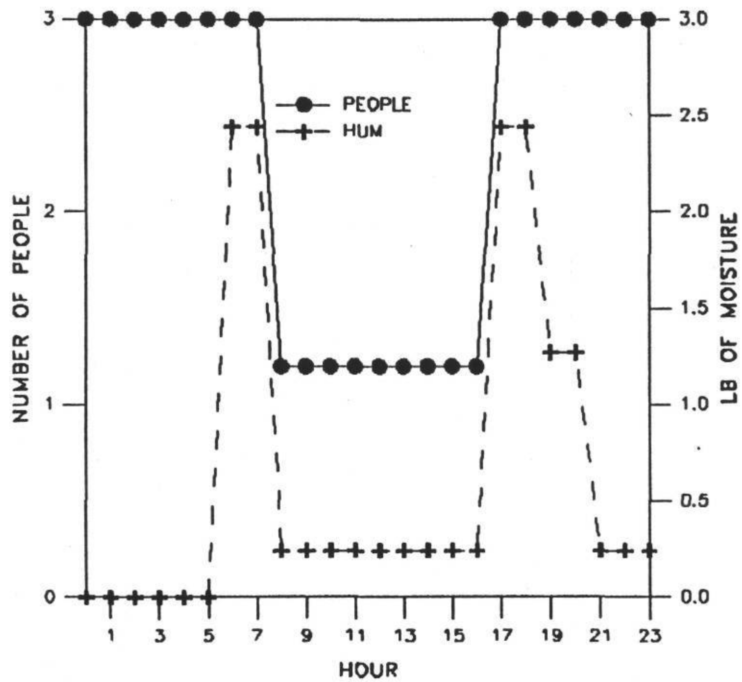

Figure 4.2 - People and Internal Latent Load Profiles for the Test House.

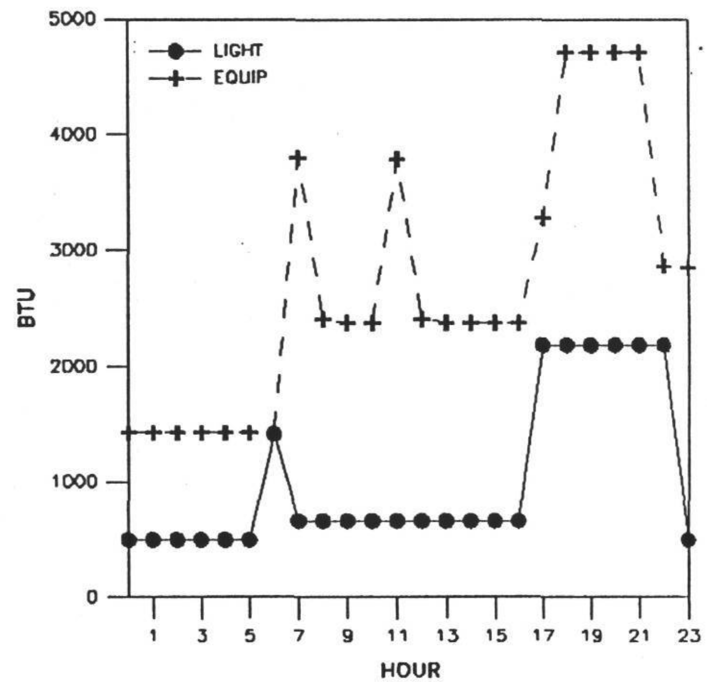

Figure 4.3 - Lighting and Equipment Profiles for the Test House

Most of this contribution occurs between 6 and 8 a.m., and 6 and 8 p.m. because of the moisture gain from the shower and cooking. The schedule of the latent loads' is shown in Figure 4.2.

The maximum amount of lighting used in the house is 660 watts (2250 Btu/hr). The peak lighting level is assumed to be 1.6 w/sf (9000 Btu/hr). The schedule of the lighting is shown in Figure 4.3. Since the occupants are asleep between 12 and 5 a.m., only 5% (450 Btu/hr) of the total lights are assumed to be ON during that period. Between 5 and 6 a.m., when the occupants wake up, 16% (1400 Btu/hr) of the lights are assumed to be ON. Between 7 a.m. and 4 p.m., when the house is not fully occupied, 7% (650 Btu/hr) of the lights are assumed to be ON. Since all the occupants are assumed to be back by 6 p.m., 25% (2250 Btu/hr) of the lights are assumed to be ON between 6 and 11 p.m.

The peak equipment wattage is assumed to be 1450 watts (4950 Btu/hr). The schedule of the equipment is shown in Figure 4.3. Some equipment like refrigerators run 24 hours, therefore, 30% (1400 Btu/hr) of the equipment is assumed to be ON even when the occupants are asleep between 12 and 5 a.m. The peak (3800 Btu/hr) occurs at 6 a.m., 12 p.m., 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, 52% (2400 Btu/hr) of the equipment is assumed to be ON.

CHAPTER 5

DEVELOPMENT OF NEW ROUTINES

The standard TRNSYS library does not have a HVAC equipment control module; therefore, a new module compatible with TRNSYS was developed. Also, a program was written to analyze the minute-by-minute outputs of TRNSYS. Both the control module and the post-processing routine are discussed in this chapter.

HVAC CONTROL MODULE

In addition to the standard TRNSYS library, an equipment simulation module was developed. The functions of the module included: (i) forcing the equipment to be ON/OFF for a specified time, (ii) scaling the capacity of the CAC unit to match the design load of the test house, (iii) forcing a specified dead band on the thermostat and (iv) controlling the operation of the two-speed CAC units.

The first function of the module was to prevent short-cycling of the compressor. This was accomplished by specifying a fixed time delay between the compressor ON and OFF period. 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 house. 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 dry-bulb temperature reaches a temperature equal to the set point minus half the specified dead band. Similarly, the equipment remains OFF until the room dry-bulb temperature reaches a temperature equal to the set point plus half the specified dead band.

The two-speed CAC units need additional control parameters to switch back and forth from high to low speed depending on the room dry-bulb temperature. The two-speed CAC unit comes ON with the compressor operating at high speed mode. The compressor remains at high speed until the room dry-bulb temperature equals the set point temperature, then the compressor is switched to low speed. The compressor will remain ON at low speed until the temperature in the conditioned space equals the set point temperature minus half the dead band. If the room dry-bulb temperature at any time during the low speed operation exceeds a temperature equal to the set point plus half the dead band, then the compressor is switched back to high speed.

The HVAC control module makes use of the manufacturers' data to simulate the cooling equipment. These data relate input power, and sensible and latent heat removal to the dry-bulb and wet-bulb temperatures of the air entering the coil and to the outdoor temperature. The two–speed units have two tables: one for the low speed and the other for the high speed operation.

POST-PROCESSING OF THE TRNSYS OUTPUT

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 is evaluated by checking the temperature and the specific humidity. The condition is said to be outside the comfort zone if the humidity ratio is greater than 0.0012 lb/lba for dry-bulb temperatures between 70 - 80 F. Second, the equipment ON time and the number of ON/OFF cycles in each hour are evaluated. Finally, the integrated hourly values for indoor-relative humidity, supply dry-bulb temperature, relative humidity and capacity, SHR, power and energy efficiency ratio (EER) are evaluated. The EER value is adjusted for compressor cycling according to the DOE (Department of Energy) specification test procedure to account for ON/OFF cycling.

CHAPTER 6

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 is no acceptable criteria for either daily or seasonal performances of the residential CACs. In this study dehumidification performance will be based on three typical summer days.

A program was developed to select test days from the TMY weather file. Three days were selected: (i) hot, (ii) hot/humid and (iii) mild/humid. The selection criterion for the hot day was to pick the day with the maximum dry bulb temperature in the TMY Houston weather file. The hottest day had a maximum dry-bulb temperature of 99 F and was selected by scanning the entire TMY Houston weather file.

The selection criterion for the hot/humid day was: (i) to select all the days with maximum dry–bulb temperatures in the range of 90 and 95 F and (ii) then select a day (among them) with the least difference between the maximum and the minimum dry–bulb temperature. When the difference between the maximum and the minimum dry–bulb temperature is small, it is a good indication of a humid day. The maximum dry–bulb temperature for the hot/humid day was 92 F. The design dry–bulb temperature for Houston is 95 F; therefore, the range of the dry–bulb temperature for the selection of hot/humid day was set between 90 and 95 F [13].

The selection criterion for the mild/humid day was: (i) to select all the days with maximum dry-bulb temperature in the range of 80 and 86 F and (ii) then to select a day with the least difference between the maximum and the minimum dry–bulb temperatures.

The hourly dry–bulb temperature, relative humidity, solar radiation and wind speed for the three test days are shown in Figures 6.1– 6.6. The maximum dry–bulb, solar radiation and wind speed, average dry-bulb temperature and relative humidity for the three test days are shown in Table 6.1.

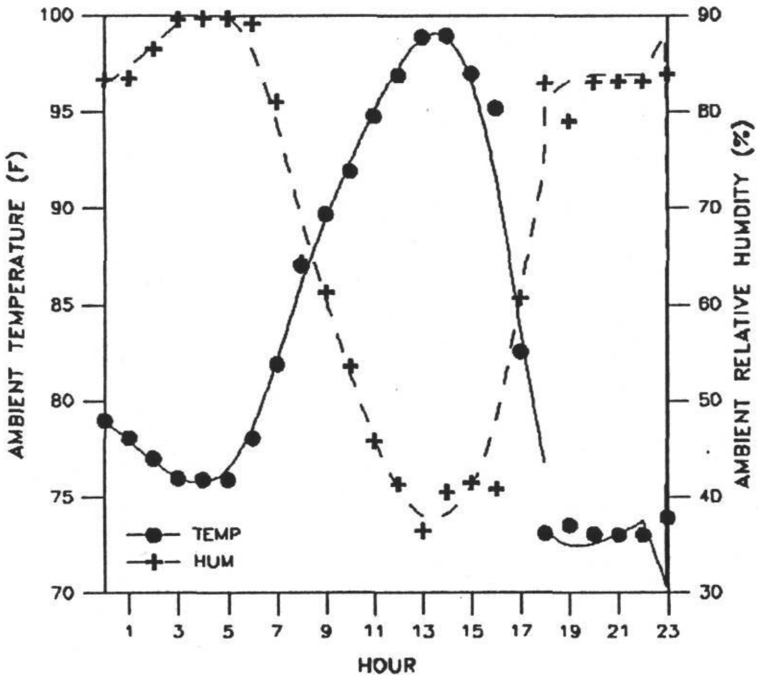

Figure 6.1 – Hourly Temperature and Humidity Profiles for the Design Day.

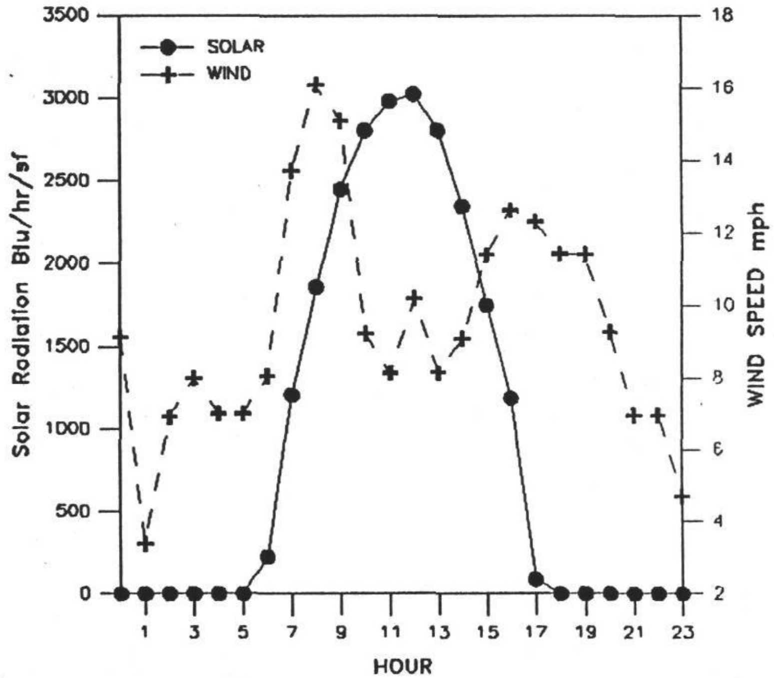

Figure 6.2 – Hourly Solar Radiation and Wind Speed Profiles for the Design Day.

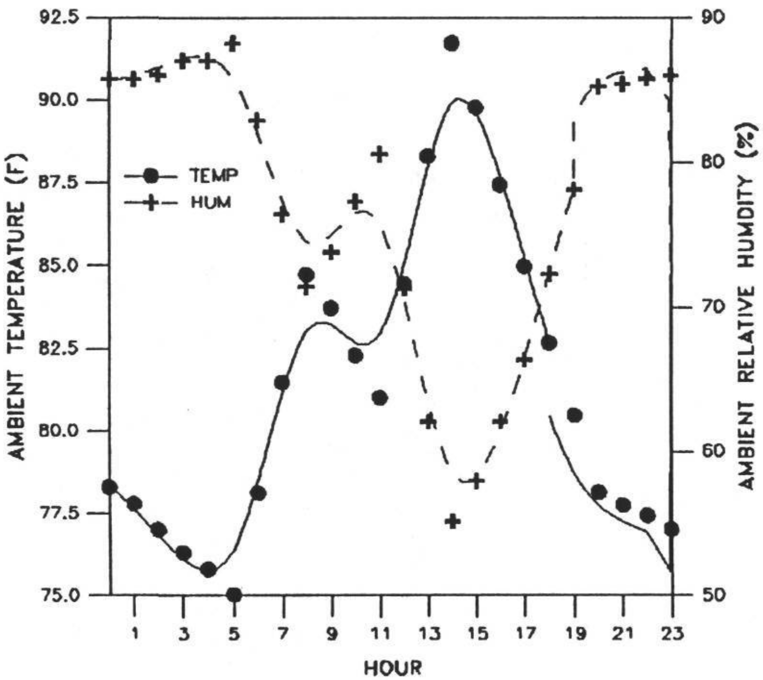

Figure 6.3 - Hourly Temperature and Humidity Profiles for the Hot/Humid Day.

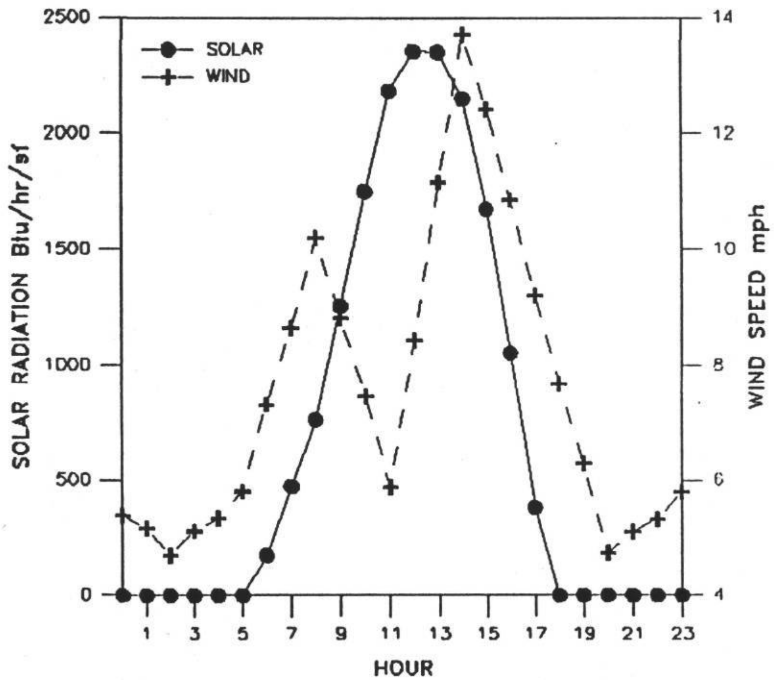

Figure 6.4 - Hourly Solar Radiation and Wind Speed Profiles for the Hot/Humid Day.

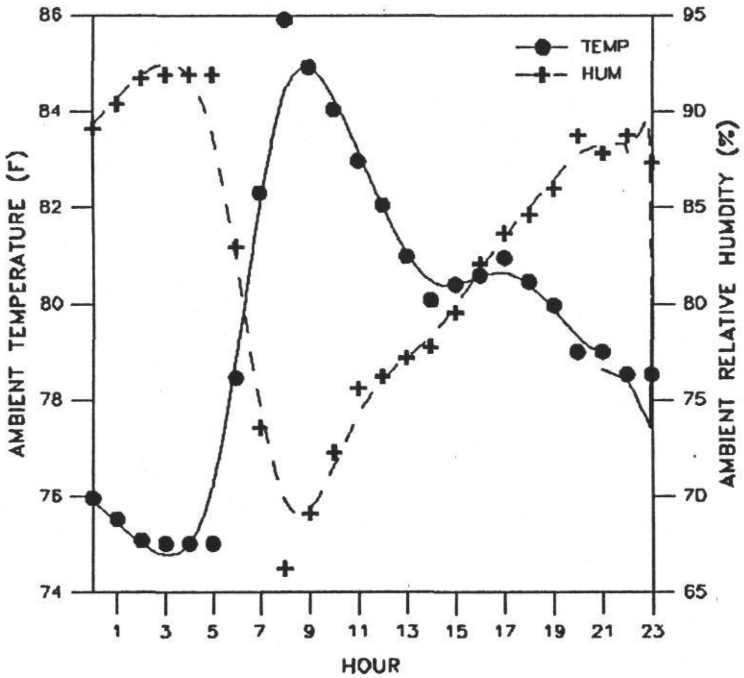

Figure 6.5 – Hourly Temperature and Humidity Profiles for the Mild/Humid Day.

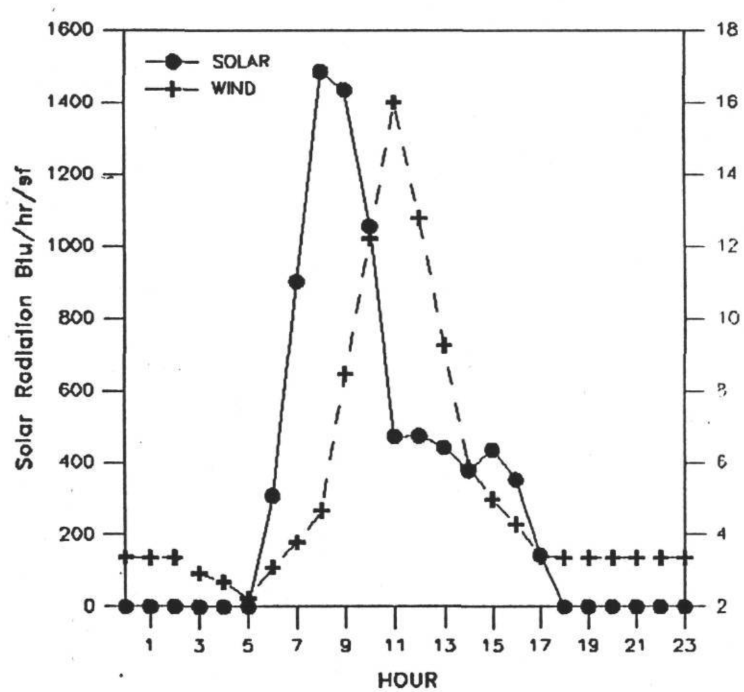

Figure 6.6 – Hourly Solar Radiation and Wind Speed Profiles for the Mild/Humid Day.

Table 6.1 - Summary of the Three Test Days.

TEST DAY	MAXIMUM DRY-BULB TEMP. (F)	MAXIMUM SOLAR RADIATION (Btu/hr/sf)	MAXIMUM WIND SPEED (mph)	AVERAGE DRY-BULB TEMP. (F)	AVERAGE RELATIVE HUMIDITY (%)
Design July 24^{th}	99	3000	16.0	83.1	69.8
Hot/Humid Aug. 14^{th}	92	2400	15.5	81.3	77.1
Mild/Humid Aug. 22^{nd}	86	1500	16.0	76.7	85.6

The design day has maximum dry-bulb temperature, solar radiation and average dry-bulb temperature. The average relative humidity is minimum on the design day. The mild/humid day has minimum solar radiation and minimum average dry-bulb temperature, and the average relative humidity is maximum. The conditions on the hot/humid day are between the design and mild/humid day. The maximum wind speed is same for all the three days. The maximum dry-bulb temperature for the design day and the hot/humid day occurs at 2 p.m., while on the mild/humid day it occurs at 9 a.m. The relative humidity for the three days is constant between 12 a.m. and 6 a.m. and 9 p.m. and 11 p.m. and is also maximum during this period. The relative humidity decreases as the dry-bulb temperature increases and is minimum when the dry-bulb temperature is maximum for all the three days. The solar radiation is maximum at noon for the design and hot/humid days while for the mild/humid day it peaks at 9 a.m.

CHAPTER 7

BASE CASE

The estimation of 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, variables which influence the dehumidifying performance of the CACs had to be identified and base values for some of them had to be assigned.

A number of variables influence the dehumidifying performance of residential CACs. Due to wide variations of these variables in the HL&P service area, a base case was formulated. Five variables were identified as the most important for estimating dehumidification performance: room temperature set point, dead band on thermostat, external load (weather), infiltration and sizing of the CAC unit. The base values for the five variables are: (i) temperature set point of 78 F, (ii) dead band of 3 F on the thermostat, (iii) design day with the maximum dry–bulb temperature for the season, (iv) infiltration rate at 0.5 air–changes per hour, and (v) sizing the CAC unit to the design load.

Subsequent to the estimation of the design cooling load, 14 CAC units were selected for evaluation of their performance in maintaining conditions within the specified comfort zone.

EVALUATION OF THE DESIGN LOAD

Two possible control modes for modeling the building energy with TRNSYS are: (i) energy rate and (ii) temperature level. In the energy rate control 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 temperature level control, the room state reflects both the ambient conditions and the HVAC equipment inputs. The advantage of the temperature rate control is that the detailed switching dynamics are not lost; however, the energy rate control gives the total load on the building which is not possible with temperature level control.

The first test was run with energy rate control (base case) to estimate the design cooling load for the test house. The ambient conditions for the design day are shown in Figures 6.1 and 6.2. Hourly zone temperature, convective and infiltration heat gains, cooling and dehumidification loads are shown in Table 7.1. The third column in the table shows the convective heat gains from all surfaces in the zone, and the fourth column shows the sensible heat gains through infiltration. The negative infiltration load indicates a heat loss from the zone. The last two columns show the sensible and latent cooling loads, respectively. The sum of sensible and latent cooling loads is the total load for each hour. The maximum ambient dry-bulb temperature on the design day occurred at 2 p.m. The cooling load at 2 p.m. is 32.5 kBtu, which is maximum for this day.

Table 7.1 - Various Loads for the Test House Energy Rate Control.

HOUR	ZONE TEMP.	CONVECTIVE HEAT GAINS	SENSIBLE INFIL. GAINS	SENSIBLE COOLING LOAD	LATENT COOLING LOAD
0	78.0	6.791	1.316	8.625	1.012
1	78.0	5.971	-1.097	7.563	0.986
2	78.0	5.223	-3.729	6.552	0.986
3	78.0	4.581	-5.045	5.778	0.986
4	78.0	4.042	-5.045	5.239	0.986
5	78.0	3.850	-2.413	5.311	1.050
6	78.0	4.596	5.045	9.177	1.137
7	78.0	6.299	1.601	1.108	1.044
8	78.0	8.234	2.566	1.327	1.012
9	78.0	9.818	3.159	1.544	0.985
10	78.0	1.101	3.773	1.970	0.857
11	78.0	1.213	4.387	1.970	0.760
12	78.0	1.336	4.870	2.070	0.675
13	78.0	1.461	5.111	2.219	0.712
14	78.0	1.563	4.870	2.296	0.955
15	78.0	1.625	4.387	2.310	0.680
16	78.0	1.562	2.566	2.131	0.687
17	78.0	1.349	-1.097	1.769	0.687
18	78.0	1.161	-1.097	1.554	0.659
19	78.0	1.025	-1.097	1.418	0.659
20	78.0	9.091	-1.206	1.291	0.686
21	78.0	8.128	-1.206	1.100	0.686
22	78.0	7.100	-1.097	9.131	0.718
23	78.0	7.100	-1.097	9.131	0.718

*Temperature in F, and Loads in kBtu

EVALUATION OF THE 14 CAC UNITS FOR BASE CASE

To get a range of the dehumidification provided by currently available CACs, 14 units were selected. The units have SHRs between 0.67 and 0.85, SEERs between 8 and 12 and indoor flow rates of 1200 to 1400 cfm. The capacity, flow rate, SEER and SHR of the units are tabulated in Table 7.2. The capacity shown in the table is at 95 F outdoor dry-bulb, 67 F indoor wet-bulb and 80 F entering dry-bulb temperature (ARI Standard 210) conditions. The SHR for the units is a steady state value at the same conditions as the capacity.

The rated capacity of the units was adjusted to match the design cooling load of the test house by scaling down the capacity and the compressor power. Dehumidifying performance for all the units were studied with temperature rate control for the base case. The test day (weather) was the hottest day in the TMY Houston weather file. The maximum dry-bulb temperature on the test day was 99 F and the average dry-bulb temperature between 6 a.m. and 11 p.m. is 85 F, which is 7 F higher than the room set point (78 F). The average relative humidity for the same time period was 64 %.

Table 7.2 - Design@ Specification for the Selected CAC Units

UNIT	CAPACITY (BTUH)	FLOW RATE (CFM)	SEER	SHR
SA	36,000	1200	8.85	0.67
SB	38,500	1200	8.40	0.70
SC	39,000	1200	9.45	0.70
SD	42,400	1200	8.70	0.72
SE	35,800	1350	9.15	0.74
SF	42,000	1350	11.85	0.75
SG	35,000	1200	8.00	0.76
SH	35,000	1200	11.00	0.76
SI	34,700	1250	8.60	0.77
SJ	35,300	1250	8.75	0.79
SK	39,300	1250	9.25	0.79
SL	32,200	1200	8.80	0.83
SM	32,700	1200	8.75	0.84
SN	34,200	1200	9.25	0.85

@ ARI Standard 210

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 which are tabulated in Tables B7.1 - B7.28 in Appendix B.

The time spent outside the comfort zone, equipment ON time, power consumed, and ON/OFF cycles for each of the 14 units integrated over the period between 6 a.m. and 11 p.m. are shown in Table 7.3. The time spent outside the comfort zone is the ratio of the number of minutes the unit was outside the comfort zone between 6 a.m. and 11 p.m. to the total number of minutes in a day (1440).

Table 7.3 – Summary of the Results of the 14 Units – Base* Case.

UNIT	SHR	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	POWER CONSUMED (kWh)	NUMBER ON/OFF CYCLES
SA	0.67	8.8	40.4	39.5	38
SB	0.69	9.8	40.3	37.5	38
SC	0.70	9.7	39.1	34.6	39
SD	0.72	10.6	38.4	34.8	42
SE	0.74	11.0	39.9	34.0	38
SF	0.75	11.1	38.6	26.9	40
SG	0.76	11.3	37.5	37.5	41
SH	0.76	11.3	37.6	31.5	41
SI	0.77	12.8	37.8	33.5	42
SJ	0.79	14.3	34.3	33.5	44
SK	0.79	14.4	35.1	29.1	42
SL	0.83	17.7	37.1	34.0	42
SM	0.84	21.0	37.8	37.1	41
SN	0.85	23.1	34.3	34.4	42

* Between 6 a.m. and 11 p.m.

The time spent outside the comfort zone increased with SHR for all the units. The time outside the comfort zone is plotted against the SHR in Figure 7.1. 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 SHR spent considerable time outside the comfort zone. The correlation matrix is shown in the Table 7.4. The correlation between the time spent outside the comfort zone and SHR is quiet significant (.92), whereas, the correlation between the time spent outside the comfort zone and the correlation between the SHR and SEER are insignificant. The quadratic correlation was in the form:

$$TOCZ = 9.5 - 2.4 * SHR + 14.4 * SHR^2 \tag{7.1}$$

Table 7.4 – Summary of the Correlation Results.

DEPENDENT VARIABLE	INDEPENDENT VARIABLE	CORRELATION COEFFICIENT
TOCZ@	SHR	0.923
TOCZ	SEER	0.117
SHR	SEER	0.014

@ Time outside the comfort zone

The minutes outside the comfort zone within each hour for some of the units are shown in Figure 7.2. Most of the discomfort time, except for the high SHR

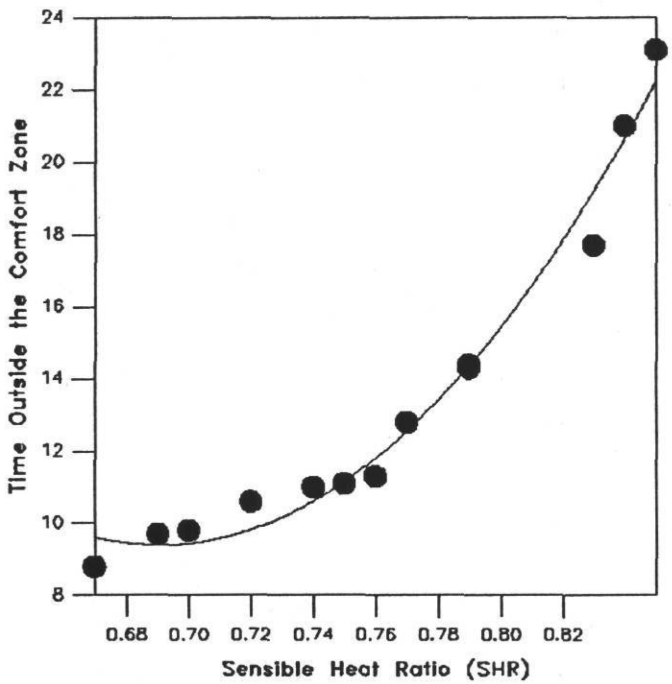

Figure 7.1 – SHR versus Time Outside the Comfort Zone for Single Speed Units.

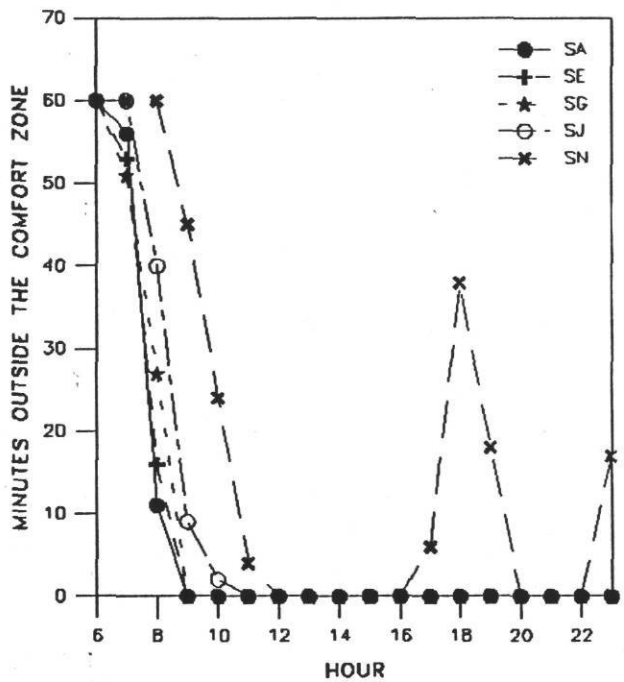

Figure 7.2 – Minutes Outside the Comfort Zone on Design Day with Base Case for Selected Units.

unit, is between 6 a.m. and 8 a.m. and this is also true of the units which are not shown in this figure. The reason for this is due to: (i) high ambient relative humidity, (ii) high internal moisture generation and (iii) low sensible loads during this period. Since the sensible loads are small during this period, the units are ON only for a short period of time, which means that the unit has little time to remove moisture from the air. In contrast, the time spent outside the comfort zone is negligible for all the units for the period between 9 a.m. and 11 p.m.; because the ambient temperatures are higher, the units run longer and remove more moisture from the zone. This trend is true for all the 14 CAC units.

Although there is a clear trend between the time spent outside the comfort zone and the SHR, there is no trend between the power consumption and the SHR. The percent ON time and the number of ON/OFF cycles were almost the same for all the units. As expected there was a clear trend between the SEER and power consumed.

DISCUSSION OF THE BASE CASE RESULTS

Although most of the unit selected for this study had a reasonably good SHRs (SHRs less than 0.80) these units have spent on a average of 12% of the time outside the comfort zone. The units with SHR greater than 0.80 have spent almost 20% of time in the day (between 6 a.m. and 11 p.m.) outside the comfort zone. If the actual performance of the units with SHR less than 0.80 is comparable to what has been modeled, then it appears that people are willing to accept a small amount of time outside the ASHRAE comfort zone. However, the units with SHRs greater than 0.80 may not be satisfactory in humid climates.

Even during the hottest part of the day (2 p.m. to 4 p.m.), most of the units were utmost ON only 60 to 65% of the time. This would indicate that the units are still somewhat oversized for the load even though they were sized using acceptable load calculations. The units that are better matched to the load would require less power, and this could have an impact on the electrical demand in the HL&P service area.

Most of the time spent outside the comfort zone for all the units was between 6 a.m. and 8 a.m., when the sensible loads are low while the outdoor and indoor relative humidities are high. The test day used in this simulation was the hottest day which had an average relative humidity much lower than the other two test days. Therefore, on a day with a high average relative humidity and low sensible gains from the external loads, it appears that the time spent outside the comfort zone may extend beyond 8 a.m. Finally, for the single speed units there is a strong correlation between the time outside the comfort zone and SHR.

CHAPTER 8

SENSITIVITY ANALYSIS

In Chapter 7, five variables were identified as being the most important in determining the dehumidifying performance of residential CAC units. It was also mentioned that there could be a wide variation of these variables in houses in the HL&P service area. Therefore, alternative values were assigned to these variables and the performance of the CAC unit was evaluated, to estimate the sensitivity of the results presented in Chapter 7.

The major variables studied included: (i) the weather, (ii) thermostat set point, (iii) thermostat dead band, (iv) infiltration and (v) sizing of the units. Important hourly simulation data for the sensitivity tests are shown in Tables B.1 - B.106 of Appendix B. The results of the sensitivity analysis are presented below.

WEATHER

The influence of weather on the dehumidifying performance of the CAC units was studied for the three typical days described in Chapter 6. These include the design (base case), hot/humid and mild/humid days. Table 8.1 shows the affect of weather on the performance of the the unit SJ. The simulation was performed at two set point temperatures (78 & 76 F) and two dead bands (3 & 6 F). The infiltration rate was assumed constant at 0.5 air changes/hr.

Table 8.1 summarizes the time spent outside the comfort zone, equipment ON time, power consumed and number of ON/OFF cycles for the three test days, between 6 a.m. and 11 p.m. The unit spends 14%, 34% and 58% of the time outside the comfort zone for the design, hot/humid, and mild/humid days, respectively.

Table 8.1 - 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	POWER 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

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.

THERMOSTAT SET POINT

Table 8.2 shows the effect of set point on the performance of the unit SJ. The table summarizes the time spent outside the comfort zone, equipment ON time, power consumed and number of ON/OFF cycles for three test days.

Table 8.2 – Sensitivity of Results to Set Point Temperature with a Dead Band of 3 F.

TEST DAY	SET POINT (F)	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	POWER 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

Decreasing the set point temperature from 78 to 76 F reduced the time spent outside the comfort zone and increased the power consumption for all cases. The decrease in time spent outside the comfort zone is greater for days with low average ambient dry–bulb temperature. The decrease in the time spent outside the comfort zone on the design day with an average ambient dry–bulb temperature of 85.1 F is 27%. In comparison, the mild/humid day with average ambient dry–

bulb temperature of 81.1 F has a decrease of 41%. Also, the equipment ON time and the power consumption increased as the set point temperature was reduced from 78 to 76 F. There was no significant change in ON/OFF cycles with the decrease in set point temperature.

THERMOSTAT DEAD BAND

Tables 8.3 and 8.4 show the effect of dead band on the performance of the unit SJ. The tables summarize the time spent outside the comfort zone, equipment ON time, power consumed and number of ON/OFF cycles for the three test days.

Table 8.3 - Sensitivity of Results to Dead Band with the Set Point Temperature at 78 F.

TEST DAY	DEAD BAND (F)	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	POWER 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

Increasing the dead band from 3 F (± 1.5 F) to 6 F (± 3 F) with a set point temperature 78 F, decreased the time spent outside the comfort zone for all the three days. Minutes spent outside the comfort zone in each hour for the 78 F set point are shown in Figure 8.1 for both 3 and 6 F dead bands for the hot/humid day. 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.

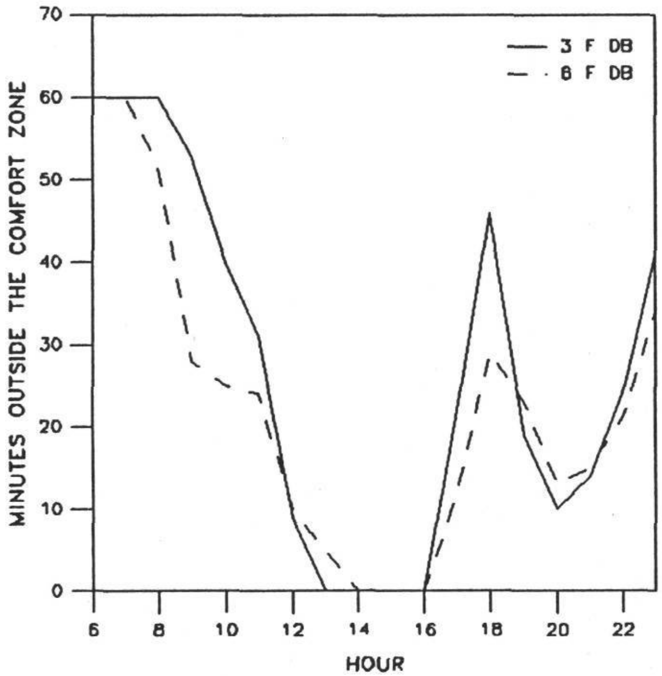

Figure 8.1 – Minutes Outside the Comfort Zone at 78 F Set Point Temperature on Hot/Humid Day.

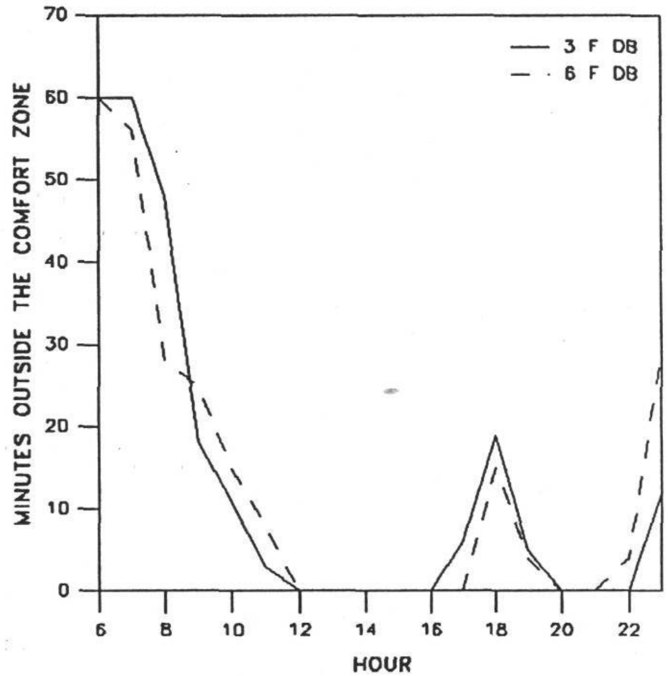

Figure 8.2 – Minutes Outside the Comfort Zone at 76 F Set Point Temperature on Hot/Humid Day.

Table 8.4 – Sensitivity of Results to Dead Band with the Set Point Temperature at 76 F.

TEST DAY	DEAD BAND (F)	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	POWER CONSUMED (kWh)	NUMBER ON/OFF CYCLES
DESIGN	3	10.5	37.5	35.5	45
DESIGN	6	9.7	40.3	38.2	19
HOT/HUMID	3	16.8	33.8	31.8	41
HOT/HUMID	6	16.9	36.7	33.9	19
MILD/HUMID	3	34.2	28.3	26.2	38
MILD/HUMID	6	30.3	30.7	28.4	17

Decrease in time spent outside the comfort zone with the increase in the dead band at 76 F temperature set point was quiet small for all the three test days (Figure 8.2). However, a unit with a 6 F dead band and 76 F set point spent lesser time outside the comfort zone than a unit with a 6 F dead band and 78 F set point. The power consumption increased with increase in the dead band. Also, there was a significant reduction in the number of ON/OFF cycles as the dead band was increased.

INFILTRATION

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 two indoor set point temperatures (78 & 76 F) with a 6 F dead band. A single unit was used for all the simulations without a change in the rated capacity. The test day was the hot/humid day; and the unit SJ was used for the simulation. The time spent outside comfort zone and the power consumption increased with infiltration (Figure 8.3). The ON time and number of ON/OFF cycles are shown in Figure 8.4. The increase in time spent outside the comfort zone is sharper than that of the equipment ON time and the power consumption. Important hourly simulation data are shown in Tables B.25 – B.48 in Appendix B. The Table 8.5 summarizes the time spent outside the comfort zone, equipment ON time, power consumed and number of ON/OFF cycles for the unit SJ at the set point temperature of 78 F.

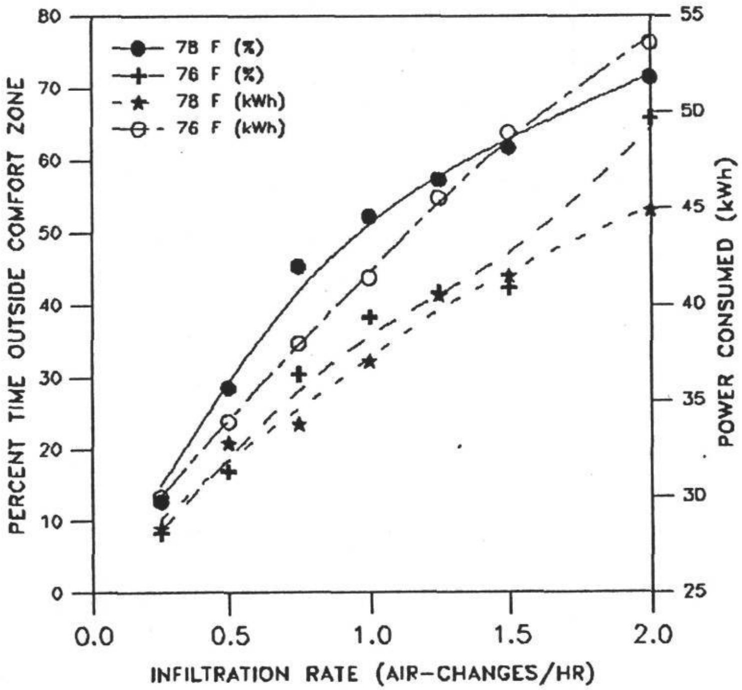

Figure 8.3 - Percent Outside the Comfort Zone and Power Consumed at Various Infiltration Rates on Hot/Humid Day with a 6 F Dead Band.

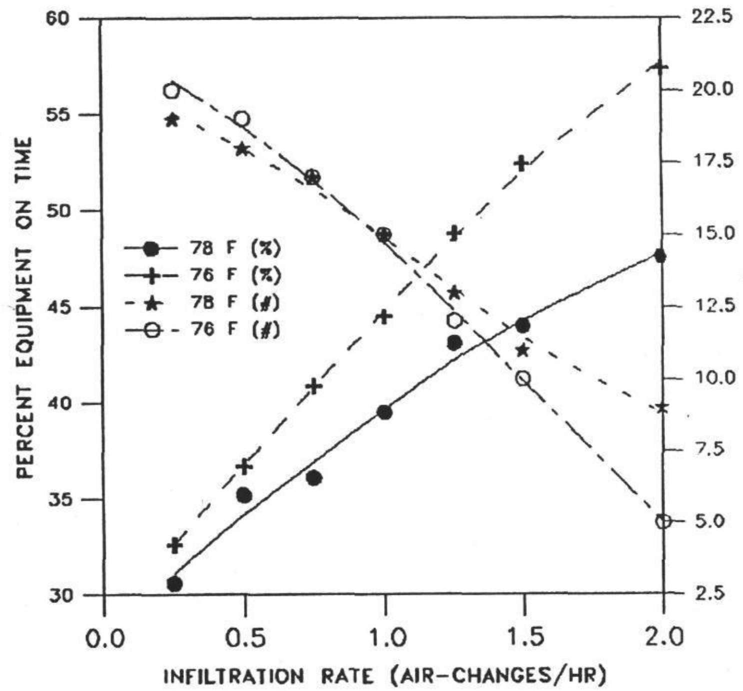

Figure 8.4 - Percent Equipment ON Time and Number of ON/OFF Cycles at Various Infiltration Rates on Hot/Humid Day with a 6 F Dead Band.

Table 8.5 - Sensitivity of Results to Infiltration
at the Set Point Temperature of 78 F.

INFILTRATION RATE AIR-CHANGES/HR	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	POWER 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 8.6 summarizes the time spent outside the comfort zone, equipment ON time, power consumed and number of ON/OFF cycles for the unit SJ at the set point temperature of 76 F.

Table 8.6 - Sensitivity of Results to Infiltration
at the Set Point Temperature of 76 F.

INFILTRATION RATE AIR-CHANGE/HR	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	POWER CONSUMED	NUMBER ON/OFF CYCLES
2.00	65.8	57.4	53.6	5
1.50	42.4	52.4	48.9	10
1.25	41.7	48.8	45.5	12
1.00	38.8	44.5	41.4	15
0.75	30.6	40.9	38.0	17
0.50	16.9	36.7	33.7	19
0.25	8.3	32.6	30.0	20

As seen earlier, the time spent outside the comfort zone decreased with decrease in set point temperature regardless of the rate of infiltration.

SIZING OF UNITS

Design Day

The influence of sizing of unit on the dehumidifying performance of CAC units was studied. A series of simulations were performed by varying the rated capacity of the unit SJ from 70 to 130 % of the design load of the test building. The simulations were repeated for two set point temperatures (78 & 76 F) and two dead bands (3 & 6 F). All the other variables (infiltration, external and internal loads, etc.) were constant at base case values.

Important hourly simulation data are shown in Tables B.49 - B.80 in Appendix B. The summary of the results at set point temperature of 78 F and 3 F dead band are shown in Table 8.7.

Table 8.7 - Sensitivity of Results to the Capacity of the Unit at the Set Point Temperature of 78 F and Dead Band of 3 F.

CAPACITY (%)	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	POWER CONSUMED (kWh)	NUMBER ON/OFF CYCLES
70	14.2	49.1	33.5	27
85	14.1	40.3	33.5	38
100	14.3	34.3	33.5	44
115	14.2	29.4	33.0	50
130	14.4	26.0	33.0	54

The time spent outside the comfort zone did not change with capacity, also the power consumption was constant for all capacities. The results for the set point temperature of 78 F and dead band of 6 F are shown in Table 8.8.

Table 8.8 - 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	POWER 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

The time spent outside the comfort zone increased with increase in capacity and corresponding to that increase there was slight decrease in power consumption. The results for the set point of 76 F and dead band of 3 F are shown in Table 8.9.

Table 8.9 – Sensitivity of Results to the Capacity of the Unit at the Set Point Temperature of 76 F and Dead Band 3 F.

CAPACITY (%)	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	POWER CONSUMED (kWh)	NUMBER ON/OFF CYCLES
70	6.2	60.3	37.5	18
85	10.1	46.8	35.4	31
100	10.5	37.5	35.6	45
115	10.3	34.9	35.8	47
130	10.6	30.6	35.5	52

The time outside the comfort zone did not change with the change in capacity with exception to the capacity of 70%. The power consumption also remained constant. The results for the set point temperature of 76 F and 6 F dead band are shown in table 8.10.

Table 8.10 – Sensitivity of Results to the Capacity of the Unit at the Set Point Temperature of 76 F and Dead Band 6 F.

CAPACITY (%)	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	POWER CONSUMED (kWh)	NUMBER ON/OFF CYCLES
70	4.7	69.4	42.9	3
85	8.1	56.8	42.9	8
100	16.9	36.7	33.9	19
115	10.7	35.6	35.5	23
130	10.1	31.2	36.2	23

The plot of percent time outside comfort zone and power consumed versus the ratio of the capacity of the unit to the design load for two set point temperatures and dead bands is shown in Figures 8.5 and 8.6. There is no significant change in the time spent outside the comfort zone with the capacity of the unit, for both 78 F and 76 F set points at the dead band of 3 F; however, there was a slight increase in time spent outside the comfort zone with capacity for both set point temperatures at 6 F dead band. The affect of capacity on comfort is not significant because of high average outdoor dry–bulb temperature and low average relative humidity on the design day. When the ambient temperatures are higher than the set point the equipment will run longer; therefore, the unit will extract more humidity from the zone. The power consumption remained constant at all capacities at 3 F dead band; however, there is a slight increase in power

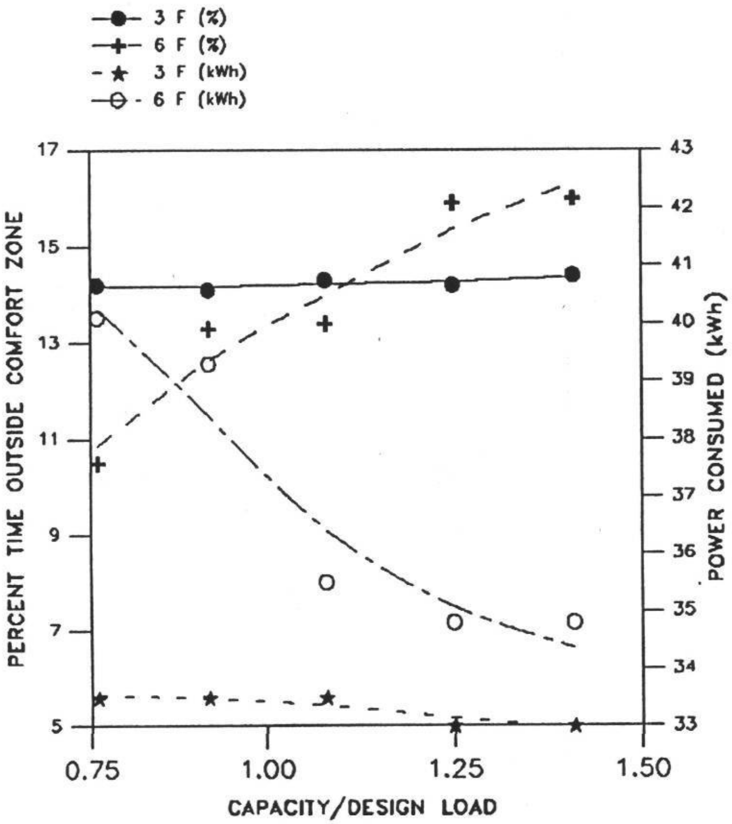

Figure 8.5 – Time Outside Comfort Zone and Power Consumed versus Capacity/Design Load on Design Day and 78 F Set Point Temperature.

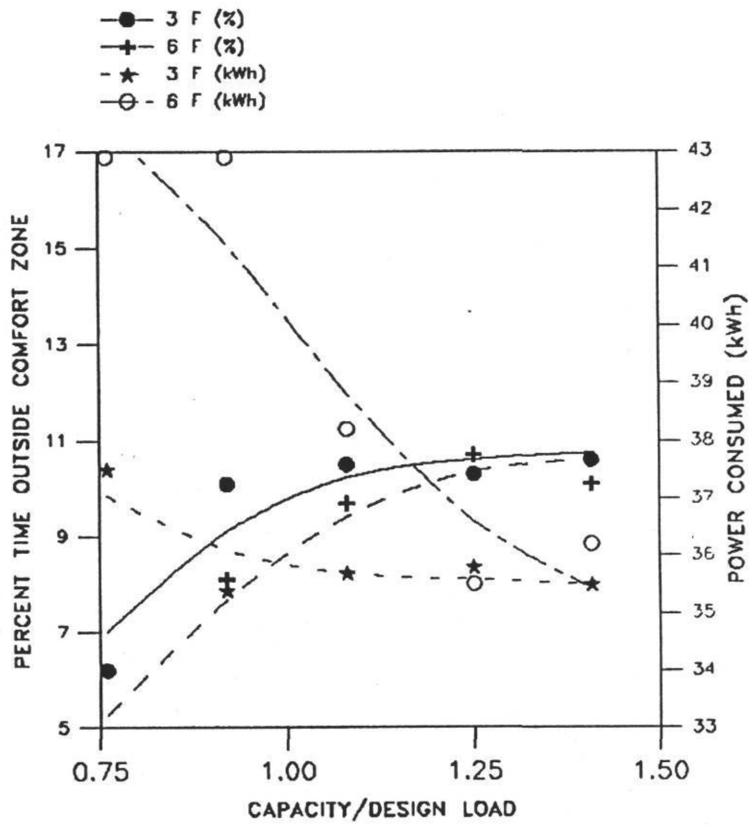

Figure 8.6 – Time Outside Comfort Zone and Power Consumed versus Capacity/Design Load on Design Day and 76 F Set Point Temperature.

consumption (10 %) for 6 F dead band. The ON/OFF cycles increased with the capacity in all cases.

The minutes outside the comfort zone in each hour are shown in Figure 8.7 and 8.8 for set point temperatures of 78 F and 76 F, respectively. Most of the discomfort time occurred between 6 and 9 a.m. The average ambient dry-bulb temperature and relative humidity during this period is 82.3 F and 78.2%, respectively. Since the outdoor dry-bulb temperatures are lower between 6 and 9 a.m., the unit comes ON for a very short time. Also, the relative humidities are much higher than the average relative humidity; therefore, the dehumidifying process is not complete. The outdoor dry-bulb temperatures between 9 a.m. and 11 p.m. are higher, so the unit runs longer. The relative humidities are lower, therefore, the time spent outside the comfort zone is insignificant. This is true even for 76 F set point temperature except the time spent outside the comfort zone is reduced.

The time spent outside the comfort zone for 76 F set point is lower because the difference between the average outdoor dry-bulb temperature and the set point is increased. Thus, the run time of the compressor is increased and more moisture is extracted from the conditioned space.

Hot/Humid Day

The same series of simulations were repeated to study the affect of sizing on the hot/humid day. All the other parameters were constant at base case values. The important hourly simulation data are shown in Tables B.81 - B.112 in Appendix B and the summary of the results for the set point temperature of 78 F and dead band of 3 F are shown in Table 8.11.

Table 8.11 - Sensitivity of Results to the Capacity of the Unit at the Set Point Temperature of 78 F and Dead Band 3 F.

CAPACITY (%)	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	POWER CONSUMED (kWh)	NUMBER ON/OFF CYCLES
70	30.3	44.7	29.8	27
85	33.2	36.3	29.4	36
100	34.0	30.6	29.2	40
115	36.7	26.0	28.6	45
130	35.4	23.5	29.2	48

There was a slight increase in time spent outside the comfort zone with

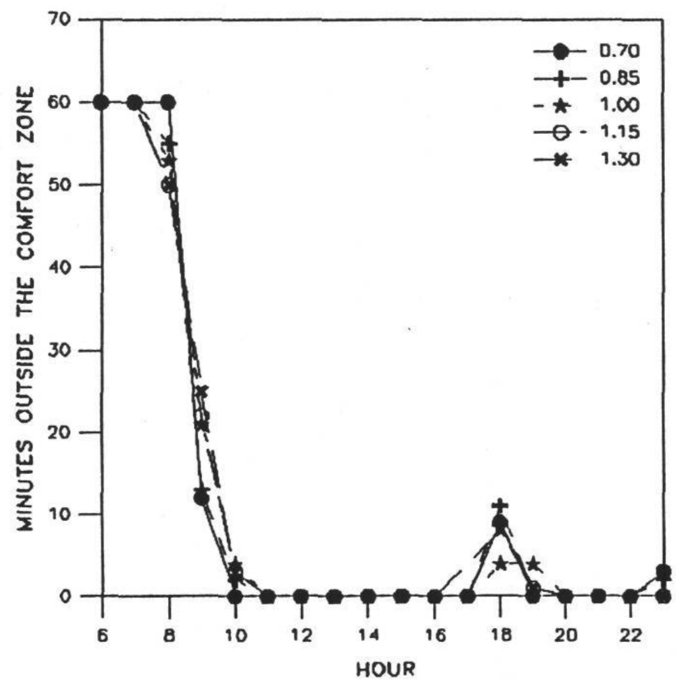

Figure 8.7 – Minutes Outside the Comfort Zone at 78 F Set Point Temperature and with a 3 F Dead Band on Design Day.

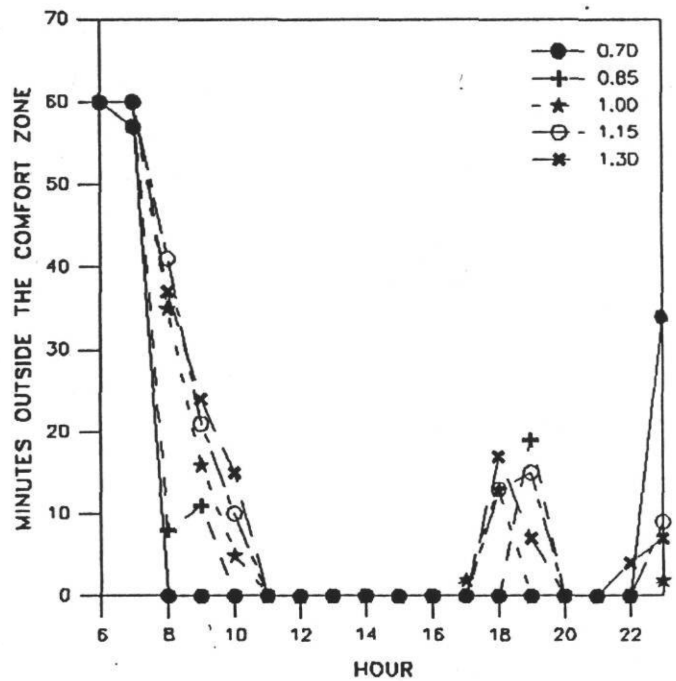

Figure 8.8 – Minutes Outside the Comfort Zone at 76 F Set Point Temperature and with a 3 F Dead Band on Design Day.

capacity; however, the power consumption was almost constant. The results for the set point temperature of 78 F and dead band of 6 F are shown in Table 8.12.

Table 8.12 - Sensitivity of Results to the Capacity of the Unit at the Set Point Temperature of 78 F and Dead Band 6 F.

CAPACITY (%)	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	POWER CONSUMED (kWh)	NUMBER ON/OFF CYCLES
70	14.2	61.7	35.7	3
85	20.9	46.5	34.5	10
100	28.5	35.2	32.8	18
115	29.5	31.3	31.5	19
130	31.2	27.4	31.3	20

The change in time outside the comfort zone with capacity is significant. It increased from 14.2% at 70% capacity to 31.2% at 130% capacity. However, corresponding to this increase there was a decrease in power consumption from 35.7 kWh at 70% to 31.3 kWh at 130% capacity. The results for the set point temperature of 76 F and dead band of 3 F are shown in Table 8.13.

Table 8.13 - Sensitivity of Results to the Capacity of the Unit at the Set Point Temperature of 76 F and Dead Band 3 F.

CAPACITY (%)	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	POWER CONSUMED (kWh)	NUMBER ON/OFF CYCLES
70	17.0	51.3	31.2	23
85	16.0	43.3	32.0	32
100	16.8	33.8	31.8	41
115	16.9	31.5	31.6	44
130	17.6	27.4	31.1	48

The change in time spent was not significant with capacity, also the power consumption was constant for all the capacities. The results for the set point temperature of 76 F and dead band of 3 F are shown in Table 8.14.

Table 8.14 - Sensitivity of Results to the Capacity of the Unit at the Set Point Temperature of 76 F and Dead Band 6 F.

CAPACITY (%)	PERCENT OUTSIDE COMFORT ZONE	PERCENT EQUIP. ON TIME	POWER CONSUMED (kWh)	NUMBER ON/OFF CYCLES
70	12.3	65.5	39.5	2
85	11.9	51.3	37.8	9
100	16.9	36.7	33.9	19
115	18.5	33.3	33.4	20
130	19.6	29.3	33.2	21

The time spent outside the comfort zone increases as the capacity of the unit increases for both 78 and 76 F set points at 6 F dead band. The units with the lower capacity run longer than the units with higher capacity; therefore, the lower capacity units remove more humidity from the zone and spend less time outside the comfort zone. Although the time outside the comfort zone decreases as the capacity of the units is decreased, the power consumption decreases with the increasing capacity. However, the decrease in time outside the comfort zone in greater than the increase in the power consumption. Decreasing the set point temperature from 78 to 76 F increases the unit ON time; therefore, the time spent outside the comfort zone decreases.

Increasing the dead band on the thermostat from 3 to 6 F at 78 F set point reduced the time spent outside the comfort zone. The reduction in the time spent outside the comfort zone is significant (53%) at lower capacities as shown in Figure 8.9. However, the decrease in time spent outside the comfort zone is only 27% when the the dead band was increased at 76 F set point temperature (Figure 8.10).

The time spent outside the comfort zone in every hour for 78 and 76 F set point temperatures (3 F dead band) is shown in Figures 8.11 and 8.12. The ambient dry-bulb temperatures are low and relative humidities high between 6 a.m. and 12 noon. Therefore, a large fraction of the time the comfort conditions were not achieved for the capacities considered. The same plots for 6 F dead band are shown in Figures 8.13 and 8.14. There is more fluctuation of the time spent outside the comfort zone. Also, the time spent outside the comfort zone was considerably reduced.

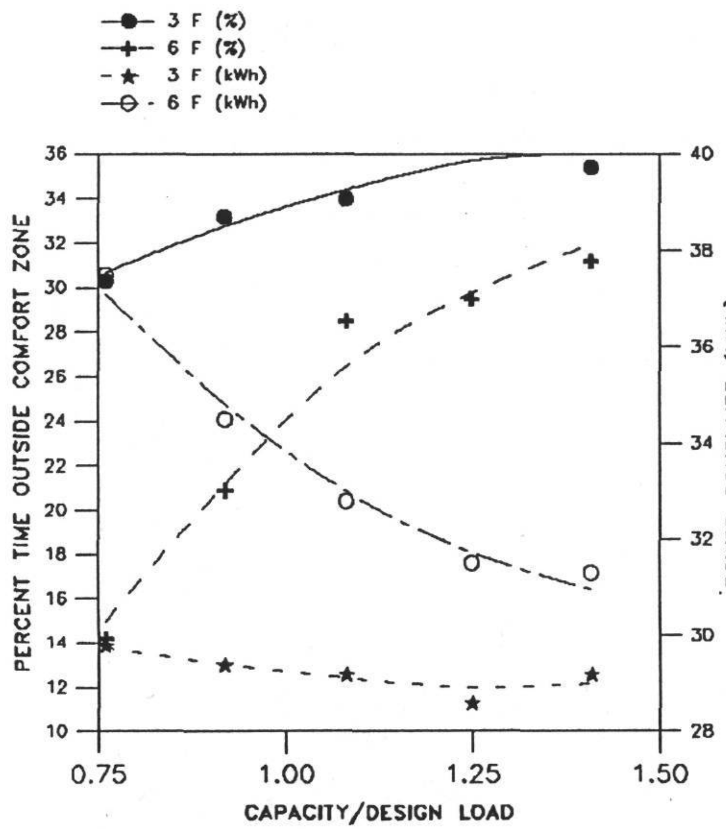

Figure 8.9 – Time Outside Comfort Zone and Power Consumed versus Capacity/Design Load on Hot/Humid Day at 78 F Set Point Temperature.

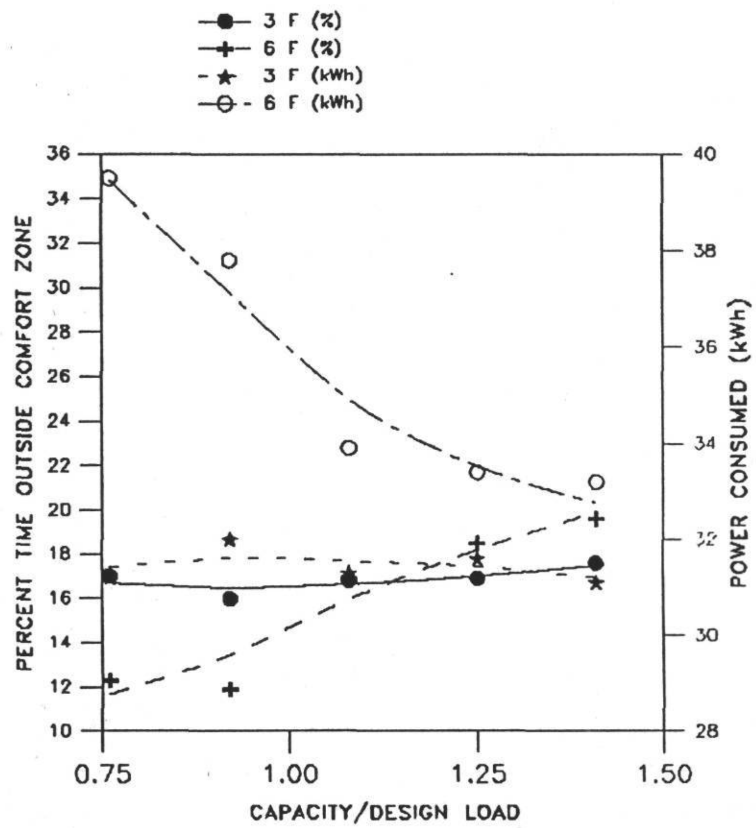

Figure 8.10 – Time Outside Comfort Zone and Power Consumed versus Capacity/Design Load on Mild/Humid Day at 78 F Set Point Temperature.

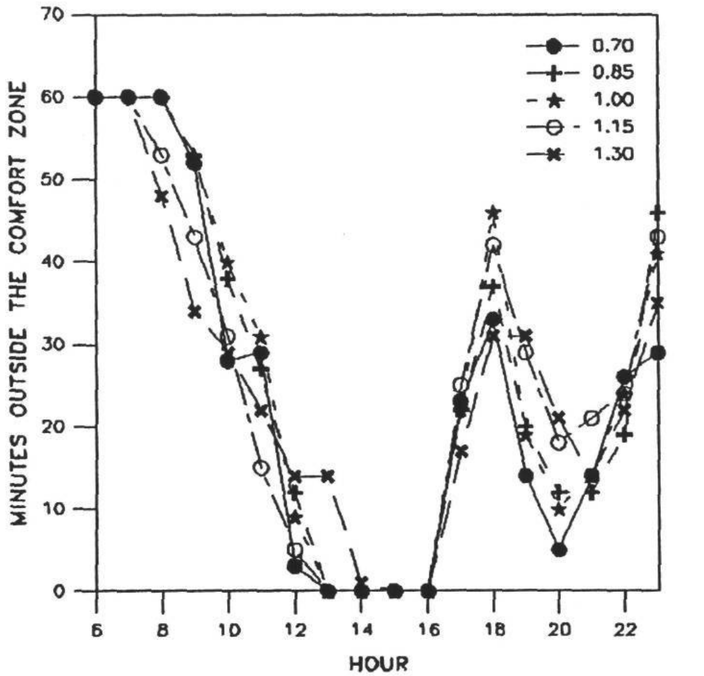

Figure 8.11 - Minutes Outside the Comfort Zone at 78 F Set Point Temperature and with a 3 F Dead Band on Hot/Humid Day.

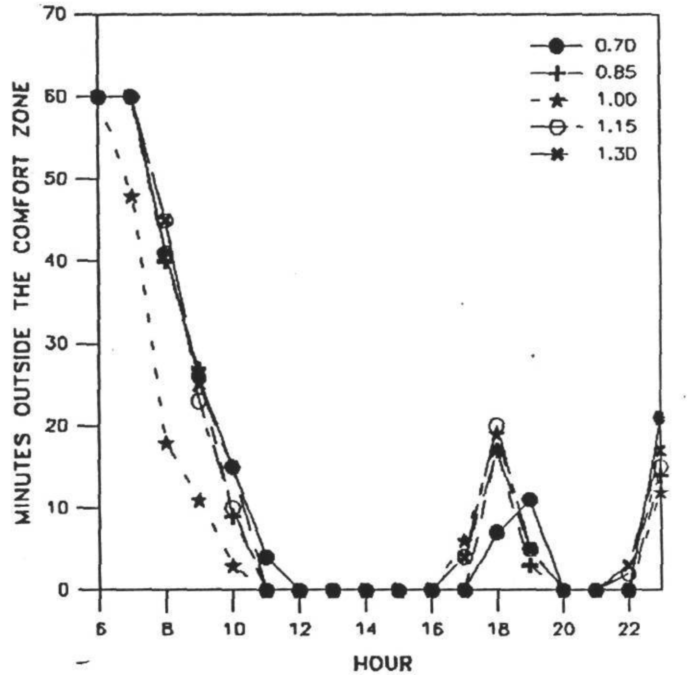

Figure 8.12 - Minutes Outside the Comfort Zone at 76 F Set Point Temperature and with a 3 F Dead Band on Hot/Humid Day.

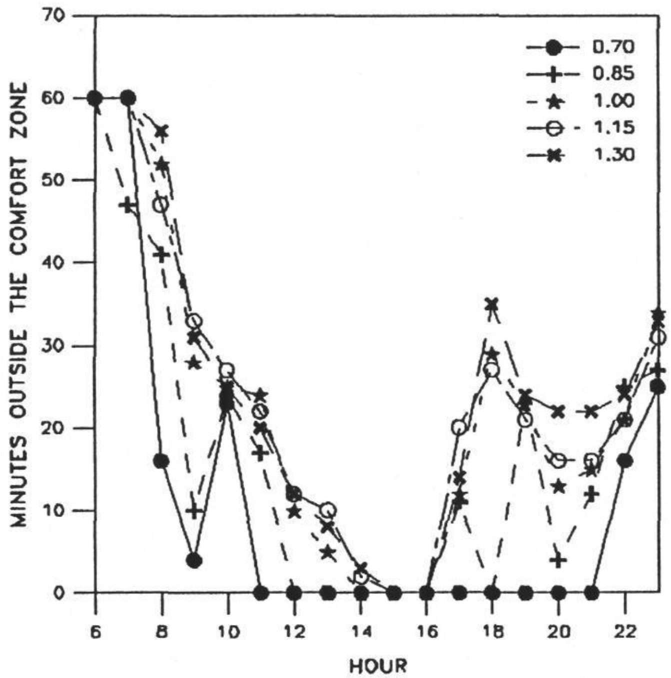

Figure 8.13 - Minutes Outside the Comfort Zone at 78 F Set Point Temperature and with a 6 F Dead Band on Hot/Humid Day.

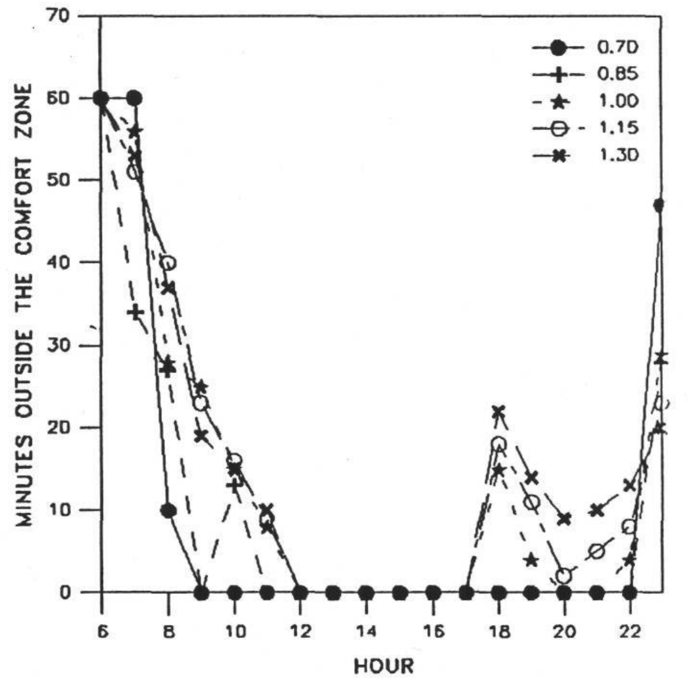

Figure 8.14 - Minutes Outside the Comfort Zone at 76 F Set Point Temperature and with a 6 F Dead Band on Hot/Humid Day.

DISCUSSION OF SENSITIVITY ANALYSIS

The external loads (weather) influence a great deal the time spent outside the comfort zone. The unit SJ spent 14% of the time outside the comfort zone on the design day compared to 58% on the mild/humid day. The primary reason for a 300% increase in time spent outside the comfort zone is the increase in ambient specific humidity ratio which means an increase in the latent cooling load. Decreasing the set point temperature from 78 to 76 F reduced the time spent outside the comfort zone and increased the power consumption. Increasing the dead band from 3 F to 6 F decreased the time spent outside the comfort zone for 78 F set point. Another benefit from increasing the dead band is reduction in number of ON/OFF cycles. Since the compressor life is based on the number of ON/OFF cycles, a unit operating on a 3 F dead band will deteriorate faster than a unit on a 6 F dead band.

Infiltration is another parameter which plays a major role by increasing the latent load on CACs. A house with a properly sized unit will spend more than 60% of the time outside the comfort zone if the infiltration rate exceeds 1 air-change/hr. Although the test house was sized using acceptable load calculations, it appears that under sizing the unit capacity would infact reduce the time spent outside the comfort zone. Over sizing the unit increased time spent outside the comfort zone. Although under sizing the units decreased the time spent outside the comfort zone, there was an increase in power consumption. The increase in power consumption was smaller than the decrease in the time outside the comfort zone.

CHAPTER 9

EVALUATION OF TWO-SPEED CAC UNITS

In recent years, manufacturers have placed a high priority on improving the energy efficiency of residential CACs. One method of improving the CAC efficiency has been to increase the heat exchanger surface area on both the evaporator and the condenser. Another method of improving the efficiency is by operating the compressor at a variable speed.

The two-speed residential CAC units have SEERs as high as 15. The major feature of these units is a two-speed compressor with a two-stage indoor thermostat corresponding to low and high speed operations. The unit operates on low speed under light and medium loads and at high speed for heavy loads. The two-stage thermostat is driven by the indoor dry-bulb temperature; therefore, the compressor will operate at high speed only if the sensible load is high.

In this chapter the dehumidifying performance of seven two-speed residential CAC units are evaluated. The design ratings of the seven units are shown in Table 9.1. The capacity is at 95 F outdoor dry-bulb entering the compressor, 67 F indoor wet-bulb and 80 F dry-bulb temperature entering the evaporator. The SHR for the units is a steady state value at the same conditions as the capacity.

Table 9.1 - Design@ Specification for the Selected Two-Speed CAC Units.

UNIT	CAPACITY BTUH (High/Low)	FLOW RATE CFM (High/Low)	SEER	SHR (High/Low)
TA	39,000/22,800	1200/725	10.00	0.65/0.68
TB	43,000/24,000	1400/850	10.25	0.73/0.75
TC	44,000/25,000	1400/850	10.50	0.72/0.77
TD	36,400/24,400	1200/1200	12.15	0.75/0.94
TE	38,900/25,500	1200/1200	12.45	0.75/0.95
TF	40,600/26,200	1200/1200	15.00	0.74/0.95
TG	41,200/26,400	1200/1200	13.30	0.74/0.96

@ ARI Standard 210

The performance of all units was evaluated for their ability to maintain conditions in the specified comfort zone. Simulation was performed for two typical summer days (design & hot/humid), two indoor set point temperatures (76 & 78 F) and with a 6 F dead band on the two-stage thermostat. The rated capacity of all the units was scaled down to match the design load of the test house (32.5 kBtu). The infiltration rate, internal loads, etc., were at the base case values. The base values were: (i) infiltration rate of 0.5 air-changes per hour, (ii) internal loads as shown in Figures 4.2 and 4.3, and (iii) sizing the equipment to the design load. The time spent outside the comfort zone for the two-speed CAC units is the ratio of the number of minutes the units spent outside the comfort zone in the entire day (24 hours) to the total number of minutes in a day (1440), while the single speed units were evaluated only between 6 a.m. and 11 p.m. The two-speed units had to be studied on a 24 hour basis because of fewer ON/OFF cycles due to the long periods of low speed compressor operation. The time outside the comfort zone between 6 a.m. and 11 p.m. will be summarized in the tables so that the performance of the single speed units can be compared to that of two-speed units. The comparisons between two two-speed units, however, will always be made on a 24 hour basis.

DESIGN DAY

The performance of the seven two-speed units was studied for two set point temperatures for the design day. The important hourly results are tabulated in Tables B9.1 through B9.28 in Appendix B. The summary of the results for units with a 78 F set point temperature are shown in Table 9.2.

Table 9.2 - Performance of Two-Speed Units at 78 F Set Point Temperature and a Dead Band of 6 F on a Design Day.

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

@ 12 a.m. to 11 p.m.
* 6 a.m. to 11 p.m.

The time spent outside the comfort zone increased with SHR for all the units (Figure 9.1). 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 power consumption, in general, decreased with the increasing SHR at low speed. The units TA - TC had only one ON/OFF cycle, while the

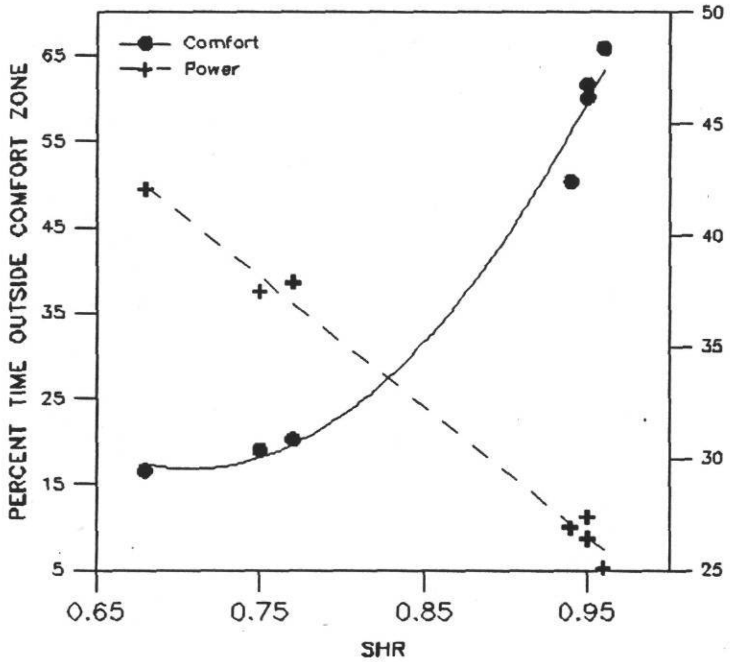

Figure 9.1 - Percent Outside the Comfort zone and Power Consumption vs SHR (low speed) at 78 F Set Point Temperature on the Design Day.

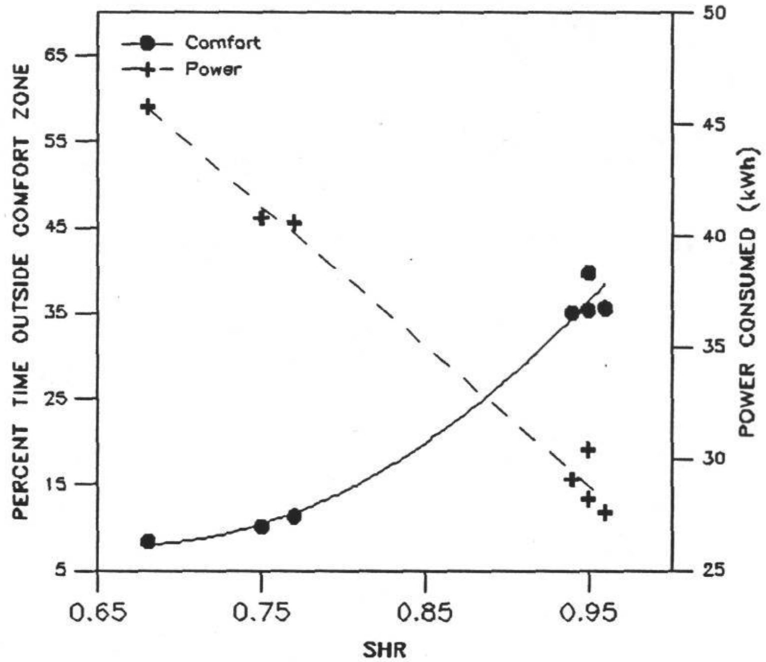

Figure 9.2 - Percent Outside the Comfort zone and Power Consumption vs SHR (low speed) at 76 F Set Point Temperature on the Design Day.

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.

The summary of results for units with a 76 F set point are shown in Table 9.3.

Table 9.3 - Performance of Two-Speed Units at 76 F Set Point Temperature and a Dead Band of 6 F on a Design Day.

UNIT	SHR (High/Low)	PERCENT OUTSIDE COMFORT ZONE		PERCENT EQUIP. ON TIME		POWER CONSUMED (kWh)		NUMBER ON/OFF CYCLES	
		A	B	A	B	A	B	A	B
TA	0.65/0.68	8.3	8.0	91.8	69.8	45.8	35.5	1	1
TB	0.73/0.75	10.1	8.6	91.3	69.9	40.8	32.0	1	1
TC	0.72/0.77	11.3	8.9	90.6	70.1	40.6	32.0	1	1
TD	0.75/0.94	35.1	15.8	69.9	61.7	29.1	25.4	5	5
TE	0.75/0.95	39.7	20.4	71.2	62.4	28.2	24.4	5	5
TF	0.74/0.95	35.4	17.6	74.0	62.9	30.4	25.6	5	5
TG	0.74/0.96	35.6	19.0	74.1	64.9	27.6	24.2	4	4

The time spent outside the comfort zone increased with SHR at low speed for all units with exception to the unit TE (Figure 9.2). The average ON time for the units TA - TC was about 91%, and for the units TD - TG it was 72%. The power consumption, in general, decreased with increase in the SHR at low speed. The units TA - TC had only one ON/OFF cycle, while the units TD - TG had 5 ON/OFF cycles.

Two variables which influence the time spent outside the comfort zone are: (i) SHR of the unit and (ii) ON time of the unit. Decreasing the set point temperature from 78 to 76 F increased the ON time for all the units. Also, the SHRs for the units with a 76 F set point temperature are lower than the units with a 78 F set point temperature (Figure 9.3 and 9.4). The SHR decreases with decreasing dry-bulb temperature of the return air. Since the average dry-bulb temperature of the return air is lower for the unit with a 76 F set point temperature, the SHRs are also lower; hence, the latent capacity for the units with a 76 F set point temperature is higher than that of the units with a 78 F set point temperature. The units with 76 F set point temperature remove more moisture from the conditioned space, because of higher ON time and lower SHRs. The power consumption for the units with a 76 F set point temperature increased when compared to the units with a 78 F set point temperature.

HOT/HUMID DAY

The performance of all the seven two-speed units was studied for two set point temperatures on the hot/humid day. The summary of the results for the units with a 78 F set point temperature are tabulated in Table 9.4.

Table 9.4 - Performance of Two-Speed Units at 78 F Set Point Temperature and a Dead Band of 6 F on a Hot/Humid Day.

UNIT	SHR (High/Low)	PERCENT OUTSIDE COMFORT ZONE		PERCENT EQUIP. ON TIME		POWER CONSUMED (kWh)		NUMBER ON/OFF CYCLES	
		A	B	A	B	A	B	A	B
TA	0.65/0.68	38.6	20.4	74.4	69.7	35.7	33.5	1	1
TB	0.73/0.75	49.8	28.3	73.1	70.0	31.2	29.9	1	1
TC	0.72/0.77	42.4	19.9	71.3	70.1	31.7	31.2	1	1
TD	0.75/0.94	51.3	26.3	50.3	59.9	24.3	24.1	3	3
TE	0.75/0.95	57.3	32.3	59.7	59.7	22.8	23.8	3	3
TF	0.74/0.95	63.6	38.6	63.2	63.2	25.2	25.2	3	3
TG	0.74/0.96	71.0	46.0	61.5	61.5	22.5	22.5	3	3

The time spent outside the comfort zone for all the units increased with SHR at low speed (Figure 9.5). The average ON time for the units TA - TC was 73% and for the units TD - TG it was 59%. The power consumption decreased with increasing SHR at low speed. The unit TA - TC had one ON/OFF cycle, while the units TD - TG had three.

All the units on the hot/humid day spent more time outside the comfort zone

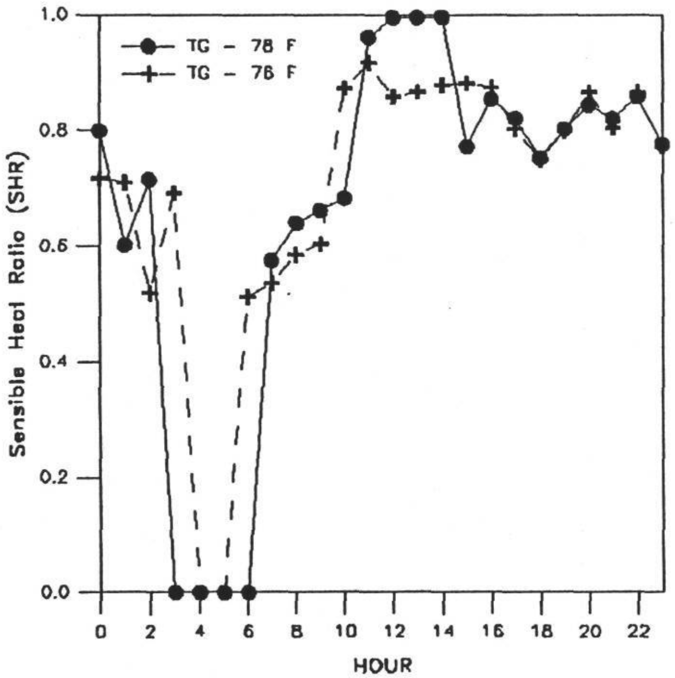

Figure 9.3 - Comparisons of SHRs for the Unit TG at 78 and 76 F Set Point Temperatures on the Design Day.

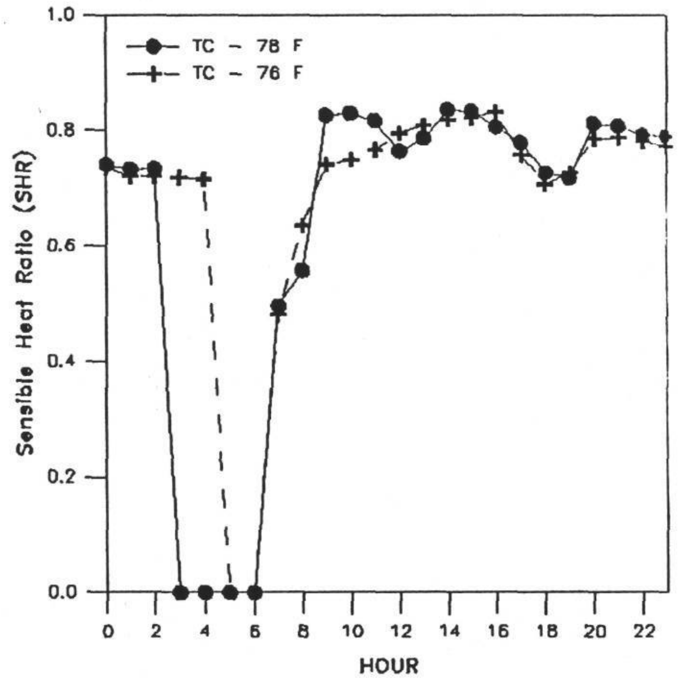

Figure 9.4 - Comparisons of SHRs for the Unit TC with 78 F and 76 F Set Point Temperatures on the Design Day.

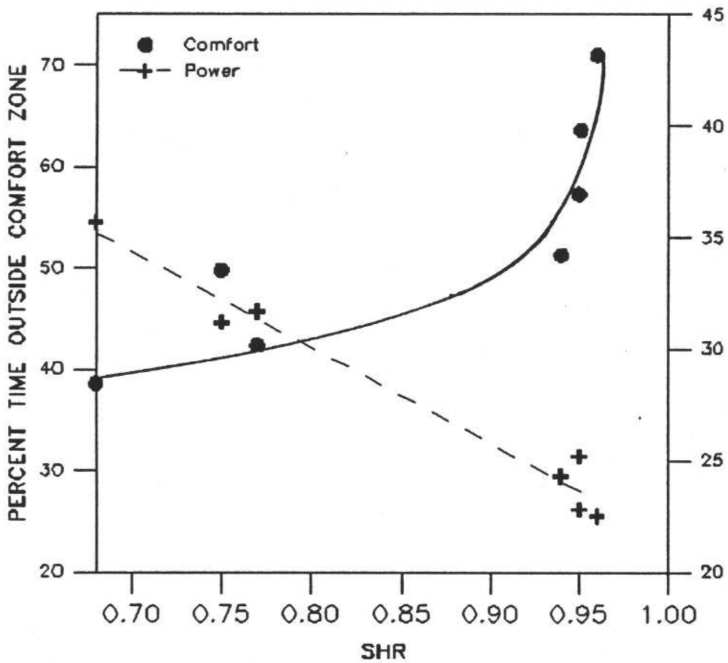

Figure 9.5 – Percent Outside the Comfort zone and Power Consumption vs SHR (low speed) at 78 F Set Point Temperature on the Hot/Humid Day.

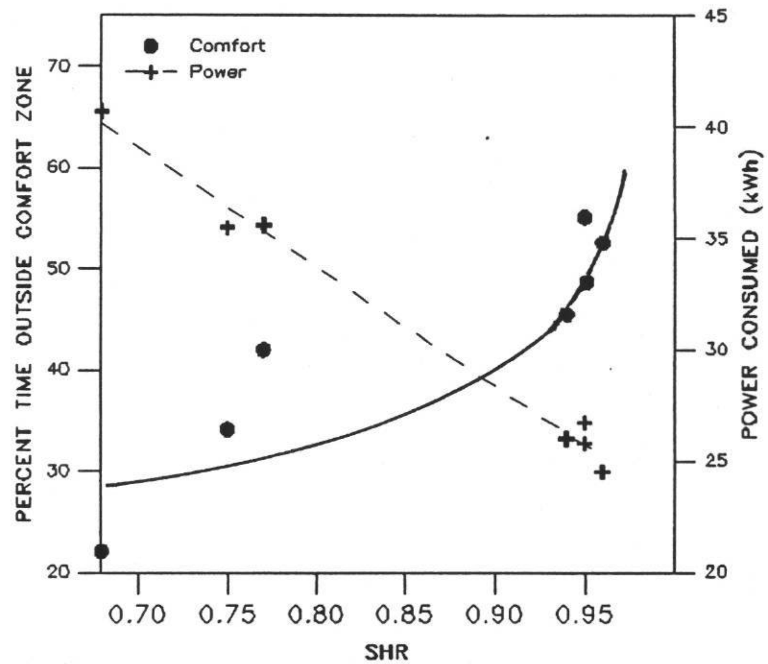

Figure 9.6 – Percent Outside the Comfort zone and Power Consumption vs SHR (low speed) at 76 F Set Point Temperature for the Hot/Humid Day.

compared to the units on the design day. The increase in time spent outside the comfort zone was significant for the units TA – TC (100%), and for the units TD – TG the increase was 2 – 5%. The moisture gains on the hot/humid day are significantly higher than on the design day; therefore, the time spent outside the comfort zone increased.

The ON time of the units decreased (13%) on the hot/humid day compared to the units on the design day. The decrease in the ON time was due to lower ambient dry-bulb temperatures on the hot/humid day. The power consumption was also reduced due to lower ON time for all the units.

The summary of the results for the units with a 76 F set point temperature on the hot/humid day are shown in Table 9.5.

Table 9.5 – Performance of Two-Speed Units at 76 F Set Point Temperature and a Dead Band of 6 F on a Hot/Humid Day.

UNIT	SHR (High/Low)	PERCENT OUTSIDE COMFORT ZONE A	PERCENT OUTSIDE COMFORT ZONE B	PERCENT EQUIP. ON TIME A	PERCENT EQUIP. ON TIME B	POWER CONSUMED (kWh) A	POWER CONSUMED (kWh) B	NUMBER ON/OFF CYCLES A	NUMBER ON/OFF CYCLES B
TA	0.65/0.68	22.1	15.4	85.1	69.7	40.7	33.5	1	1
TB	0.73/0.75	34.1	35.0	83.7	70.0	35.5	29.8	1	1
TC	0.72/0.77	42.0	29.0	80.3	70.4	35.6	31.3	1	1
TD	0.75/0.94	45.5	21.6	65.3	64.4	26.0	25.8	3	3
TE	0.75/0.95	55.1	30.4	67.3	64.3	25.8	24.6	3	3
TF	0.74/0.95	48.7	26.7	66.9	64.5	26.7	25.7	3	3
TG	0.74/0.96	52.6	26.3	66.6	63.3	24.5	23.2	3	3

The time spent outside the comfort zone increased with the SHR at low speed with exception to unit TE (Figure 9.6). The average ON time for the units TA – TC was 83% and for the units TD – TG it was 66%. The power consumption decreased with increasing SHR at the low speed.

All the units show a decrease in time spent outside the comfort zone compared to the units with a 78 F set point temperature. In comparison to the units with a 76 F set point temperature on the design day, all the units showed a significant increase (40 - 275%) in the time spent outside the comfort zone. Again, this was primarily due to an increase in the moisture gains through infiltration, and partly due to reduced ON times.

DISCUSSION OF RESULTS FOR THE TWO-SPEED UNITS

A series of simulations were performed to assesses the dehumidifying performance of the high efficiency two-speed residential CACs. The units were evaluated on their ability to maintain conditions in the specified comfort zone in a typical residence in Houston, Texas.

The minute-by-minute simulation was performed for two typical summer days; because the primary concern was only with dehumidifying performance of residential CACs rather than annual energy consumptions. Based on the design cooling load seven two-speed CACs units were selected and their performance was analyzed.

Although the SHR at the design conditions at high speed for all the seven two-speed units appear to be reasonable, these units spent considerable amounts of time outside the comfort zone; because all the units were operating at low speed for 90% of the ON time. The units TD - TG have very high SHRs at low speed; therefore, these units spent more than 50% of the time outside the comfort zone even on design day (at 78 F set point). The plots for time outside the comfort zone versus SHR shown in Figures 9.1,9.2,9.5 and 9.6 are only probable trends; since the selection of units were made from more than one manufacturer the exact relationship between comfort and SHR is difficult. However the time outside the comfort zone in almost all cases increased with increasing SHR.

The hot/humid day has higher specific humidity levels than the design day. Since an increase in specific humidity increases the latent loads from infiltration, the time spent outside the comfort zone also increases. Therefore, on a day with mild ambient dry-bulb temperatures and high specific humidity levels, the time spent outside the comfort zone will increase further.

In almost all cases the two-speed units spent considerably more time outside the comfort zone as compared to the single speed unit SJ. However, the power consumption of the two-speed units was lower than single speed units. Finally, the results indicate a strong correlation between the SHR at low speed and the time spent outside the comfort zone for all the seven units.

CHAPTER 10

CONCLUSIONS AND RECOMMENDATIONS

In recent years, air conditioning manufactures have placed a high priority in improving the efficiency of split system central air conditioners. Units with SEERs as high as 15 are now available in the market. The manufactures are improving their efficiency by increasing the surface area of the heat exchanger on both evaporator and the condenser and/or running the compressor at variable speed.

The results from this study did not indicate any relation between comfort and SEER; however, the relation between comfort and SHR was very strong for the single speed units. Also, the correlation between SHR and SEER was insignificant.

Most of the single speed units with rated SHRs less than 0.75 considered for this study performed adequately on hot and hot/humid days; however, the units with rated SHRs between 0.75 and 0.80 may not adequately dehumidify on mild/ humid days. The units with SHRs greater than 0.80 spent considerable time outside the comfort zone even on hot days; therefore, these units may not perform satisfactorily in Houston and Galveston climates.

In addition to SHR the variables that influence the comfort include: (i) the set point temperature, (ii) dead band on the thermostat, (iii) external load, (iv) infiltration and (v) sizing of the CAC units.

The simulated results indicated that by decreasing the set point temperature and/or increasing the dead band on thermostat, will improve comfort level; however, the power consumption increased in both cases. Infiltration not only increases the sensible but causes an additional burden on the latent capacity of the unit. If the infiltration rate is more than 1 air-change/hr, a properly sized CAC unit with reasonable good SHR would spent more than 50% of the time outside the comfort zone during the day. The results also indicated that undersizing the CAC units slightly (80% of the design load) would improve the comfort level and also reduce the peak demand.

The simulated results of the two–speed units indicated a strong correlation between comfort and SHR at low speed. The relation between SEER and comfort was also strong. The units having lower indoor flow rates at low speed performed well on the design day; however, their performance on hot/humid day was not satisfactory. The units which had constant flow rates at both the high and low speed did not perform well even on the design day.

If the actual test results are comparable with the simulated results obtained in this study, then it appears that: (i) the single speed units with SHR greater than 0.80 and (ii) most of the two–speed units will not adequately dehumidify in hot/humid climates. Therefore, these units may not provide the required comfort for Houston or Galveston areas.

The data used for the simulation of the units were steady state values provided by the manufacturers. In order to get a clear picture of the transient dehumidification performance of the CAC units it is recommended that actual cycling dehumidification performance should be studied in a tightly controlled environment.

Also the results from this study should be experimentally verified. The verification can be done by selecting a few typical residence in Houston area and measuring variables such as: infiltration, indoor dry–bulb temperature and humidity, power consumption, ON time of units, etc. in order to determine their effects on comfort levels.

Finally, it must be pointed out that TRNSYS software and the additional subroutines developed for this study can also be used to simulate the transient performance of residential heat pumps, both under cooling and heating modes. Also studying the transient performance of the heat pumps in a controlled environment is recommended. Such a study would illustrate the potential of the heat pumps and significantly enhance the understanding of the heat pump operation.

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TRNSYS INPUT FILE

```
SIM 1 48 .0166
WIDTH 72
*READING INPUTS  DRAD,AMB TEMP.,HUMR,WIND SPEED:
UNIT 9 TYPE 9 DATA READER
PAR 7
4 1 -1 1 0 -1 1
(T14,F5.0,1X,F4.1,F6.4,F3.1)
*CONVERSION SI TO BTU FOR OUTPUT ONLY:
UNIT 1 TYPE 15
PAR 14
-11 -12 1 -4 -13 -14 1 -15 3 -4 -16 -17 1 -4
INP 7
9,1 0,0   9,2 0,0  0,0  9,4  0,0
 0. 0.95   0.  1.8  32.  0.  2.237
* OUTPUTS DRAD TEMP HUMR WIND SPEED
UNIT 2 TYPE 25
PAR 4
1 25. 48 1
INP 4
1,1 1,2 9,3 1,3
 0.  0.  0.  0.
*RADIATION PROCESSOR 204(DESIGN),225(HOT/HUMID),233(MILD/HUMID):
UNIT 16 TYPE 16
PAR 7
3 1 225 29.7 4871 0 -1
INP 12
9,1  9,19  9,20  0,0 0,0 0,0 0,0 0,0  0,0 0,0  0,0 0,0
 0.  0.     0.    0.2 90. 45. 90. 135. 90. 225. 90. -45.
* PEOPLES SCHEDULES FOR TYPICAL RESIDENTIAL HOUSE:
UNIT 4 TYPE 14
PAR 12
0. 3. 8. 3. 8. 1.2 17. 1.2 17. 3. 24 3.
*LIGHTING LOAD SCHEDULES:
UNIT  5  TYPE  14
PAR 18
0. 0. 6. 0. 7. 1500. 7. 700. 17. 700. 17. 2300. 23. 2300. 23. 0. 24. 0.
*EQUIPMENT LOAD SCHEDULES:
UNIT  6  TYPE  14
PAR 28
0. 1500. 7. 1500. 7. 4000. 8. 4000. 9. 2500. 11. 2500 12. 4000. 13. 2500.
17. 2500. 18. 3500. 19. 5000. 22. 5000. 23. 3000. 24. 3000.
*LATENT LOAD SCHEDULES IN TERMS OF SP. HUMIDITY:
UNIT 7 TYPE 14
PAR  24
0. 0. 6. 0. 6. 1. 8. 1. 8. 0.025 17. 0.025 17. 1. 19. 1. 19. 0.25
20 0.25 20. 0.025 24. 0.025
*  DAILY SCHEDULE LOAD PROFILE FOR PEOPLE, LIGHT, EQUIPMENT,
*  AND LATENT LOAD:
UNIT  41  TYPE  41  DAILY LOAD PROFILE
PAR 8
4 1 1 1 1 1 1 1
INPUT 4
4,1  5,1  6,1  7,1
0.    0.   0.   0.
*CONVERSION OF LOAD SCHEDULES FROM KJ/HR TO BTU/HR FOR OUTPUT ONLY:
UNIT 10 TYPE 15
PAR 8
-11 -12 1 -4 -13 -12 1 -4
INP 3
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41,2 0,0  41,3
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 0. 0.95  0.
*PRINTING PEOPLE, LIGHTING, EQUIPMENT, AND LATENT LOAD SCHEDULES:
UNIT 11 TYPE 25
PAR 4
1 25. 48. 2
INP 4
41,1 10,1 10,2 41,4
 0.  0.   0.    0.
************DESCRIPTION OF THE HOUSE*****************
UNIT 19 TYPE 19
* ZONE DESCRIPTION
PAR 10
*TEMPERATURE CONTROL METHOD BASE CASE:
2 1 379 .50 0.0 0.0 2000. 11. 24 0.009
INP 11
9,2 9,3 42,2 42,1 42,3 41,4 41,1 0,0 41,2 41,3 9,4
0.   0.  0.   0.   0.   0.  0.   5.   0.    0.  0.
* WALLS
PAR 16
1 1 28.25 .7 .8 2 97 2 -1 31.22 3 -1 28.25 4 -1 31.22
INP 4
16,6 16,11 16,14 16,17
0.   0.    0.    0.
* FLOOR AND CEILING
PAR 14
10 2 156. .7 .8 3 16
11 2 156. .7 .8 3 19
* WINDOW
PAR 8
5 5 6 1  .8 30. 1 10
INP 5
16,6 16,7 0,0 0,0 0,0
 0.   0.   .8 12.3 1
*
PAR 8
6 5 1.86 1 .8 30. 1 10
INP 5
16,6 16,7 0,0 0,0 0,0
0.   0.  .8 12.3 1
*
PAR 8
7 5 4.75 1 .8 30. 1 10
INP 5
16,11 16,12 0,0 0,0 0,0
0.   0.   .8   12.3 1
*
PAR 8
8 5 7.2  1 .8 30. 1 10
INP 5
16,14 16,15 0,0 0,0 0,0
 0.   0.   .8   12.3 1
*
PAR 8
9 5 4.75 1 .8 30. 1 10
INP 5
16,17 16,18 0,0 0,0 0,0
0.   0.     .8  12.3 1
* VIEW FACTORS
PAR 41
1 2.43 11.58 12.8 3 4 1 2 10 11 5
```

```
5 1 3.05 .61 1.52 3.96
6 1 6.    0. 2.03 .914
7 2 3.05 .61 1.52 3.12
8 3 3.05 .61 1.52 4.72
9 4 3.05 .61 1.52 3.12
* PSYCHROMETRICS
UNIT 30 TYPE 33
PAR 3
4 1 1.0
INP 2
9,2 9,3
0.   0.
UNIT 33 TYPE 33
PAR 3
4 1 1.0
INP 2
19,1 19,2
0.    0.
* THERMOSTAT CONTROL (25.55[78 F] ,24.45 [76F])
UNIT 8 TYPE 8
PAR 6
4 0 19 24.45 20 20
INP 2
19,1 0,0
 0.   0.
* HVAC CONTROL AND SIMULATION:
UNIT 42 TYPE 42
PAR 16
* LENNOX  SINGLE SPEED UNITS
1 1200 0. .5 20. 9 4 3 4 2. 0. 24.45 1.6 1.20 0. 0.
* CARRIER SINGLE SPEED UNITS
* 2 1200 0. .5 20. 9 4 4 4 2. 0. 25.55 .8 .93 0. 0.
* TRANE SINGLE SPEED UNITS
* 2 1200 0. .5 20. 9 5 4 5 2. 0. 24.45 .8 .83 0. 0.
* LENNOX TWO SPEED UNITS
* 3 1200 0. .5 20. 9 4 3 4 2. 0. 24.45 1.6 .74 1. 0.
* TRANE TWO SPEED UNITS
* 4 850 0. .5 20. 9 5 4 5 2. 0. 25.55 1.6 .74 1. 1400
*
INP 7
8,3 9,2 30,2  9,3  19,1  33,2 19,2
 0. 0.   0.    0.   0.    0.   0.
*
*CONVERSION OF TOTAL SOLAR RADIATION TO BTU/HR
UNIT 12 TYPE 15
PAR 16
-11 -12 1 -4 -13 -12 1 -4 -14 -12 1 -4 -15 -12 1 -4
INP 5
16,6 0,0 16,11 16,14 16,17
 0.   0.95 0.    0.    0.
*
UNIT 15 TYPE 15
PAR 22
-11 -12 1 -13 3 -4 -14 -15 1 -4 -15 -16 1 -4 -15 -17 1 -4 -18 -19
1 -4
INP  9
19,1 0,0 0,0 19,3 0,0   19,4 19,5 19,6  0,0
 0. 1.8  32   0.  0.95  0.    0.   0.   7.898E-5
*
UNIT 14 TYPE 15
```

```
PAR 16
-11 -12 1 -4 -13 -14 1 -15 3 -4 -16 -14 1 -15 3 -4
INP 6
42,1 0,0  42,2  0,0   0,0 42,7
 0.  2.2    0.  1.8  32.   0.
*
UNIT 20 TYPE 28
PAR 17
1 24 47 -1 2 -11 -4 -12 -4 -13 -4 -14 -4 -15 -4 -16 -4
INP 6
15,1 19,2 15,2 15,3 15,4 15,5
*
*SLB/HR  STEMP SHUMR SCAP SHR POW:
UNIT 25 TYPE 25
PAR 4
0.0166 24. 47.984 6
INP 6
42,1 42,2 42,3 42,4 42,5 42,6
 0.   0.   0.   0.   0.   0.
*OUTPUT ROOM TEMP., ROOM HUM. RATIO, EQUI. ON/OFF:
UNIT 18 TYPE 25
PAR 4
0.0166 24. 47.984 5
INP 3
15,1 19,2 42,9
 0.   0.   0.
END
```