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GARD Project No. ASH330

ASHRAE 1254-RP EVALUATING THE ABILITY OF UNITARY EQUIPMENT TO MAINTAIN ADEQUATE SPACE HUMIDITY LEVELS, PHASE II

FINAL REPORT

Results of Cooperative Research between the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., and GARD Analytics, Inc.

Prepared for

Project Monitoring Subcommittee
ASHRAE Technical Committee TC 8.11
Unitary and Room Air Conditioners and Heat Pumps
(formerly TC 7.6 Unitary Air Conditioners and Heat Pumps)

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EXECUTIVE SUMMARY

The objectives of ASHRAE Research Project 1254-RP "Evaluating the Ability of Unitary Equipment to Maintain Adequate Space Humidity Levels, Phase II" were: a) to compare various unitary air conditioning system humidity control configurations in terms of humidity control performance and operating costs; and b) develop guidelines to help identify the important application characteristics and climate factors that determine which option is most appropriate. This research project builds on the Phase I project, ASHRAE 1121-RP, which was completed in June 2001 (Brandemuehl and Katejanekarn 2001). The Phase I final report describes the Evaluation Plan which guided this work.

The EnergyPlus whole-building energy simulation software (EnergyPlus 2005) was used to perform a parametric analysis of eighteen HVAC system types in seven commercial building types (Small Office, Large Retail, Classroom 9-month, Classroom 12-month, Restaurant Dining Area, Small Hotel/Motel Guest Room, and Theater) using two sets of ventilation rates (ASHRAE Standards 62-2001 and 62.1-2004) in 10 locations. To meet the needs of this project, new equipment models were developed and added to EnergyPlus to provide advanced modeling capabilities for multi-stage multi-mode DX cooling coils, and enhanced component configuration and control options. The systems types included single-path and dual-path DX with and without enhancements such as enthalpy wheel, demand controlled ventilation, desiccant dehumidifier, subcool reheat, hot gas reheat, and air-to-air heat exchangers around the cooling coil.

The relative performance of each system type was compared on the basis of humidity control (occupied hours >65% RH) and annual energy use, including heating energy. The systems were also compared on a life cycle cost basis using approximate installed equipment costs and HVAC annual energy costs. The following guidelines, issues, and conclusions resulted from this analysis.

<u>*Guidelines*</u>

- In nearly all cases, simple variations in the Base DX system (lower airflow, lower SHR) do little to improve humidity control but may be useful to save fan energy. The exception to this rule is Standard 2004 ventilation rates with the Retail application in the most humid climates.

- Demand controlled ventilation (DCV) saves energy, but does little to improve humidity control in most cases.

- Semi-active humidity control systems (Case 5 Subcool Reheat and Case 7 and Coil Bypass) can help but often fall short, especially in the most humid climates.

- Certain applications, such as the Theater, Restaurant and Motel, in very humid climates have high humidity issues primarily at times when there is no sensible load on the coil due to cool moist outside air. Only active humidity control systems (desiccants and reheat) can control humidity at such times. Depending on the control settings, enthalpy wheels may not operate at such times, and therefore provide less benefit for humidity control.

- For all of the systems without direct humidity control (all cases except desiccant Cases 8 and 14 and reheat Case 17), system capacity vs. load profile is crucial. The poor humidity control performance of many of these system options can be attributed primarily to a high percentage of hours operating at low part loads. 2-stage systems with a 60% stage 1 capacity help significantly, but do not overcome this issue. Case 6 Base DX w/o Latent Coil Degradation represents the ideal in capacity staging where the coil never evaporates condensed moisture back into the supply air stream.

- For the Office, humidity control is not an issue.

- For the Restaurant, Theater, and Schools, systems with direct humidity control (desiccant Cases 8 and 14 and reheat Case 17) are the only systems which can provide adequate humidity control in the most humid climates. In less humid climates, enthalpy wheel systems (Cases 9 and 12) can also provide adequate control.

- For the Motel, continuous operation and single-stage equipment result in excessive hours of high humidity. Only Case 14 Dual path w/Desiccant provides adequate (or near-adequate) humidity control in the most humid climates. Reheat and dual path systems can help significantly, and are sufficient in moderate climates.

- For the Retail Store, a wider range of options can be beneficial.

- The enthalpy wheel and DCV options generally provide equal or better humidity control compared to the base system, with significant energy cost and life cycle cost savings. Significantly better humidity control (but not necessarily adequate control) is found in the Restaurant with the 2004 Standard, Retail with both standards, and School with both standards. Worse humidity control is found in the Restaurant and Theater in certain locations.

Issues
The results of this analysis raise several issues for further investigation:

- Would adequate capacity staging solve humidity control problems in all but the most extreme cases? Case 6 Base DX w/o Latent Coil Degradation results show that better staging might help in cases with moderate humidity control issues, but it makes little difference in the Theater, Restaurant, School, and Motel in the most humid climates. .

- Do the dual path systems in this analysis perform better because they are dual path, or simply because they have four stages of cooling available in the outside air stream? Would the same four-stage system in a single path unit provide similar results?

- For some applications in high humidity climates, there are times when a zero SHR is required, because humidity is high but there is no need for sensible cooling. This requires a system such as hot gas reheat, essentially a dehumidifier. How much of the total system cooling capacity is needed at these times? Would it be more cost effective to add a small dehumidifier in the outside air stream?

- Fan power issues are significant. Would generally lower fan cfm/ton be beneficial if combined with adequate capacity staging to improve humidity control and save energy? How can the year-round fan power penalty of some of these systems be minimized?

- The outdoor air preconditioning system was not the typical application. This should be examined in combination with subcool or hot gas reheat.

- Would alternative desiccant dehumidifier configurations, such as placing the desiccant wheel after the DX cooling coil, provide adequate humidity control at lower costs and energy use?

- Additional data mining may reveal trends related to design SHR, ventilation load index, or other defining characteristics of the loads.

Conclusions
This research project has provided the following benefits:

- Comprehensive analysis of humidity control performance of a wide range of DX system configurations.

- Significant advancement in whole building energy simulation capabilities for modeling DX equipment by adding new capabilities to EnergyPlus. This provides access to designers and analysts to study specific projects and extend the results of this analysis.

- Identification of key issues for further exploration to better understand some of the key drivers and possibly develop some simple new system configurations that can efficiently control humidity.

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Hugh Henderson, CDH Energy Corporation, Chair
Charles Bullock, Carrier Corporation
Elizabeth Jones, ARI
Doug Kosar, University of Illinois at Chicago (Chair, TC 8.12)
Leon Tang, Outokumpu

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1 Background

The ASHRAE technical program often reflects the significant concerns faced by the HVAC design community about humidity control in buildings. Typically, there are several symposiums, seminars or forums related to the impacts of moisture and moisture related problems in buildings. And quite often these presentations or discussions will focus on ASHRAE Standards 62 and 90 and their contribution to the problem. Clearly the connection between moisture related problems and the increased ventilation mandated by the use of Standard 62 or by changes in HVAC system designs for energy conservation is widely suspected. Standard 90.1 and federal regulations such as NAECA and the 1992 Energy Policy Act have often seen raising efficiency levels on unitary cooling equipment being one way to save energy. Unfortunately, this can exacerbate the humidity control problem since manufacturers of unitary equipment may elect to increase the evaporator coil size. Requests have come from several ASHRAE members to the 90.1 committee that the efficiency levels on unitary equipment be reduced in order to boost the dehumidification effectiveness. These requests have been rejected but cost-effective energy-efficient alternatives that provide sufficient dehumidification have not yet been suggested.

Furthermore, according to the DOE/EIA Commercial Building Energy Consumption Survey for 1999, packaged air conditioning units cool 65% to 70% of the floor space built since 1980 and this fraction seems to be growing. In some specific applications, such as retail and restaurants, packaged air conditioning units serve 75% to almost 90% of the cooled floor space.

Many designers today feel that they are faced with a perplexing choice, either:

- Specify inexpensive unitary products to cool their building and expect humidity problems to follow, or

- Specify expensive chilled-water based systems.

In reality, designers have a third option,

- Unitary products with enhancements to improve dehumidification performance.

Unfortunately, little is known about this third option. Because the design process in specifying unitary equipment is focused on off-the-shelf product selection, little is usually “engineered” in the design besides the duct layout. Today designers have almost no guidance in making informed selections of unitary products concerning dehumidification. Many options exist today for better unitary dehumidification technologies including:

- Evaporator coils with more rows and lower air flow rates,
- Lower air flow rates,
- Air-to-air heat exchangers around the evaporator coil,
- Condenser reheat coil in series with evaporator,
- Prevention of re-evaporation by cycling the fan,
- Bypassing a fraction of the air flow around the coil,
- Pre-drying using active desiccant prior to the evaporator coil,

• Enthalpy recovery wheel using return air,

• Dedicated preconditioning DX system for outside air ,

• Separate outside air conditioning dual path system,

• Enthalpy recovery wheel with a separate outside air conditioning system,

• Air-to-air heat exchangers with a separate outside air conditioning system,

• Separate outside air conditioning with desiccant system,

• Reducing ventilation by using carbon dioxide monitoring (demand-controlled ventilation),

• Separate outside air conditioning with reducing the air flow by carbon dioxide monitoring.

Unfortunately, since little is known about these options, they tend to be specified only for niche applications even when they may be broadly applicable. Given the large number of options, it is difficult or impossible to choose which enhanced unitary technology is appropriate for a given application. This ASHRAE research project was initiated to improve our knowledge about these choices. By examining these applications from the perspective of what is the most cost effective way to control humidity, designers will be able to use the guidance provided to quickly select the equipment configuration for the specific building. Some anecdotal information exists regarding application of these enhancements, but until this comprehensive study was completed, no quantitative study across a wide range of available options existed.

2 Project Objective

This research project seeks to assess the ability of various HVAC technologies and systems to maintain proper space humidity levels while meeting the ventilation requirements of ASHRAE Standard 62.

The objectives of this ASHRAE research project were to:

1. Compare various unitary air conditioning system humidity control configurations for application to commercial buildings in terms of humidity control performance, operating costs, and lifecycle costs to each other as well as to conventional unitary equipment.

2. Develop guidelines to help HVAC engineers and practitioners identify the important application characteristics and climate factors that determine which option is most appropriate.

The technical approach was to use simulations to evaluate the ability of unitary equipment to maintain adequate space humidity levels and to develop design guidelines based on the results of this evaluation. Humidity control is a significant issue for a variety of reasons, including occupant comfort, microbial growth, and physical damage to buildings and equipment. This research project builds on the Phase I project ASHRAE 1121-RP which was completed in June 2001 (Brandemuehl and Katejanekarn 2001). The Phase I final report describes the Evaluation Plan which guided this work.

3 Scope of Work

During Phase I of this project an Evaluation Plan was developed which was to guide the work to be accomplished under Phase II. The Evaluation Plan identified combinations of commercial buildings, ventilation rates, locations and HVAC system configurations that were to be analyzed for their humidity control ability and operating cost effectiveness. With the approval of the Project Monitoring Subcommittee (PMS) the Evaluation Plan was adjusted and extended to better meet the goals of the project. The scope of work under Phase II included the following combinations which were analyzed using the EnergyPlus whole-building energy simulation software (EnergyPlus 2005):

- Seven commercial building types
  - Small Office
  - Large Retail
  - Classroom 9-month
  - Classroom 12-month
  - Restaurant Dining Area
  - Small Hotel (Motel) Guest Room
  - Theater

- Two ventilation rates
  - Based on Standard 62-2001 which is referenced by many building codes
  - Based on Standard 62.1-2004 which is the current version of the standard

- Ten locations

Atlanta, GA	New York, NY
Chicago, IL	Portland, OR
Dallas/Fort Worth, TX	Shreveport, LA
Houston, TX	St. Louis, MO
Miami, FL	Washington, DC

- Eighteen HVAC equipment options
  - Case 0 - Conventional DX System (typical HVAC design practice)
  - Case 1 - Base DX System (good dehumidification design practice)
  - Case 2 - DX Design for Improved Dehumidification (modified coil, compressor, etc.)
  - Case 3 - Base DX System with Lower Airflow
  - Case 4 - DX System with Air-to-Air Heat Exchanger (AAHX)
  - Case 5 - DX System with Subcooling Reheat Coil
  - Case 6 - Base DX System with No Latent Degradation
  - Case 7 - DX System with Airflow Control Using Bypass Damper
  - Case 8 - Hybrid DX and Desiccant System (condition outdoor air)
  - Case 9 - DX System with Enthalpy Recovery
  - Case 10 - DX System with Outdoor Air Preconditioning
  - Case 11 - DX Dual Path System (separate systems for outdoor and recirculated air)
  - Case 12 - DX Dual Path with Enthalpy Recovery
  - Case 13 - DX Dual Path with AAHX
  - Case 14 - DX and Desiccant Dual Path System (desiccant system for outdoor air)
  - Case 15 - DX System with Demand Controlled Ventilation
  - Case 16 - DX Dual Path System with Demand Controlled Ventilation
  - Case 17 – Base DX System with Free Reheat (hot gas reheat)

4 Weather Data

The Evaluation Plan called for simulations to be run in the following eight climates:

- Atlanta, GA
- Chicago (O'Hare Airport), IL
- Dallas/Ft. Worth (Airport)
- Miami (International Airport)
- New York (LaGuardia Airport)
- Portland, OR
- St. Louis, MO
- Washington, DC (the Evaluation Plan indicates National Airport, however the TMY2 data for DC is actually for Dulles Airport in Sterling, VA.)

Based on prior work, it seemed that two significant humid climate areas had been missed in this set. Figures 4.1 and 4.2 show the ASHRAE humid climate area map and a map of latent ventilation cooling load index (LVCLI) (Hedrick and Shirey 1998). Even though Houston falls in the same region as Miami on both maps, the distribution of temperature vs. humidity conditions shown in Figures 4.3 and 4.4 is dramatically different since Miami's dry-bulb temperatures are quite moderate. Houston was added to the suite in order to capture the impact of a very humid climate with a wide range of dry bulb temperatures.

The LVCLI varies from 6 in Atlanta to 18 in Miami. It is quite likely that a different set of humidity control options may be most cost-effective in the LVCLI 10-12 region than in either Atlanta or Miami. Figures 4.5 through 4.7 reveal that Atlanta and Dallas humidity levels top out in the range of 120 gr/lb, while Shreveport has a significant number of hours in the 130 gr/lb range. Shreveport, LA was therefore added to represent this moderately humid climate zone.

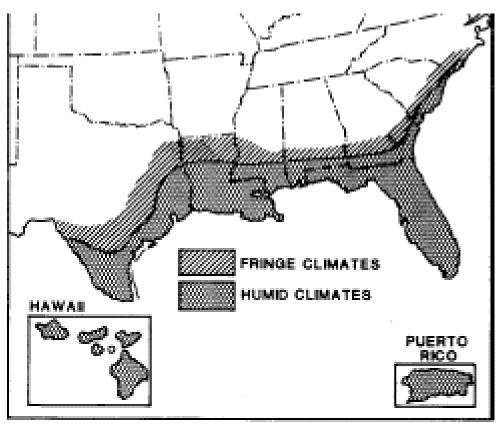

Figure 4.1 - ASHRAE Humid Climate Map (ASHRAE 1993)

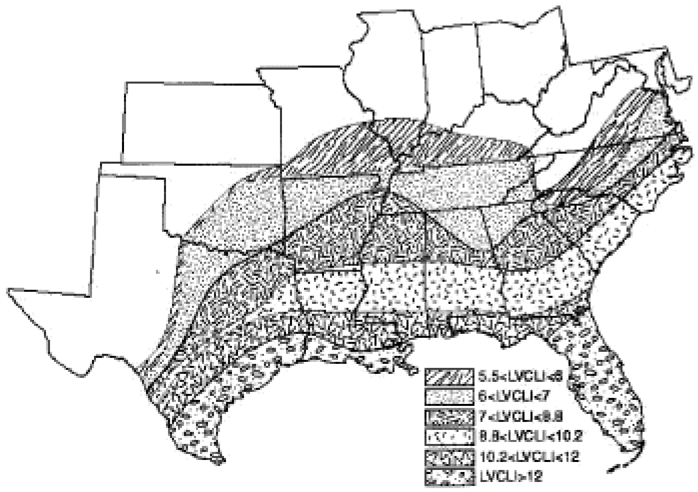

Figure 4.2 - Latent Ventilation Cooling Load Index Map (Hedrick and Shirey 1998)

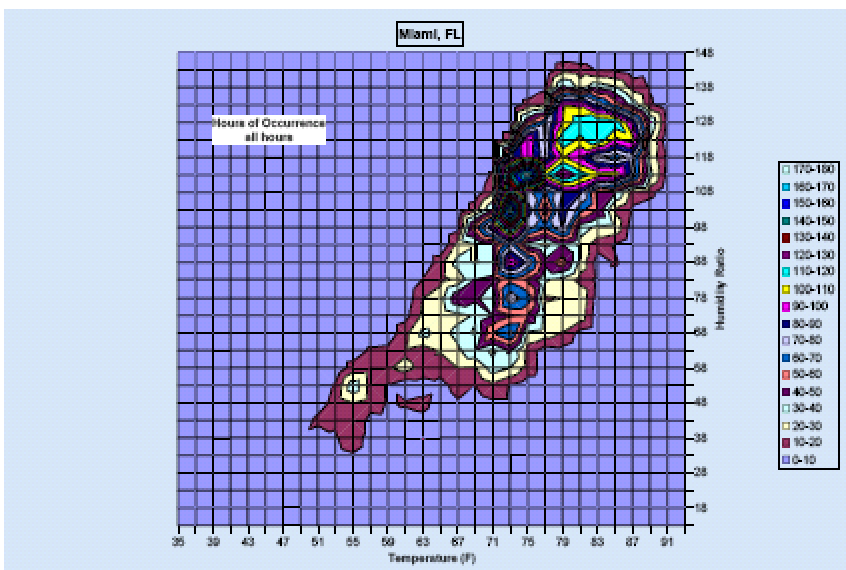

Figure 4.3 - Joint Frequency Chart for Miami, FL TMY2 Weather Data

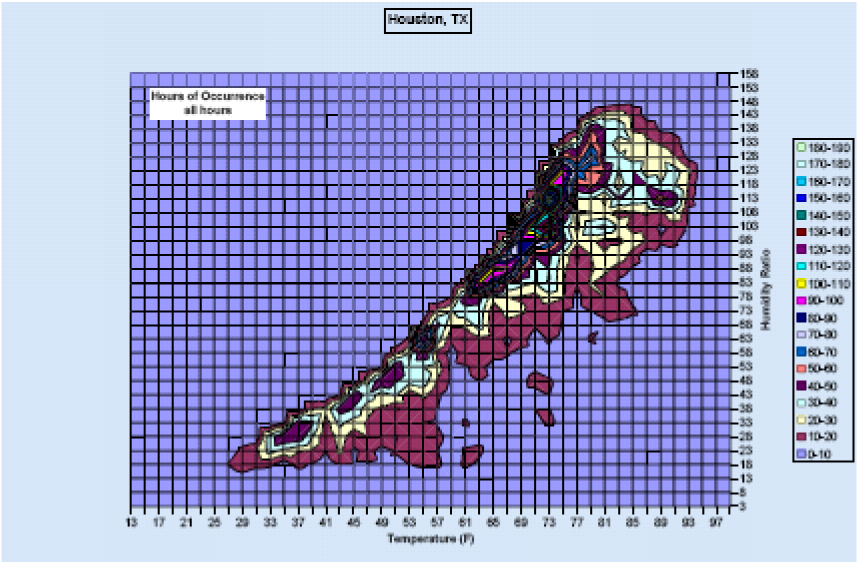

Figure 4.4 - Joint Frequency Chart for Houston, TX TMY2 Weather Data

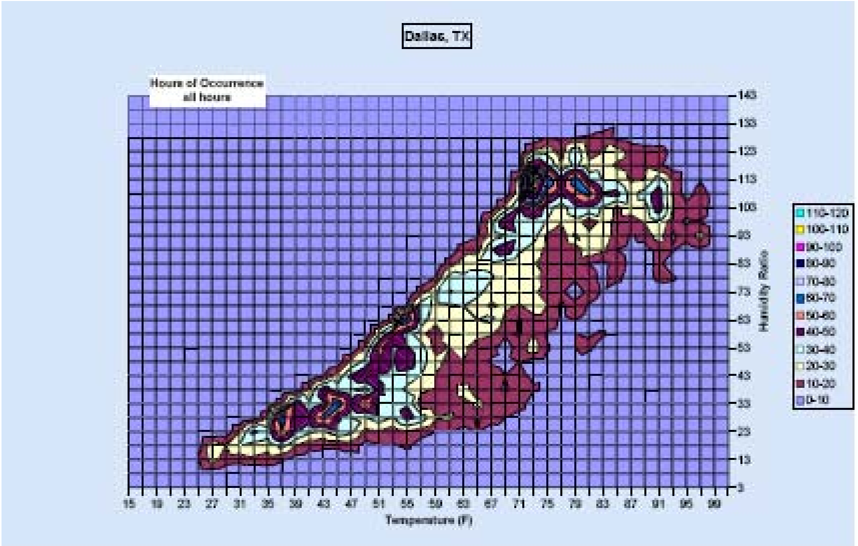

Figure 4.5 - Joint Frequency Chart for Dallas, TX TMY2 Weather Data

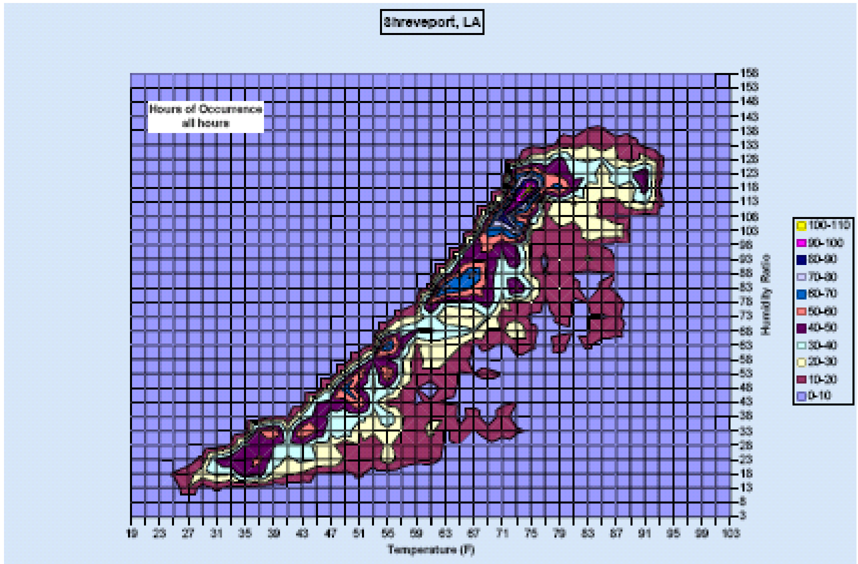

Figure 4.6 - Joint Frequency Chart for Shreveport, LA TMY2 Weather Data

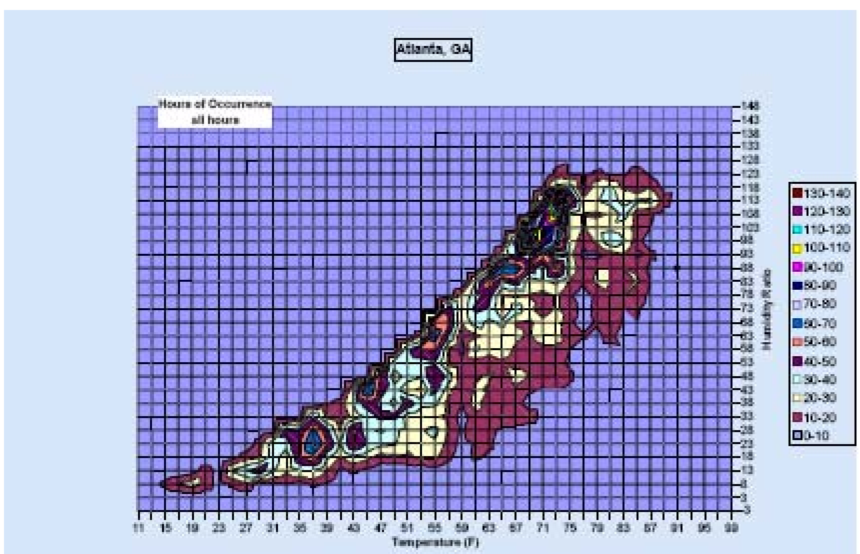

Figure 4.7 - Joint Frequency Chart for Atlanta, GA TMY2 Weather Data

Analyses were performed for all 10 U.S. locations which are representative of a broad range of climates and dehumidification challenges. Design load calculations for each location and application were done using the ASHRAE (ASHRAE 2001) 0.4% cooling design conditions. Hourly simulations for each location were performed using TMY2 weather files.

5 Building Models

EnergyPlus building models were constructed based on information presented in Table 3-1 of the Phase 1 Evaluation Plan as shown below with some modifications. The prototypical building physical characteristics in the Phase I Evaluation Plan were taken from a study done by Huang and Franconi of LBNL (Huang and Franconi 1999). The theater has been added and its characteristics were taken from a set of design drawings for an actual theater built within the last several years. Its envelope construction was considered to be similar to the large retail building.

Table 5.1 Prototype Building Characteristics

Characteristic	Small Office	Large Retail	Classroom	Restaurant Dining Room	Motel Room	Theater
Floor Area (ft2)	6,000	79,000	1,000	5,250	350	9,000
No. Floors	1	1	2	1	2	1
Ceiling Height (ft)	10	15	10	10	8	24
Percent Glass (%)	15	15	18	15	21	0
Window R-Value	1.6	1.7	1.7	1.5	1.71	1.7
Window SC	0.75	0.76	0.73	0.80	0.76	0.76
Wall R-Value	5.6	4.8	5.7	4.9	5.32	4.8
Roof R-Value	12.6	12.0	13.3	13.2	13.16	12.0
Wall Material	Masonry	Masonry	Masonry	Masonry	Masonry	Masonry
Roof Material	Built-up	Built-up	Built-up	Built-up	Built-up	Built-up

Each building was modeled in EnergyPlus as a single zone (square configuration) with the floor area as indicated with the following exceptions: the classroom was modeled as a single classroom of 1000 sq. ft. rather than a group of classrooms totaling 16,000 sq. ft. as originally specified and the motel room also was modeled as one room of 350 sq. ft. rather than a group of rooms totaling 12,000 sq. ft. in the original plan. The following additional assumptions were made:

1) Constructions do not vary between locations

2) Material layers and constructions were taken from LBNL DOE-2 models developed by Huang for a Gas Research Institute study referenced in the Evaluation Plan (Huang, et al, 1991). In some cases there was not a one-to-one correspondence between the building types from the LBNL/GRI study and the Evaluation Plan as indicated below.

1254-RP Bldg	LBNL/GRI Bldg
Small Office	Large Office
Large Retail	Large Retail
Classroom	Secondary School
Restaurant Dining Room	Sit Down Restaurant
Motel	Small Hotel
Theater	Large Retail

3) Constructions taken from the LBNL/GRI study and used for the 1254-RP buildings are as follows (layer names refer to those used in DOE-2 Materials Library dataset for EnergyPlus):

Large Office	Wall	(ST01,Insul,AL21,GP02)
	Roof	(BR01,CC24,Insul,AL33,AC02)
	Slab	(CC15)
Large Retail & Theater	Wall	(CC26,Insul,AL21,GP02)
	Roof	(BR01,CC25,Insul,AL33,AC02)
	Slab	(CC15)
Secondary School	Wall	(CB31,Insul,GP02)
	Roof	(BR01,PW04,Insul,AL33,GP02)
	Slab	(CC15)
Sit Down Restaurant	Wall	(BK04,PW03,Insul,GP02)
	Roof	(BR01,PW05,Insul,AL33,AC02)
	Slab	(CC14,LT01)
Small Hotel	Wall	(PW04,Insul,GP02)
	Roof	(AR02,PW05,Insul,AL33,AC02)
	Slab	(CC15,CP01)

Per e-mail communication with Joe Huang of LBNL, the Wall and Roof R-value shown in LBNL report is for the insulation layer only.

4) For the two buildings which are specified as 2 story (school and motel), the space was modeled as a first floor zone with only one wall exposed to outdoor conditions. The other walls and ceiling were considered to be interior and therefore adiabatic.

5) The PMS requested that for the school and motel two different exposures be examined – south and west. A series of sensitivity simulations were performed and the results examined. The zone relative humidity frequency occurrences for both exposures were almost identical for each application. The cooling equipment electric consumption changed less than 1% for the school in Atlanta. The cooling equipment electric consumption for the motel with the west exposure was 24% higher than the motel with the south exposure in Atlanta. Since the scope of work called for analysis of only one type of each application, the motel and school with a southern exposure were selected.

6) All buildings are assumed to have a square floor plan. For the office, retail and restaurant, windows are assumed to be on all four walls in the center of the wall. For the school and motel, there is a window only on one exterior wall. The theater has no windows.

7) Windows are clear double pane.

8) Internal load levels for each building type for lights and equipment were set as described in Table 5.2 below which comes directly out of the Evaluation Plan.

Table 5.2 Prototype Building Internal Loads

Characteristic	Small Office	Large Retail	Classroom	Restaurant Dining Room	Motel	Theater
Lighting						
Power (W/ft2)	1.7	1.6	1.8	2.1	1.06	1.0
Equipment						
Power (W/ft2)	0.5	0.4	0.8	0.0	0.69	0.0

9) Two sets of occupant density and ventilation rates were analyzed as represented in Standard 62-2001 (currently referenced by many building codes) and Standard 62.1-2004 (most current version). See Section 6 for further details regarding this.

10) Hourly operating schedules for occupancy, lighting and equipment were taken from Tables 3-3 through 3-5 of the Evaluation Plan and shown on the next two pages. In some cases the occupancy schedule specified seemed atypical, e.g., the weekend occupancy schedule for the office shows the building occupied for 4 hours from 9AM till 1PM, and the weekend occupancy schedule for the retail store shows occupancy for only 5 hours from noon till 5PM. They were left as is.

Occupancy Profiles

Hr	Office WD	Office WE	Retail WD	Retail WE	School WD	School Sat	School Sum	Restaurant WD	Restaurant WE	Motel WD	Motel WE
1	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1.00	0.80
2	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1.00	0.80
3	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1.00	0.80
4	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1.00	0.80
5	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1.00	0.80
6	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1.00	0.80
7	0.33	0.00	0.00	0.00	0.11	0.00	0.50	0.25	0.38	1.00	0.80
8	0.67	0.00	0.33	0.00	1.00	0.00	0.50	0.38	0.50	0.60	0.70
9	1.00	0.20	0.67	0.00	1.00	0.10	0.50	0.13	0.63	0.30	0.50
10	1.00	0.20	1.00	0.00	1.00	0.10	0.50	0.06	0.25	0.10	0.30
11	1.00	0.20	1.00	0.00	1.00	0.10	0.50	0.25	0.25	0.10	0.10
12	0.75	0.20	1.00	1.00	0.89	0.10	0.50	0.63	0.38	0.10	0.10
13	1.00	0.00	1.00	1.00	0.89	0.10	0.50	0.63	0.63	0.10	0.10
14	1.00	0.00	1.00	1.00	0.89	0.00	0.50	0.50	0.63	0.10	0.10
15	1.00	0.00	1.00	1.00	0.89	0.00	0.50	0.25	0.63	0.20	0.20
16	1.00	0.00	1.00	1.00	0.50	0.00	0.00	0.06	0.44	0.30	0.20
17	0.67	0.00	1.00	0.00	0.17	0.00	0.00	0.13	0.31	0.40	0.30
18	0.33	0.00	1.00	0.00	0.06	0.00	0.00	0.50	0.63	0.50	0.40
19	0.00	0.00	0.67	0.00	0.37	0.00	0.00	0.75	1.00	0.60	0.50
20	0.00	0.00	0.33	0.00	0.37	0.00	0.00	0.63	1.00	0.70	0.60
21	0.00	0.00	0.00	0.00	0.37	0.00	0.00	0.50	0.88	0.80	0.80
22	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.25	0.50	1.00	0.80
23	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.13	0.25	1.00	0.80
24	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1.00	0.80

Lighting Profiles

Hr	Office WD	Office WE	Retail WD	Retail WE	School WD	School Sat	School Sum	Restaurant WD	Restaurant WE	Motel WD	Motel WE
1	0.05	0.05	0.15	0.15	0.00	0.00	0.00	0.05	0.05	0.05	0.05
2	0.05	0.05	0.15	0.15	0.00	0.00	0.00	0.05	0.05	0.05	0.05
3	0.05	0.05	0.15	0.15	0.00	0.00	0.00	0.05	0.05	0.05	0.05
4	0.05	0.05	0.15	0.15	0.00	0.00	0.00	0.05	0.05	0.05	0.05
5	0.05	0.05	0.15	0.15	0.00	0.00	0.00	0.05	0.05	0.30	0.20
6	0.05	0.05	0.15	0.15	0.00	0.00	0.00	1.00	1.00	0.67	0.50
7	0.80	0.05	1.00	0.15	1.00	0.00	0.50	1.00	1.00	1.00	0.80
8	1.00	0.05	1.00	0.15	1.00	0.00	0.50	1.00	1.00	1.00	0.80
9	1.00	0.20	1.00	0.15	1.00	0.10	0.50	1.00	1.00	0.30	0.80
10	1.00	0.20	1.00	0.15	1.00	0.10	0.50	1.00	1.00	0.20	0.30
11	1.00	0.20	1.00	1.00	1.00	0.10	0.50	1.00	1.00	0.20	0.20
12	1.00	0.20	1.00	1.00	1.00	0.10	0.50	1.00	1.00	0.20	0.20
13	1.00	0.05	1.00	1.00	1.00	0.10	0.50	1.00	1.00	0.20	0.20
14	1.00	0.05	1.00	1.00	1.00	0.00	0.50	1.00	1.00	0.20	0.20
15	1.00	0.05	1.00	1.00	1.00	0.00	0.50	1.00	1.00	0.20	0.20
16	1.00	0.05	1.00	1.00	0.40	0.00	0.00	1.00	1.00	0.30	0.20
17	1.00	0.05	1.00	1.00	0.40	0.00	0.00	1.00	1.00	0.40	0.30
18	0.80	0.05	1.00	0.15	0.40	0.00	0.00	1.00	1.00	0.50	0.40
19	0.05	0.05	1.00	0.15	0.40	0.00	0.00	1.00	1.00	0.60	0.50
20	0.05	0.05	1.00	0.15	0.40	0.00	0.00	1.00	1.00	0.80	0.60
21	0.05	0.05	1.00	0.15	0.40	0.00	0.00	1.00	1.00	1.00	0.80
22	0.05	0.05	0.15	0.15	0.00	0.00	0.00	1.00	1.00	1.00	0.80
23	0.05	0.05	0.15	0.15	0.00	0.00	0.00	1.00	1.00	0.50	0.40
24	0.05	0.05	0.15	0.15	0.00	0.00	0.00	1.00	1.00	0.05	0.05

Equipment Profiles

	Office		Retail		School			Restaurant		Motel	
Hr	WD	WE	WD	WE	WD	Sat	Sum	WD	WE	WD	WE
1	0.05	0.05	0.30	0.30	0.00	0.00	0.00	0.00	0.00	0.00	0.00
2	0.05	0.05	0.30	0.30	0.00	0.00	0.00	0.00	0.00	0.00	0.00
3	0.05	0.05	0.30	0.30	0.00	0.00	0.00	0.00	0.00	0.00	0.00
4	0.05	0.05	0.30	0.30	0.00	0.00	0.00	0.00	0.00	0.00	0.00
5	0.05	0.05	0.30	0.30	0.00	0.00	0.00	0.00	0.00	0.30	0.20
6	0.05	0.05	0.30	0.30	0.00	0.00	0.00	0.00	0.00	0.67	0.50
7	0.80	0.05	1.00	0.30	0.10	0.00	0.10	0.00	0.00	1.00	0.80
8	1.00	0.05	1.00	0.30	0.50	0.00	0.10	0.00	0.00	1.00	0.80
9	1.00	0.20	1.00	0.30	1.00	0.10	0.10	0.00	0.00	0.50	0.80
10	1.00	0.20	1.00	0.30	0.50	0.10	0.10	0.00	0.00	0.10	0.10
11	1.00	0.20	1.00	1.00	0.50	0.10	0.10	0.00	0.00	0.10	0.10
12	1.00	0.20	1.00	1.00	0.50	0.10	0.10	0.00	0.00	0.10	0.10
13	1.00	0.05	1.00	1.00	1.00	0.10	0.10	0.00	0.00	0.10	0.10
14	1.00	0.05	1.00	1.00	0.50	0.00	0.10	0.00	0.00	0.10	0.10
15	1.00	0.05	1.00	1.00	0.50	0.00	0.10	0.00	0.00	0.10	0.10
16	1.00	0.05	1.00	1.00	0.10	0.00	0.00	0.00	0.00	0.10	0.10
17	1.00	0.05	1.00	1.00	0.10	0.00	0.00	0.00	0.00	0.20	0.30
18	0.80	0.05	1.00	0.30	0.10	0.00	0.00	0.00	0.00	0.30	0.40
19	0.05	0.05	1.00	0.30	0.10	0.00	0.00	0.00	0.00	0.30	0.50
20	0.05	0.05	1.00	0.30	0.10	0.00	0.00	0.00	0.00	0.40	0.60
21	0.05	0.05	1.00	0.30	0.10	0.00	0.00	0.00	0.00	0.80	0.80
22	0.05	0.05	0.30	0.30	0.00	0.00	0.00	0.00	0.00	0.80	0.80
23	0.05	0.05	0.30	0.30	0.00	0.00	0.00	0.00	0.00	0.70	0.80
24	0.05	0.05	0.30	0.30	0.00	0.00	0.00	0.00	0.00	0.00	0.00

11) Operating profiles for the theater were assumed to be as follows:

	Occupants		Lighting		Equipment	
Hr	WD	WE	WD	WE	WD	WE
1	0.00	0.10	0.00	1.00	0.30	0.30
2	0.00	0.00	0.00	0.00	0.30	0.30
3	0.00	0.00	0.00	0.00	0.30	0.30
4	0.00	0.00	0.00	0.00	0.30	0.30
5	0.00	0.00	0.00	0.00	0.30	0.30
6	0.00	0.00	0.00	0.00	0.30	0.30
7	0.00	0.00	0.00	0.00	1.00	0.30
8	0.00	0.00	0.00	0.00	1.00	0.30
9	0.00	0.00	0.00	0.00	1.00	0.30
10	0.00	0.10	0.00	0.00	1.00	0.30
11	0.00	0.10	0.00	1.00	1.00	1.00
12	0.00	0.40	0.00	1.00	1.00	1.00
13	0.00	0.70	0.00	0.20	1.00	1.00
14	0.10	0.70	1.00	0.20	1.00	1.00
15	0.10	0.70	1.00	0.20	1.00	1.00
16	0.40	0.90	0.20	0.20	1.00	1.00
17	0.40	0.90	0.20	0.20	1.00	1.00
18	0.70	1.00	0.20	0.20	1.00	0.30
19	0.70	1.00	0.20	0.20	1.00	0.30
20	0.70	1.00	0.20	0.20	1.00	0.30
21	0.70	1.00	0.20	0.20	1.00	0.30
22	0.30	0.70	0.20	0.20	0.30	0.30
23	0.10	0.50	1.00	0.20	0.30	0.30
24	0.00	0.30	0.00	0.20	0.30	0.30

12) Two different types of school operations were studied – traditional school year which runs 9 months (September through May) and year-round school year which runs 12 months with several vacation breaks during the year. For the year-round school, the following schedule was assumed (taken from Virginia Beach City Public Schools):

Jan 1 through March 28	classes
March 29 through April 18	vacation
April 19 through June 20	classes
June 21 through July 25	vacation
July 26 through September 26	classes
September 27 through October 10	vacation
October 11 through December 22	classes
December 23 through December 31	vacation

13) Internal load sensible/radiant fractions

People	0.50 radiant from *ASHRAE 2001 HOF*, page 29.4, Table 1 (ASHRAE 2001)
Lights	0.59 radiant including 0.20 visible from page 235 of McQuiston, Parker, and Spitler, 2003
Equipment	0.70 radiant from page 236 of McQuiston, Parker, and Spitler, 2003

14) EMPD Model
The effective mean penetration depth (EMPD) model in EnergyPlus was used to account for moisture storage in building constructions. The properties used for each type of interior surface layer are summarized in Table 5.3 below. The EnergyPlus Engineering Reference provides a full description of these coefficients and the EMPD method.

Table 5.3 EMPD Moisture Properties

Interior Construction	EMPD Value	Coeff. A	Coeff. B	Coeff. C	Coeff. D
Gypsum Wall Board	0.004	0.072549	0.397173	0.007774	11.7057
1/16 inch Floor Tile	0.002	0.06246	4.516	0.07096	0.4883
Acoustical Ceiling Tile	0.004	0.072549	0.397173	0.007774	11.7057
Carpet	0.004	0.178457	0.583749	0.095156	3.51718

15) Ventilation (outside air flow in the HVAC system) is assumed to occur at a constant rate any time occupants are present except for the cases where demand controlled ventilation is being simulated.

16) An infiltration rate of 0.038 cfm/ft2 of gross exterior wall area was used as specified in Section 13.7.3.2 of ASHRAE/IES Standard 90.1-1989 (ASHRAE 1989). Also, as per this standard, when the HVAC system is ON, no infiltration rate for the building occurs and when the HVAC system is OFF, infiltration does occur. Motels and hotels have infiltration occurring whether the HVAC is ON or OFF. As per the PMS's recommendation, the infiltration was assumed to vary with wind speed. The EnergyPlus algorithm for infiltration is:

$$Infiltration = (I_{design})\,(F_{schedule})\,[A + B\,abs(T_{zone} - T_{odb}) + C(WindSpeed) + D\,(Windspeed)^2]$$

where

I_{design} = 0.038 cfm/ft2 at 4.47 m/s (10 mph)

$F_{schedule}$ is equal to 1.0 when HVAC system is OFF and
0.0 when HVAC system is ON

Wind speed is an hourly value from the weather file, m/s

T_{zone} is the current hour zone temperature, C

T_{odb} is current hour outdoor temperature, C

A, B, C and D are coefficients taken from the DOE-2 energy analysis program converted to SI units

A = 0
B = 0
C = 0.224
D = 0

6 Occupant Density and Ventilation Rates

The Evaluation Plan called for two different levels of occupant densities and ventilation rates to be analyzed. In consultation with the PMS, it was decided to evaluate the occupant densities and ventilation rates as specified in ANSI/ASHRAE Standards 62-2001 (ASHRAE 2001b) and 62.1-2004 (ASHRAE 2004). Standard 62-2001 is currently referenced by several model building codes. Standard 62.1-2004 is the latest version of Standard 62 for commercial buildings. Table 6.1 compares the requirements of these two standards for the applications studied as part of this research project. There are significant differences between the two Standards.

Table 6.1 Comparison of Ventilation Rates for Two Versions of Standard 62

									Demand Controlled	
	Floor Area (ft2)	Std 62-2001 Design Occ. Density (ft2/per)	Max #People	Std 62-2001 Vent Rate (cfm/ft2)	Std 62-2001 Vent Rate (cfm/per)	Std 62-2001 Vent Rate (cfm/room)	Total Vent Rate (cfm)	Total Vent Rate (cfm/ft2)	Minimum Vent Rate (cfm)	Minimum Fraction (%)
Office	6600	143	46		20		923	0.14	56	6%
Retail	79000	33	2394	0.3			23700	0.30	23700	100%
School	1000	20	50		15		750	0.75	9	1%
Restaurant	5250	14	375		20		7500	1.43	126	2%
Motel	350					30	30	0.09	30	100%
Theater*	9000	7	1286		15		19286	2.14	965	5%

Standard 62.1-2004 Occupancy Density and Ventilation Rates

									Demand Controlled	
	Floor Area (ft2)	Std 62-2004 Design Occ. Density (ft2/per)	Max #People	Std 62-2004 Vent Rate (cfm/per)	Std 62-2004 Vent Rate (cfm/ft2)	% change vs. 2001	Total Vent Rate (cfm)	Total Vent Rate (cfm/ft2)	Minimum Vent Rate (cfm)	Minimum Fraction (%)
Office	6600	200	33	5	0.06	-39%	561	0.09	410	73%
Retail	79000	67	1179	7.5	0.12	-23%	18323	0.23	9981	54%
School	1000	40	25	10	0.12	-51%	370	0.37	126	34%
Restaurant	5250	14	375	7.5	0.18	-50%	3758	0.72	992	26%
Motel	350	100	3.5	5	0.06	28%	39	0.11	22	58%
Theater*	9000	7	1286	5	0.06	-64%	6969	0.77	862	12%

* Theater was not in the Eval Plan. It was added based on plans from a 9000 ft2 theater in a multi-theater complex.

7 HVAC System Models

7.1 HVAC System Descriptions

This section presents descriptions and schematics of the HVAC systems that were analyzed. Much of this material was excerpted from Section 4 Humidity Control Options of the Phase I Evaluation Plan (Brandemuehl and Katejanekarn. 2001). A complete copy of Section 4 from the Evaluation Plan is included as Appendix A to this report.

7.1.1 Case 0 – Conventional DX System

This base system is considered to be a conventional, high-efficiency, packaged DX rooftop unit (RTU), shown schematically in Figure 7.1. It has two single-stage scroll compressors, each with a dedicated refrigeration circuit. The evaporators each have three tube rows and are configured in a face-split configuration.

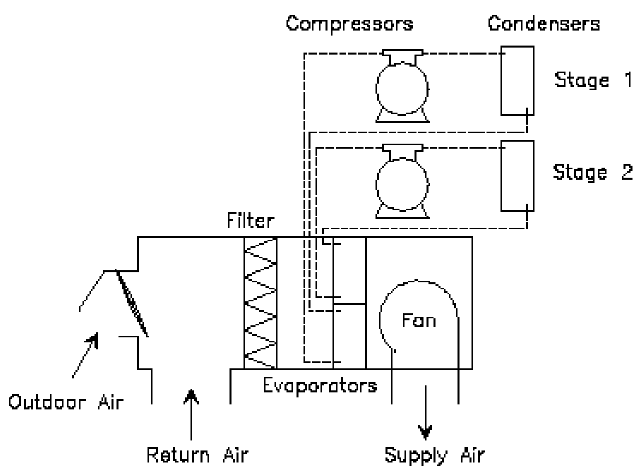

Figure 7.1 - Schematic of DX RTU

Rather than identify the performance of a generic, unspecified rooftop unit, it was desired to select a specific rooftop unit with readily available performance characteristics as the base case. It was further desired that this unit be in the size range of 6 - 15 tons cooling capacity, representing the largest market segment. The Carrier Model 48HJ line of packaged rooftop units with gas heating was selected as representative. The Model 48HJ008, with a nominal capacity of 7½ tons and an ARI rated airflow rate of 400 cfm/ton was selected as the typical conventional design.

7.1.2 Case 1 – Base DX System

The conventional DX system of Case 0 is designed for a standard airflow of 400 cfm/ton. While this design may be representative of typical practice, it is not representative of good HVAC design for applications in which humidity control is a high priority. A more enlightened design would use a packaged system for which the design airflow is 350 cfm/ton and the compressor is slightly oversized for the evaporator, ensuring low evaporator temperatures. Coincidentally, the Carrier Model 48HJ009 (8½ tons) is representative of such a system. Compared to the 7½ ton size, the 8½ ton unit uses the same evaporator and condenser, and the same rated airflow, giving about 350 cfm/ton. This system represents good performance for selection in commercial building applications in humid climates. It will serve as the base case for comparison with all subsequent humidity control options. The system schematic for Case 1 is the same as Case 0 (Figure 7.1).

7.1.3 Case 2 – DX System with Improved Dehumidification

This case considers modification of the base DX system (Case 1) to further improve dehumidification performance. Reduced airflow and larger compressor (relative to evaporator size) can improve dehumidification performance. Another approach involves increasing the number of cooling coil rows, increasing contact time between the moist air and coil surface. The option considered here is a combination of some of these effects.

The Carrier 10-ton Model 48HJ012 unit offers a further improvement over the 7½ ton and 8½ ton models. Specifically, the coil is four rows deep rather than three rows and the ARI rated airflow is 320 cfm/ton. This case considers the 10-ton model operating at a design airflow rate of 300 cfm/ton. The system schematic for Case 2 is the same as Case 0 (Figure 7.1).

7.1.4 Case 3 – Base DX System with Lower Airflow

The base DX system of Case 1 operates at a design airflow of 350 cfm/ton. While the previous case considered a design with 300 cfm/ton and a four-row cooling coil, this case examines the base DX system of Case 1 operating at 300 cfm/ton. The results of the analysis should also give an indication of the level of variability among DX system designs operating at the same airflow rate. The system schematic for Case 3 is the same as Case 0 (Figure 7.1).

7.1.5 Case 4 – DX System with AAHX

An air-to-air heat exchanger (AAHX) can be used to improve the dehumidification performance of a DX system through a clever combination of reheating and precooling. These systems were introduced as *runaround coils* in the 1940s using water as an indirect heat transfer medium. Today, the most common systems employ heat pipes or compact air-to-air devices. The basic system is shown in Figure 7.2.

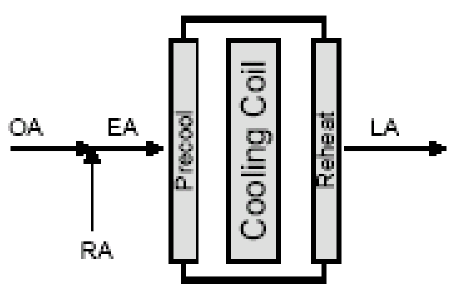

Figure 7.2 - Schematic of DX Cooling Coil with Air-to-Air Heat Exchanger

7.1.6 Case 5 – DX System with Subcooling Reheat Coil

Reheat is an effective method of increasing the latent fraction of system capacity. One alternative is to modify the vapor compression cycle to selectively draw reheat energy from the condenser and, at the same time, improve DX system performance. A subcooling reheat coil provides such a solution. A schematic diagram of a particular subcooling coil arrangement is shown in Figure 7.3.

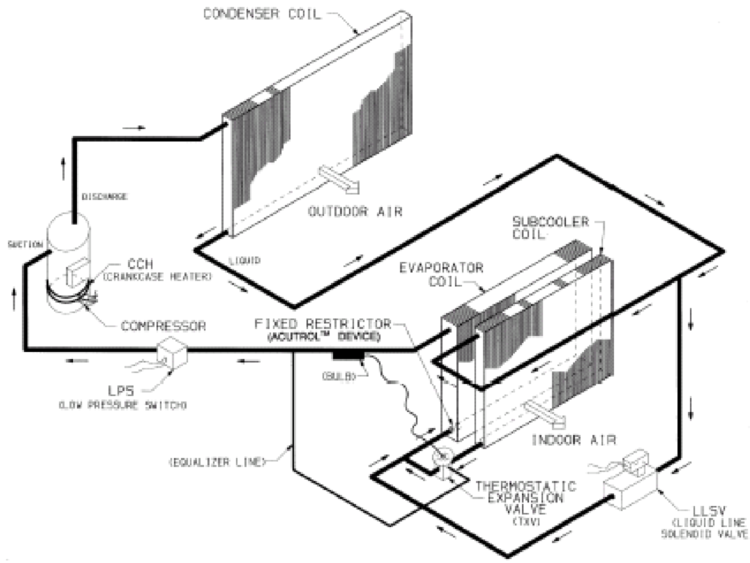

Figure 7.3 - Schematic of DX System with MoistureMiser (from Carrier)

This system, the MoistureMiser manufactured by Carrier, uses a controllable subcooling coil. The system is available as an option to standard Carrier rooftop units 48HJ and 50HJ. When additional dehumidification is required, as indicated by a space humidistat, refrigerant leaving the condenser is directed to an additional coil downstream of the evaporator. There, the refrigerant is further subcooled and the air is heated. The additional subcooling increases the capacity of the DX system. If additional dehumidification is not required the subcooling coil is deactivated.

7.1.7 Case 6 – Base DX System with No Latent Degradation

All DX systems described thus far assume continuous fan operation with a part-load latent performance degradation due to re-evaporation during the off-cycle of the DX compressor (Henderson and Rengarajan, 1996). This ideal case ignores moisture re-evaporation by disabling it in the EnergyPlus simulation. The system schematic for Case 6 is the same as Case 0 (Figure 7.1).

7.1.8 Case 7 – Airflow Control with Bypass Damper

The dehumidification performance of a DX system at part load can be enhanced by control of the airflow. One of the disadvantages of simply reducing the airflow of a DX system with constant fan speed is that dehumidification does not always drive system operation. In addition, it is sometimes required to maintain a higher supply airflow rate than desired for cooling coil performance.

This case analyzes the scenario for which the total supply airflow must be maintained at 350 cfm/ton (base DX system), but the airflow over the coil can be controlled using a two position damper as shown in Figure 7.4. Regardless of damper position, the airflow through the fan is 350 cfm/ton. When the damper is open, only 300 cfm/ton is delivered across the coil, with the remainder bypassing the coil through the damper. The approach of this improvement is to reduce airflow across the coil when the indoor humidity is above a desired setpoint. A humidistat controls the damper to be open or closed, while a thermostat cycles the compressor stages to meet the sensible load. Moisture re-evaporation occurs when the compressor cycles off regardless of the bypass damper position.

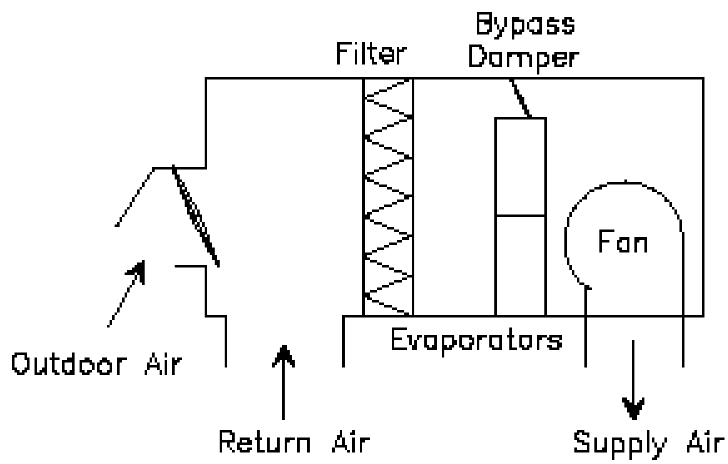

Figure 7.4 - Schematic of DX System with Two-Position Bypass Damper

7.1.9 Case 8 – Hybrid DX/Desiccant System

Dehumidification loads in most buildings are accompanied by some sensible cooling requirements. One approach is to combine the dehumidification capabilities of a desiccant system with the cooling capabilities of a DX system. While the two technologies can be applied in independent packages in the same building, it is possible to integrate desiccant dehumidification and mechanical cooling into a single package and process stream. Such systems are known as integrated or hybrid systems. The one potential advantage of integrating a DX and desiccant system is that the DX system can be designed to meet only sensible loads. A DX system specifically designed for sensible cooling could operate at a higher coil temperature, improving DX system efficiency. Figure 7.5 shows a schematic diagram of a hybrid system. It consists of a rotary desiccant dehumidifier, with a rotary heat exchanger for heat recovery, coupled to a standard DX system. The heat exchanger helps performance by reducing the load on the DX coil.

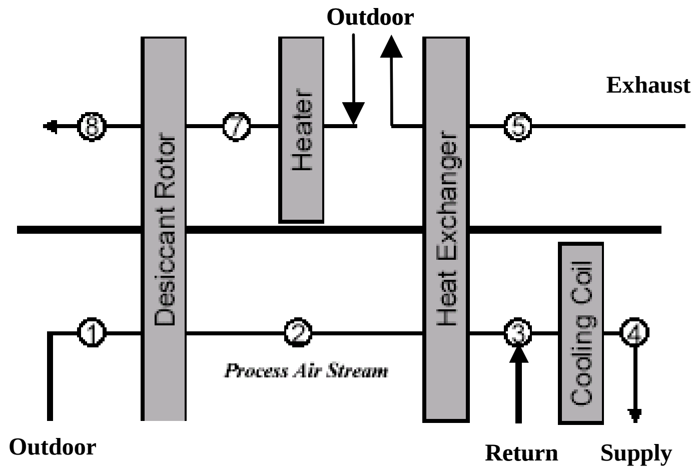

Figure 7.5 - Schematic of Hybrid Integrated Desiccant and DX System

7.1.10 Case 9 – DX System with Enthalpy Recovery

Enthalpy, or total energy, exchangers reduce the dehumidification load associated with ventilation air by transferring moisture from the humid outdoor ventilation air to the relatively drier exhaust air from the building. The moisture transport is analogous to heat transfer; while heat moves from hot to cold, water vapor moves from high vapor pressure to low vapor pressure. Moisture transport typically occurs through either direct transfer through a membrane that separates the two air streams or indirect transfer to a matrix that rotates between the two air streams. Figure 7.6 shows a schematic representation of an enthalpy exchanger integrated with a DX rooftop unit.

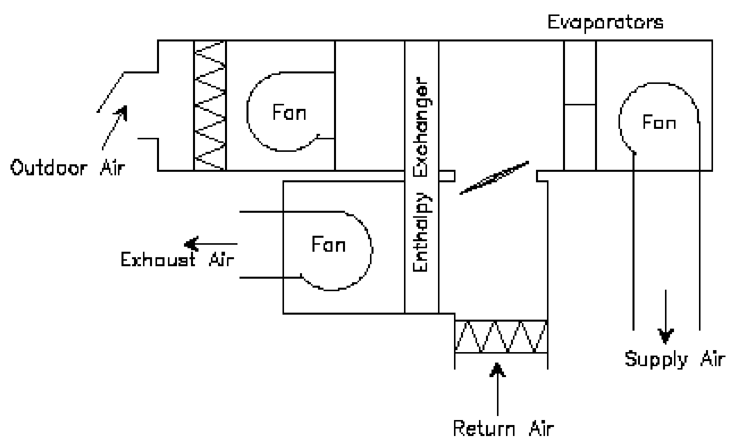

Figure 7.6 - Schematic of Enthalpy Recovery in DX System

7.1.11 Case 10 – DX System with Outdoor Air Preconditioning

Conventional DX unitary equipment introduces ventilation air into the unit and mixes it with return air upstream of the DX coil. One approach to reducing the load on the main cooling and dehumidifying coil is to precondition the outdoor air with a separate DX system. The system considered here preconditions the outdoor air before it mixes with return air and passes over the main DX coil.

The system can be particularly helpful in retrofit applications where it is necessary to increase outdoor airflow for improved air quality. In such retrofit applications, the existing cooling coil may be unable to meet the increased cooling and dehumidification demands of the increased ventilation. An outdoor air preconditioning system could be designed to bolt onto the ventilation air intake of the existing system. In new installations, the preconditioning system could be used to reduce the size of the main unit.

Two different system designs might be considered for the outdoor air preconditioning system. One approach involves the use of a conventional DX packaged air conditioner for preconditioning in which condenser heat is rejected to ambient air. Figure 7.7 shows an alternative arrangement to be evaluated here which is based on the Carrier Energy$Recycler accessory. In this case, the preconditioning system rejects heat to the exhaust air from the building, giving higher efficiency due to cooler air entering the condenser. The other advantage of the configuration of Figure 7.7 is that the preconditioning system can be a heat pump, offering preheating of outdoor air in winter to reduce heating requirements. The heat pump arrangement is particularly appealing when electric resistance heating is applied to the main unit.

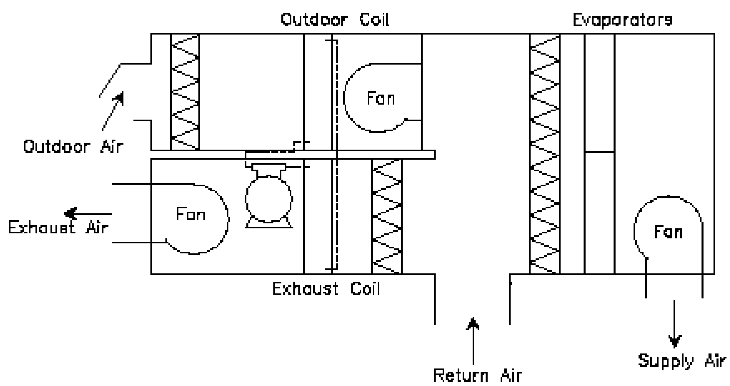

Figure 7.7 - Schematic of DX System with Outdoor Air Preconditioning

7.1.12 Case 11 – Base DX Dual Path System

Conventional DX unitary equipment introduces ventilation air into the unit and mixes it with return air upstream of the cooling coil. A dual path system relies on separate conditioning of the ventilation air stream before mixing with the return air. A schematic representation of a dual path system is shown in Figure 7.8. The figure shows a true dual path arrangement with separate coils for return and outdoor air streams. However, the improvements in dehumidification are largely based on the improved dehumidification performance of a system that conditions 100% outdoor air as well as the ability to maintain a lower leaving air temperature.

These systems are also known as 100% outdoor air units or make-up air units. However, the cooling and dehumidification loads of a commercial building almost always require a combination of outdoor and return air conditioning. For this reason, a building system that provides separate conditioning of outdoor air will be referred to as a dual path system. Dehumidification improvements of dual path systems are largely based on the low SHR available by conditioning high humidity outdoor air. Dual path systems can be applied in two main configurations.

- **Independent Systems.** One approach is to treat the outdoor and return air systems as independent of each other. The outdoor air system can have separate fan and ductwork to deliver conditioned outdoor air to the zone.
- **Common Supply System**. The outdoor and return air systems could share a common supply fan and ductwork. The discharge air from the two systems can be mixed immediately upstream of the supply fan. Airflow through the two systems can be controlled with dampers, or an injection fan can be used to help control outdoor airflow. This approach allows colder supply air temperatures to be used.

For the purposes of this analysis, the performance of these two main configurations will be considered identical. In actual applications, the independent systems may have higher initial costs, but may be easier to control. The outdoor air system is modeled with four stages of cooling which are operated to meet the space cooling load, but with a lower limit on leaving supply air temperature of 7.22C (45F). The return air compressor is cycled on as the last stage of cooling when needed.

A well-designed dual path system will also exhibit improved efficiency, especially for multi-stage systems. Since the outdoor air DX system has the same air conditions entering the evaporator and condenser, the required compressor lift is lower and efficiency can be higher. However, the control of the outdoor air DX unit is particularly challenging. While the return air system typically sees nearly constant entering air conditions and airflow rate, the outdoor air system sees wide variations in outdoor temperature and humidity. More advanced systems can also modulate outdoor airflow rate to match indoor air quality needs, further complicating the capacity control challenge.

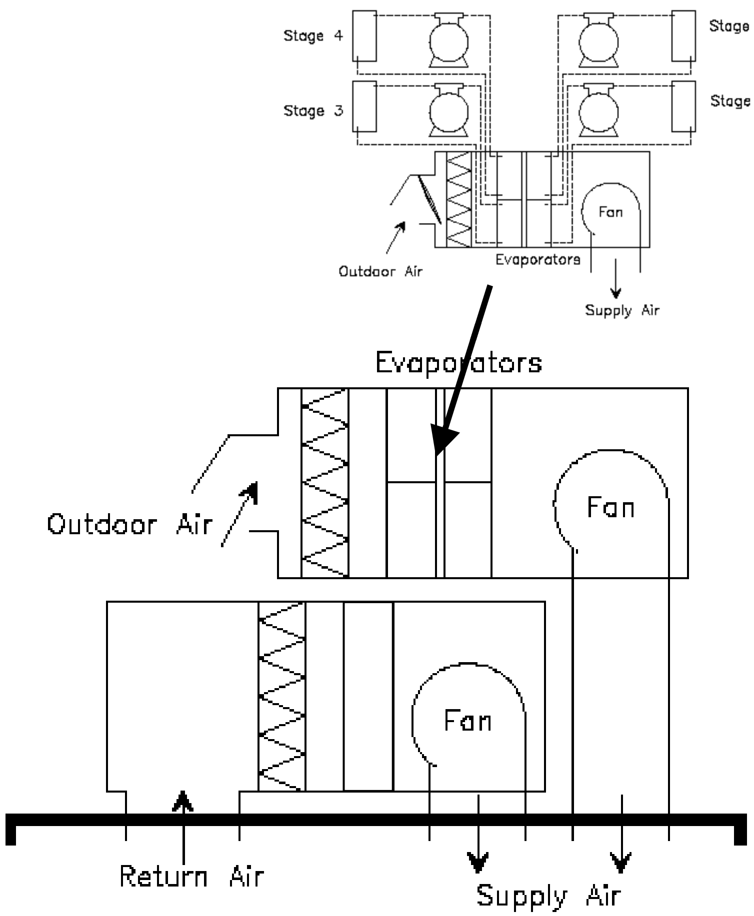

Figure 7.8 - Schematic of Dual Path System

7.1.13 Case 12 – DX Dual Path System with Enthalpy Recovery

The dual path system of the previous case can be combined with enthalpy recovery equipment to precondition the outdoor air before the DX cooling system. Figure 7.9 shows a schematic depiction of the system. Note that the enthalpy exchanger will dramatically reduce the load on the DX cooling coil, eliminating the need for four stages of capacity used in the dual path system without recovery.

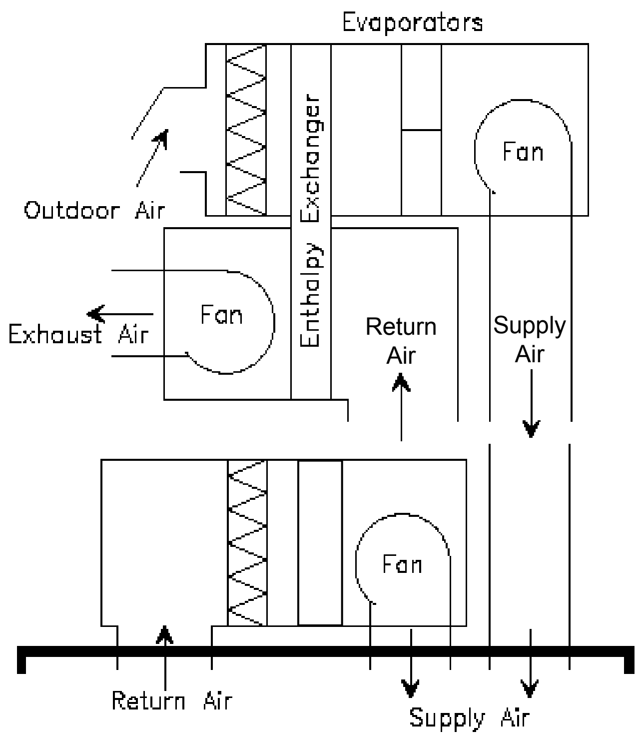

Figure 7.9 - Schematic of Dual Path System with Enthalpy Recovery

7.1.14 Case 13 – DX Dual Path System with AAHX

The dual path system of Case 11 can be combined with AAHX equipment to precondition the outdoor air before the DX cooling system. Figure 7.10 shows a schematic depiction of the system.

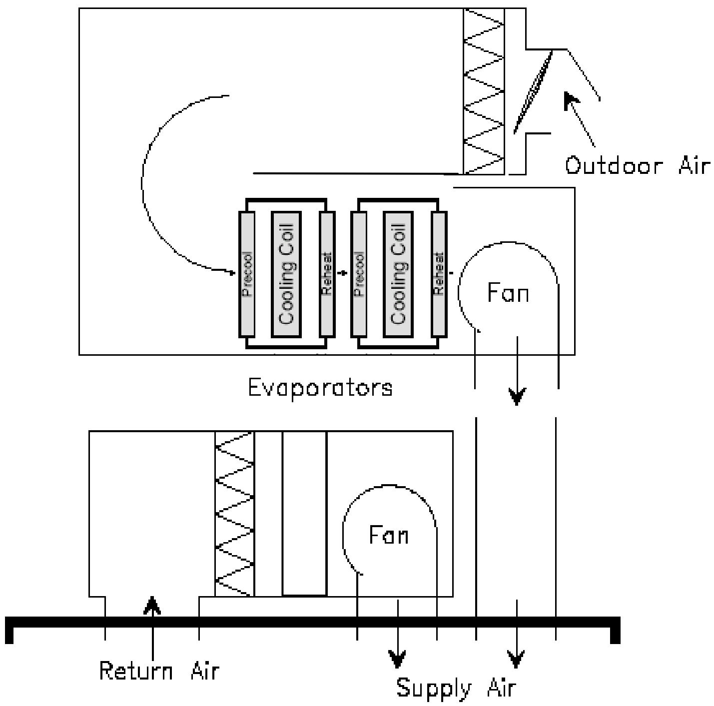

Figure 7.10 - Schematic of Dual Path System with AAHX

7.1.15 Case 14 – Hybrid DX and Desiccant Dual Path System

The hybrid desiccant dual path system is a variation of the dual path and hybrid systems discussed in Case 11 and Case 8, except that the hybrid DX/desiccant system is used to condition outdoor air and a separate DX system is used to condition the return air. A schematic diagram is shown in Figure 7.11. Conceptually, the hybrid DX/desiccant system will meet all dehumidification requirements of the building and introduce conditioned outdoor air to the zone at room temperature.

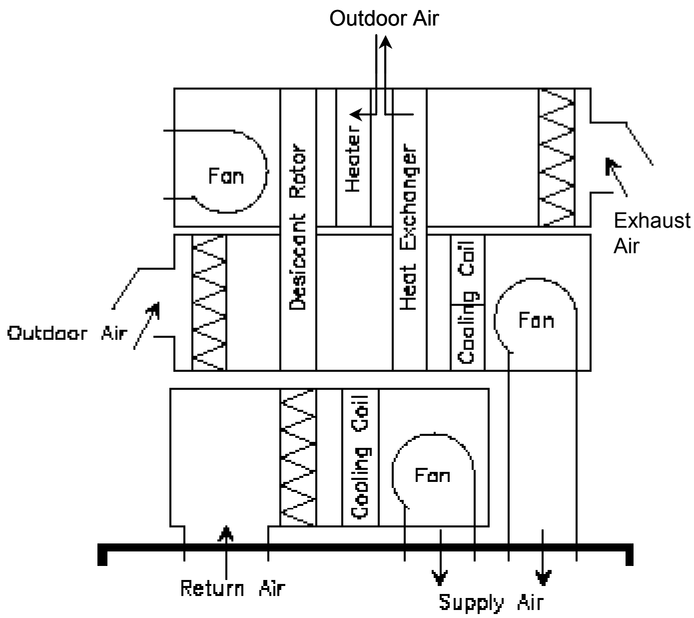

Figure 7.11 – Hybrid DX/Desiccant Dual Path System

7.1.16 Case 15 – DX System with Demand Controlled Ventilation

In most commercial buildings, dehumidification loads are dictated by ventilation requirements. Since ventilation is dictated by indoor air quality concerns, it is possible to modulate ventilation airflow in response to air quality demands. For commercial buildings in which occupant-generated contaminants dictate ventilation requirement, ASHRAE Standard 62-1999 allows modulation of ventilation air to maintain the concentration of CO_2 at 1000 ppm in the occupied zone. Of the building applications for this study, all buildings except the retail store are considered to have ventilation requirements dictated by occupant-generated contaminants. This humidity control option involves control of ventilation airflow rate to maintain 1000 ppm CO_2. As a limiting case, the control should also be applied to the retail building. The system schematic for Case 15 is the same as Case 0 (Figure 7.1).

7.1.17 Case 16 – Dual Path System with Demand Controlled Ventilation

This system applies demand controlled ventilation (see description in Case 15 above) with the dual-path system of Case 11. The system schematic for Case 16 is the same as Case 11 (Figure 7.8).

7.1.18 Case 17 – Base DX System with Free Reheat

This system is the same as the base DX system (Case 1) with free reheat from the condenser coil.

7.2 HVAC System Simulation Models and Assumptions

EnergyPlus HVAC system models were developed to conform to the Carrier Model 48HJ Single-Package Rooftop Unit configuration which has DX cooling coils, gas heat and a draw-thru fan. All single path systems were modeled using the EnergyPlus "DXSystem:AirLoop" object with options such as air-to-air heat exchanger, bypass damper, desiccant dehumidification, enthalpy wheel, multi-mode coils, etc. added as required. For dual path systems, the "Air Conditioner:Window:Cycling" object was used to condition return air only. The EnergyPlus input (idf) files for each simulation case are available upon request from ASHRAE. The following assumptions were made regarding the HVAC systems:

1) As specified in the Evaluation Plan, equipment was sized for 110% of peak design sensible load.

2) As specified in Evaluation Plan, the space cooling setpoint was set to 75F. During unoccupied hours, cooling was off.

3) No heating setpoint was specified but was assumed to be 70F. A 10F setback to 60F during heating was simulated.

4) The fan power in terms of watts/cfm as given for each case was used as specified in the Evaluation Plan to calculate the corresponding delta P (see Table 7.1). Supply fans run continuously during normal HVAC system operation hours. Supply fans cycle during heating setback operation. In all systems, the additional fan power requirements for extra components were added to the main supply fan. For the enthalpy wheel systems (Case 9 and Case 12) and the OA pretreat system (Case 10), fan heat in the exhaust stream was not added to the supply air stream. For the desiccant systems (Case 8 and Case 14), the regeneration fan was modeled as a separate fan.

5) System performance data was taken from Carrier catalog data and was curve fit by first converting the temperatures and capacities from IP to SI units and then curve fitting. Although catalog data includes performance for a range of supply air flows for each different model, curve fits were done only for one set of air flow rates. Based on catalog data, curve fits were done for cooling capacity (CoolCapFT) as a function of condenser entering air dry-bulb temperature and evaporator entering air wet-bulb temperature and for energy input ratio (CoolEIRFT) as a function of condenser entering air dry-bulb temperature and evaporator entering air wet-bulb temperature. Other performance curves required by EnergyPlus were taken from the IEA HVAC BESTEST specification (Neymark and Judkoff 2002), i.e., energy input ratio as a function of part load ratio. The performance data and resulting curve fits are included in Appendix C.

6) Heating and cooling was assumed to be available 2 hours before occupancy but the ventilation schedule still followed occupancy schedule.

7) All systems modeled as a single unit, regardless of size.

8) Gas heat was assumed with a constant efficiency of 80%.

Table 7.1 Fan Pressure Drops Used for HVAC Systems

System Code	Description	Nominal Supply Flow (CFM/ton)	Supply Fan Power (W/CFM)	Nominal Supply Fan Power (W/ton)	Fan Efficiency (%)	Motor Efficiency (%)	Total Fan Efficiency (%)	Calculated Total Fan Pressure Drop (in. water)
0	Conventional DX	400	0.550	220	24%	80%	19.2%	0.90
1	Base DX	353	0.550	194	24%	80%	19.2%	0.90
2	DX w/Improved Dehumid.	300	0.479	144	24%	80%	19.2%	0.78
3	Base DX w/Lower Airflow	300	0.476	143	24%	80%	19.2%	0.78
4	Base DX w/AAHX	353	0.675	238	24%	80%	19.2%	1.10
5	Base DX w/Subcool Reheat	353	0.603	213	24%	80%	19.2%	0.99
6	Base DX	353	0.550	194	24%	80%	19.2%	0.90
7	Base DX w/Bypass	353	0.550	194	24%	80%	19.2%	0.90
8Main	Conventional DX w/Desiccant	400	0.550	220	24%	80%	19.2%	1.39
8OA	Desiccant Outdoor Air Fan		0.300		24%	80%	19.2%	0.49
8Regen	Desiccant Regen Fan		0.400		24%	80%	19.2%	0.65
9Main	Base DX w/Enthalpy Wheel	353	0.550	194	24%	80%	19.2%	1.39
9OA	Enth Whl Outdoor Air Fan		0.150		24%	80%	19.2%	0.25
9EA	Enth Whl Exhaust Air Fan		0.150		24%	80%	19.2%	0.25
10Main	Base DX w/OA Precool	353	0.550	194	24%	80%	19.2%	0.90
10OA	Precool Outdoor Air Fan		0.300		24%	80%	19.2%	0.49
10EA	Precool Exhaust Air Fan		0.300		24%	80%	19.2%	0.49
11Main	Dual Path	300	0.479	144	24%	80%	19.2%	0.78
11RA	Return Air System Fan	400	0.550	220	24%	80%	19.2%	0.90
12Main	Dual Path w/Enthalpy Wheel	353	0.479	169	24%	80%	19.2%	0.78
12RA	Return Air System Fan	400	0.550	220	24%	80%	19.2%	0.90
12OA	Enth Whl Outdoor Air Fan		0.150	0	24%	80%	19.2%	1.02
12EA	Enth Whl Exhaust Air Fan		0.150		24%	80%	19.2%	1.02
13Main	Dual Path w/AAHX	300	0.604	181	24%	80%	19.2%	1.10
13RA	Return Air System Fan	400	0.550	220	24%	80%	19.2%	0.90
14Main	Dual Path w/Desiccant	353	0.479	169	24%	80%	19.2%	0.78
14RA	Return Air System Fan	400	0.550	220	24%	80%	19.2%	0.90
14OA	Desiccant Outdoor Air Fan		0.300		24%	80%	19.2%	1.02
14Regen	Desiccant Regen Fan		0.400		24%	80%	19.2%	0.65
15	Base DX w/DCV	353	0.550	194	24%	80%	19.2%	0.90
16OA	Dual Path w/DCV	300	0.479	144	24%	80%	19.2%	0.78
16RA	Return Air System Fan	400	0.550	220	24%	80%	19.2%	0.90
17	Condenser Reheat	353	0.550	194	24%	80%	19.2%	0.90

Main = main supply air fan (OA system in dual-path)
OA = outside air fan
RA = return air system fan (dual-path)
EA = exhaust air fan
Regen = regeneration fan

9) For cases with cooling coil latent degradation turned on, the following parameters were set:
   Twet Nominal Time for Condensate Removal to Begin (s)
   1.5 Ratio of Initial Moisture Evaporation Rate and Steady-state Latent Capacity
   3 Maximum ON/OFF Cycling Rate
   45 Latent Capacity Time Constant (s)

   See Table 9.1 for details about Twet.

   The model for treating latent degradation in EnergyPlus is based on a methodology developed by Henderson and Rengarajan (1996).

10) Two stage cooling operation was simulated for all systems and applications except the motel and systems with air-to-air heat exchangers (Case 4 and Case 13). The two-stage modeling assumes face-split coils (see Figure 7.1) with stage 1 capacity being 60% of the total capacity. For the unitary equipment being modeled, units with capacity less than 6 tons are usually single stage units and since the motel room cooling capacities were always less than one ton, single stage cooling equipment for the motel was a good assumption. The use of single stage cooling for the systems with air-to-air heat exchangers was due to modeling limitations within EnergyPlus. The sensitivity of results to using one-stage versus two-stage cooling is shown below for the retail application in Atlanta using the base HVAC system Case 1.

Application	Location	HVAC System	Number Stages of Cooling	Number of Humidity Hours >65%	Electric Cooling Consumption (kWh)
Retail	Atlanta	S01	1	2453	82306
Retail	Atlanta	S01	2	580	93930

11) For Cases 8 and 14 with desiccant dehumidification, the following parameters were set:
   50% Zone RH Setpoint
   3.556 Nominal Process Air Velocity (m/s)
   1/40 HP/7000 cfm or 5.642 W/m3/s Rotor Power (W)
   121 Nominal Regen Temperature (C)
   EnergyPlus desiccant system default performance curves were used.

12) For Cases 9 and 12 with enthalpy wheel, the evaluation plan specified an enthalpy effectiveness of 0.88 and a humidity ratio effectiveness of 0.85. The simulation inputs for the enthalpy exchanger were:
   0.91 Sensible effectiveness at 100% and 75% airflow heating/cooling condition
   0.85 Latent effectiveness at 100% and 75% airflow heating/cooling condition
   Rotary Heat exchanger type
   1/40 HP per 7000 cfm or 5.642 W/m3/s Nominal electric power (W)
   The rotor power was based on data for another project.

13) For all dual-path systems (Cases 11, 12, 13, 14, and 16), the available stages of cooling in the outside air system are operated first to meet the space cooling load, but with a lower limit on leaving supply air temperature of 7.22C (45F). The return air system compressor is cycled on as the last stage of cooling when needed. Cases 11 (Base DX Dual Path), and 16 (Dual Path System with Demand Controlled Ventilation) have four stages of cooling in the outside air systems (two 2-stage coils in series). Case 12 (DX Dual Path System with Enthalpy Recovery) has one 2-stage coil in the outside air system. Case 13 (DX Dual Path System with AAHX) has two 1-stage AAHX-assisted coils in series in the outside air system. The return air system for all dual-path systems is one single-stage DX coil.

14) Enthalpy wheel controls were set to bypass the enthalpy wheel at times when outdoor air is beneficial, using the following rules: bypass when OA dry bulb temperature < return air temperature (RETURN AIR TEMP LIMIT in EnergyPlus), bypass when OA enthalpy < return air enthalpy (RETURN AIR ENTHALPY LIMIT), do not bypass if OA dry bulb below 5C (41F) or above 23.9C (75F).

    An alternative enthalpy wheel control rule was also investigated: Bypass the enthalpy wheel when outdoor dry bulb (ODB) is 1 to 2 degrees F below the dewpoint of the zone temperature and relative humidity setpoint. This strategy was compared for selected cases and found to be the same or worse than the above strategy in terms of humidity control and energy use.

15) For Cases 15 and 16 with demand controlled ventilation (DCV), two different scenarios as specified in ASHRAE Standard 62-2001 and 62.1-2004 were simulated using the "VENTILATION:MECHANICAL" object. Under DCV, anytime a building was scheduled to be occupied, the occupant-based portion of the ventilation rate was varied in accordance with occupancy level. However, the total ventilation to the space was never allowed to go below the minimum required based on floor space. See Table 6.1 for a comparison of ventilation requirements between the two standards.

16) Case 17, Base DX w/Free Reheat, was designed to provide a comparison point with a free source of reheat. In the simulation, reheat was modeled using the EnergyPlus "COIL:Desuperheater:Heating" object. The "Heat Reclaim Recovery Efficiency" field for this object was set to 1.0 to simulate sufficient reheat capacity. In some cases, this may be more full condenser reheat than simply desuperheating. Setting the recovery efficiency to 1, requires editing the maximum limit for this field in the Energy+.idd file. The reheat coil was controlled using the "SET POINT MANAGER:SINGLE ZONE REHEAT" object. In the version of EnergyPlus used for this analysis (1.2.2.030), this applied reheat to maintain the cooling setpoint. The behavior of this control may change in a future version of EnergyPlus.

17) For Case 17, Base DX w/Free Reheat, the controls in the EnergyPlus "DXSystem:AirLoop" object were set as follows: Dehumidification Control Type = CoolReheat, Run on Sensible Load = Yes, Run on Latent Load = No. This means that the compressor would operate only when there was a sensible cooling load. It would not operate for a latent-only load. Once the compressor was on to meet a sensible load, the controls would also operate to meet the humidistat setpoint by overcooling and reheating if necessary.

18) The characteristics of the unitary systems for each of the cases analyzed are summarized in Table 7.2.

Table 7.2 Characteristics of HVAC Systems

System Code	Description	Carrier Model No.	Nominal Catalog Cooling Capacity (tons)	Nominal & Rated Airflow (CFM)	Nominal Airflow per Cooling Capacity (CFM/ton)
0, 8, 11RA, 12RA, 13RA, 14RA, 16RA	Conventional DX	48HJ008	7.5	3000	400
1, 5Conv, 6, 7Conv, 9, 10Main, 12Main, 14Main, 15, 17	Base DX	48HJ009	8.5	3000	353
2, 7BYPASS	DX w/Improved Dehumid.	48HJ012	10.0	3000	300
3, 11Main, 13Main, 16Main	Base DX w/Lower Airflow	48HJ009	8.5	2550	300
4	Base DX w/AAHX	48HJ009	8.5	3000	353
5MoistMiser	Base DX w/Subcool Reheat	48HJ009	8.5	3000	353
10OA	OA Preconditioner	62AQ Energy$Recycler	1.9	1000	533

System Code	Description	Rated Compressor Power (kW)	Rated Condenser Fan Power (kW)	Rated Total Gross Cooling Capacity (kBtu)	Rated Sensible Gross Cooling Capacity (kBtu)	Rated Airflow per Cooling Capacity (CFM/ton)	Rated Gross SHR	Rated Gross COP
0, 8, 11RA, 12RA, 13RA, 14RA, 16RA	Conventional DX	6.61	0.65	92.9	67.2	388	0.72	3.75
1, 5Conv, 6, 7Conv, 9, 10Main, 12Main, 14Main, 15, 17	Base DX	7.68	0.65	101.2	68.9	356	0.68	3.56
2, 7BYPASS	DX w/Improved Dehumid.	9.18	0.65	120.7	79.6	298	0.66	3.60
3, 11Main, 13Main, 16Main	Base DX w/Lower Airflow	7.62	0.65	97.7	62.8	313	0.64	3.46
4	Base DX w/AAHX	7.75	0.65	101.2	68.9	356	0.68	3.53
5MoistMiser	Base DX w/Subcool Reheat	7.75	0.65	94.8	57.0	380	0.60	3.31
10OA	OA Preconditioner	1.37	0	19.9	15.9	603	0.80	4.26

Main = main supply air unit (OA system in dual-path)
Conv = conventional mode
RA = return air unit (dual-path)
OA = outside air unit
BYPASS = DX coil bypass mode
MoistMiser = subcooling reheat mode

Rated data is based on values from the equipment performance tables for the ARI rating point of 67F entering web bulb and 95F outdoor dry bulb

7.3 HVAC System Characteristics Summary

The significant differences between the various HVAC systems described in the previous section as implemented in EnergyPlus are summarized below.

Case 0 - Conventional DX System

- 400 cfm/ton
- Represents typical HVAC design practice
- System schematic - Figure 7.1

Case 1 - Base DX System

- 350 cfm/ton
- Different equipment from Case 0
- Better dehumidification design practice
- System schematic - Figure 7.1

Case 2 - DX System with Improved Dehumidification

- 300 cfm/ton
- Modified coil, compressor, etc.
- System schematic - Figure 7.1

Case 3 - Base DX System with Lower Airflow

- 300 cfm/ton
- Same coil and compressor as Case 1
- System schematic - Figure 7.1

Case 4 – DX System with Air-to-Air Heat Exchanger (AAHX)

- 350 cfm/ton
- Wrap-around HX
- Sensible effectiveness 0.4
- No latent transfer across HX
- Single-stage coil in all applications
- System schematic - Figure 7.2

Case 5 – DX System with Subcooling Reheat Coil

- 350 cfm/ton
- Normal mode same as Case 1
- Enhanced dehumidification mode
- Carrier “MoistureMiser”
- Switch modes if 50% RH setpoint not met
- System schematic - Figure 7.3

Case 6 – Base DX System with No Latent Degradation

- 350 cfm/ton
- Normal mode same as Case 1
- Fan off short time when compressor cycles off
- No moisture re-evaporation
- Modeled by turning off latent degradation in DX coil model
- System schematic - Figure 7.1

Case 7 – Bypass Damper

- 350 cfm/ton
- Normal mode same as Case 1
- 300 cfm/ton in bypass mode
- 50 cfm/ton bypassed as needed
- Switch modes if 50% RH setpoint not met
- System schematic - Figure 7.4

Case 8 – Hybrid DX with Desiccant

- 400 cfm/ton (Case 0)
- Desiccant dehumidifier with sensible heat recovery to exhaust air
- Desiccant conditions outside air stream
- Mixed air to cooling coil
- Control to meet 50% RH setpoint
- System schematic - Figure 7.5

Case 9 – Enthalpy Recovery Wheel

- 350 cfm/ton (Case 1)
- Enthalpy heat recovery OA and exhaust
- Bypassed when not beneficial
- 0.91 sensible effectiveness (constant)
- 0.85 latent effectiveness (constant)
- System schematic - Figure 7.6

Case 10 – DX Outdoor Air Preconditioning

- DX Preconditioner
    - Evaporator in OA stream
    - Condenser in relief air stream
    - 580 cfm/ton
    - Carrier Energy$Recycler
    - Run 1st
- Main DX Coil
    - 350 cfm/ton
    - Run as needed
- System schematic - Figure 7.7

Case 11 – Base Dual Path

- Outdoor air system
    - 2 DX coils in series
    - 300 cfm/ton (Case 3)
    - 150 cfm/ton overall
    - 2 stages each
    - 7.22C (45F) minimum supply air temperature
- Return air system
    - 1 DX coil
    - 400 cfm/ton (Case 0)
    - 1 stage (last stage of cooling)
- System schematic - Figure 7.8

Case 12 – Dual Path plus Enthalpy Recovery

- Outdoor air system
    - 1 DX coil
    - 350 cfm/ton (Case 1)
    - 2 stages each
    - Enthalpy heat recovery OA and exhaust
    - 7.22C (45F) minimum supply air temperature
- Return air system
    - 1 DX coil
    - 400 cfm/ton (Case 0)
    - 1 stage (last stage of cooling)
- System schematic - Figure 7.9

Case 13 – Dual Path plus Air-to-Air Heat Exchanger

- Outdoor air system
    - 2 DX coils in series
    - AAHX, 0.4 sensible effectiveness, no latent
    - 300 cfm/ton (Case 3)
    - 150 cfm/ton overall
    - 1 stage each
    - 7.22C (45F) minimum supply air temperature
- Return air system
    - 1 DX coil
    - 400 cfm/ton (Case 0)
    - 1 stage (last stage of cooling)
- System schematic - Figure 7.10

Case 14 – Dual Path plus Desiccant

- Outdoor air system
    - 1 DX coil
    - 350 cfm/ton (Case 1)
    - 2 stages each
    - Desiccant dehumidifier with sensible heat recovery to exhaust air
    - Control desiccant to meet 50% RH setpoint
    - 7.22C (45F) minimum supply air temperature
- Return air system
    - 1 DX coil
    - 400 cfm/ton (Case 0)
    - 1 stage (last stage of cooling)
- System schematic - Figure 7.11

Case 15 – Demand Controlled Ventilation

- 350 cfm/ton
- Case 1 equipment
- Minimum outdoor air tracks occupancy schedule
- System schematic - Figure 7.1

Case 16 – Dual Path Plus Demand Controlled Ventilation

- Case 11 equipment
- Minimum outdoor air tracks occupancy schedule
- OA system is constant volume, unit processes mixed air when DCV reduces outdoor air flow
- System schematic - Figure 7.8

Case 17 – Base DX with Free Reheat

- 350 cfm/ton
- Case 1 equipment
- Condenser heat available for reheat (hot gas reheat)
- Reheat capacity 100% of cooling capacity plus compressor power
- Controlled for sensible and humidity
- Dehumidify only when there is a sensible cooling load (no operation for latent-only load)
- System schematic - Figure 7.1

8 Design Loads and Equipment Sizing

Cooling design loads were calculated based on simulation of the 0.4% drybulb design day weather for each location (ASHRAE 2001). The peak sensible cooling load that occurred was increased by a 10% oversizing factor. Supply air flow was then calculated based on each system's rated sensible cfm/ton as determined from catalog data as summarized in Table 8.1 below. For dual-path systems, the 100% OA system cooling capacity was determined by taking the OA flow rate and applying the rated sensible cfm/ton and rated SHR. Any remaining sensible capacity requirement was then applied to the return air system and the supply air flow rate was then calculated by applying the rated sensible cfm/ton. For the enthalpy wheel systems, the design sensible capacity was reduced by the enthalpy wheel sensible capacity at design conditions based on the outside air flow rate and dry bulb temperature.

Table 8.1 Characteristics of Modeled Unitary Cooling Equipment

Case	Carrier Model	- - -Total Capacity - - - Nominal (Tons)	Rated (Tons)	Airflow (CFM)	- - - - Airflow/Capacity - - - - Nominal (CFM/TCap)	Rated (CFM/TCap)	Rated SHR	Rated Airflow/ (CFM/SCap)
0	48HJ008	7.5	7.74	3000	400	388	0.72	539
1	48HJ009	8.5	8.43	3000	350	356	0.68	524
2	48HJ012	10.0	10.06	3000	300	298	0.66	452
3	48HJ009	8.5	8.14	2550	300	313	0.64	489

The resulting sensible and total cooling capacity and supply airflow rate for the Case 1 Base DX system for each application in ten climate locations are presented in Tables 8.2 and 8.3. Table 8.2 presents design loads for each application based on ASHRAE Standard 62-2001 ventilation rates while Table 8.3 shows the design loads based on ASHRAE Standard 62.1-2004 ventilation rates.

Using this methodology, a different supply air flow rate was computed for every system type for every combination of building and location. The constant across all systems was the net sensible capacity. For dual-path systems, this was the combined net sensible capacity of the main (OA) system and the secondary return air system with a few exceptions. For enthalpy wheel systems (which were given a capacity reduction), and in some high ventilation rate applications (such as the Restaurant and the Theater) this resulted in significantly more than the capacity being available in the outside air portion of the dual-path systems. This occurred when the outside air flow rate applied to the specified cfm/ton for the dual path systems resulted in more than enough capacity to meet the outside air load and the space load. No design outdoor dry bulb temperature adjustment was made to equipment capacities.

Table 8.2
Results of Design Load Calculations Using ASHRAE Standard 62-2001 Ventilation Rates

					Case 0-Base DX (10% oversize)				
				Net	*400 cfm/ton (nominal)*				Area per
			Minimum	Sensible	*536 cfm/sensible-ton (rated)*			Net Rated	Net Rated
		Zone	Outside	Cooling	Supply	Outside	Supply Air	Total	Total
		Area	Air Flow	Capacity	Air Flow	Air	Rate	Capacity	Capacity
Building	Location	(sq ft)	(cfm)	(tons)	(cfm)	(%)	(cfm/sq ft)	(tons)	(sf/ton)
Motel-South	ATLANTA GA	350	30	0.26	169	18%	0.48	0.41	857
Motel-South	CHICAGO IL	350	30	0.29	189	16%	0.54	0.45	778
Motel-South	FORT_WORTH TX	350	30	0.29	188	16%	0.54	0.46	767
Motel-South	MIAMI FL	350	30	0.23	146	20%	0.42	0.35	992
Motel-South	NEW YORK CITY NY	350	30	0.30	195	15%	0.56	0.47	752
Motel-South	PORTLAND OR	350	30	0.33	215	14%	0.61	0.52	669
Motel-South	ST._LOUIS MO	350	30	0.29	191	16%	0.54	0.45	774
Motel-South	STERLING VA	350	30	0.29	184	16%	0.52	0.45	783
Motel-South	HOUSTON TX	350	30	0.26	166	18%	0.47	0.40	875
Motel-South	SHREVEPORT LA	350	30	0.27	173	17%	0.50	0.42	829
Office	ATLANTA GA	6600	923	5.52	3555	26%	0.54	8.65	763
Office	CHICAGO IL	6600	923	5.47	3539	26%	0.54	8.57	770
Office	FORT_WORTH TX	6600	923	6.68	4299	21%	0.65	10.47	630
Office	MIAMI FL	6600	923	5.49	3550	26%	0.54	8.60	767
Office	NEW YORK CITY NY	6600	923	5.98	3846	24%	0.58	9.36	705
Office	PORTLAND OR	6600	923	5.77	3714	25%	0.56	9.04	730
Office	ST._LOUIS MO	6600	923	5.85	3763	25%	0.57	9.16	720
Office	STERLING VA	6600	923	5.78	3717	25%	0.56	9.05	729
Office	HOUSTON TX	6600	923	5.92	3812	24%	0.58	9.28	711
Office	SHREVEPORT LA	6600	923	5.99	3857	24%	0.58	9.39	703
Retail	ATLANTA GA	79000	23700	88.34	57509	41%	0.73	138.46	571
Retail	CHICAGO IL	79000	23700	88.30	56823	42%	0.72	138.35	571
Retail	FORT_WORTH TX	79000	23700	109.03	70164	34%	0.89	170.84	462
Retail	MIAMI FL	79000	23700	96.04	56367	42%	0.71	150.16	526
Retail	NEW YORK CITY NY	79000	23700	94.44	60773	39%	0.77	147.97	534
Retail	PORTLAND OR	79000	23700	90.10	57981	41%	0.73	141.17	560
Retail	ST._LOUIS MO	79000	23700	93.82	61069	39%	0.77	147.05	537
Retail	STERLING VA	79000	23700	92.05	59241	40%	0.75	144.24	548
Retail	HOUSTON TX	79000	23700	95.60	62389	38%	0.79	149.84	527
Retail	SHREVEPORT LA	79000	23700	93.76	62995	38%	0.80	147.07	537
Restaurant	ATLANTA GA	5250	7500	18.99	12285	61%	2.34	29.76	176
Restaurant	CHICAGO IL	5250	7500	17.97	11573	65%	2.20	28.16	186
Restaurant	FORT_WORTH TX	5250	7500	23.82	16927	44%	3.22	37.42	140
Restaurant	MIAMI FL	5250	7500	18.10	11645	64%	2.22	28.35	185
Restaurant	NEW YORK CITY NY	5250	7500	19.26	12417	60%	2.37	30.18	174
Restaurant	PORTLAND OR	5250	7500	17.97	11578	65%	2.21	28.15	186
Restaurant	ST._LOUIS MO	5250	7500	20.37	13133	57%	2.50	31.93	164
Restaurant	STERLING VA	5250	7500	19.09	12308	61%	2.34	29.91	176
Restaurant	HOUSTON TX	5250	7500	21.03	13534	55%	2.58	32.95	159
Restaurant	SHREVEPORT LA	5250	7500	21.35	13782	54%	2.63	33.46	157

Table 8.2 (Continued)
Results of Design Load Calculations Using ASHRAE Standard 62-2001 Ventilation Rates

					Case 0-Base DX (10% oversize)				
				Net	*400 cfm/ton (nominal)*				Area per
			Minimum	Sensible	*536 cfm/sensible-ton (rated)*			Net Rated	Net Rated
		Zone	Outside	Cooling	Supply	Outside	Supply Air	Total	Total
		Area	Air Flow	Capacity	Air Flow	Air	Rate	Capacity	Capacity
Building	Location	(sq ft)	(cfm)	(tons)	(cfm)	(%)	(cfm/sq ft)	(tons)	(sf/ton)
School-9 Month-South	ATLANTA GA	1000	750	2.21	1424	53%	1.42	3.47	288
School-9 Month-South	CHICAGO IL	1000	750	2.13	1373	55%	1.37	3.34	299
School-9 Month-South	FORT_WORTH TX	1000	750	2.73	1756	43%	1.76	4.27	234
School-9 Month-South	MIAMI FL	1000	750	2.05	1326	57%	1.33	3.22	311
School-9 Month-South	NEW YORK CITY NY	1000	750	2.30	1481	51%	1.48	3.61	277
School-9 Month-South	PORTLAND OR	1000	750	2.25	1448	52%	1.45	3.53	284
School-9 Month-South	ST._LOUIS MO	1000	750	2.33	1534	49%	1.53	3.65	274
School-9 Month-South	STERLING VA	1000	750	2.24	1443	52%	1.44	3.51	285
School-9 Month-South	HOUSTON TX	1000	750	2.41	1548	48%	1.55	3.77	265
School-9 Month-South	SHREVEPORT LA	1000	750	2.45	1574	48%	1.57	3.83	261
School-12 Month-South	ATLANTA GA	1000	750	2.21	1424	53%	1.42	3.47	288
School-12 Month-South	CHICAGO IL	1000	750	2.13	1373	55%	1.37	3.34	299
School-12 Month-South	FORT_WORTH TX	1000	750	2.73	1756	43%	1.76	4.27	234
School-12 Month-South	MIAMI FL	1000	750	2.05	1326	57%	1.33	3.22	311
School-12 Month-South	NEW YORK CITY NY	1000	750	2.30	1481	51%	1.48	3.61	277
School-12 Month-South	PORTLAND OR	1000	750	2.25	1448	52%	1.45	3.53	284
School-12 Month-South	ST._LOUIS MO	1000	750	2.33	1534	49%	1.53	3.65	274
School-12 Month-South	STERLING VA	1000	750	2.24	1443	52%	1.44	3.51	285
School-12 Month-South	HOUSTON TX	1000	750	2.41	1548	48%	1.55	3.77	265
School-12 Month-South	SHREVEPORT LA	1000	750	2.45	1574	48%	1.57	3.83	261
Theater	ATLANTA GA	9000	19286	46.63	30007	64%	3.33	73.06	123
Theater	CHICAGO IL	9000	19286	43.72	28134	69%	3.13	68.50	131
Theater	FORT_WORTH TX	9000	19286	58.90	37903	51%	4.21	92.29	98
Theater	MIAMI FL	9000	19286	44.64	28726	67%	3.19	69.94	129
Theater	NEW YORK CITY NY	9000	19286	47.19	30366	64%	3.37	73.94	122
Theater	PORTLAND OR	9000	19286	43.89	28242	68%	3.14	68.77	131
Theater	ST._LOUIS MO	9000	19286	50.54	32524	59%	3.61	79.19	114
Theater	STERLING VA	9000	19286	46.61	29994	64%	3.33	73.03	123
Theater	HOUSTON TX	9000	19286	52.19	33587	57%	3.73	81.78	110
Theater	SHREVEPORT LA	9000	19286	53.16	34211	56%	3.80	83.30	108

Table 8.3
Results of Design Load Calculations Using ASHRAE Standard 62.1-2004 Ventilation Rates

					Case 0-Base DX (10% oversize)				
					400 cfm/ton (nominal)				
					536 cfm/sensible-ton (rated)				
Building	Location	Zone Area (sq ft)	Minimum Outside Air Flow (cfm)	Net Sensible Cooling Capacity (tons)	Supply Air Flow (cfm)	Outside Air (%)	Supply Air Rate (cfm/sq ft)	Net Rated Total Capacity (tons)	Area per Net Rated Total Capacity (sf/ton)
Motel-South	ATLANTA GA	350	39	0.27	179	22%	0.51	0.43	813
Motel-South	CHICAGO IL	350	39	0.30	196	20%	0.56	0.47	750
Motel-South	FORT_WORTH TX	350	39	0.31	202	19%	0.58	0.49	718
Motel-South	MIAMI FL	350	39	0.24	156	25%	0.45	0.37	934
Motel-South	NEW YORK CITY NY	350	39	0.31	204	19%	0.58	0.49	720
Motel-South	PORTLAND OR	350	39	0.35	224	17%	0.64	0.55	642
Motel-South	ST._LOUIS MO	350	39	0.31	201	19%	0.57	0.49	716
Motel-South	STERLING VA	350	39	0.30	192	20%	0.55	0.47	749
Motel-South	HOUSTON TX	350	39	0.27	177	22%	0.51	0.43	821
Motel-South	SHREVEPORT LA	350	39	0.29	185	21%	0.53	0.45	781
Office	ATLANTA GA	6600	561	4.93	3170	18%	0.48	7.72	855
Office	CHICAGO IL	6600	561	4.58	2948	19%	0.45	7.18	919
Office	FORT_WORTH TX	6600	561	5.81	3755	15%	0.57	9.11	725
Office	MIAMI FL	6600	561	4.90	3154	18%	0.48	7.68	859
Office	NEW YORK CITY NY	6600	561	5.32	3423	16%	0.52	8.33	792
Office	PORTLAND OR	6600	561	5.17	3327	17%	0.50	8.10	815
Office	ST._LOUIS MO	6600	561	5.14	3307	17%	0.50	8.05	820
Office	STERLING VA	6600	561	5.13	3301	17%	0.50	8.04	821
Office	HOUSTON TX	6600	561	5.17	3325	17%	0.50	8.10	815
Office	SHREVEPORT LA	6600	561	5.23	3366	17%	0.51	8.20	805
Retail	ATLANTA GA	79000	18323	64.99	41821	44%	0.53	101.83	776
Retail	CHICAGO IL	79000	18323	64.04	41213	44%	0.52	100.35	787
Retail	FORT_WORTH TX	79000	18323	80.57	51849	35%	0.66	126.24	626
Retail	MIAMI FL	79000	18323	70.18	41823	44%	0.53	109.77	720
Retail	NEW YORK CITY NY	79000	18323	68.90	44338	41%	0.56	107.96	732
Retail	PORTLAND OR	79000	18323	65.54	42176	43%	0.53	102.69	769
Retail	ST._LOUIS MO	79000	18323	67.89	45515	40%	0.58	106.50	742
Retail	STERLING VA	79000	18323	66.84	43012	43%	0.54	104.73	754
Retail	HOUSTON TX	79000	18323	69.51	46466	39%	0.59	109.03	725
Retail	SHREVEPORT LA	79000	18323	70.33	47160	39%	0.60	110.32	716
Restaurant	ATLANTA GA	5250	3758	13.58	8737	43%	1.66	21.27	247
Restaurant	CHICAGO IL	5250	3758	13.02	8408	45%	1.60	20.40	257
Restaurant	FORT_WORTH TX	5250	3758	16.19	11461	33%	2.18	25.43	206
Restaurant	MIAMI FL	5250	3758	13.00	8364	45%	1.59	20.36	258
Restaurant	NEW YORK CITY NY	5250	3758	13.75	8884	42%	1.69	21.54	244
Restaurant	PORTLAND OR	5250	3758	13.08	8462	44%	1.61	20.50	256
Restaurant	ST._LOUIS MO	5250	3758	14.20	9140	41%	1.74	22.26	236
Restaurant	STERLING VA	5250	3758	13.58	8788	43%	1.67	21.28	247
Restaurant	HOUSTON TX	5250	3758	14.50	9329	40%	1.78	22.72	231
Restaurant	SHREVEPORT LA	5250	3758	14.71	9463	40%	1.80	23.04	228

Table 8.3 (Continued)
Results of Design Load Calculations Using ASHRAE Standard 62.1-2004 Ventilation Rates

					Case 0-Base DX (10% oversize)				
				Net	*400 cfm/ton (nominal)*				Area per
			Minimum	Sensible	*536 cfm/sensible-ton (rated)*			Net Rated	Net Rated
		Zone	Outside	Cooiling	Supply	Outside	Supply Air	Total	Total
		Area	Air Flow	Capacity	Air Flow	Air	Rate	Capacity	Capacity
Building	Location	(sq ft)	(cfm)	(tons)	(cfm)	(%)	(cfm/sq ft)	(tons)	(sf/ton)
School-9 Month-South	ATLANTA GA	1000	370	1.30	839	44%	0.84	2.04	490
School-9 Month-South	CHICAGO IL	1000	370	1.29	829	45%	0.83	2.02	496
School-9 Month-South	FORT_WORTH TX	1000	370	1.58	1015	36%	1.01	2.47	405
School-9 Month-South	MIAMI FL	1000	370	1.13	724	51%	0.72	1.76	567
School-9 Month-South	NEW YORK CITY NY	1000	370	1.35	881	42%	0.88	2.11	473
School-9 Month-South	PORTLAND OR	1000	370	1.39	894	41%	0.89	2.18	459
School-9 Month-South	ST._LOUIS MO	1000	370	1.40	902	41%	0.90	2.20	455
School-9 Month-South	STERLING VA	1000	370	1.34	861	43%	0.86	2.10	477
School-9 Month-South	HOUSTON TX	1000	370	1.39	896	41%	0.90	2.18	459
School-9 Month-South	SHREVEPORT LA	1000	370	1.42	912	41%	0.91	2.22	450
School-12 Month-South	ATLANTA GA	1000	370	1.30	839	44%	0.84	2.04	490
School-12 Month-South	CHICAGO IL	1000	370	1.29	829	45%	0.83	2.02	496
School-12 Month-South	FORT_WORTH TX	1000	370	1.58	1015	36%	1.01	2.47	405
School-12 Month-South	MIAMI FL	1000	370	1.13	724	51%	0.72	1.76	567
School-12 Month-South	NEW YORK CITY NY	1000	370	1.35	881	42%	0.88	2.11	473
School-12 Month-South	PORTLAND OR	1000	370	1.39	894	41%	0.89	2.18	459
School-12 Month-South	ST._LOUIS MO	1000	370	1.40	902	41%	0.90	2.20	455
School-12 Month-South	STERLING VA	1000	370	1.34	861	43%	0.86	2.10	477
School-12 Month-South	HOUSTON TX	1000	370	1.39	896	41%	0.90	2.18	459
School-12 Month-South	SHREVEPORT LA	1000	370	1.42	912	41%	0.91	2.22	450
Theater	ATLANTA GA	9000	6969	28.96	18649	37%	2.07	45.38	198
Theater	CHICAGO IL	9000	6969	27.79	17887	39%	1.99	43.55	207
Theater	FORT_WORTH TX	9000	6969	33.82	21765	32%	2.42	52.99	170
Theater	MIAMI FL	9000	6969	27.91	17966	39%	2.00	43.74	206
Theater	NEW YORK CITY NY	9000	6969	29.38	18943	37%	2.10	46.03	196
Theater	PORTLAND OR	9000	6969	28.47	18322	38%	2.04	44.61	202
Theater	ST._LOUIS MO	9000	6969	30.17	19419	36%	2.16	47.28	190
Theater	STERLING VA	9000	6969	28.86	18570	38%	2.06	45.21	199
Theater	HOUSTON TX	9000	6969	30.75	19789	35%	2.20	48.18	187
Theater	SHREVEPORT LA	9000	6969	31.13	20034	35%	2.23	48.78	185

9 DX Coil Latent Degradation Model

This analysis makes extensive use of the latent capacity degradation model which is a standard feature of EnergyPlus. This model adjusts the latent capacity of DX cooling coils which are operated with continuous supply fan and cycling compressors to simulate the effect of re-evaporation of moisture from the wet coil when the compressor cycles off. The following excerpt from the EnergyPlus Engineering Reference (EnergyPlus 2005b) describes the basis of this model. These excerpts are generally a direct quote, however figure numbers have been changed to match the numbering of this report.

Latent Capacity Degradation with Continuous Supply Air Fan Operation

The latent (dehumidification) capacity of a direct-expansion (DX) cooling coil is strongly affected by part-load, or cyclic, operation. This is especially true in applications where the supply air fan operates continuously while the cooling coil cycles on and off to meet the cooling load. During constant fan operation, moisture condenses on the cooling coil when the compressor operates, but part or all of the moisture that is held by the coil evaporates back into the airstream when the cooling coil is deactivated (Figure 9.1). The net effect is that the amount of moisture removed from the air is degraded at part-load conditions as compared to steady-state conditions when the compressor operates continuously (Figure 9.2). EnergyPlus is able to model latent capacity degradation based on algorithms developed by Henderson and Rengarajan (1996). The model is applicable to single-stage cooling units, like residential and small commercial air conditioners or heat pumps with less than 19 kW of nominal cooling capacity. The model inputs are described in the EnergyPlus Input/Output Reference for the object Coil:DX:CoolingBypassFactorEmpirical. The model is enabled only if the four numerical inputs are defined (values greater than zero, see IO Reference) and the field “Supply Air Fan Operation Mode” must be “ContFanCycComp”.

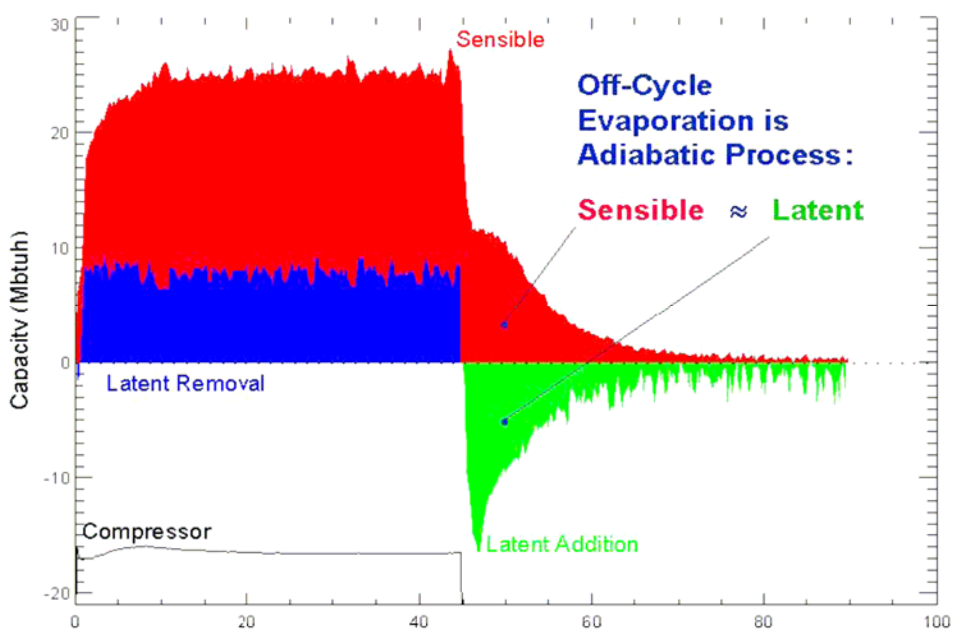

Figure 9.1 Transient Sensible and Latent Capacity of a Cooling Coil Over an Operating Cycle

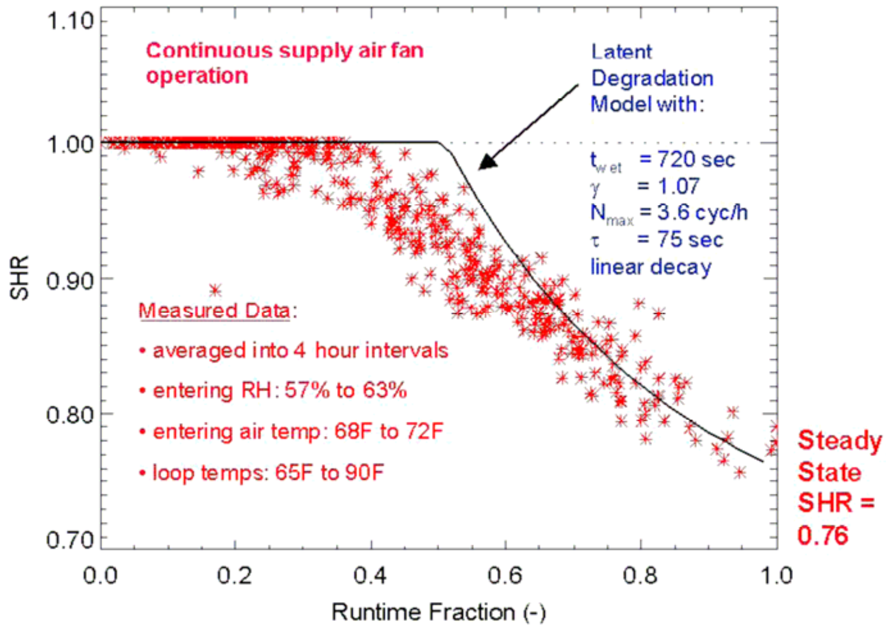

Figure 9.2 Field Data Showing the Net Impact of Part-Load Operation on Sensible Heat Ratio

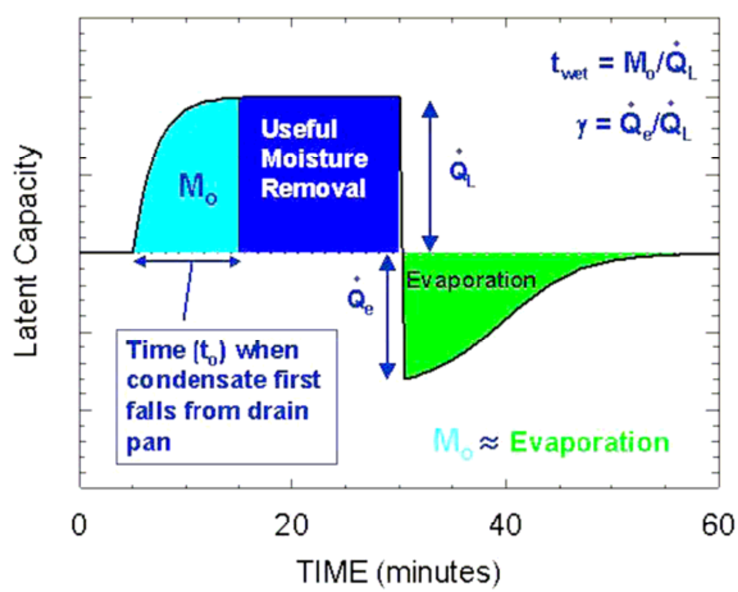

Figure 9.3 Concepts of Moisture Buildup and Evaporation

Figure 9.3 graphically depicts the latent degradation concepts and defines several key model parameters. After the cooling coil starts to operate, the coil temperature is eventually reduced below the dewpoint temperature of the entering air. Moisture from the air then builds on the surface of the coil until time t_o has elapsed and the total moisture mass on the coil is M_o. After this time (t_o), moisture begins to fall from the coil and all of the latent capacity provided by the coil is "useful" since this condensate is collected and removed from the unit. When the coil cycles off and the supply air fan continues to operate, the initial moisture mass buildup on the coil

(M_o) evaporates back into the supply air stream. If the cooling coil cycles back on before all of the moisture has evaporated, then the time until the first condensate removal (t_o) is shorter for this cooling cycle since the coil is already partially wetted. Figure 9.3 also shows several parameters that are used in the latent degradation model. The ratio of the coil's moisture holding capacity (M_o) and the steady-state latent capacity (Q_L) is defined as *twet* : the nominal time for moisture to fall from the coil (ignoring transient effects at startup and starting with a dry coil). The ratio of the initial moisture evaporation rate (Q_e) and the steady-state latent capacity (Q_L) is defined as γ . Both *twet* and γ at the rated air volume flow rate and temperature conditions are required model inputs. Two other model inputs are the Maximum ON/OFF Cycling Rate (cycles per hour, *Nmax*) and the time constant (τ , in seconds) for the cooling coil's latent capacity to reach steady state after startup. The development of the latent degradation model is fully described by Henderson and Rengarajan (1996). The model implemented in EnergyPlus is for their "linear decay" evaporation model. During the simulation, all of the steady-state calculations described previously . . . are completed. The latent degradation model then modifies the steadystate sensible heat ratio for the coil . . .

Further details, including equations, for this model may be found in the EnergyPlus Engineering Reference. The description above states that this model is for single-stage equipment. The multi-mode DX cooling coil model in EnergyPlus models each stage individually and combines the results. Table 9.1 lists the values of *twet* which were used for the various systems in this analysis.

Table 9.1 Latent Degradation Parameters for DX Cooling Coils

System Code	Description	Carrier Model No.	Latent Load (kBtu)	Estimated Fin Area (ft^2)	Moisture Holding Capacity (lb)	twet (min)	gamma
0, 8, 11RA, 12RA, 13RA, 14RA, 16RA	Conventional DX	48HJ008	25.7	750.0	6.75	16.7	1.5
1, 5Conv, 6, 7Conv, 9, 10Main, 12OA, 14OA, 15, 17	Base DX	48HJ009	32.3	850.0	7.65	15.1	1.5
2, 7BYPASS	DX w/Improved Dehumid.	48HJ012	41.1	1000.0	9.00	13.9	1.5
3, 11OA, 13OA, 16OA	Base DX w/Lower Airflow	48HJ009	34.9	850.0	7.65	13.9	1.5
4	Base DX w/AAHX	48HJ009	32.3	850.0	7.65	15.1	1.5
5MoistMiser	Base DX w/MoistureMiser	48HJ009	37.8	850.0	7.65	12.9	1.5
10OA	OA Preconditioner	62AQ Energy$Recycler	4.0	187.5	1.69	26.8	1.5

As noted in the earlier list of assumptions, for all system types, the following additional latent degradation model assumptions were used:

- Maximum ON/OFF Cycling Rate = 3 cycles/hr. This is the maximum on-off cycling rate for the compressor which occurs at 50% run time fraction.

- Latent Capacity Time Constant = 45 seconds. This is the time constant for the cooling coil's latent capacity to reach steady state after startup.

Figures 9.4 and 9.5 show the impact of the latent degradation model on the SHR of the Case 1 Base DX equipment in the Retail Store in Atlanta.

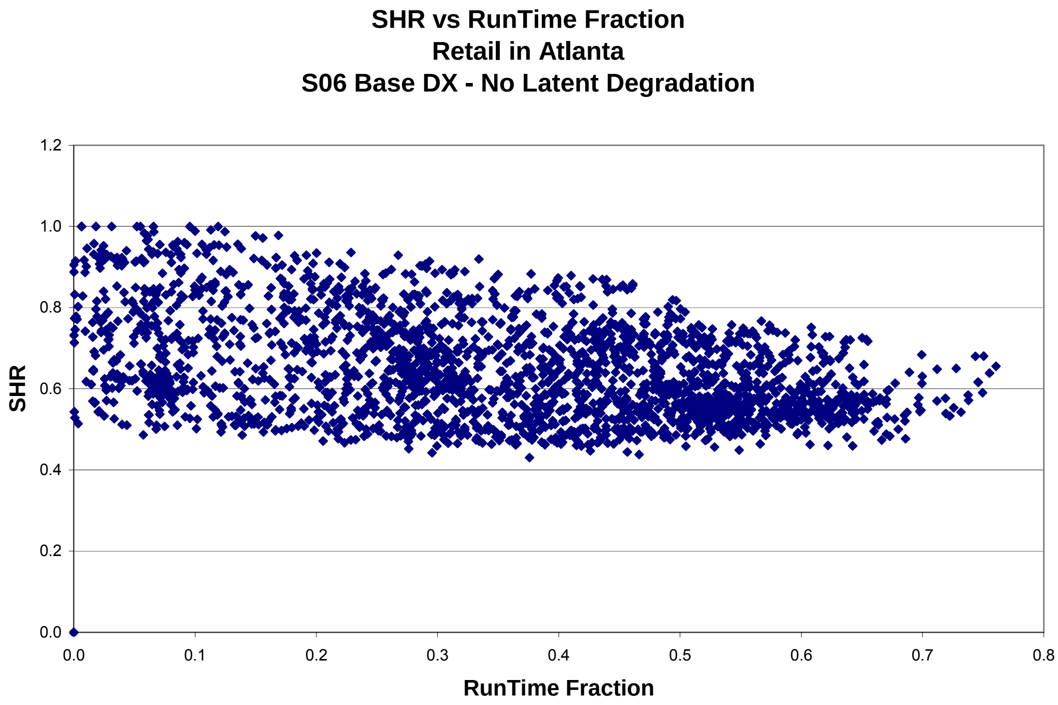

Figure 9.4 Sensible Heat Ratio (SHR) vs. Runtime Fraction <u>With No</u> Latent Degradation.

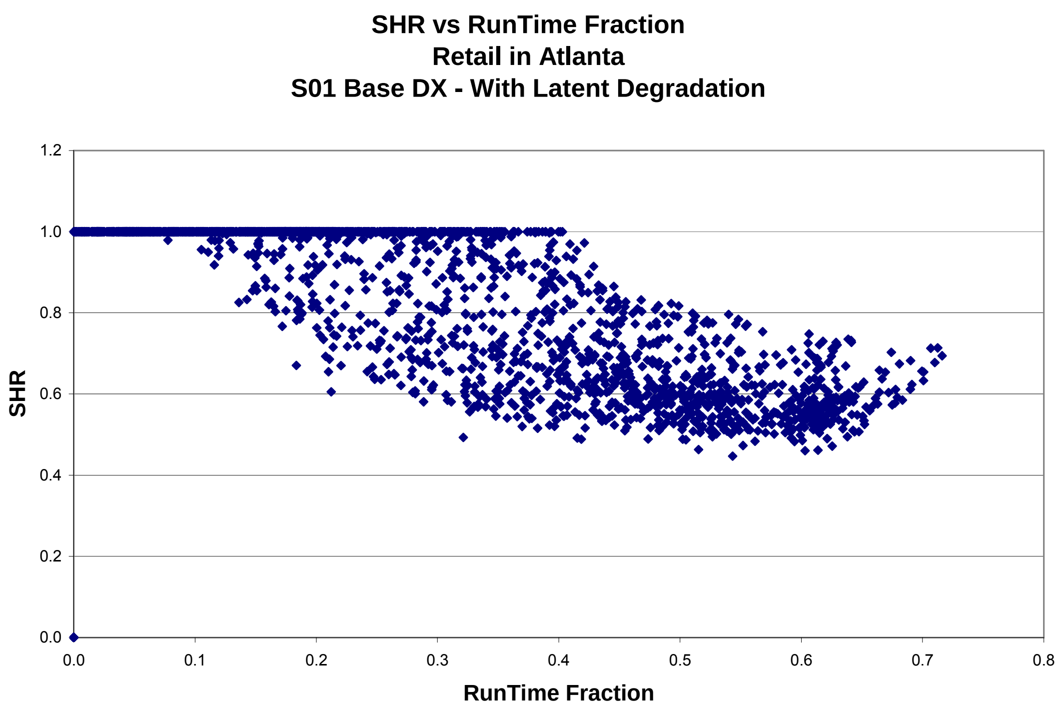

Figure 9.5 Sensible Heat Ratio (SHR) vs. Runtime Fraction <u>With</u> Latent Degradation.

10 EnergyPlus HVAC System New Feature Development

The following new EnergyPlus capabilities were developed as part of this research project:

- New 2-stage DX coil with controllable enhanced dehumidification mode option named COIL:DX:MultiMode:CoolingEmpirical – This DX coil model is able to model a 1-stage or 2-stage DX cooling coil with an optional enhanced dehumidification mode such as subcool reheat. The model uses four sets of performance curves for the following operation modes: stage 1 normal mode, stages 1&2 normal mode, stage 1 enhanced dehumidification mode, and stages 1&2 enhanced dehumidification mode. This parallels the way that some manufacturers present performance data for such systems.

- Additional DESICCANT DEHUMIDIFIER:SOLID humidity setpoint control option – Previously the desiccant dehumidifier could only be controlled to a fixed leaving humidity ratio which was entered as a single value in the desiccant dehumidifier object. This new option allows the dehumidifier to read the humidity ratio setpoint from a control node, thus allowing the use of a set point manager to control the dehumidifier. The existing SET POINT MANAGER:SINGLE ZONE MAX HUM can be used along with the new SET POINT MANAGER:OUTSIDE AIR PRETREAT (see below) to control the desiccant dehumidifier based on the relative humidity of a particular simulation zone.

- New set point managers for controlling single zone equipment named SET POINT MANAGER:SINGLE ZONE HEATING, and SET POINT MANAGER:SINGLE ZONE COOLING – These set point managers allow independent control of heating and cooling coils based on the current zone heating and cooling requirement, offering more flexibility for this analysis than the existing EnergyPlus set point managers.

- New set point manager for controlling outside air equipment named SET POINT MANAGER:OUTSIDE AIR PRETREAT – This set point manager takes a supply air temperature or humidity and determines the equivalent set point required by outside air pretreatment equipment to meet this requirement. For example, if the temperature set point at the mixed air node is 15C, the return air temperature is 20C, and the outside air flow fraction is 0.5, the Outside Air Pretreat setpoint would be set to 10C. In this analysis, this was used to control the desiccant dehumidifier to meet the entire dehumidification load if possible.

- New humidity control options for DXSystem:AirLoop, one of the EnergyPlus DX system types – DXSystem:AirLoop is one of the EnergyPlus objects that can be used to control a DX cooling coil, and it is the approach that was used for this analysis. Previously, this component controlled only for leaving dry bulb temperature (sensible load). The new options allow this component to control the DX coil for humidity in one of two ways: CoolReheat in which the DX coil will overcool the supply air in order to meet the humidity setpoint, or Multimode in which the DX coil is switched into enhanced dehumidification mode if the humidity setpoint is not met (see new component COIL:DX:MultiMode:CoolingEmpirical above). The new control options also allow the unit to activate the DX coil for sensible loads only, latent loads only, or both.

- Enable DXSystem:AirLoop to be used in the outside air stream, allowing DX coil pretreatment of OA – EnergyPlus now allows DX coils to be used in the outside air stream prior to the mixing box.

All of these enhancements to EnergyPlus are part of the public release version of EnergyPlus as of version 1.2.2.030, released in April 2005 with acknowledgement to ASHRAE for supporting this work. These enhancements provide a combination of modeling capabilities never seen before in whole building energy simulation software. With this capability, the results of this study can be extended and customized as required to meet the specific needs of designers and researchers.

The EnergyPlus input (idf) files for each simulation case are available upon request from ASHRAE.

The majority of the simulation results were run using EnergyPlus version 1.2.2.030 (April 2005). A handful of cases failed in version 1.2.2.030 due to a DX system part-load-ratio bug. This bug was fixed in version 1.2.2.032 (June 2005, Change Request 6675). For cases which did not encounter the bug, results were identical in versions 1.2.2.030 and 1.2.2.032, so only the affected cases were re-run using 1.2.2.032. In February 2006 all of the desiccant and enthalpy wheel cases were run again using version 1.2.2.032 with revised inputs to correct some heat recovery control problems. The bug fix from version 1.2.2.032 is now part of the standard release versions of EnergyPlus. In May 2006 another bug was discovered which caused Case 8 – Base DX w/Desiccant to produce incorrect results for some combinations of application type and location. In order to maintain consistency with prior results, version 1.2.2.032B was created based on 1.2.2.032 source code with a fix for this bug. All Case 8 simulations were re-run using version 1.2.2.032B. This bug fix (Change Request 7002) will become part of the next public release version of EnergyPlus.

11 Economic Analysis

A basic economic analysis was performed accounting for installed equipment costs and HVAC energy costs. A set of life cycle cost results have been computed for this report, but they are only examples based on generic cost data. It is strongly recommended that any equipment selection be based on actual equipment and cost quotes for a specific project. Energy use and equipment capacity results in Appendix B are presented in a form that provides a way to calculate a rough estimate of energy costs which can then be applied to first cost data acquired for a given project. Performing full energy simulations with the applicable utility rates for the particular design under consideration would, of course, provide better energy cost estimates.

11.1 Cost Assumptions

First cost data is highly variable, because many of these systems are not standard production units or may be units which are sold in low volumes. Proprietary concerns also limit the availability of cost data. In addition, first costs can vary widely depending on contractor relationships, project size, national accounts vs. single projects and many other bid variables. Several sources ranging from Means to a past Florida Solar Energy Center report were used to estimate the relative costs of the systems in this analysis. Table 11.1 summarizes the equipment cost assumptions.

Equipment costs were varied by city using the "City Cost Indexes" published by Means. State by state average commercial electric and gas costs were obtained from EIA. Table 11.2 summarizes the location dependent cost assumptions.

Energy cost escalation and equipment life assumptions are show in Table 11.3. Projected national average commercial electric and gas prices were obtained from EIA in constant 2004 dollars. This price series, in constant dollars, embodies both energy cost escalation and discount rate for future payments. The 15-year series of prices is summed and then divided by the first year energy price to obtain an energy cost factor.

11.2 Life Cycle Cost Method

Equipment and energy cost assumptions are in 2004 dollars (US$). The life cycle cost is computed as follows:

LCC = EquipmentCost +
AnnualElecUse * ElectricPrice * ElecCostFactor +
AnnualGasUse * GasPrice * GasCostFactor

where:

EquipmentCost	=	Installed equipment cost
AnnualElecUse	=	Annual HVAC electric energy use
AnnualGasUse	=	Annual HVAC gas energy use
ElectricPrice	=	Average first-year electric price (see Table 11.2)
GasPrice	=	Average first year gas price (see Table 11.2)
ElecCostFactor	=	15-yr electric cost factor (see Table 11.3)
GasCostFactor	=	15-yr gas cost factor (see Table 11.3)

Table 11.1 Equipment Installed Cost Assumptions

Case	System Description	(A) Base DX Cost Net Total ($/ton) (Ref. 1)	(B) Custom or Dual Path Premium (%) (Ref. 2)	(C) Extras per Primary ($/ton)	(D) Extras per Secndy ($/ton)	(E) Extras per OA cfm ($/cfm)	(F) Extras per unit ($)	Ref.	Comments
00	Conventional DX	$2,130						1	Standard unit
01	Base DX	$2,130						1	Standard unit, different airflow
02	DX w/Improved Dehumid.	$2,130	21%					2	Custom single-path cost premium
03	Base DX w/Lower Airflow	$2,130						1	Standard unit, different airflow
04	Base DX w/AAHX	$2,130		$475				2	Heat pipe
05	Base DX w/Subcool Reheat	$2,130		$133				4	Subcool reheat option.
06	Base DX w/o Lat. Coil Degrad.	$2,130					$640	2	Similar to CO2 "Controls and Wiring" cost
07	Base DX w/Bypass Damper	$2,130		$215				2	Bypass damper and controls
08	Base DX w/Desiccant	$2,130				$7.50		3	Commercial solid desiccant
09	Base DX w/Enthalpy Wheel	$2,130				$3.75		0	Enthalpy wheel assumed to be 50% of desiccant system cost
10	Base DX w/OA Precool	$2,130	21%					0, 2	OA system equipment assumed to be same cost per ton as standard unit, plus custom single-path cost premium
11	Dual Path	$2,130	27%					2	Dual-path cost premium
12	Dual Path w/Enthalpy Wheel	$2,130	27%			$3.75		2, 0	Dual-path, plus enthalpy wheel assumed to be 50% of desiccant system cost
13	Dual Path w/AAHX	$2,130	27%	$475				2	Dual-path, plus heat pipe.
14	Dual Path w/Desiccant	$2,130	27%			$7.50		2, 3	Dual-path, plus commercial solid desiccant
15	Base DX w/DCV	$2,130				$1.08		2	CO2 sensor, damper, and controls
16	Dual Path w/DCV	$2,130	27%			$1.08		2	Dual-path, plus CO2 sensor, damper, and controls
17	Base DX w/Free Reheat	$2,130		$133				0, 4	Reheat coil and controls assumed to be similar to Subcool Reheat cost.

Equipment Cost = BaseDXCost*(PrimCap+SecCap)*(1+Premium)
+ExtraPerPrim*PrimCap+ExtraPerSec*SecCap+ExtraPerOA*Oacfm+ExtraPerUnit

Where:

(A)	BaseDXCost	$/ton	Cost per Net Total Cooling Capacity for Base DX Unit, installed
(B)	Premium	%	Cost premium for custom or dual-path unit, percent of base cost
(C)	ExtraPerPrim	$/ton	Cost for extra options per Net Total Primary System Cooling Capacity
(D)	ExtraPerSec	$/ton	Cost for extra options per Net Total Secondary System Cooling Capacity
(E)	ExtraPerOA	$/cfm	Cost for extra options per outside air flow rate
(F)	ExtraPerUnit	$	Cost for extra options per unit (regardless of capacity)
	PrimCap	tons	Primary system Net Total Cooling Capacity
	SecCap	tons	Secondary system Net Total Cooling Capacity

References:

0. Engineering judgement inferred from other cost data.

1. RSMeans Mechanical Cost Data, 27th Ed., Reed Construction Data, Kingston, MA, 2004.
Table D3050 150, pp. 482-483, "Rooftop Single Zone Unit Systems - Electric cooling and gas heat".

2. Henderson, H.I., and D.B. Shirey, "Impacts of ASHRAE Standard 62-1989 on Florida Supermarkets",
Appendix A, Florida Solar Energy Center, FSEC-CR-910-96, October 1996.
1995 costs escalated by CPI change of 28%.

3. The Midwest CHP Application Center, "Spreadsheet for Evaluating Economics of CHP Systems", September 2004.
http://www.chpcentermw.org/docs/20040913CHPAssessorMAC.xls
"Equipment Factors" tab, Commercial Solid Desiccant in the 2000 cfm size range.

4. Manufacturer price quote, April 2002, escalated 6% to 2004.

Table 11.2 Location Dependent Cost Assumptions

City		Miami	Houston	Shreveport	Fort Worth	Atlanta	Sterling	St. Louis	New York	Chicago	Portland
State		FL	TX	LA	TX	GA	VA (DC)	MO	NY	IL	OR
City Code		MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Means Cost Factor		88.9%	87.7%	80.1%	82.9%	90.8%	95.6%	102.3%	129.6%	110.8%	103.3%
Average Electric Cost	$/kWh	$ 0.0761	$ 0.0790	$ 0.0758	$ 0.0790	$ 0.0688	$ 0.0745	$ 0.0580	$ 0.1298	$ 0.0754	$ 0.0645
2004 Average Gas Cost	$/Mcf	$ 11.46	$ 8.37	$ 10.32	$ 8.37	$ 11.60	$ 13.20	$ 10.13	$ 10.49	$ 9.12	$ 8.98
Heating Value	Btu/Mcf	10100	10100	10100	10100	10100	10100	10100	10100	10100	10100
2004 Average Gas Cost	$/therm	$ 1.16	$ 0.85	$ 1.04	$ 0.85	$ 1.17	$ 1.33	$ 1.02	$ 1.06	$ 0.92	$ 0.91
2004 Average Gas Cost	$/kWh	$ 0.0396	$ 0.0290	$ 0.0355	$ 0.0290	$ 0.0399	$ 0.0454	$ 0.0348	$ 0.0362	$ 0.0314	$ 0.0311

References:

RSMeans Mechanical Cost Data, 27th Ed., Reed Construction Data, Kingston, MA, 2004.
"City Cost Indexes," pp. 587ff

EIA, *Electric Sales, Revenue, and Average Price 2004,* "2004 Average Commercial Electricity Cost by State"
http://www.eia.doe.gov/cneaf/electricity/esr/esr_sum.html

EIA, *Gas Prices* , "2004 Average Commercial Gas Cost by State"
http://tonto.eia.doe.gov/dnav/ng/ng_pri_sum_dcu_nus_a.htm

Table 11.3 Energy Cost Escalation Assumptions

Discount Rate	0%	(Already accounted for in energy price projections)
Electric Cost Factor	14.3	(Sum of prices from 2004 thru 2018 divided by 2004 price)
Gas Cost Factor	14.6	(Sum of prices from 2004 thru 2018 divided by 2004 price)
Life	15	(years)

	2004	**2005**	**2006**	**2007**	**2008**	**2009**	**2010**	**2011**	**2012**	**2013**	**2014**	**2015**	**2016**	**2017**	**2018**
Commercial Electric Prices	8.2	8.0	8.7	8.6	8.2	7.9	7.8	7.6	7.5	7.4	7.5	7.5	7.4	7.4	7.4
Commercial Gas Prices	9.38	10.98	10.38	9.82	9.59	9.27	9.03	8.8	8.69	8.75	8.62	8.37	8.33	8.36	8.5

References:

From EIA Annual Energy Outlook 2006 (Early Release)
http://www.eia.doe.gov/oiaf/aeo/pdf/aeotab_13.pdf
Prices are in constant 2004 dollars, so no discount rate should be applied to calculate total payments over life.

The detailed results tables in Appendix B provide the following data for each combination of building type, location, and ventilation standard:

- Occupied hours when RH>65% [Annual Hrs]
- HVAC Electric Energy per Base DX S01 Net Total Capacity [Annual kWh/ton]
- HVAC Electric Demand per Base DX S01 Net Total Capacity [Annual Peak kW/ton]
- Heating+Regen Gas per Base DX S01 Net Total Capacity [Annual kWh/sensible ton]

A designer may use these results to estimate the annual energy use of all 17 system types given the capacity of a Case 1 system for a given project.

If heating is not provided by natural gas, the gas use can be adjusted by first dividing by 80% to account for the gas heating efficiency used in the analysis, and then multiplied by the seasonal efficiency of the desired heating system.

12 Results

12.1 Detailed Results

The results for the 2001 Ventilation Standard, Retail Store, Atlanta and the 2004 Ventilation Standard, Retail Store, Miami will be discussed in detail in this section. Bar charts of humidity control and HVAC energy use for all combinations of standard, building type, and location may be found in Appendix B.

12.1.1 Humidity Control

Figure 12.1 compares the level of humidity control for all system types, and Figure 12.2 compares the HVAC system energy use. (Note that for Retail, the 2001 ventilation standard is specified only as cfm/sf, so the DCV cases are meaningless for this application.) Highlights of the results are discussed below.

Referring to Figure 12.1, the humidity control results for the first four systems are not intuitive. One would expect to see a trend of better humidity control from Case 0 through Case 4. There are several key factors driving these results: continuous fan and resulting re-evaporation from the DX cooling coil (latent degradation), fan power or fan heat, and high percentage of part-load hours. Figure 12.5 shows five variations for Cases 0 through 6. The first three variations are “Continuous with Fan Heat & No Latent Degradation”, “Cycling with Fan Heat (Latent Degradation n/a)” and “Continuous No Fan Heat & No Latent Degradation”. These variations show the systems behaving as expected, where lower cfm/ton means lower rated SHR which results in better humidity control. The fourth variation adds latent degradation back in but still with no fan heat. The large change in humidity control illustrates how important latent degradation is with continuous fan operation. The last variation adds the fan heat back in as well and is the same as the result in Figure 12.1. This shows the significant impact of the fan heat as a reheat source which improves humidity control, but due to the varying amounts of fan heat for each case, the impact is not the same on each system type and alters the relative pattern of humidity control. Figure 12.3 shows the humidity control for 2004 Standard, Retail Store, Miami. For this application, the results are much more as expected with a strong trend of better humidity control from Case 0 to Case 4

12.1.2 Energy Use

Referring to Figures 12.2 and 12.4, the energy use patterns are generally as expected. The supply fan energy is generally greater than the DX system energy use, because the supply fan runs continuously all year when the HVAC system is on. For the system types with significant added pressure drops due to extra heat exchangers and coils such as Case 4 with the AAHX, the impact is quite significant. This fan power penalty acts as reheat and improves humidity control, but it is a penalty paid all year long. During the heating season, this added fan heat does offset some heating gas energy use, but the cost impact is smaller since heating in these climates would generally be provided by electric heat pumps or natural gas. On the other hand, the dual path units (Cases 11-14 and 16) have a fan power advantage, because they use nearly half the fan power per unit of outside air as the standard systems. The outside air units in the dual path systems are set up with two DX coils in series for an effective supply air flow rate of 150 cfm/ton.

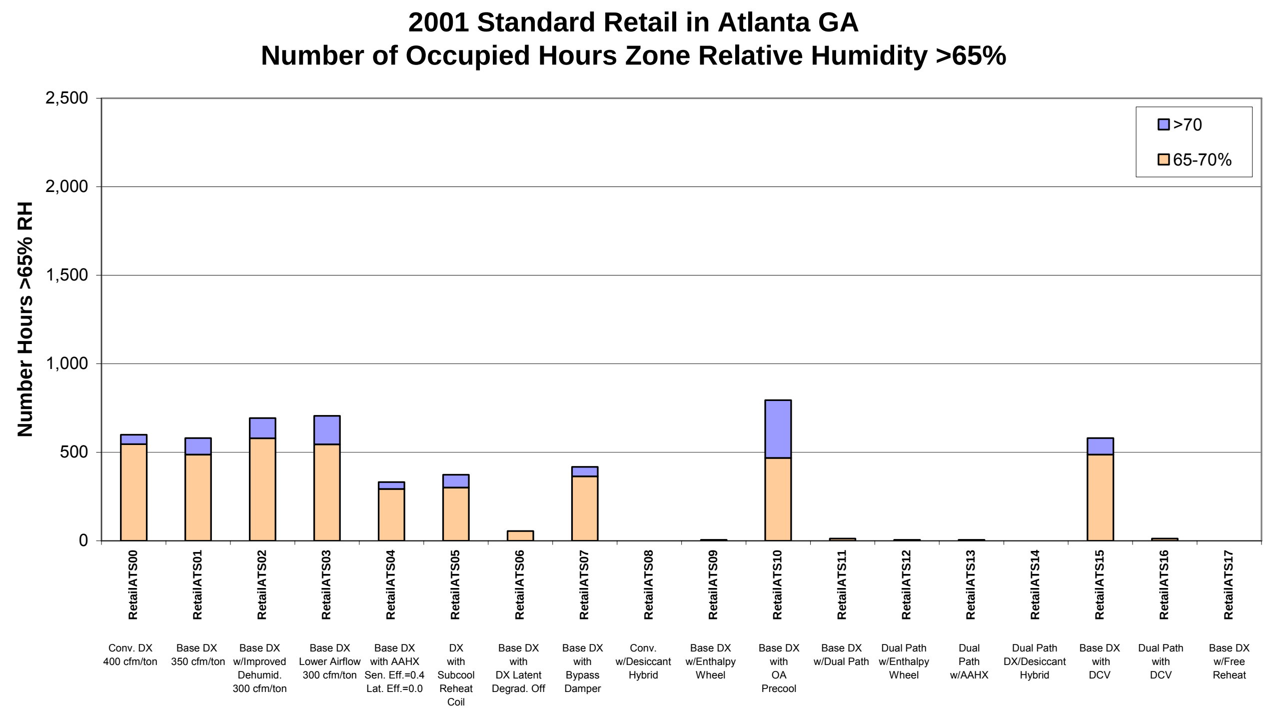

Figure 12.1 System Humidity Control for 2001 Standard Retail Store in Atlanta, GA.

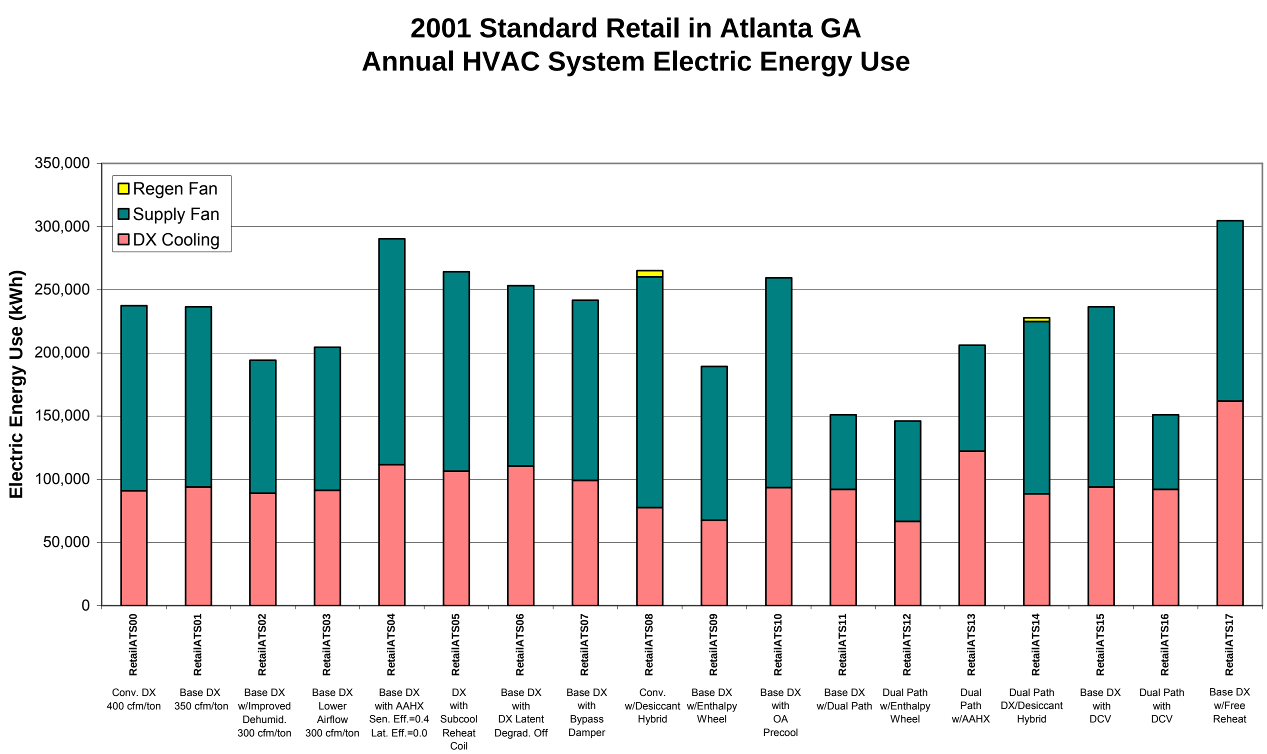

Figure 12.2 System Electricity Use for 2001 Standard Retail Store in Atlanta, GA.

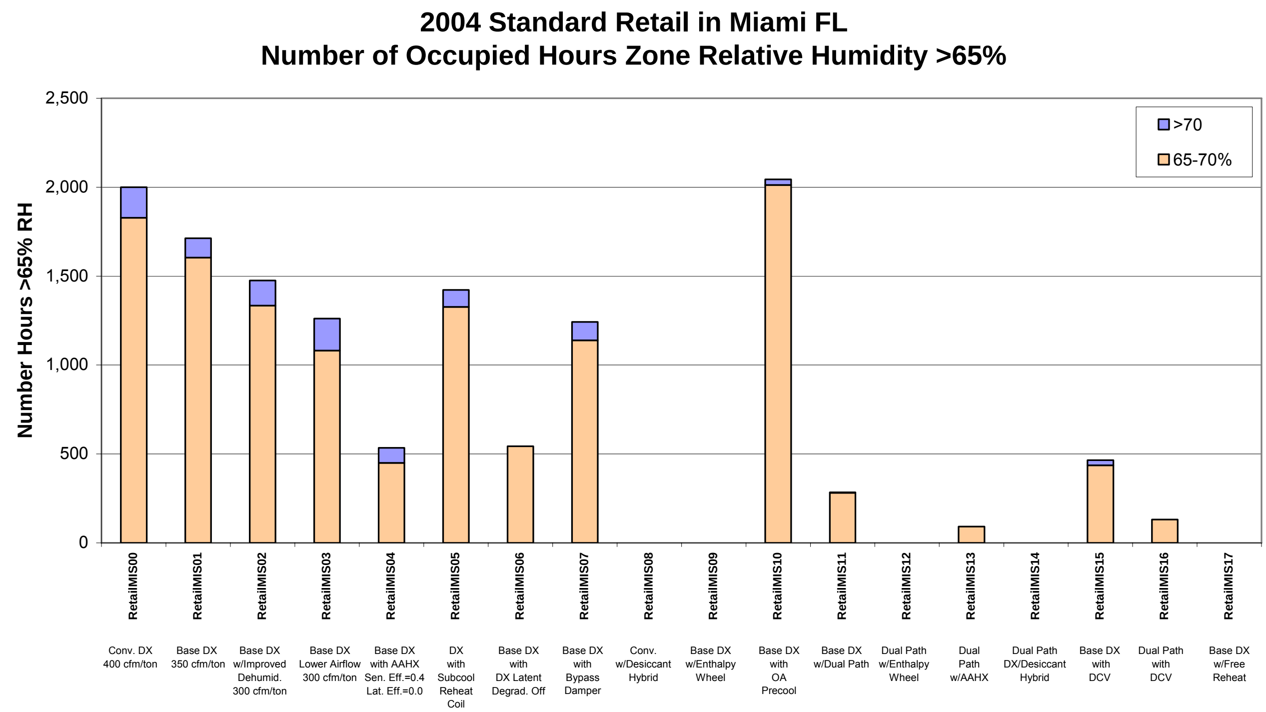

Figure 12.3 System Humidity Control for 2004 Standard Retail Store in Miami, FL.

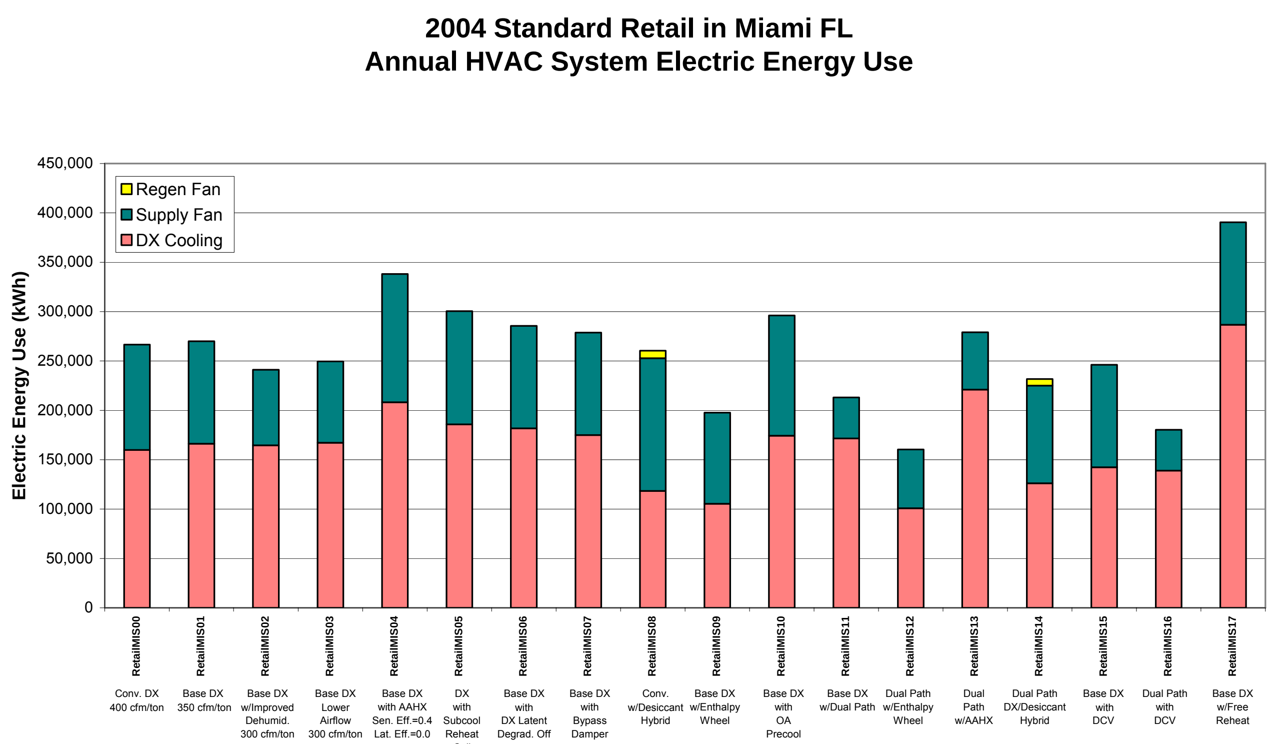

Figure 12.4 System Electricity Use for 2004 Standard Retail Store in Miami, FL.

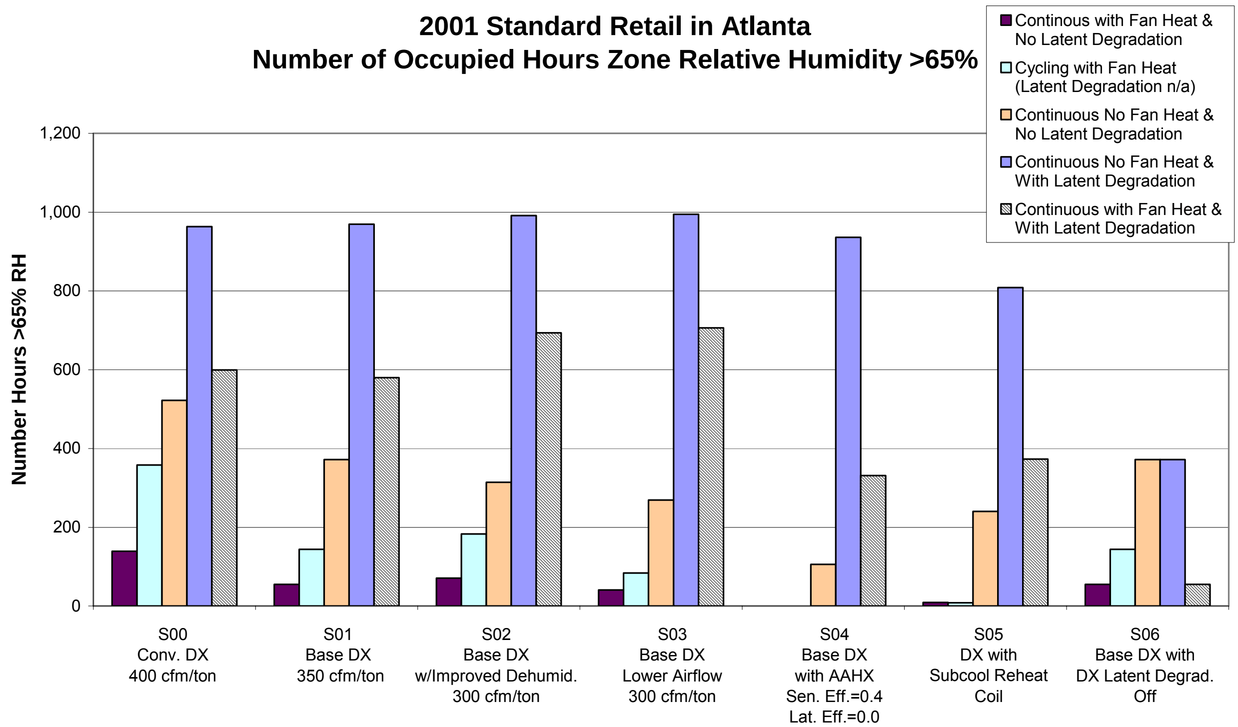

Figure 12.5 Impact of Fan Heat and Latent Degradation on Humidity Control.

12.2 Detailed Results Tables

The detailed results tables in Appendix B summarize the overall humidity control, equipment capacities, energy use, and life cycle cost results for all application types in all locations. Tables 12.1a through 12.1f are a sample of these tables for the 2001 Standard Retail. In these tables, the locations have been arranged with the most humid climates to the left and the driest climates to the right. The same order has been maintained throughout all of the tables, even though one might reorder certain locations differently for different building types. The reported results are:

Occupied Hours when RH>65%
Life Cycle Cost
Annual HVAC Energy Cost
Annual HVAC Source Energy
Net Total DX Cooling Capacity
Installed Equipment Cost
Net Sensible DX Cooling Capacity
HVAC Electric Energy per Base DX S01 Net Total Capacity
HVAC Electric Demand per Base DX S01 Net Total Capacity
Heating+Regen Gas per Base DX S01 Net Total Capacity
Net Total DX Cooling Capacity - Primary System
Net Total DX Cooling Capacity - Secondary System

These tables are primarily intended to provide detailed data. The next section discusses system performance comparisons.

Table 12.1a
Detailed Results – 2001 Standard Retail Store

Retail
2001 Standard

Occupied Hours when RH>65%
[Annual Hrs]

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	2384	1636	1169	699	599	455	471	257	233	0
01	Base DX	2279	1519	1066	675	580	438	441	280	226	0
02	DX w/Improved Dehumid.	2361	1653	1153	775	693	519	512	334	261	0
03	Base DX w/Lower Airflow	2273	1573	1082	755	706	514	483	340	266	0
04	Base DX w/AAHX	928	803	542	528	331	289	235	237	157	0
05	Base DX w/Subcool Reheat	1896	1167	777	454	373	281	312	203	179	0
06	Base DX w/o Lat. Coil Degrad.	1341	657	406	99	55	95	133	39	31	0
07	Base DX w/Bypass Damper	1958	1233	829	537	417	331	334	238	195	0
08	Base DX w/Desiccant	0	0	0	0	0	0	0	0	0	0
09	Base DX w/Enthalpy Wheel	19	5	14	1	5	4	1	2	0	0
10	Base DX w/OA Precool	2337	1724	1294	1180	794	620	730	260	217	0
11	Dual Path	653	252	78	38	13	25	2	18	6	0
12	Dual Path w/Enthalpy Wheel	4	5	11	0	5	0	0	0	0	0
13	Dual Path w/AAHX	241	169	53	19	6	0	2	6	3	0
14	Dual Path w/Desiccant	0	0	0	0	0	0	0	0	0	0
15	Base DX w/DCV	2279	1519	1066	675	580	438	441	280	226	0
16	Dual Path w/DCV	653	252	78	38	13	25	2	18	6	0
17	Base DX w/Free Reheat	3	12	10	2	0	0	1	4	0	0

Life Cycle Cost*
[1000 $2004]

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	622	676	650	711	618	779	729	1013	786	618
01	Base DX	640	695	667	732	635	797	749	1035	805	636
02	DX w/Improved Dehumid.	663	719	690	753	670	840	805	1072	857	691
03	Base DX w/Lower Airflow	624	678	652	712	627	791	750	1009	802	637
04	Base DX w/AAHX	791	846	801	879	747	912	866	1209	923	738
05	Base DX w/Subcool Reheat	701	757	722	794	678	840	791	1100	847	668
06	Base DX w/o Lat. Coil Degrad.	665	718	688	756	652	812	762	1057	818	642
07	Base DX w/Bypass Damper	679	736	704	774	669	833	788	1083	843	669
08	Base DX w/Desiccant	1457	1173	1109	1070	1008	1038	973	1282	933	793
09	Base DX w/Enthalpy Wheel	524	528	504	535	497	554	505	799	567	563
10	Base DX w/OA Precool	711	765	733	797	703	872	822	1150	887	711
11	Dual Path	636	703	677	745	663	835	808	1045	854	689
12	Dual Path w/Enthalpy Wheel	520	523	500	533	504	564	525	797	583	587
13	Dual Path w/AAHX	785	842	799	875	769	939	911	1194	961	776
14	Dual Path w/Desiccant	1425	1161	1076	1053	956	1009	976	1283	948	849
15	Base DX w/DCV	663	718	688	753	658	821	776	1068	833	662
16	Dual Path w/DCV	659	725	698	766	686	860	834	1078	882	716
17	Base DX w/Free Reheat	856	870	799	862	719	876	821	1135	867	660

** Installed Equipment Cost plus 15-yr HVAC Electric and Gas Cost in 1000s of 2004 dollars*

MI	=	Miami FL	ST	=	Washington DC
HO	=	Houston TX	SL	=	St. Louis MO
SH	=	Shreveport LA	NY	=	New York NY
FW	=	Fort Worth TX	CH	=	Chicago IL
AT	=	Atlanta GA	PO	=	Portland OR

Table 12.1b
Detailed Results – 2001 Standard Retail Store

Retail

2001 Standard

Annual HVAC Energy Cost
[1000 $2004]

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	25.35	27.42	27.09	28.59	24.14	33.63	28.05	42.00	31.81	21.31
01	Base DX	25.50	27.53	27.18	28.73	24.13	33.60	28.08	41.74	31.73	21.19
02	DX w/Improved Dehumid.	22.45	24.15	24.12	24.88	21.79	31.35	26.18	37.06	29.51	19.51
03	Base DX w/Lower Airflow	23.22	25.07	24.96	25.99	22.40	31.89	26.70	38.12	30.06	19.88
04	Base DX w/AAHX	31.61	33.21	32.01	33.88	27.35	36.60	30.70	46.95	34.41	23.07
05	Base DX w/Subcool Reheat	28.52	30.46	29.71	31.65	25.83	35.18	29.44	44.33	33.06	21.96
06	Base DX w/o Lat. Coil Degrad.	27.20	29.08	28.57	30.38	25.28	34.60	28.91	43.28	32.61	21.61
07	Base DX w/Bypass Damper	26.28	28.23	27.78	29.40	24.49	33.90	28.38	42.06	31.94	21.22
08	Base DX w/Desiccant	71.65	50.71	48.69	43.03	39.73	39.68	32.19	44.60	28.30	20.65
09	Base DX w/Enthalpy Wheel	18.72	19.19	19.10	19.78	16.84	19.02	14.78	28.20	16.98	17.39
10	Base DX w/OA Precool	27.52	29.10	28.72	29.66	25.86	35.54	29.44	45.11	33.83	22.95
11	Dual Path	18.90	21.44	21.78	22.89	19.59	29.27	24.56	32.81	27.22	17.48
12	Dual Path w/Enthalpy Wheel	15.30	15.81	16.04	16.64	14.21	16.34	12.61	23.27	14.24	15.24
13	Dual Path w/AAHX	25.39	27.32	26.76	28.38	22.98	32.32	27.24	37.55	29.84	18.96
14	Dual Path w/Desiccant	64.07	44.09	41.14	35.73	30.67	31.65	25.84	36.32	22.65	18.14
15	Base DX w/DCV	25.50	27.53	27.18	28.73	24.13	33.60	28.08	41.74	31.73	21.19
16	Dual Path w/DCV	18.90	21.44	21.78	22.89	19.59	29.27	24.56	32.81	27.22	17.48
17	Base DX w/Free Reheat	39.39	38.42	35.15	36.44	28.81	37.80	31.59	46.85	34.54	21.44

** Annual HVAC Energy Cost in 1000s of 2004 dollars using state average energy prices*

Annual HVAC Source Energy
[MWh]

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	1056	1092	1076	1134	954	1090	1179	1080	1169	882
01	Base DX	1062	1096	1079	1140	953	1087	1179	1074	1165	875
02	DX w/Improved Dehumid.	932	958	943	981	832	971	1057	961	1063	777
03	Base DX w/Lower Airflow	965	996	980	1027	862	999	1089	987	1088	798
04	Base DX w/AAHX	1320	1328	1288	1350	1112	1233	1340	1200	1287	981
05	Base DX w/Subcool Reheat	1189	1216	1188	1259	1036	1162	1262	1137	1225	918
06	Base DX w/o Lat. Coil Degrad.	1133	1159	1137	1206	1006	1130	1225	1112	1203	895
07	Base DX w/Bypass Damper	1095	1125	1104	1167	969	1100	1195	1082	1174	876
08	Base DX w/Desiccant	2218	1896	1693	1635	1386	1270	1335	1138	1085	906
09	Base DX w/Enthalpy Wheel	777	766	761	785	700	701	708	712	673	732
10	Base DX w/OA Precool	1146	1160	1142	1177	1030	1165	1249	1158	1252	958
11	Dual Path	781	846	837	898	713	858	947	860	954	657
12	Dual Path w/Enthalpy Wheel	632	628	628	656	571	575	579	592	552	615
13	Dual Path w/AAHX	1055	1085	1051	1122	879	1001	1105	975	1071	740
14	Dual Path w/Desiccant	1978	1651	1442	1368	1104	1030	1092	926	872	774
15	Base DX w/DCV	1062	1096	1079	1140	953	1087	1179	1074	1165	875
16	Dual Path w/DCV	781	846	837	898	713	858	947	860	954	657
17	Base DX w/Free Reheat	1645	1537	1415	1451	1170	1267	1373	1200	1284	887

** Source Energy = Gas Energy + Electric Energy/31.3%*
Electricity delivery efficiency of 31.3% from DOE 2004 Buildings Energy Databook, p. 6-4

MI	=	Miami FL	ST	=	Washington DC
HO	=	Houston TX	SL	=	St. Louis MO
SH	=	Shreveport LA	NY	=	New York NY
FW	=	Fort Worth TX	CH	=	Chicago IL
AT	=	Atlanta GA	PO	=	Portland OR

Table 12.1c
Detailed Results – 2001 Standard Retail Store

Retail

2001 Standard

Net Total DX Cooling Capacity*
[tons]

Case	System / Location ==>	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	137.2	151.9	153.4	170.8	140.0	144.2	148.7	148.0	138.3	141.2
01	Base DX	145.7	161.3	162.9	181.4	148.7	153.1	157.9	157.1	146.9	149.9
02	DX w/Improved Dehumid.	149.2	165.2	166.8	185.8	152.3	156.8	161.7	160.9	150.4	153.5
03	Base DX w/Lower Airflow	154.4	170.9	172.5	192.1	157.5	162.2	167.2	166.4	155.6	158.8
04	Base DX w/AAHX	146.7	162.4	164.0	182.7	149.7	154.2	159.0	158.2	147.9	150.9
05	Base DX w/Subcool Reheat	146.1	161.8	163.3	181.9	149.1	153.6	158.3	157.6	147.3	150.3
06	Base DX w/o Lat. Coil Degrad.	145.7	161.3	162.9	181.4	148.7	153.1	157.9	157.1	146.9	149.9
07	Base DX w/Bypass Damper	145.7	161.3	162.9	181.4	148.7	153.1	157.9	157.1	146.9	149.9
08	Base DX w/Desiccant	138.1	152.8	154.2	171.7	140.9	145.1	149.5	148.8	139.2	142.0
09	Base DX w/Enthalpy Wheel	93.8	93.7	93.0	100.5	90.4	96.0	92.6	101.1	95.0	100.3
10	Base DX w/OA Precool	138.5	154.1	155.6	174.2	141.5	145.9	150.7	149.9	139.7	142.7
11	Dual Path	152.0	166.7	168.2	185.6	154.8	159.0	163.5	162.8	153.2	156.0
12	Dual Path w/Enthalpy Wheel	92.3	92.2	91.5	98.6	89.0	94.3	91.2	99.2	93.4	98.4
13	Dual Path w/AAHX	151.5	166.2	167.6	185.1	154.3	158.5	162.9	162.2	152.6	155.4
14	Dual Path w/Desiccant	140.7	155.4	156.8	174.3	143.5	147.7	152.1	151.4	141.8	144.6
15	Base DX w/DCV	145.7	161.3	162.9	181.4	148.7	153.1	157.9	157.1	146.9	149.9
16	Dual Path w/DCV	152.0	166.7	168.2	185.6	154.8	159.0	163.5	162.8	153.2	156.0
17	Base DX w/Free Reheat	145.7	161.3	162.9	181.4	148.7	153.1	157.9	157.1	146.9	149.9

** Capacity of Primary plus Secondary systems where applicable (Case 10-14 & 16)*

Installed Equipment Cost*
[1000 $2004]

Case	System / Location ==>	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	260	284	262	302	271	294	324	408	327	311
01	Base DX	276	301	278	320	288	312	344	434	347	330
02	DX w/Improved Dehumid.	342	373	344	397	356	386	426	537	430	409
03	Base DX w/Lower Airflow	292	319	294	339	305	330	364	459	367	349
04	Base DX w/AAHX	340	371	342	394	354	384	424	534	427	406
05	Base DX w/Subcool Reheat	294	321	296	341	306	332	367	462	369	351
06	Base DX w/o Lat. Coil Degrad.	276	302	278	321	288	312	345	435	347	330
07	Base DX w/Bypass Damper	304	332	306	353	317	343	379	477	382	363
08	Base DX w/Desiccant	420	441	406	451	434	465	508	641	526	496
09	Base DX w/Enthalpy Wheel	257	253	230	251	256	280	293	394	323	312
10	Base DX w/OA Precool	317	348	321	372	331	360	397	501	399	380
11	Dual Path	366	395	364	416	380	411	452	571	459	436
12	Dual Path w/Enthalpy Wheel	301	297	269	295	299	329	343	463	378	367
13	Dual Path w/AAHX	422	451	415	468	438	472	517	652	529	501
14	Dual Path w/Desiccant	496	524	482	538	514	552	603	761	622	588
15	Base DX w/DCV	299	324	298	342	311	336	370	467	375	356
16	Dual Path w/DCV	388	418	385	438	404	436	479	604	487	462
17	Base DX w/Free Reheat	293	320	295	340	306	331	366	461	368	350

** Installed Equipment Cost in 1000s of 2004 dollars (Representative costs only, get current quotes.)*

MI	=	Miami FL	ST	=	Washington DC
HO	=	Houston TX	SL	=	St. Louis MO
SH	=	Shreveport LA	NY	=	New York NY
FW	=	Fort Worth TX	CH	=	Chicago IL
AT	=	Atlanta GA	PO	=	Portland OR

Table 12.1d
Detailed Results – 2001 Standard Retail Store

Retail

2001 Standard

Net Sensible DX Cooling Capacity*
[tons]

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
01	Base DX	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
02	DX w/Improved Dehumid.	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
03	Base DX w/Lower Airflow	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
04	Base DX w/AAHX	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
05	Base DX w/Subcool Reheat	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
06	Base DX w/o Lat. Coil Degrad.	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
07	Base DX w/Bypass Damper	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
08	Base DX w/Desiccant	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
09	Base DX w/Enthalpy Wheel	61.3	61.3	60.8	65.8	59.1	62.8	60.6	66.2	62.1	65.6
10	Base DX w/OA Precool	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
11	Dual Path	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
12	Dual Path w/Enthalpy Wheel	62.3	62.3	61.8	66.8	60.1	63.8	61.6	67.2	63.1	66.6
13	Dual Path w/AAHX	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
14	Dual Path w/Desiccant	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
15	Base DX w/DCV	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
16	Dual Path w/DCV	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1
17	Base DX w/Free Reheat	96.3	106.6	107.7	119.9	98.3	101.3	104.4	103.9	97.1	99.1

** Capacity of Primary plus Secondary systems where applicable (Case 10-14 & 16)*

HVAC Electric Energy per Base DX S01 Net Total Capacity*
[Annual kWh/ton]

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
	S01 Net Cap==>	145.7	161.2	162.8	181.3	148.6	153.1	157.8	157.1	146.9	149.9
00	Conventional DX	2241	1933	1810	1737	1598	1468	1548	1305	1342	1172
01	Base DX	2254	1939	1814	1744	1592	1458	1542	1288	1328	1151
02	DX w/Improved Dehumid.	1971	1652	1528	1445	1306	1180	1262	1014	1054	889
03	Base DX w/Lower Airflow	2043	1728	1604	1530	1376	1246	1333	1077	1119	947
04	Base DX w/AAHX	2811	2404	2240	2131	1953	1795	1901	1585	1636	1425
05	Base DX w/Subcool Reheat	2529	2176	2034	1960	1778	1626	1723	1432	1475	1264
06	Base DX w/o Lat. Coil Degrad.	2407	2062	1927	1859	1704	1546	1633	1364	1407	1194
07	Base DX w/Bypass Damper	2324	1994	1863	1791	1627	1484	1575	1304	1347	1154
08	Base DX w/Desiccant	2204	1946	1862	1777	1785	1664	1701	1536	1577	1440
09	Base DX w/Enthalpy Wheel	1638	1369	1294	1206	1273	1184	1173	1089	1135	1026
10	Base DX w/OA Precool	2434	2058	1928	1801	1746	1608	1671	1443	1497	1309
11	Dual Path	1640	1412	1296	1278	1016	898	998	752	750	566
12	Dual Path w/Enthalpy Wheel	1324	1092	1023	970	983	905	899	824	847	745
13	Dual Path w/AAHX	2231	1888	1723	1678	1387	1215	1336	1014	1037	777
14	Dual Path w/Desiccant	1941	1723	1640	1586	1533	1400	1450	1266	1291	1133
15	Base DX w/DCV	2254	1939	1814	1744	1592	1458	1542	1288	1328	1151
16	Dual Path w/DCV	1640	1412	1296	1278	1016	898	998	752	750	566
17	Base DX w/Free Reheat	3508	2798	2461	2282	2050	1826	1927	1539	1582	1177

**All systems are normalized by the same tons in a given city to provide common comparison point.*

MI	=	Miami FL	ST	=	Washington DC
HO	=	Houston TX	SL	=	St. Louis MO
SH	=	Shreveport LA	NY	=	New York NY
FW	=	Fort Worth TX	CH	=	Chicago IL
AT	=	Atlanta GA	PO	=	Portland OR

Table 12.1e
Detailed Results – 2001 Standard Retail Store

Retail

2001 Standard

HVAC Electric Demand per Base DX S01 Net Total Capacity*
[Annual Peak kW/ton]

		Location ==>	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Case	**System**	**S01 Net Cap==>**	145.7	161.2	162.8	181.3	148.6	153.1	157.8	157.1	146.9	149.9
00	Conventional DX		1.0	1.0	1.0	0.9	1.0	1.0	1.0	0.9	0.9	0.7
01	Base DX		1.0	1.0	1.0	1.0	1.0	1.1	1.1	0.9	1.0	0.7
02	DX w/Improved Dehumid.		1.0	1.0	1.0	0.9	1.0	1.0	1.0	0.8	0.9	0.7
03	Base DX w/Lower Airflow		1.0	1.0	1.0	0.9	1.0	1.1	1.1	0.9	0.9	0.7
04	Base DX w/AAHX		1.3	1.3	1.4	1.2	1.3	1.4	1.4	1.1	1.2	0.9
05	Base DX w/Subcool Reheat		1.1	1.1	1.2	1.1	1.2	1.3	1.4	1.0	1.1	0.8
06	Base DX w/o Lat. Coil Degrad.		1.0	1.0	1.1	1.0	1.1	1.1	1.1	0.9	1.0	0.8
07	Base DX w/Bypass Damper		1.0	1.0	1.1	1.0	1.1	1.1	1.1	0.9	1.0	0.8
08	Base DX w/Desiccant		0.8	0.7	0.7	0.6	0.7	0.8	0.8	0.6	0.7	0.6
09	Base DX w/Enthalpy Wheel		0.6	0.5	0.6	0.5	0.6	0.6	0.6	0.5	0.6	0.5
10	Base DX w/OA Precool		1.1	1.0	1.0	0.8	1.0	1.1	1.1	0.8	1.0	0.7
11	Dual Path		0.9	0.9	0.9	0.8	0.9	1.0	1.0	0.7	0.8	0.6
12	Dual Path w/Enthalpy Wheel		0.5	0.5	0.5	0.4	0.5	0.5	0.5	0.5	0.5	0.4
13	Dual Path w/AAHX		1.1	1.1	1.2	1.1	1.2	1.2	1.2	1.0	1.1	0.8
14	Dual Path w/Desiccant		0.7	0.7	0.7	0.6	0.7	0.8	0.7	0.6	0.7	0.6
15	Base DX w/DCV		1.0	1.0	1.0	1.0	1.0	1.1	1.1	0.9	1.0	0.7
16	Dual Path w/DCV		0.9	0.9	0.9	0.8	0.9	1.0	1.0	0.7	0.8	0.6
17	Base DX w/Free Reheat		1.2	1.3	1.3	1.3	1.3	1.3	1.3	1.2	1.2	0.8

**All systems are normalized by the same tons in a given city to provide common comparison point.*

Heating+Regen Gas per Base DX S01 Net Total Capacity*
[Annual kWh/ton]

		Location ==>	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Case	**System**	**S01 Net Cap==>**	145.7	161.2	162.8	181.3	148.6	153.1	157.8	157.1	146.9	149.9
00	Conventional DX		88	598	822	704	1315	2428	2528	2707	3675	2141
01	Base DX		89	604	829	714	1324	2441	2542	2721	3693	2160
02	DX w/Improved Dehumid.		103	663	910	793	1421	2574	2663	2882	3866	2342
03	Base DX w/Lower Airflow		100	653	894	772	1404	2543	2640	2845	3831	2302
04	Base DX w/AAHX		78	553	756	638	1243	2320	2421	2574	3534	1994
05	Base DX w/Subcool Reheat		84	586	797	678	1290	2393	2489	2660	3626	2090
06	Base DX w/o Lat. Coil Degrad.		89	601	828	713	1324	2441	2542	2721	3692	2159
07	Base DX w/Bypass Damper		89	604	829	714	1324	2441	2542	2721	3693	2160
08	Base DX w/Desiccant		8184	5542	4448	3342	3621	2977	3025	2335	2350	1444
09	Base DX w/Enthalpy Wheel		97	374	541	475	644	795	735	1053	959	1604
10	Base DX w/OA Precool		92	618	852	733	1349	2475	2575	2762	3741	2211
11	Dual Path		125	737	1001	870	1551	2738	2809	3075	4101	2576
12	Dual Path w/Enthalpy Wheel		108	407	591	521	702	865	798	1139	1053	1724
13	Dual Path w/AAHX		113	698	951	826	1483	2655	2734	2968	3982	2457
14	Dual Path w/Desiccant		7376	4734	3615	2474	2527	2255	2288	1849	1810	1543
15	Base DX w/DCV		89	604	829	714	1324	2441	2542	2721	3693	2160
16	Dual Path w/DCV		125	737	1001	870	1551	2738	2809	3075	4101	2576
17	Base DX w/Free Reheat		87	594	826	711	1324	2441	2541	2721	3692	2160

**All systems are normalized by the same tons in a given city to provide common comparison point.*

MI	=	Miami FL	ST	=	Washington DC
HO	=	Houston TX	SL	=	St. Louis MO
SH	=	Shreveport LA	NY	=	New York NY
FW	=	Fort Worth TX	CH	=	Chicago IL
AT	=	Atlanta GA	PO	=	Portland OR

Table 12.1f
Detailed Results – 2001 Standard Retail Store

Retail

2001 Standard

Net Total DX Cooling Capacity - Primary System
[tons]

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	137.2	151.9	153.3	170.8	140.0	144.2	148.6	147.9	138.3	141.1
01	Base DX	145.7	161.2	162.8	181.3	148.6	153.1	157.8	157.1	146.9	149.8
02	DX w/Improved Dehumid.	149.2	165.1	166.7	185.7	152.2	156.8	161.6	160.9	150.4	153.5
03	Base DX w/Lower Airflow	154.3	170.8	172.5	192.1	157.4	162.2	167.2	166.4	155.6	158.7
04	Base DX w/AAHX	146.7	162.4	163.9	182.6	149.7	154.2	158.9	158.2	147.9	150.9
05	Base DX w/Subcool Reheat	146.1	161.7	163.3	181.9	149.1	153.6	158.3	157.5	147.3	150.3
06	Base DX w/o Lat. Coil Degrad.	145.7	161.2	162.8	181.3	148.6	153.1	157.8	157.1	146.9	149.8
07	Base DX w/Bypass Damper	145.7	161.2	162.8	181.3	148.6	153.1	157.8	157.1	146.9	149.8
08	Base DX w/Desiccant	138.1	152.7	154.2	171.6	140.8	145.0	149.5	148.8	139.2	142.0
09	Base DX w/Enthalpy Wheel	93.8	93.7	93.0	100.5	90.3	96.0	92.6	101.1	94.9	100.2
10	Base DX w/OA Precool	99.2	114.8	116.4	134.9	102.2	106.7	111.4	110.6	100.4	103.4
11	Dual Path	148.5	147.6	147.5	146.4	148.3	148.1	147.8	147.8	148.4	148.3
12	Dual Path w/Enthalpy Wheel	61.6	61.6	61.7	61.2	61.8	61.5	61.7	61.2	61.5	61.2
13	Dual Path w/AAHX	136.8	135.8	135.7	134.6	136.6	136.3	136.0	136.1	136.7	136.5
14	Dual Path w/Desiccant	58.5	57.6	57.5	56.3	58.3	58.0	57.8	57.8	58.4	58.2
15	Base DX w/DCV	145.7	161.2	162.8	181.3	148.6	153.1	157.8	157.1	146.9	149.8
16	Dual Path w/DCV	148.5	147.6	147.5	146.4	148.3	148.1	147.8	147.8	148.4	148.3
17	Base DX w/Free Reheat	145.7	161.2	162.8	181.3	148.6	153.1	157.8	157.1	146.9	149.8

Net Total DX Cooling Capacity - Secondary System
[tons]

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
01	Base DX	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
02	DX w/Improved Dehumid.	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
03	Base DX w/Lower Airflow	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
04	Base DX w/AAHX	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
05	Base DX w/Subcool Reheat	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
06	Base DX w/o Lat. Coil Degrad.	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
07	Base DX w/Bypass Damper	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
08	Base DX w/Desiccant	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
09	Base DX w/Enthalpy Wheel	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
10	Base DX w/OA Precool	39.2	39.2	39.2	39.2	39.2	39.2	39.2	39.2	39.2	39.2
11	Dual Path	3.5	19.1	20.6	39.2	6.4	10.9	15.7	14.9	4.7	7.7
12	Dual Path w/Enthalpy Wheel	30.6	30.6	29.8	37.4	27.2	32.8	29.5	38.0	31.8	37.1
13	Dual Path w/AAHX	14.7	30.3	31.9	50.4	17.7	22.1	26.9	26.1	15.9	18.9
14	Dual Path w/Desiccant	82.2	97.8	99.3	117.9	85.1	89.6	94.3	93.6	83.3	86.3
15	Base DX w/DCV	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
16	Dual Path w/DCV	3.5	19.1	20.6	39.2	6.4	10.9	15.7	14.9	4.7	7.7
17	Base DX w/Free Reheat	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

MI	=	Miami FL	ST	=	Washington DC
HO	=	Houston TX	SL	=	St. Louis MO
SH	=	Shreveport LA	NY	=	New York NY
FW	=	Fort Worth TX	CH	=	Chicago IL
AT	=	Atlanta GA	PO	=	Portland OR

12.3 Comparative Results and Trends

Tables 12.2 through 12.8 compare the humidity control, energy cost, and life cycle cost results for all 18 system types in all locations. Each table presents results for a single combination of ventilation standard and building type. In these tables, as in the detailed results tables, the locations have been arranged with the most humid climates to the left and the driest climates to the right. The following sub-tables are presented:

- **Humidity control** is ranked based on occupied hours with RH>65%. The number of hours is shown, and four shades of color are used to rank the performance. The shading ranges are listed at the bottom of the table.

- **Relative Annual HVAC Energy Cost** is shown as percent change compared to the Base DX Case 1. Negative values indicate energy cost savings, and positive values indicate higher energy cost than Case 1.

- **Best Cases** presents the lowest cost system type for each of the following criteria:

  Minimum Energy Cost (EC) – The system with the lowest annual HVAC energy cost.

  Minimum EC, <=150 hrs RH>65% – The system with the lowest annual HVAC energy cost selected only from systems capable of controlling humidity to 150 occupied hours or less above 65%RH. For some cases, none of the systems were capable of this level of control, and "NA" is reported. The 150 hour criterion is an arbitrary number chosen to represent good humidity control.

  Minimum Life Cycle Cost (LCC) – The system with the lowest life cycle cost (installed equipment cost plus 15-yr HVAC energy costs).

  Minimum LCC, <=150 hrs RH>65% – The system with the lowest life cycle cost selected only from systems capable of controlling humidity to 150 occupied hours or less above 65%RH. For some cases, none of the systems were capable of this level of control, and "NA" is reported. The 150 hour criterion is an arbitrary number chosen to represent good humidity control.

  Ratio Min LCC<=150 to Case 01 LCC – This is the ratio of the LCC of the system which meets the "Minimum LCC, <=150 hrs RH>65%" criterion to the LCC of the Case 01 Base DX system. A value less than 1.0 indicates a system with a lower LCC than Case 01.

Table 12.2a
System Performance Comparison – 2001 Standard Office

Office

2001 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	229	134	45	41	10	12	6	2	1	0
01	Base DX	228	111	44	43	7	14	5	5	1	0
02	DX w/Improved Dehumid.	284	135	49	66	8	17	7	5	0	0
03	Base DX w/Lower Airflow	314	148	58	84	10	23	9	9	1	0
04	Base DX w/AAHX	113	54	21	28	2	4	3	0	0	0
05	Base DX w/Subcool Reheat	128	63	29	29	2	9	2	0	0	0
06	Base DX w/o Lat. Coil Degrad.	8	5	2	0	0	0	0	0	0	0
07	Base DX w/Bypass Damper	160	81	30	32	3	8	3	1	0	0
08	Base DX w/Desiccant	0	0	0	0	0	0	0	0	0	0
09	Base DX w/Enthalpy Wheel	0	0	0	0	0	0	0	0	0	0
10	Base DX w/OA Precool	450	349	142	238	19	18	69	36	4	0
11	Dual Path	0	4	0	0	0	0	0	0	0	0
12	Dual Path w/Enthalpy Wheel	0	0	0	0	0	0	0	0	0	0
13	Dual Path w/AAHX	0	1	0	0	0	0	0	0	0	0
14	Dual Path w/Desiccant	0	0	0	0	0	0	0	0	0	0
15	Base DX w/DCV	0	0	0	0	0	0	0	0	0	0
16	Dual Path w/DCV	0	0	0	0	0	0	0	0	0	0
17	Base DX w/Free Reheat	0	1	0	0	0	0	0	0	0	0

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	-2%	-1%	-1%	-1%	-1%	0%	-1%	0%	0%	0%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-11%	-11%	-10%	-12%	-8%	-7%	-7%	-11%	-7%	-9%
03	Base DX w/Lower Airflow	-7%	-7%	-7%	-8%	-6%	-5%	-5%	-8%	-5%	-7%
04	Base DX w/AAHX	20%	19%	17%	18%	13%	10%	10%	13%	9%	10%
05	Base DX w/Subcool Reheat	10%	9%	8%	8%	5%	4%	4%	6%	4%	4%
06	Base DX w/o Lat. Coil Degrad.	5%	4%	3%	4%	3%	2%	2%	2%	2%	2%
07	Base DX w/Bypass Damper	3%	2%	2%	2%	1%	1%	1%	1%	1%	0%
08	Base DX w/Desiccant	88%	40%	35%	24%	21%	4%	1%	2%	-10%	-2%
09	Base DX w/Enthalpy Wheel	-17%	-19%	-19%	-18%	-18%	-26%	-30%	-19%	-27%	-11%
10	Base DX w/OA Precool	4%	4%	4%	3%	4%	4%	4%	5%	4%	6%
11	Dual Path	-5%	-7%	-7%	-6%	-7%	-6%	-5%	-10%	-8%	-10%
12	Dual Path w/Enthalpy Wheel	-24%	-27%	-27%	-26%	-26%	-33%	-36%	-27%	-34%	-19%
13	Dual Path w/AAHX	6%	3%	2%	3%	1%	0%	0%	-2%	-2%	-4%
14	Dual Path w/Desiccant	70%	27%	22%	11%	9%	-7%	-9%	-10%	-20%	-10%
15	Base DX w/DCV	-12%	-13%	-13%	-11%	-16%	-23%	-25%	-16%	-25%	-16%
16	Dual Path w/DCV	-19%	-21%	-22%	-19%	-26%	-32%	-33%	-29%	-35%	-30%
17	Base DX w/Free Reheat	26%	18%	12%	10%	6%	5%	5%	4%	2%	0%

Occupied Hours >65%RH		% Change in Annual HVAC Energy Cost	
(white)	<= 150 hrs	(white)	< 0% (less energy use)
(yellow)	151 to 1000 hrs	(yellow)	1% to 25% (more energy use)
(cyan)	1001 to 2000 hrs	(cyan)	26% to 50% (more energy use)
(magenta)	> 2000 hrs	(magenta)	>50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	12	12	12	12	16	12	12	16	16	16
Minimum EC, <=150 hrs RH>65%	12	12	12	12	16	12	12	16	16	16
Minimum Life Cycle Cost (LCC)	09	09	09	09	09	09	09	09	09	15
Minimum LCC, <=150 hrs RH>65%	09	09	09	09	09	09	09	09	09	15
Ratio Min LCC<=150 to Case 01 LCC	0.9	0.8	0.8	0.8	0.9	0.8	0.8	0.9	0.8	0.9

MI	=	Miami FL	ST	=	Washington DC
HO	=	Houston TX	SL	=	St. Louis MO
SH	=	Shreveport LA	NY	=	New York NY
FW	=	Fort Worth TX	CH	=	Chicago IL
AT	=	Atlanta GA	PO	=	Portland OR

Table 12.2b
System Performance Comparison – 2004 Standard Office

Office
2004 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	15	9	2	1	0	1	0	0	0	0
01	Base DX	20	14	2	1	0	1	0	0	0	0
02	DX w/Improved Dehumid.	26	21	4	2	0	1	0	0	0	0
03	Base DX w/Lower Airflow	45	22	4	2	0	1	0	0	0	0
04	Base DX w/AAHX	13	8	0	0	0	0	0	0	0	0
05	Base DX w/Subcool Reheat	9	6	0	0	0	0	0	0	0	0
06	Base DX w/o Lat. Coil Degrad.	0	0	0	0	0	0	0	0	0	0
07	Base DX w/Bypass Damper	12	9	0	1	0	1	0	0	0	0
08	Base DX w/Desiccant	0	0	0	0	0	0	0	0	0	0
09	Base DX w/Enthalpy Wheel	0	0	0	0	0	0	0	0	0	0
10	Base DX w/OA Precool	142	63	14	43	1	3	8	2	0	0
11	Dual Path	0	0	0	0	0	0	0	0	0	0
12	Dual Path w/Enthalpy Wheel	0	0	0	0	0	0	0	0	0	0
13	Dual Path w/AAHX	0	0	0	0	0	0	0	0	0	0
14	Dual Path w/Desiccant	0	0	0	0	0	0	0	0	0	0
15	Base DX w/DCV	4	0	0	0	0	0	0	0	0	0
16	Dual Path w/DCV	0	0	0	0	0	0	0	0	0	0
17	Base DX w/Free Reheat	0	0	0	0	0	0	0	0	0	0

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	-2%	-2%	-1%	-1%	-1%	0%	-1%	0%	0%	0%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-11%	-11%	-10%	-11%	-9%	-7%	-7%	-12%	-7%	-9%
03	Base DX w/Lower Airflow	-6%	-7%	-7%	-8%	-6%	-5%	-5%	-9%	-5%	-7%
04	Base DX w/AAHX	19%	18%	17%	18%	14%	11%	11%	14%	10%	11%
05	Base DX w/Subcool Reheat	9%	8%	7%	7%	5%	4%	4%	6%	4%	4%
06	Base DX w/o Lat. Coil Degrad.	4%	3%	3%	3%	2%	2%	2%	2%	1%	2%
07	Base DX w/Bypass Damper	3%	2%	1%	1%	1%	1%	1%	0%	0%	0%
08	Base DX w/Desiccant	52%	23%	18%	12%	9%	1%	-2%	1%	-8%	-1%
09	Base DX w/Enthalpy Wheel	-12%	-14%	-13%	-13%	-12%	-17%	-20%	-12%	-18%	-7%
10	Base DX w/OA Precool	2%	2%	2%	2%	3%	3%	2%	3%	3%	4%
11	Dual Path	-3%	-5%	-5%	-5%	-5%	-5%	-5%	-7%	-7%	-8%
12	Dual Path w/Enthalpy Wheel	-18%	-21%	-20%	-19%	-19%	-23%	-26%	-19%	-26%	-14%
13	Dual Path w/AAHX	4%	1%	0%	1%	0%	-1%	-1%	-2%	-3%	-3%
14	Dual Path w/Desiccant	48%	18%	14%	8%	8%	-3%	-5%	-6%	-14%	-8%
15	Base DX w/DCV	-2%	-2%	-2%	-2%	-3%	-4%	-5%	-3%	-5%	-3%
16	Dual Path w/DCV	-5%	-7%	-8%	-6%	-8%	-10%	-10%	-11%	-12%	-11%
17	Base DX w/Free Reheat	16%	11%	6%	5%	2%	2%	2%	2%	1%	0%

Occupied Hours >65%RH	*% Change in Annual HVAC Energy Cost*
(white) <= 150 hrs	(white) < 0% (less energy use)
(yellow) 151 to 1000 hrs	(yellow) 1% to 25% (more energy use)
(cyan) 1001 to 2000 hrs	(cyan) 26% to 50% (more energy use)
(magenta) > 2000 hrs	(magenta) >50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	12	12	12	12	12	12	12	12	12	12
Minimum EC, <=150 hrs RH>65%	12	12	12	12	12	12	12	12	12	12
Minimum Life Cycle Cost (LCC)	09	09	09	09	09	09	09	09	09	09
Minimum LCC, <=150 hrs RH>65%	09	09	09	09	09	09	09	09	09	09
Ratio Min LCC<=150 to Case 01 LCC	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	1.0

MI = Miami FL; ST = Washington DC
HO = Houston TX; SL = St. Louis MO
SH = Shreveport LA; NY = New York NY
FW = Fort Worth TX; CH = Chicago IL
AT = Atlanta GA; PO = Portland OR

Table 12.3a
System Performance Comparison – 2001 Standard Restaurant

Restaurant
2001 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	4550	3669	2662	2092	2011	1489	1911	947	819	40
01	Base DX	4554	3656	2650	2039	2015	1489	1459	946	813	40
02	DX w/Improved Dehumid.	4598	3698	2722	2167	2063	1540	1530	1007	855	45
03	Base DX w/Lower Airflow	4583	3658	2684	2114	2049	1520	1489	987	844	44
04	Base DX w/AAHX	3992	3218	2273	1939	1887	1361	1245	926	711	44
05	Base DX w/Subcool Reheat	4442	3486	2505	1729	1888	1369	1340	872	755	39
06	Base DX w/o Lat. Coil Degrad.	4140	3287	2304	1398	1650	1206	1171	730	623	36
07	Base DX w/Bypass Damper	4500	3563	2572	1853	1948	1432	1389	910	783	40
08	Base DX w/Desiccant	53	234	63	4	0	4	51	0	7	0
09	Base DX w/Enthalpy Wheel	4485	3675	2621	1821	2057	1542	1459	1042	774	44
10	Base DX w/OA Precool	4521	3810	3017	2250	2012	1508	1960	965	838	40
11	Dual Path	4525	3581	2604	1989	1936	1436	1431	921	808	47
12	Dual Path w/Enthalpy Wheel	4551	3760	2834	2255	2279	1690	1716	1130	913	49
13	Dual Path w/AAHX	3433	2812	1988	1131	1445	1112	1023	793	662	44
14	Dual Path w/Desiccant	0	8	0	0	0	0	0	0	0	0
15	Base DX w/DCV	4549	3645	2861	3580	2290	1825	1891	1329	1295	27
16	Dual Path w/DCV	4575	3675	2880	2529	2273	1803	1867	1343	1277	25
17	Base DX w/Free Reheat	465	625	430	249	416	336	266	257	232	37

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	1%	1%	1%	0%	0%	0%	1%	1%	0%	0%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-10%	-10%	-9%	-12%	-7%	-4%	-5%	-9%	-4%	-5%
03	Base DX w/Lower Airflow	-10%	-8%	-7%	-9%	-5%	-3%	-4%	-7%	-4%	-4%
04	Base DX w/AAHX	26%	19%	15%	15%	9%	5%	6%	9%	6%	5%
05	Base DX w/Subcool Reheat	12%	9%	7%	9%	4%	3%	3%	4%	3%	2%
06	Base DX w/o Lat. Coil Degrad.	7%	5%	4%	6%	3%	2%	2%	2%	1%	1%
07	Base DX w/Bypass Damper	3%	2%	2%	2%	1%	0%	1%	0%	0%	0%
08	Base DX w/Desiccant	326%	140%	114%	71%	79%	14%	14%	9%	-18%	-6%
09	Base DX w/Enthalpy Wheel	-25%	-31%	-30%	-36%	-31%	-49%	-51%	-36%	-52%	-18%
10	Base DX w/OA Precool	20%	10%	8%	-1%	11%	8%	7%	11%	10%	10%
11	Dual Path	-19%	-21%	-17%	-26%	-10%	-6%	-7%	-13%	-6%	-7%
12	Dual Path w/Enthalpy Wheel	-46%	-47%	-43%	-49%	-42%	-56%	-59%	-49%	-61%	-27%
13	Dual Path w/AAHX	4%	-6%	-6%	-12%	-3%	-2%	-3%	-7%	-2%	-4%
14	Dual Path w/Desiccant	298%	119%	97%	53%	64%	5%	5%	-7%	-28%	-15%
15	Base DX w/DCV	-29%	-38%	-45%	-35%	-56%	-69%	-66%	-53%	-67%	-65%
16	Dual Path w/DCV	-50%	-61%	-66%	-67%	-70%	-78%	-77%	-69%	-76%	-76%
17	Base DX w/Free Reheat	60%	38%	26%	23%	16%	8%	9%	9%	5%	0%

Occupied Hours >65%RH		*% Change in Annual HVAC Energy Cost*	
(white)	<= 150 hrs	(white)	< 0% (less energy use)
(yellow)	151 to 1000 hrs	(yellow)	1% to 25% (more energy use)
(cyan)	1001 to 2000 hrs	(cyan)	26% to 50% (more energy use)
(magenta)	> 2000 hrs	(magenta)	>50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	16	16	16	16	16	16	16	16	16	16
Minimum EC, <=150 hrs RH>65%	14	14	14	14	14	14	14	14	14	16
Minimum Life Cycle Cost (LCC)	12	12	15	12	15	15	15	15	15	15
Minimum LCC, <=150 hrs RH>65%	14	14	14	14	14	14	08	14	08	15
Ratio Min LCC<=150 to Case 01 LCC	3.4	2.1	2.0	1.6	1.8	1.3	1.3	1.2	1.1	0.6

MI = Miami FL; ST = Washington DC
HO = Houston TX; SL = St. Louis MO
SH = Shreveport LA; NY = New York NY
FW = Fort Worth TX; CH = Chicago IL
AT = Atlanta GA; PO = Portland OR

Table 12.3b

System Performance Comparison – 2004 Standard Restaurant

Restaurant

2004 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System \ Location ==>	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	4473	3518	2518	1903	1827	1354	1308	862	661	2
01	Base DX	4483	3498	2506	1917	1839	1363	1296	885	670	2
02	DX w/Improved Dehumid.	4549	3598	2612	2117	1955	1445	1395	957	738	6
03	Base DX w/Lower Airflow	4548	3541	2560	2070	1932	1427	1369	946	729	6
04	Base DX w/AAHX	3790	3013	2101	1835	1689	1229	1083	876	593	2
05	Base DX w/Subcool Reheat	4230	3175	2281	1551	1627	1177	1126	793	618	2
06	Base DX w/o Lat. Coil Degrad.	3727	2713	1892	886	1083	831	815	477	346	0
07	Base DX w/Bypass Damper	4362	3308	2372	1709	1729	1276	1205	841	638	2
08	Base DX w/Desiccant	17	97	3	1	0	0	5	0	0	0
09	Base DX w/Enthalpy Wheel	2498	1852	1049	1327	740	655	461	645	307	6
10	Base DX w/OA Precool	4302	3759	3080	2634	2183	1936	2146	813	649	0
11	Dual Path	4254	3259	2264	1104	1519	1115	1094	698	544	10
12	Dual Path w/Enthalpy Wheel	674	649	308	110	270	162	149	213	121	6
13	Dual Path w/AAHX	1729	1300	795	249	616	510	344	381	282	7
14	Dual Path w/Desiccant	0	0	0	0	0	0	0	0	0	0
15	Base DX w/DCV	4521	3532	2680	2435	2026	1592	1606	1095	909	2
16	Dual Path w/DCV	2542	1628	884	202	620	562	418	446	284	0
17	Base DX w/Free Reheat	85	217	155	40	116	40	49	91	30	0

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System \ Location ==>	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	1%	1%	0%	0%	0%	0%	0%	1%	0%	1%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-13%	-12%	-11%	-14%	-9%	-6%	-6%	-10%	-6%	-7%
03	Base DX w/Lower Airflow	-10%	-10%	-8%	-10%	-7%	-4%	-5%	-8%	-5%	-5%
04	Base DX w/AAHX	24%	20%	16%	17%	11%	7%	8%	11%	7%	7%
05	Base DX w/Subcool Reheat	12%	10%	8%	9%	6%	3%	4%	5%	3%	3%
06	Base DX w/o Lat. Coil Degrad.	8%	6%	5%	7%	5%	2%	3%	3%	2%	1%
07	Base DX w/Bypass Damper	3%	2%	2%	2%	1%	1%	1%	0%	0%	0%
08	Base DX w/Desiccant	238%	112%	99%	61%	75%	17%	16%	10%	-15%	-4%
09	Base DX w/Enthalpy Wheel	-27%	-31%	-31%	-31%	-32%	-49%	-51%	-35%	-51%	-21%
10	Base DX w/OA Precool	10%	8%	7%	5%	9%	7%	6%	9%	6%	8%
11	Dual Path	-27%	-26%	-22%	-21%	-17%	-11%	-12%	-22%	-13%	-14%
12	Dual Path w/Enthalpy Wheel	-42%	-44%	-42%	-42%	-43%	-56%	-59%	-48%	-60%	-30%
13	Dual Path w/AAHX	-8%	-6%	-6%	-5%	-7%	-5%	-5%	-13%	-8%	-10%
14	Dual Path w/Desiccant	211%	92%	78%	40%	51%	2%	1%	-7%	-27%	-15%
15	Base DX w/DCV	-16%	-22%	-28%	-22%	-36%	-48%	-46%	-35%	-46%	-46%
16	Dual Path w/DCV	-43%	-49%	-52%	-45%	-57%	-62%	-60%	-59%	-60%	-64%
17	Base DX w/Free Reheat	61%	42%	30%	26%	20%	11%	12%	12%	8%	1%

Occupied Hours >65%RH		*% Change in Annual HVAC Energy Cost*	
(white)	<= 150 hrs	(white)	< 0% (less energy use)
(yellow)	151 to 1000 hrs	(yellow)	1% to 25% (more energy use)
(cyan)	1001 to 2000 hrs	(cyan)	26% to 50% (more energy use)
(magenta)	> 2000 hrs	(magenta)	>50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	16	16	16	16	16	16	16	16	16	16
Minimum EC, <=150 hrs RH>65%	17	14	14	12	17	14	12	14	12	16
Minimum Life Cycle Cost (LCC)	12	12	12	12	12	12	09	12	12	15
Minimum LCC, <=150 hrs RH>65%	17	14	14	12	17	17	12	17	12	15
Ratio Min LCC<=150 to Case 01 LCC	1.4	1.9	1.8	0.7	1.2	1.1	0.6	1.1	0.6	0.7

MI	=	Miami FL	ST	=	Washington DC
HO	=	Houston TX	SL	=	St. Louis MO
SH	=	Shreveport LA	NY	=	New York NY
FW	=	Fort Worth TX	CH	=	Chicago IL
AT	=	Atlanta GA	PO	=	Portland OR

Table 12.4a
System Performance Comparison – 2001 Standard Retail

Retail

2001 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System (Location ==>)	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	2384	1636	1169	699	599	455	471	257	233	0
01	Base DX	2279	1519	1066	675	580	438	441	280	226	0
02	DX w/Improved Dehumid.	2361	1653	1153	775	693	519	512	334	261	0
03	Base DX w/Lower Airflow	2273	1573	1082	755	706	514	483	340	266	0
04	Base DX w/AAHX	928	803	542	528	331	289	235	237	157	0
05	Base DX w/Subcool Reheat	1896	1167	777	454	373	281	312	203	179	0
06	Base DX w/o Lat. Coil Degrad.	1341	657	406	99	55	95	133	39	31	0
07	Base DX w/Bypass Damper	1958	1233	829	537	417	331	334	238	195	0
08	Base DX w/Desiccant	0	0	0	0	0	0	0	0	0	0
09	Base DX w/Enthalpy Wheel	19	5	14	1	5	4	1	2	0	0
10	Base DX w/OA Precool	2337	1724	1294	1180	794	620	730	260	217	0
11	Dual Path	653	252	78	38	13	25	2	18	6	0
12	Dual Path w/Enthalpy Wheel	4	5	11	0	5	0	0	0	0	0
13	Dual Path w/AAHX	241	169	53	19	6	0	2	6	3	0
14	Dual Path w/Desiccant	0	0	0	0	0	0	0	0	0	0
15	Base DX w/DCV	2279	1519	1066	675	580	438	441	280	226	0
16	Dual Path w/DCV	653	252	78	38	13	25	2	18	6	0
17	Base DX w/Free Reheat	3	12	10	2	0	0	1	4	0	0

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System (Location ==>)	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	-1%	0%	0%	-1%	0%	0%	0%	1%	0%	1%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-12%	-12%	-11%	-13%	-10%	-7%	-7%	-11%	-7%	-8%
03	Base DX w/Lower Airflow	-9%	-9%	-8%	-10%	-7%	-5%	-5%	-9%	-5%	-6%
04	Base DX w/AAHX	24%	21%	18%	18%	13%	9%	9%	12%	8%	9%
05	Base DX w/Subcool Reheat	12%	11%	9%	10%	7%	5%	5%	6%	4%	4%
06	Base DX w/o Lat. Coil Degrad.	7%	6%	5%	6%	5%	3%	3%	4%	3%	2%
07	Base DX w/Bypass Damper	3%	3%	2%	2%	1%	1%	1%	1%	1%	0%
08	Base DX w/Desiccant	181%	84%	79%	50%	65%	18%	15%	7%	-11%	-3%
09	Base DX w/Enthalpy Wheel	-27%	-30%	-30%	-31%	-30%	-43%	-47%	-32%	-46%	-18%
10	Base DX w/OA Precool	8%	6%	6%	3%	7%	6%	5%	8%	7%	8%
11	Dual Path	-26%	-22%	-20%	-20%	-19%	-13%	-13%	-21%	-14%	-18%
12	Dual Path w/Enthalpy Wheel	-40%	-43%	-41%	-42%	-41%	-51%	-55%	-44%	-55%	-28%
13	Dual Path w/AAHX	0%	-1%	-2%	-1%	-5%	-4%	-3%	-10%	-6%	-11%
14	Dual Path w/Desiccant	151%	60%	51%	24%	27%	-6%	-8%	-13%	-29%	-14%
15	Base DX w/DCV	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
16	Dual Path w/DCV	-26%	-22%	-20%	-20%	-19%	-13%	-13%	-21%	-14%	-18%
17	Base DX w/Free Reheat	54%	40%	29%	27%	19%	12%	13%	12%	9%	1%

Occupied Hours >65%RH		*% Change in Annual HVAC Energy Cost*	
(white)	<= 150 hrs	(white)	< 0% (less energy use)
(yellow)	151 to 1000 hrs	(yellow)	1% to 25% (more energy use)
(cyan)	1001 to 2000 hrs	(cyan)	26% to 50% (more energy use)
(magenta)	> 2000 hrs	(magenta)	>50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	12	12	12	12	12	12	12	12	12	12
Minimum EC, <=150 hrs RH>65%	12	12	12	12	12	12	12	12	12	12
Minimum Life Cycle Cost (LCC)	12	12	12	12	09	09	09	12	09	09
Minimum LCC, <=150 hrs RH>65%	12	12	12	12	09	09	09	12	09	09
Ratio Min LCC<=150 to Case 01 LCC	0.8	0.8	0.7	0.7	0.8	0.7	0.7	0.8	0.7	0.9

MI	=	Miami FL	ST	=	Washington DC
HO	=	Houston TX	SL	=	St. Louis MO
SH	=	Shreveport LA	NY	=	New York NY
FW	=	Fort Worth TX	CH	=	Chicago IL
AT	=	Atlanta GA	PO	=	Portland OR

Table 12.4b
System Performance Comparison – 2004 Standard Retail

Retail
2004 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	2000	1201	846	356	196	200	292	78	54	0
01	Base DX	1713	961	610	314	150	165	220	82	59	0
02	DX w/Improved Dehumid.	1475	897	574	357	135	161	208	104	67	0
03	Base DX w/Lower Airflow	1261	842	556	390	188	176	208	129	85	0
04	Base DX w/AAHX	534	247	134	162	41	42	27	46	6	0
05	Base DX w/Subcool Reheat	1422	682	416	218	95	116	148	62	42	0
06	Base DX w/o Lat. Coil Degrad.	543	283	155	12	0	35	28	14	3	0
07	Base DX w/Bypass Damper	1242	640	408	236	85	120	144	68	39	0
08	Base DX w/Desiccant	0	0	0	0	0	0	0	0	0	0
09	Base DX w/Enthalpy Wheel	0	1	5	0	0	0	0	0	0	0
10	Base DX w/OA Precool	2044	1621	1185	1114	432	389	586	207	27	0
11	Dual Path	284	145	56	25	0	0	1	4	1	0
12	Dual Path w/Enthalpy Wheel	0	1	6	0	0	0	0	0	0	0
13	Dual Path w/AAHX	91	78	24	3	0	0	0	3	0	0
14	Dual Path w/Desiccant	0	0	0	0	0	0	0	0	0	0
15	Base DX w/DCV	465	265	139	190	27	58	41	45	3	0
16	Dual Path w/DCV	131	43	13	2	0	0	0	0	0	0
17	Base DX w/Free Reheat	0	1	5	0	0	0	0	2	0	0

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	-1%	-1%	-1%	-1%	-1%	0%	0%	0%	0%	0%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-10%	-11%	-10%	-12%	-8%	-6%	-6%	-10%	-7%	-8%
03	Base DX w/Lower Airflow	-7%	-8%	-7%	-8%	-6%	-4%	-4%	-8%	-5%	-6%
04	Base DX w/AAHX	24%	21%	19%	19%	14%	9%	10%	13%	9%	9%
05	Base DX w/Subcool Reheat	11%	10%	9%	10%	7%	5%	5%	6%	4%	4%
06	Base DX w/o Lat. Coil Degrad.	6%	5%	4%	5%	4%	2%	3%	3%	2%	2%
07	Base DX w/Bypass Damper	3%	3%	2%	3%	2%	1%	1%	1%	1%	0%
08	Base DX w/Desiccant	169%	79%	75%	47%	61%	18%	14%	6%	-11%	-2%
09	Base DX w/Enthalpy Wheel	-26%	-30%	-29%	-30%	-30%	-42%	-46%	-32%	-46%	-18%
10	Base DX w/OA Precool	10%	7%	6%	4%	8%	6%	5%	8%	7%	9%
11	Dual Path	-20%	-19%	-18%	-18%	-17%	-13%	-12%	-21%	-13%	-18%
12	Dual Path w/Enthalpy Wheel	-39%	-42%	-41%	-42%	-41%	-51%	-54%	-44%	-55%	-28%
13	Dual Path w/AAHX	4%	1%	1%	1%	-2%	-3%	-1%	-9%	-5%	-10%
14	Dual Path w/Desiccant	137%	53%	44%	20%	20%	-9%	-11%	-14%	-30%	-15%
15	Base DX w/DCV	-9%	-11%	-14%	-12%	-18%	-26%	-25%	-20%	-26%	-25%
16	Dual Path w/DCV	-32%	-33%	-34%	-32%	-38%	-40%	-38%	-42%	-40%	-45%
17	Base DX w/Free Reheat	44%	32%	24%	23%	15%	10%	10%	10%	7%	1%

Occupied Hours >65%RH	*% Change in Annual HVAC Energy Cost*
(white) <= 150 hrs	(white) < 0% (less energy use)
(yellow) 151 to 1000 hrs	(yellow) 1% to 25% (more energy use)
(cyan) 1001 to 2000 hrs	(cyan) 26% to 50% (more energy use)
(magenta) > 2000 hrs	(magenta) >50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	12	12	12	12	12	12	12	12	12	16
Minimum EC, <=150 hrs RH>65%	12	12	12	12	12	12	12	12	12	16
Minimum Life Cycle Cost (LCC)	12	12	12	12	12	09	09	12	09	09
Minimum LCC, <=150 hrs RH>65%	12	12	12	12	12	09	09	12	09	09
Ratio Min LCC<=150 to Case 01 LCC	0.8	0.7	0.7	0.7	0.8	0.7	0.7	0.8	0.7	0.9

MI	=	Miami FL	ST	=	Washington DC
HO	=	Houston TX	SL	=	St. Louis MO
SH	=	Shreveport LA	NY	=	New York NY
FW	=	Fort Worth TX	CH	=	Chicago IL
AT	=	Atlanta GA	PO	=	Portland OR

Table 12.5a
System Performance Comparison – 2001 Standard Theater

Theater
2001 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	3193	2700	2069	1507	1439	1137	1336	832	597	35
01	Base DX	3240	2698	2069	1660	1448	1130	1335	765	595	35
02	DX w/Improved Dehumid.	3260	2726	2107	1743	1490	1160	1370	800	622	37
03	Base DX w/Lower Airflow	3261	2711	2097	1676	1471	1148	1146	789	606	37
04	Base DX w/AAHX	3231	2551	2022	1377	1414	1081	1022	743	544	40
05	Base DX w/Subcool Reheat	3208	2625	1998	1423	1378	1069	1022	793	542	33
06	Base DX w/o Lat. Coil Degrad.	3061	2480	1841	1075	1303	1094	896	703	471	32
07	Base DX w/Bypass Damper	3232	2666	2045	1282	1416	1097	1073	806	565	35
08	Base DX w/Desiccant	123	280	167	14	9	15	91	0	9	0
09	Base DX w/Enthalpy Wheel	3238	2658	2017	1780	1631	1225	1255	898	680	78
10	Base DX w/OA Precool	3277	2707	2090	1788	1458	1143	1340	838	604	33
11	Dual Path	3253	2712	2059	1482	1423	1113	1079	852	601	33
12	Dual Path w/Enthalpy Wheel	3270	2679	2046	1810	1646	1237	1272	911	703	90
13	Dual Path w/AAHX	2967	2317	1660	929	1277	1013	901	791	523	37
14	Dual Path w/Desiccant	57	70	21	0	0	9	36	0	9	0
15	Base DX w/DCV	3275	3036	2664	2305	2040	1904	2155	965	817	62
16	Dual Path w/DCV	3335	3115	2113	2381	2044	1921	2199	989	824	73
17	Base DX w/Free Reheat	537	548	311	159	248	264	243	199	208	30

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	2%	1%	1%	0%	0%	0%	0%	1%	0%	0%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-12%	-11%	-9%	-12%	-7%	-4%	-4%	-8%	-4%	-4%
03	Base DX w/Lower Airflow	-11%	-9%	-8%	-9%	-5%	-3%	-4%	-7%	-3%	-3%
04	Base DX w/AAHX	27%	19%	15%	15%	9%	5%	5%	9%	5%	5%
05	Base DX w/Subcool Reheat	13%	10%	7%	9%	5%	3%	2%	5%	2%	2%
06	Base DX w/o Lat. Coil Degrad.	8%	6%	4%	5%	4%	3%	1%	3%	1%	1%
07	Base DX w/Bypass Damper	3%	2%	2%	2%	1%	0%	0%	1%	0%	0%
08	Base DX w/Desiccant	360%	153%	124%	76%	94%	21%	20%	15%	-18%	-2%
09	Base DX w/Enthalpy Wheel	-23%	-29%	-27%	-32%	-27%	-47%	-49%	-33%	-51%	-16%
10	Base DX w/OA Precool	22%	10%	8%	6%	13%	9%	7%	14%	11%	12%
11	Dual Path	-21%	-22%	-18%	-23%	-10%	-5%	-8%	-12%	-6%	-6%
12	Dual Path w/Enthalpy Wheel	-47%	-47%	-41%	-47%	-40%	-55%	-57%	-48%	-60%	-26%
13	Dual Path w/AAHX	4%	-7%	-7%	-11%	-3%	-2%	-4%	-6%	-2%	-3%
14	Dual Path w/Desiccant	327%	128%	103%	54%	76%	10%	8%	-4%	-29%	-13%
15	Base DX w/DCV	-23%	-33%	-41%	-36%	-51%	-61%	-59%	-48%	-60%	-60%
16	Dual Path w/DCV	-45%	-56%	-62%	-61%	-63%	-69%	-68%	-62%	-67%	-69%
17	Base DX w/Free Reheat	71%	46%	31%	27%	21%	10%	10%	12%	6%	1%

Occupied Hours >65%RH	*% Change in Annual HVAC Energy Cost*
(white) <= 150 hrs	(white) < 0% (less energy use)
(yellow) 151 to 1000 hrs	(yellow) 1% to 25% (more energy use)
(cyan) 1001 to 2000 hrs	(cyan) 26% to 50% (more energy use)
(magenta) > 2000 hrs	(magenta) >50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	12	16	16	16	16	16	16	16	16	16
Minimum EC, <=150 hrs RH>65%	14	14	14	14	14	14	14	14	14	16
Minimum Life Cycle Cost (LCC)	12	12	15	12	15	15	15	15	15	15
Minimum LCC, <=150 hrs RH>65%	14	14	14	14	14	08	08	14	08	15
Ratio Min LCC<=150 to Case 01 LCC	3.4	2.2	2.0	1.7	1.9	1.4	1.4	1.4	1.2	0.7

MI = Miami FL
HO = Houston TX
SH = Shreveport LA
FW = Fort Worth TX
AT = Atlanta GA
ST = Washington DC
SL = St. Louis MO
NY = New York NY
CH = Chicago IL
PO = Portland OR

Table 12.5b
System Performance Comparison – 2004 Standard Theater

Theater
2004 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	3359	2851	2034	1664	1523	1221	1675	880	720	50
01	Base DX	3367	2860	2049	1666	1544	1230	1267	893	724	55
02	DX w/Improved Dehumid.	3387	2878	2085	2072	1874	1254	1306	922	761	80
03	Base DX w/Lower Airflow	3383	2871	2078	2051	1878	1249	1296	923	757	77
04	Base DX w/AAHX	3344	2660	2010	1628	1510	1199	1181	884	690	49
05	Base DX w/Subcool Reheat	3340	2785	1944	1523	1445	1157	1147	822	657	46
06	Base DX w/o Lat. Coil Degrad.	3142	2292	1988	865	1059	806	763	494	372	17
07	Base DX w/Bypass Damper	3358	2700	2016	1482	1501	1208	1220	872	687	53
08	Base DX w/Desiccant	451	437	297	79	13	17	117	0	9	0
09	Base DX w/Enthalpy Wheel	3352	2731	2123	1911	1671	1274	1343	1035	794	154
10	Base DX w/OA Precool	3382	2849	2313	2020	1877	1627	1666	1025	644	57
11	Dual Path	3279	2624	1872	1376	1378	1120	1048	752	621	51
12	Dual Path w/Enthalpy Wheel	3229	2641	1890	1258	1382	1083	1057	786	524	51
13	Dual Path w/AAHX	2331	1462	904	349	632	588	435	386	322	41
14	Dual Path w/Desiccant	17	44	6	0	0	5	28	0	0	0
15	Base DX w/DCV	3626	2965	2490	2286	2057	1682	2728	1393	1307	983
16	Dual Path w/DCV	3649	2978	2236	1373	1683	1529	1541	1171	1109	607
17	Base DX w/Free Reheat	26	120	78	23	68	11	43	56	10	6

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	1%	1%	1%	0%	1%	0%	1%	1%	1%	1%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-14%	-14%	-12%	-14%	-10%	-6%	-7%	-12%	-7%	-8%
03	Base DX w/Lower Airflow	-12%	-11%	-10%	-10%	-7%	-5%	-6%	-9%	-5%	-6%
04	Base DX w/AAHX	24%	20%	17%	17%	13%	8%	9%	13%	8%	9%
05	Base DX w/Subcool Reheat	12%	10%	9%	10%	6%	4%	4%	6%	4%	4%
06	Base DX w/o Lat. Coil Degrad.	8%	7%	7%	7%	6%	3%	3%	4%	2%	2%
07	Base DX w/Bypass Damper	2%	2%	1%	2%	1%	0%	1%	0%	0%	0%
08	Base DX w/Desiccant	229%	116%	107%	65%	94%	35%	32%	22%	-4%	11%
09	Base DX w/Enthalpy Wheel	-23%	-27%	-26%	-29%	-25%	-41%	-43%	-29%	-45%	-17%
10	Base DX w/OA Precool	8%	7%	7%	4%	8%	7%	6%	9%	6%	8%
11	Dual Path	-29%	-24%	-22%	-22%	-19%	-12%	-13%	-22%	-14%	-17%
12	Dual Path w/Enthalpy Wheel	-36%	-39%	-37%	-40%	-36%	-49%	-51%	-41%	-54%	-27%
13	Dual Path w/AAHX	-6%	-6%	-7%	-6%	-7%	-5%	-5%	-12%	-8%	-11%
14	Dual Path w/Desiccant	206%	97%	86%	46%	71%	19%	16%	3%	-18%	-6%
15	Base DX w/DCV	-13%	-18%	-23%	-20%	-30%	-45%	-42%	-31%	-45%	-39%
16	Dual Path w/DCV	-42%	-44%	-47%	-44%	-52%	-60%	-57%	-55%	-60%	-61%
17	Base DX w/Free Reheat	76%	56%	42%	38%	30%	18%	19%	19%	12%	5%

Occupied Hours >65%RH	*% Change in Annual HVAC Energy Cost*
(white) <= 150 hrs	(white) < 0% (less energy use)
(yellow) 151 to 1000 hrs	(yellow) 1% to 25% (more energy use)
(blue) 1001 to 2000 hrs	(blue) 26% to 50% (more energy use)
(magenta) > 2000 hrs	(magenta) >50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	16	16	16	16	16	16	16	16	16	16
Minimum EC, <=150 hrs RH>65%	17	17	17	17	17	17	14	14	14	12
Minimum Life Cycle Cost (LCC)	12	12	12	12	09	09	09	12	09	15
Minimum LCC, <=150 hrs RH>65%	17	17	17	17	17	17	17	17	17	12
Ratio Min LCC<=150 to Case 01 LCC	1.5	1.3	1.3	1.2	1.2	1.1	1.1	1.1	1.1	0.9

MI	=	Miami FL	ST	=	Washington DC
HO	=	Houston TX	SL	=	St. Louis MO
SH	=	Shreveport LA	NY	=	New York NY
FW	=	Fort Worth TX	CH	=	Chicago IL
AT	=	Atlanta GA	PO	=	Portland OR

Table 12.6a
System Performance Comparison – 2001 Standard School - 9 Month - South

School-9 Month-South

2001 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	2595	1858	1343	1110	923	638	624	371	341	0
01	Base DX	2601	1868	1345	1104	939	643	630	375	351	0
02	DX w/Improved Dehumid.	2645	1916	1391	1167	978	685	683	409	370	0
03	Base DX w/Lower Airflow	2638	1908	1383	1150	977	671	676	406	368	0
04	Base DX w/AAHX	2449	1792	1267	1067	906	596	597	359	329	0
05	Base DX w/Subcool Reheat	2532	1783	1248	944	877	574	565	343	314	0
06	Base DX w/o Lat. Coil Degrad.	2285	1566	1038	719	688	454	448	225	210	0
07	Base DX w/Bypass Damper	2574	1825	1300	1031	914	603	601	360	331	0
08	Base DX w/Desiccant	59	146	32	0	0	0	15	0	0	0
09	Base DX w/Enthalpy Wheel	1599	1100	808	708	657	397	510	212	210	0
10	Base DX w/OA Precool	2555	1818	1326	1090	902	633	637	349	323	0
11	Dual Path	2509	1785	1231	992	850	564	566	314	301	0
12	Dual Path w/Enthalpy Wheel	1595	1097	800	687	651	388	491	189	201	0
13	Dual Path w/AAHX	1824	1249	858	441	594	344	324	215	202	0
14	Dual Path w/Desiccant	5	3	0	0	0	0	0	0	0	0
15	Base DX w/DCV	2312	1673	1314	1280	911	670	792	375	383	0
16	Dual Path w/DCV	1422	888	690	663	574	393	586	290	311	0
17	Base DX w/Free Reheat	118	155	105	19	72	49	25	45	16	0

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	1%	1%	1%	1%	1%	0%	0%	1%	0%	0%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-12%	-12%	-10%	-13%	-7%	-4%	-5%	-9%	-5%	-5%
03	Base DX w/Lower Airflow	-10%	-10%	-8%	-10%	-6%	-3%	-4%	-7%	-4%	-4%
04	Base DX w/AAHX	23%	18%	14%	15%	9%	5%	6%	9%	5%	5%
05	Base DX w/Subcool Reheat	12%	9%	7%	8%	4%	3%	3%	4%	3%	2%
06	Base DX w/o Lat. Coil Degrad.	7%	5%	4%	5%	3%	1%	2%	2%	1%	0%
07	Base DX w/Bypass Damper	2%	2%	1%	1%	1%	0%	0%	0%	0%	0%
08	Base DX w/Desiccant	280%	112%	92%	67%	59%	2%	1%	-1%	-23%	-8%
09	Base DX w/Enthalpy Wheel	-29%	-35%	-33%	-35%	-34%	-51%	-55%	-39%	-54%	-19%
10	Base DX w/OA Precool	13%	9%	8%	8%	9%	7%	6%	9%	7%	8%
11	Dual Path	-12%	-16%	-14%	-20%	-8%	-5%	-6%	-11%	-5%	-6%
12	Dual Path w/Enthalpy Wheel	-37%	-42%	-39%	-42%	-40%	-55%	-59%	-46%	-58%	-26%
13	Dual Path w/AAHX	8%	-3%	-3%	-9%	-2%	-1%	-2%	-6%	-2%	-4%
14	Dual Path w/Desiccant	254%	93%	74%	47%	44%	-8%	-9%	-16%	-33%	-17%
15	Base DX w/DCV	-27%	-34%	-40%	-32%	-50%	-65%	-64%	-50%	-65%	-59%
16	Dual Path w/DCV	-40%	-52%	-58%	-56%	-62%	-73%	-73%	-64%	-73%	-69%
17	Base DX w/Free Reheat	63%	41%	29%	28%	17%	9%	10%	9%	6%	0%

Occupied Hours >65%RH		*% Change in Annual HVAC Energy Cost*	
(white)	<= 150 hrs	(white)	< 0% (less energy use)
(yellow)	151 to 1000 hrs	(yellow)	1% to 25% (more energy use)
(cyan)	1001 to 2000 hrs	(cyan)	26% to 50% (more energy use)
(magenta)	> 2000 hrs	(magenta)	>50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	16	16	16	16	16	16	16	16	16	16
Minimum EC, <=150 hrs RH>65%	17	14	17	17	17	14	14	14	14	16
Minimum Life Cycle Cost (LCC)	09	09	09	09	15	15	09	15	15	15
Minimum LCC, <=150 hrs RH>65%	17	08	17	17	17	17	17	17	17	15
Ratio Min LCC<=150 to Case 01 LCC	1.4	1.9	1.2	1.2	1.1	1.1	1.1	1.1	1.1	0.7

MI = Miami FL; HO = Houston TX; SH = Shreveport LA; FW = Fort Worth TX; AT = Atlanta GA; ST = Washington DC; SL = St. Louis MO; NY = New York NY; CH = Chicago IL; PO = Portland OR

Table 12.6b

System Performance Comparison – 2004 Standard School - 9 Month - South

School-9 Month-South

2004 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	2399	1638	1126	977	754	516	540	266	252	0
01	Base DX	2383	1637	1115	971	761	506	550	285	259	0
02	DX w/Improved Dehumid.	2450	1688	1179	1032	801	540	600	323	293	0
03	Base DX w/Lower Airflow	2427	1684	1170	1023	804	533	591	323	297	0
04	Base DX w/AAHX	1723	1409	945	869	672	433	482	265	209	0
05	Base DX w/Subcool Reheat	2222	1518	1001	780	680	440	454	228	213	0
06	Base DX w/o Lat. Coil Degrad.	1965	1182	753	452	426	280	329	80	84	0
07	Base DX w/Bypass Damper	2295	1572	1054	869	724	463	499	260	228	0
08	Base DX w/Desiccant	34	121	1	0	0	0	0	0	0	0
09	Base DX w/Enthalpy Wheel	665	686	552	483	363	294	256	52	33	0
10	Base DX w/OA Precool	2367	1585	1110	944	732	520	575	259	217	0
11	Dual Path	1724	1077	686	489	444	315	349	104	91	0
12	Dual Path w/Enthalpy Wheel	513	367	152	9	23	19	6	0	0	0
13	Dual Path w/AAHX	805	529	262	46	127	101	84	24	19	0
14	Dual Path w/Desiccant	0	0	0	0	0	0	0	0	0	0
15	Base DX w/DCV	2006	1363	999	947	689	458	572	269	234	0
16	Dual Path w/DCV	859	656	481	211	228	139	171	11	15	0
17	Base DX w/Free Reheat	39	30	14	0	0	0	2	4	0	0

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	0%	0%	0%	0%	0%	0%	0%	1%	0%	1%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-11%	-12%	-11%	-13%	-9%	-6%	-6%	-10%	-6%	-7%
03	Base DX w/Lower Airflow	-8%	-9%	-8%	-10%	-7%	-4%	-5%	-8%	-5%	-6%
04	Base DX w/AAHX	23%	19%	16%	17%	11%	7%	7%	11%	7%	7%
05	Base DX w/Subcool Reheat	11%	9%	8%	9%	5%	3%	4%	5%	3%	3%
06	Base DX w/o Lat. Coil Degrad.	5%	5%	4%	5%	3%	2%	2%	2%	2%	1%
07	Base DX w/Bypass Damper	3%	2%	1%	1%	1%	0%	1%	0%	0%	0%
08	Base DX w/Desiccant	229%	96%	84%	59%	60%	9%	7%	4%	-17%	-5%
09	Base DX w/Enthalpy Wheel	-27%	-31%	-30%	-31%	-31%	-46%	-50%	-34%	-48%	-18%
10	Base DX w/OA Precool	11%	8%	8%	7%	9%	6%	6%	9%	6%	8%
11	Dual Path	-2%	-11%	-11%	-17%	-6%	-5%	-6%	-10%	-5%	-8%
12	Dual Path w/Enthalpy Wheel	-26%	-34%	-34%	-36%	-34%	-50%	-53%	-40%	-52%	-25%
13	Dual Path w/AAHX	18%	3%	1%	-4%	2%	0%	-1%	-4%	-1%	-5%
14	Dual Path w/Desiccant	205%	76%	63%	39%	39%	-5%	-7%	-12%	-29%	-16%
15	Base DX w/DCV	-13%	-18%	-22%	-18%	-28%	-38%	-37%	-29%	-38%	-35%
16	Dual Path w/DCV	-16%	-29%	-33%	-35%	-36%	-44%	-44%	-40%	-44%	-44%
17	Base DX w/Free Reheat	50%	40%	29%	29%	19%	11%	12%	11%	8%	1%

Occupied Hours >65%RH		*% Change in Annual HVAC Energy Cost*	
(white)	<= 150 hrs	(white)	< 0% (less energy use)
(yellow)	151 to 1000 hrs	(yellow)	1% to 25% (more energy use)
(cyan)	1001 to 2000 hrs	(cyan)	26% to 50% (more energy use)
(magenta)	> 2000 hrs	(magenta)	>50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	09	12	12	12	16	12	12	12	12	16
Minimum EC, <=150 hrs RH>65%	17	17	17	12	12	12	12	12	12	16
Minimum Life Cycle Cost (LCC)	09	09	09	09	09	09	09	09	09	15
Minimum LCC, <=150 hrs RH>65%	17	17	17	12	12	12	12	09	09	15
Ratio Min LCC<=150 to Case 01 LCC	1.3	1.2	1.2	0.8	1.0	0.8	0.8	0.8	0.7	0.9

MI = Miami FL; HO = Houston TX; SH = Shreveport LA; FW = Fort Worth TX; AT = Atlanta GA; ST = Washington DC; SL = St. Louis MO; NY = New York NY; CH = Chicago IL; PO = Portland OR

Table 12.7a

System Performance Comparison – 2001 Standard School - 12 Month - South

School-12 Month-South

2001 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	2540	1927	1386	1166	977	695	728	417	363	0
01	Base DX	2546	1922	1384	1128	986	700	727	416	371	0
02	DX w/Improved Dehumid.	2583	1966	1421	1197	1033	733	779	457	401	2
03	Base DX w/Lower Airflow	2580	1950	1413	1163	1028	721	759	450	399	1
04	Base DX w/AAHX	2301	1745	1232	1027	920	638	639	396	335	0
05	Base DX w/Subcool Reheat	2492	1829	1293	936	925	628	646	373	343	0
06	Base DX w/o Lat. Coil Degrad.	2289	1626	1142	716	739	482	527	243	238	0
07	Base DX w/Bypass Damper	2528	1871	1339	1019	963	667	690	395	359	0
08	Base DX w/Desiccant	24	53	13	0	0	1	10	0	1	0
09	Base DX w/Enthalpy Wheel	1608	1027	694	369	413	256	276	94	66	2
10	Base DX w/OA Precool	2507	1903	1389	1203	961	687	729	416	354	0
11	Dual Path	2473	1867	1293	980	902	609	637	341	315	2
12	Dual Path w/Enthalpy Wheel	1605	1025	678	321	408	247	272	66	61	2
13	Dual Path w/AAHX	1687	1147	792	382	522	346	327	192	175	1
14	Dual Path w/Desiccant	12	4	0	0	0	0	0	0	0	0
15	Base DX w/DCV	2180	1639	1293	1190	841	622	793	436	326	0
16	Dual Path w/DCV	1177	449	227	20	9	22	73	9	3	0
17	Base DX w/Free Reheat	249	230	125	30	65	46	28	42	21	0

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	1%	1%	1%	0%	1%	0%	0%	1%	0%	0%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-12%	-11%	-10%	-13%	-8%	-5%	-5%	-9%	-5%	-5%
03	Base DX w/Lower Airflow	-11%	-9%	-8%	-10%	-6%	-4%	-4%	-7%	-4%	-4%
04	Base DX w/AAHX	24%	19%	15%	16%	9%	6%	7%	10%	6%	6%
05	Base DX w/Subcool Reheat	12%	9%	8%	9%	5%	3%	3%	5%	3%	2%
06	Base DX w/o Lat. Coil Degrad.	7%	5%	4%	5%	3%	2%	2%	2%	2%	1%
07	Base DX w/Bypass Damper	3%	2%	1%	2%	1%	0%	1%	0%	0%	0%
08	Base DX w/Desiccant	288%	118%	102%	68%	71%	14%	13%	6%	-15%	-6%
09	Base DX w/Enthalpy Wheel	-32%	-37%	-35%	-37%	-34%	-49%	-53%	-37%	-51%	-18%
10	Base DX w/OA Precool	13%	9%	8%	7%	9%	7%	6%	9%	7%	8%
11	Dual Path	-12%	-16%	-14%	-20%	-8%	-5%	-6%	-11%	-5%	-7%
12	Dual Path w/Enthalpy Wheel	-39%	-43%	-41%	-43%	-40%	-53%	-57%	-45%	-56%	-25%
13	Dual Path w/AAHX	9%	-2%	-2%	-8%	-1%	-1%	-1%	-5%	-1%	-4%
14	Dual Path w/Desiccant	264%	100%	85%	49%	55%	3%	2%	-10%	-26%	-15%
15	Base DX w/DCV	-30%	-36%	-41%	-34%	-50%	-64%	-63%	-49%	-64%	-58%
16	Dual Path w/DCV	-43%	-54%	-58%	-58%	-62%	-72%	-73%	-63%	-71%	-68%
17	Base DX w/Free Reheat	61%	41%	30%	29%	19%	10%	12%	11%	7%	0%

	Occupied Hours >65%RH	*% Change in Annual HVAC Energy Cost*
White	<= 150 hrs	< 0% (less energy use)
Yellow	151 to 1000 hrs	1% to 25% (more energy use)
Cyan	1001 to 2000 hrs	26% to 50% (more energy use)
Magenta	> 2000 hrs	>50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	16	16	16	16	16	16	16	16	16	16
Minimum EC, <=150 hrs RH>65%	14	14	17	16	16	16	16	16	16	16
Minimum Life Cycle Cost (LCC)	09	09	09	09	15	15	09	15	15	15
Minimum LCC, <=150 hrs RH>65%	14	08	17	16	16	16	16	09	09	15
Ratio Min LCC<=150 to Case 01 LCC	2.9	1.9	1.2	0.9	1.0	0.8	0.9	0.7	0.6	0.7

MI	=	Miami FL	ST	=	Washington DC
HO	=	Houston TX	SL	=	St. Louis MO
SH	=	Shreveport LA	NY	=	New York NY
FW	=	Fort Worth TX	CH	=	Chicago IL
AT	=	Atlanta GA	PO	=	Portland OR

Table 12.7b

System Performance Comparison – 2004 Standard School - 12 Month - South

School-12 Month-South

2004 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System (Location ==>)	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	2373	1739	1192	959	793	547	592	278	247	0
01	Base DX	2369	1719	1185	912	789	523	586	279	252	0
02	DX w/Improved Dehumid.	2429	1769	1239	989	823	558	621	304	269	0
03	Base DX w/Lower Airflow	2412	1737	1230	960	823	553	611	308	274	0
04	Base DX w/AAHX	1449	1219	815	656	551	374	394	226	160	0
05	Base DX w/Subcool Reheat	2206	1563	1082	708	695	429	471	231	202	0
06	Base DX w/o Lat. Coil Degrad.	2005	1196	871	351	370	217	306	101	86	0
07	Base DX w/Bypass Damper	2300	1619	1126	776	730	470	507	250	214	0
08	Base DX w/Desiccant	16	23	2	0	0	0	0	0	0	0
09	Base DX w/Enthalpy Wheel	315	228	198	84	78	73	62	2	13	0
10	Base DX w/OA Precool	2363	1693	1211	1063	789	559	636	301	207	0
11	Dual Path	1639	904	623	205	247	168	160	79	65	0
12	Dual Path w/Enthalpy Wheel	239	166	139	68	52	67	44	1	10	0
13	Dual Path w/AAHX	586	325	186	58	91	89	40	35	26	0
14	Dual Path w/Desiccant	3	1	0	0	0	0	0	0	0	0
15	Base DX w/DCV	2037	1426	1007	817	601	390	507	170	111	0
16	Dual Path w/DCV	591	281	213	94	116	88	93	10	26	0
17	Base DX w/Free Reheat	200	113	39	5	10	17	2	13	0	0

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System (Location ==>)	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	0%	0%	0%	0%	0%	0%	0%	1%	0%	1%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-11%	-12%	-11%	-13%	-9%	-6%	-6%	-10%	-6%	-7%
03	Base DX w/Lower Airflow	-9%	-9%	-8%	-10%	-7%	-5%	-5%	-8%	-5%	-6%
04	Base DX w/AAHX	23%	19%	16%	17%	11%	8%	8%	11%	7%	7%
05	Base DX w/Subcool Reheat	11%	9%	8%	9%	5%	4%	4%	5%	3%	3%
06	Base DX w/o Lat. Coil Degrad.	5%	5%	4%	5%	4%	2%	2%	2%	2%	1%
07	Base DX w/Bypass Damper	3%	2%	2%	2%	1%	1%	1%	1%	0%	0%
08	Base DX w/Desiccant	240%	101%	91%	59%	69%	19%	17%	8%	-10%	-4%
09	Base DX w/Enthalpy Wheel	-29%	-32%	-32%	-32%	-31%	-44%	-48%	-32%	-45%	-17%
10	Base DX w/OA Precool	11%	8%	7%	6%	8%	6%	6%	8%	6%	8%
11	Dual Path	-1%	-11%	-10%	-16%	-6%	-5%	-5%	-10%	-5%	-8%
12	Dual Path w/Enthalpy Wheel	-28%	-36%	-35%	-38%	-34%	-48%	-51%	-38%	-49%	-24%
13	Dual Path w/AAHX	19%	4%	2%	-3%	3%	1%	1%	-3%	0%	-4%
14	Dual Path w/Desiccant	218%	82%	73%	40%	48%	5%	3%	-8%	-22%	-14%
15	Base DX w/DCV	-16%	-20%	-23%	-19%	-29%	-38%	-37%	-28%	-37%	-34%
16	Dual Path w/DCV	-18%	-31%	-34%	-36%	-36%	-44%	-44%	-39%	-42%	-43%
17	Base DX w/Free Reheat	48%	39%	29%	29%	20%	12%	13%	11%	8%	1%

	Occupied Hours >65%RH	*% Change in Annual HVAC Energy Cost*
White	<= 150 hrs	< 0% (less energy use)
Yellow	151 to 1000 hrs	1% to 25% (more energy use)
Blue	1001 to 2000 hrs	26% to 50% (more energy use)
Magenta	> 2000 hrs	>50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	09	12	12	12	16	12	12	16	12	16
Minimum EC, <=150 hrs RH>65%	14	17	12	12	16	12	12	16	12	16
Minimum Life Cycle Cost (LCC)	09	09	09	09	09	09	09	09	09	15
Minimum LCC, <=150 hrs RH>65%	14	17	12	09	09	09	09	09	09	15
Ratio Min LCC<=150 to Case 01 LCC	2.7	1.2	0.9	0.7	0.8	0.7	0.7	0.8	0.7	0.8

MI = Miami FL
HO = Houston TX
SH = Shreveport LA
FW = Fort Worth TX
AT = Atlanta GA
ST = Washington DC
SL = St. Louis MO
NY = New York NY
CH = Chicago IL
PO = Portland OR

Table 12.8a
System Performance Comparison – 2001 Standard Motel-South

Motel-South

2001 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	7212	5438	4098	3602	3129	2484	2592	1831	1680	9
01	Base DX	7296	5500	4200	3656	3183	2531	2632	1850	1748	10
02	DX w/Improved Dehumid.	7394	5591	4286	3711	3229	2559	2668	1878	1798	11
03	Base DX w/Lower Airflow	7420	5610	4309	3729	3237	2568	2678	1880	1811	11
04	Base DX w/AAHX	5014	4119	2892	1997	1929	1656	1578	1223	827	2
05	Base DX w/Subcool Reheat	7255	5458	4103	3636	3173	2509	2608	1844	1736	9
06	Base DX w/o Lat. Coil Degrad.	4041	2149	915	167	101	163	156	113	46	0
07	Base DX w/Bypass Damper	7277	5471	4146	3638	3172	2523	2621	1850	1742	10
08	Base DX w/Desiccant	2248	1612	939	393	42	11	276	5	14	0
09	Base DX w/Enthalpy Wheel	7174	5214	4048	3721	3239	2521	2657	1968	1773	38
10	Base DX w/OA Precool	6882	5135	3673	3287	2748	2085	2216	1467	1177	0
11	Dual Path	954	574	203	60	0	0	2	16	0	0
12	Dual Path w/Enthalpy Wheel	3639	1947	713	342	1	3	293	21	16	0
13	Dual Path w/AAHX	670	503	186	54	0	0	1	13	0	0
14	Dual Path w/Desiccant	240	187	0	0	0	0	38	0	0	0
15	Base DX w/DCV	7296	5500	4200	3656	3183	2531	2632	1850	1748	10
16	Dual Path w/DCV	954	574	203	60	0	0	2	16	0	0
17	Base DX w/Free Reheat	466	610	280	76	5	0	22	48	6	0

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	1%	1%	1%	1%	1%	1%	1%	1%	1%	1%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-13%	-13%	-13%	-14%	-11%	-9%	-9%	-13%	-9%	-11%
03	Base DX w/Lower Airflow	-11%	-11%	-10%	-11%	-9%	-7%	-7%	-10%	-7%	-8%
04	Base DX w/AAHX	27%	23%	20%	21%	16%	12%	12%	15%	11%	12%
05	Base DX w/Subcool Reheat	9%	8%	8%	8%	6%	5%	5%	6%	4%	5%
06	Base DX w/o Lat. Coil Degrad.	14%	12%	11%	12%	9%	6%	7%	6%	4%	3%
07	Base DX w/Bypass Damper	1%	1%	1%	1%	1%	0%	0%	0%	0%	0%
08	Base DX w/Desiccant	147%	73%	69%	44%	60%	25%	22%	13%	-1%	4%
09	Base DX w/Enthalpy Wheel	-12%	-15%	-16%	-15%	-17%	-28%	-28%	-17%	-28%	-11%
10	Base DX w/OA Precool	7%	6%	6%	5%	6%	5%	4%	5%	4%	4%
11	Dual Path	87%	67%	59%	55%	55%	37%	33%	44%	32%	27%
12	Dual Path w/Enthalpy Wheel	61%	45%	37%	35%	31%	7%	2%	24%	2%	15%
13	Dual Path w/AAHX	95%	73%	64%	61%	60%	40%	37%	48%	35%	31%
14	Dual Path w/Desiccant	197%	112%	98%	73%	82%	40%	35%	38%	16%	17%
15	Base DX w/DCV	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
16	Dual Path w/DCV	87%	67%	59%	55%	55%	37%	33%	44%	32%	27%
17	Base DX w/Free Reheat	69%	54%	44%	45%	34%	23%	25%	25%	16%	9%

Occupied Hours >65%RH
- (white) <= 150 hrs
- (yellow) 151 to 1000 hrs
- (cyan) 1001 to 2000 hrs
- (magenta) > 2000 hrs

% Change in Annual HVAC Energy Cost
- (white) < 0% (less energy use)
- (yellow) 1% to 25% (more energy use)
- (cyan) 26% to 50% (more energy use)
- (magenta) >50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	02	09	09	09	09	09	09	09	09	09
Minimum EC, <=150 hrs RH>65%	NA	NA	14	17	06	12	17	06	08	09
Minimum Life Cycle Cost (LCC)	09	09	09	09	09	09	09	09	09	09
Minimum LCC, <=150 hrs RH>65%	NA	NA	14	17	17	17	17	08	08	09
Ratio Min LCC<=150 to Case 01 LCC	0.0	0.0	2.1	1.3	1.2	1.2	1.2	1.1	1.0	0.9

MI	=	Miami FL	ST	=	Washington DC
HO	=	Houston TX	SL	=	St. Louis MO
SH	=	Shreveport LA	NY	=	New York NY
FW	=	Fort Worth TX	CH	=	Chicago IL
AT	=	Atlanta GA	PO	=	Portland OR

Table 12.8b
System Performance Comparison – 2004 Standard Motel-South

Motel-South

2004 Standard

Humidity Control (Occupied Hours >65%RH)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	7051	5369	4019	3542	3047	2409	2492	1739	1516	3
01	Base DX	7091	5406	4077	3579	3083	2451	2529	1767	1577	4
02	DX w/Improved Dehumid.	7165	5499	4169	3624	3126	2479	2576	1801	1639	4
03	Base DX w/Lower Airflow	7172	5505	4179	3634	3134	2485	2582	1800	1655	4
04	Base DX w/AAHX	5319	4321	3039	2174	2157	1695	1664	1222	839	1
05	Base DX w/Subcool Reheat	7048	5369	3987	3555	3067	2430	2519	1766	1574	3
06	Base DX w/o Lat. Coil Degrad.	4880	2886	1573	409	342	338	377	208	143	0
07	Base DX w/Bypass Damper	7074	5388	4034	3562	3075	2439	2523	1767	1574	3
08	Base DX w/Desiccant	1861	1211	620	162	22	7	208	0	13	0
09	Base DX w/Enthalpy Wheel	7004	5121	3975	3577	3166	2441	2554	1851	1624	16
10	Base DX w/OA Precool	6753	5046	3579	3203	2671	1955	2097	1358	1028	0
11	Dual Path	1146	634	252	64	0	0	7	29	9	0
12	Dual Path w/Enthalpy Wheel	1104	516	130	33	0	0	49	22	0	0
13	Dual Path w/AAHX	728	558	225	60	0	0	3	26	0	0
14	Dual Path w/Desiccant	2	18	0	0	0	0	0	0	0	0
15	Base DX w/DCV	7381	5558	4270	3698	3207	2559	2650	1862	1771	11
16	Dual Path w/DCV	1010	597	213	62	0	0	2	14	1	0
17	Base DX w/Free Reheat	555	635	301	80	17	0	38	55	11	0

Relative Annual HVAC Energy Cost vs. Base DX (Case 1)

Case	System	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
00	Conventional DX	2%	1%	1%	1%	1%	1%	1%	1%	1%	1%
01	Base DX	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%
02	DX w/Improved Dehumid.	-13%	-13%	-12%	-14%	-10%	-8%	-8%	-12%	-8%	-10%
03	Base DX w/Lower Airflow	-11%	-11%	-10%	-11%	-8%	-6%	-6%	-10%	-6%	-8%
04	Base DX w/AAHX	28%	23%	20%	21%	16%	11%	11%	14%	10%	11%
05	Base DX w/Subcool Reheat	10%	8%	7%	8%	6%	4%	4%	6%	4%	5%
06	Base DX w/o Lat. Coil Degrad.	14%	12%	11%	12%	9%	5%	6%	6%	4%	2%
07	Base DX w/Bypass Damper	2%	1%	1%	1%	1%	0%	0%	0%	0%	0%
08	Base DX w/Desiccant	173%	84%	77%	51%	65%	24%	21%	13%	-4%	1%
09	Base DX w/Enthalpy Wheel	-14%	-17%	-19%	-18%	-20%	-33%	-33%	-20%	-32%	-13%
10	Base DX w/OA Precool	9%	8%	7%	6%	7%	5%	5%	6%	5%	5%
11	Dual Path	85%	64%	55%	51%	51%	33%	30%	40%	29%	23%
12	Dual Path w/Enthalpy Wheel	56%	39%	30%	29%	24%	-2%	-5%	17%	-5%	9%
13	Dual Path w/AAHX	95%	71%	61%	58%	57%	37%	34%	45%	32%	27%
14	Dual Path w/Desiccant	217%	116%	100%	72%	80%	33%	30%	33%	9%	11%
15	Base DX w/DCV	-3%	-4%	-5%	-4%	-6%	-8%	-8%	-6%	-9%	-7%
16	Dual Path w/DCV	82%	60%	51%	47%	45%	25%	22%	33%	20%	16%
17	Base DX w/Free Reheat	71%	55%	45%	45%	34%	22%	24%	24%	15%	7%

Occupied Hours >65%RH		*% Change in Annual HVAC Energy Cost*	
(white)	<= 150 hrs	(white)	< 0% (less energy use)
(yellow)	151 to 1000 hrs	(yellow)	1% to 25% (more energy use)
(cyan)	1001 to 2000 hrs	(cyan)	26% to 50% (more energy use)
(magenta)	> 2000 hrs	(magenta)	>50% (more energy use)

Best Cases (Case ID Number)

Criteria	MI	HO	SH	FW	AT	ST	SL	NY	CH	PO
Minimum Energy Cost (EC)	09	09	09	09	09	09	09	09	09	09
Minimum EC, <=150 hrs RH>65%	14	14	12	12	12	12	12	08	12	09
Minimum Life Cycle Cost (LCC)	09	09	09	09	09	09	09	09	09	09
Minimum LCC, <=150 hrs RH>65%	14	14	12	17	17	17	17	08	08	09
Ratio Min LCC<=150 to Case 01 LCC	3.1	2.3	1.6	1.3	1.2	1.2	1.2	1.1	1.0	0.9

MI = Miami FL; HO = Houston TX; SH = Shreveport LA; FW = Fort Worth TX; AT = Atlanta GA; ST = Washington DC; SL = St. Louis MO; NY = New York NY; CH = Chicago IL; PO = Portland OR

The following general trends have been observed:

- The relative ranking of humidity control across system types remains fairly constant from location to location for a given combination of building type and ventilation standard. The overall number of high humidity hours changes across the board when changing locations, but the relative pattern of humidity control remains very similar.

- The relative ranking of humidity control across system types does change significantly from application to application.

- Portland OR has no significant humidity control issues for all applications, however there are significant cost differences across the system types.

- Simple variations on the base DX system, Cases 0-3, show varied results in humidity control and energy costs. In some cases, there is clear advantage to some of these systems, in other cases, it makes little difference in humidity control.

- Demand controlled ventilation (DCV) saves energy, but does little to improve humidity control, because the system must still be sized to handle the full ventilation load. Whenever the DCV reduces the ventilation flow rate, the part-load on the system falls which increases the latent degradation impact on the system.

- The semi-active humidity control systems, Case 5 and Case 7 (subcool reheat and coil bypass) provide some humidity control benefits in certain cases, but generally leave a significant number of high humidity hours.

- The results for Case 6 (no latent degradation) represent the best humidity control case for most of the single-path systems without special options. If this case still shows many hours of high humidity, it indicates that many of these hours occur at times when there is no sensible cooling load to activate the cooling system. This is prevalent in the Restaurant, Theater, and Motel, because of the high number of evening and nighttime operating hours.

- The OA pretreatment system (Case 10) does not control humidity well as modeled here. The typical application of this system type is in combination with subcool or hot gas reheat. According to the manufacturer, this combination can be very effective.

- Dual path systems often provide better humidity control. This configuration also reduces the total supply air flow rate and results in significant fan power savings.

- For applications with little or no humidity controls problems, including Office and Retail, Enthalpy Wheel or DCV systems can reduce overall life cycle costs compared to Case 1 Base DX.

- For applications with substantial humidity control problems, including Restaurant, Theater, School and Motel, better control generally comes at a significantly higher life cycle cost using dual path, reheat or desiccant systems, but there are some opportunities for cost savings in certain climates under the 2004 Standard.

Trends for each building type are discussed below.

12.3.1 Office

(Overall Ventilation Rates 2001: 0.14 cfm/sf ~25% OA, 2004: 0.09 cfm/sf ~20% OA)

For the Office, there are almost no humidity control issues, except perhaps in Miami with the 2001 Standard. For the most humid climates with the 2001 Standard, Cases 2 & 3, the conventional choices for better humidity control, actually result in somewhat poorer humidity control, but they do save energy due to reduced fan power. With the 2004 Standard, there are no humidity control issues, so lower airflow rates may be desirable for energy savings.

In all climates with the 2004 Standard, Case 12 Dual Path w/Enthalpy Wheel offers the lowest energy costs, and Case 9 Base DX w/Enthalpy Wheel offers the lowest life cycle costs with savings of 10-20%. With the 2001 Standard, in the drier climates, Case 16 Dual Path w/DCV is lowest in energy costs, but Case 9 Base DX w/Enthalpy Wheel is still the lowest in life cycle costs. Portland with the 2001 Standard is an exception, where Case 15 Base DX w/DCV is lowest in life cycle cost.

12.3.2 Restaurant

(Overall Ventilation Rates 2001: 1.43 cfm/sf ~60% OA, 2004: 0.72 cfm/sf ~45% OA)

The Restaurant is a challenging application in all but the driest climates due to low internal gains in the dining room, high ventilation rates, and a large number of evening operating hours. This results in the vast majority of hours out of range occurring when there is little or no sensible cooling load in the space. Only the desiccant and reheat systems have been controlled to operate when there is only a latent cooling requirement, and so they are generally the only systems which can adequately control humidity in this application. In addition, with the 2004 Standard, Case 12 (dual path with enthalpy wheel) is able to eliminate a high percentage of the high humidity hours at a significantly lower life cycle cost. 2004 Case 9, single path with enthalpy wheel reduces them by approximately 50%, but in the humid climates there are still approximately 2000 hours of high humidity.

Of the systems which provide adequate humidity control, Case 14 Dual Path w/Desiccant offers the lowest energy costs and life cycle costs with the 2001 Standard, except for some of the drier locations in which Case 8 Base DX w/Desiccant and Case 15 Base DX w/DCV offer lower life cycle costs. With the 2004 Standard, the results are more varied, with Case 17 Base DX w/Free Reheat, Case 14 Dual Path w/Desiccant, Case 12 Dual Path w/Enthalpy Wheel, and Case 15 Base DX w/DCV offering lowest costs depending on the location. For the 2001 standard, better humidity control comes at a significantly higher life cycle cost compared to Case 1 Base DX, ranging from a factor of 1.1 to 3.4. For the 2004 Standard, the cost ratios vary from 0.6 to 1.9, generally higher in the most humid climates.

12.3.3 Retail

(Overall Ventilation Rates 2001: 0.30 cfm/sf ~40% OA, 2004: 0.23 cfm/sf ~40% OA)

For the Retail Store the enthalpy wheel systems control humidity well, save energy, and have the lowest life cycle costs. (Note that for the 2001 Standard, the minimum ventilation rate does not have a per person component, so the DCV controls are inactive.) The desiccant systems control humidity well, but with higher energy and life cycle costs.

With both the 2001 and 2004 Standards, Case 12 Dual Path w/Enthalpy Wheel shows the lowest costs in the more humid climates while Case 9 Base DX w/Enthalpy Wheel shows lower life cycle costs in the drier climates. DCV cases did not apply with the 2001 Standard, because it has a straight cfm/sf ventilation requirement for Retail. Under the 2004 Standard, DCV systems have life cycle costs equal to or less than Case 1 Base DX.

12.3.4 Theater

(Overall Ventilation Rates 2001: 2.14 cfm/sf ~65% OA, 2004: 0.77 cfm/sf ~40% OA)

The Theater is similar to the restaurant with small internal gains other than people, and a high number of evening operating hours resulting in 1000s of high humidity hours. Again, the desiccant and reheat systems are the only ones able to control humidity. For the 2004 Standard, with significantly lower ventilation rates, Cases 13 Dual Path w/AAHX can also significantly reduce humidity levels and save energy. Unlike the Restaurant, the enthalpy wheel systems do not provide better humidity control.

With the 2001 Standard, of the systems which offer adequate humidity control, Case 14 Dual Path w/Desiccant has the lowest energy costs in all climates except Portland. Case 14 also has the lowest life cycle costs in the more humid climates, with Case 8 Base DX w/Desiccant showing lower costs in the drier climates. With the 2004 Standard, Case 17 Base DX w/Free Reheat is the lowest life cycle cost option in all cities except Portland, while Case 14 Dual Path w/Desiccant offers lower energy costs in the drier climates. For both standards, better humidity control comes at a significantly higher life cycle cost compared to Case 1 Base DX, ranging from a factor of 1.1 to 3.4.

12.3.5 School-9 Month-South

(Overall Ventilation Rates 2001: 0.75 cfm/sf ~50% OA, 2004: 0.37 cfm/sf ~40% OA)

The School-9 Month-South has significant humidity control problems, but not as severe as the Theater and Restaurant. Again, the desiccant and reheat systems are the only systems which fully control the humidity levels. The enthalpy wheel systems (Case 9 and 12) are highly effective, but still have significant hours of high humidity in the most humid climates.

With the 2001 Standard, of the systems which offer adequate humidity control, Case 17 Base DX w/Free Reheat is the lowest life cycle cost system in all locations except Portland and Houston. Case 14 Dual Path w/Desiccant offers lower energy costs in the drier climates but higher life cycle costs. With the 2004 Standard, Case 17 Base DX w/Free Reheat is the lowest energy and life cycle cost in the most humid climates. In moderate climates, Case 12 Dual Path w/Enthalpy Wheel takes over as the lowest cost, and in the driest climates Case 9 Base DX w/Enthalpy Wheel offers lower life cycle cost. For both standards, better humidity control comes at a significantly higher life cycle cost compared to Case 1 Base DX, ranging from a factor of 1.1 to 1.9, except moderate and dry cities show 20% to 30% life cycle cost reduction under the 2004 Standard.

12.3.6 School-12 Month-South

(Overall Ventilation Rates 2001: 0.75 cfm/sf ~50% OA, 2004: 0.37 cfm/sf ~40% OA)

The School-12 Month model follows a year-round school schedule with multiple 2-3 week breaks rather than one long summer break. The total number of occupied hours is very similar, 3610 for the School-12 Month and 3764 for the School-9 Month, but the distribution of weather conditions is significantly different. The relative humidity control performance of the various systems in School-12 Month-South is nearly identical to School-9 Month-South, but the minimum cost system differs.

With the 2001 Standard, of the systems which offer adequate humidity control, Case 14 Dual Path w/Desiccant, Case 17 Base DX w/Free Reheat, and Case 8 Base DX w/Desiccant are each the lowest energy and life cycle cost in one of the most humid climates. Case 16 Dual Path w/DCV is the lowest cost system in the moderately humid climates, and Case 9 Base DX w/enthalpy Wheel is lowest cost in northern climates. With the 2004 Standard, Case 14 Dual Path w/Desiccant, Case 17 Base DX w/Free Reheat, and Case 12 Dual Path w/Enthalpy Wheel are each the lowest energy and life cycle cost system in one humid climate. For the moderately humid and drier climates, Case 12 Dual Path w/Enthalpy Wheel and Case 16 Dual Path w/DCV have the lowest energy costs, but Case 9 Base DX w/Enthalpy Wheel is the lowest life cycle cost. For both standards, better humidity control comes at a significantly higher life cycle cost compared to Case 1 Base DX in the most humid climates, ranging from a factor of 1.2 to 2.9, while other cities show the potential for cost savings.

12.3.7 Motel-South

(Overall Ventilation Rates 2001: 0.09 cfm/sf ~16% OA, 2004: 0.11 cfm/sf ~20% OA)

The Motel is by far the most challenging application with over 1000 hours of high humidity in all locations except Portland OR, and more than 7000 hours in Miami. This is because the Motel systems are running at all times, infiltration is active at all times, the internal gains are low, and the equipment is single-stage. Because of the low OA fraction, this is one application where Case 8 (base DX with desiccant) is not able to control the humidity in the most humid climates. With a few exceptions, Case 6 (no latent degradation), Cases 11-14 and 16 (dual path variations), and Case 17 (reheat) provide substantially better humidity control, but with higher energy use. In the less humid climates, these systems are able to provide adequate humidity control, possibly because they provide multiple stages of cooling more than because of the dual path configuration.

With the 2001 Standard, none of the systems provide adequate humidity control in the most humid climates (Miami and Houston). In the moderate climates, except Shreveport, Case 17 Base DX w/Free Reheat offers the lowest life cycle cost. In Shreveport, Case 14 Dual Path w/Desiccant is the lowest life cycle cost of systems which provide adequate humidity control. With the 2004 Standard, Case 14 Dual Path w/Desiccant is the only system which provides adequate humidity control in the most humid climates (Miami and Houston). In most other climates, Case 17 Base DX w/Free Reheat offers the lowest life cycle cost, with Case 12 Dual Path w/Enthalpy Wheel being the lowest cost in Shreveport, and Case 8 Base DX w/Desiccant being the lowest cost in New York and Chicago. For both standards, better humidity control comes at a significantly higher life cycle cost compared to Case 1 Base DX, ranging from a factor of 1.1 to 3.1. For the cases where adequate humidity control cannot be obtained, Case 14 Dual Path w/Desiccant offers the best option with 180-250 hrs >65%RH and life cycle cost factors ranging from approximately 2.3 to 3.0.

13 Guidelines, Issues and Conclusions

13.1 Guidelines

The ultimate goal of this work was to develop design guidelines. The following general principles may be posed from the results of this analysis:

1. In nearly all cases, simple variations in the Base DX system (lower airflow, lower SHR) do little to improve humidity control but may be useful to save fan energy. The exception to this rule is Standard 2004 ventilation rates with the Retail application in the most humid climates.

2. Demand controlled ventilation (DCV) saves energy, but does little to improve humidity control in most cases.

3. Semi-active humidity control systems (Case 5 Subcool Reheat and Case 7 and Coil Bypass) can help but often fall short, especially in the most humid climates.

4. Certain applications, such as the Theater, Restaurant and Motel, in very humid climates have high humidity issues primarily at times when there is no sensible load on the coil due to cool moist outside air. Only active humidity control systems (desiccants and reheat) can control humidity at such times. Depending on the control settings, enthalpy wheels may not operate at such times, and therefore provide less benefit for humidity control.

5. For all of the systems without direct humidity control (all cases except desiccant Cases 8 and 14 and reheat Case 17), system capacity vs. load profile is crucial. The poor humidity control performance of many of these system options can be attributed primarily to a high percentage of hours operating at low part loads. 2-stage systems with a 60% stage 1 capacity help significantly, but do not overcome this issue. Case 6 Base DX w/o Latent Coil Degradation represents the ideal in capacity staging where the coil never evaporates condensed moisture back into the supply air stream.

6. For the Office, humidity control is not an issue.

7. For the Restaurant, Theater, and Schools, systems with direct humidity control (desiccant Cases 8 and 14 and reheat Case 17) are the only systems which can provide adequate humidity control in the most humid climates. In less humid climates, enthalpy wheel systems (Cases 9 and 12) can also provide adequate control.

8. For the Motel, continuous operation and single-stage equipment result in excessive hours of high humidity. Only Case 14 Dual path w/Desiccant provides adequate (or near-adequate) humidity control in the most humid climates. Reheat and dual path systems can help significantly, and are sufficient in moderate climates.

9. For the Retail Store, a wider range of options can be beneficial.

10. The enthalpy wheel and DCV options generally provide equal or better humidity control compared to the base system, with significant energy cost and life cycle cost savings. Significantly better humidity control (but not necessarily adequate control) is found in the Restaurant with the 2004 Standard, Retail with both standards, and School with both standards. Worse humidity control is found in the Restaurant and Theater in certain locations.

13.2 Issues

The results of this analysis raise several issues for further investigation:

- Would adequate capacity staging solve humidity control problems in all but the most extreme cases? Case 6 Base DX w/o Latent Coil Degradation results show that better staging might help in cases with moderate humidity control issues, but it makes little difference in the Theater, Restaurant, School, and Motel in the most humid climates. .

- Do the dual path systems in this analysis perform better because they are dual path, or simply because they have four stages of cooling available in the outside air stream? Would the same four-stage system in a single path unit provide similar results?

- For some applications in high humidity climates, there are times when a zero SHR is required, because humidity is high but there is no need for sensible cooling. This requires a system such as hot gas reheat, essentially a dehumidifier. How much of the total system cooling capacity is needed at these times? Would it be more cost effective to add a small dehumidifier in the outside air stream?

- Fan power issues are significant. Would generally lower fan cfm/ton be beneficial if combined with adequate capacity staging to improve humidity control and save energy? How can the year-round fan power penalty of some of these systems be minimized?

- The outdoor air preconditioning system was not the typical application. This should be examined in combination with subcool or hot gas reheat.

- Would alternative desiccant dehumidifier configurations, such as placing the desiccant wheel after the DX cooling coil, provide adequate humidity control at lower costs and energy use?

- Additional data mining may reveal trends related to design SHR, ventilation load index, or other defining characteristics of the loads.

13.3 Conclusions

This research project has provided the following benefits:

- Comprehensive analysis of humidity control performance of a wide range of DX system configurations.

- Significant advancement in whole building energy simulation capabilities for modeling DX equipment by adding new capabilities to EnergyPlus. This provides access to designers and analysts to study specific projects and extend the results of this analysis.

- Identification of key issues for further exploration to better understand some of the key drivers and possibly develop some simple new system configurations that can efficiently control humidity.

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