Contract Report

# Desiccant Dehumidification Enhancement of Electric Air Conditioning Units

## Final Report

FSEC-CR-651-93

October 8, 1993

Solar Cooling Research Project
DOE Contract # DE-FC03-86SF16305

Prepared for:
U.S. Department of Energy

Prepared by:
Bruce G. Nimmo
Kannan Rengarajan

**Florida Solar Energy Center**
300 State Road 401
Cape Canaveral, Florida 32920

# Desiccant Dehumidification Enhancement of Electric Air Conditioning Units

FINAL REPORT
October 1993

By

Florida Solar Energy Center
300 State Road 401
Cape Canaveral, Florida 32920

DOE Contract No. DE-FC03-86SF16305
Amendment A007 and A011

Prepared for
Department of Energy
Building Equipment Division
Office of Buildings Energy Research
Conservation and Renewable Energy
US DOE CE-422
1000 Independence Avenue., SW
Washington, DC 20585

## ACKNOWLEDGEMENTS

The authors wish to thank the several people who contributed to this study:

Kirk Collier for input to the modeling of the desiccant wheels; Hugh Henderson for providing insights to some of the simulation models he had created while at FSEC; Don Shirey for providing technical support on various issues related to this project; Richard Raustad for preparing several of the illustrations in this report; Jeanette Holder for her creative final typing of this report.

Finally, we would like to thank Esher Kweller, the DOE project manager, for his guidance, support and encouragement during the course of this project.

## TABLE OF CONTENTS

ACKNOWLEDGEMENTS . . . . . i

EXECUTIVE SUMMARY . . . . . x

1.0 INTRODUCTION . . . . . 1
- 1.1 DEAC Process . . . . . 1
- 1.2 Goals of Study . . . . . 2

2.0 APPROACH TO THE STUDY . . . . . 4
- 2.1 Overview . . . . . 4
- 2.2 Development of Component Performance Maps . . . . . 4
- 2.3 Development of AC System Modules . . . . . 4
- 2.4 Parametric Comparison of Electric AC Systems . . . . . 5
- 2.5 Development of Building Characteristics . . . . . 5
- 2.6 AC System Component Sizing . . . . . 5
- 2.7 Baseline AC Characteristics . . . . . 5
- 2.8 Seasonal Simulations . . . . . 6

3.0 AC SYSTEMS REVIEW . . . . . 7
- 3.1 Conventional Single-Speed AC . . . . . 7
- 3.2 Single-Speed AC with Heat Pipes . . . . . 7
- 3.3 Desiccant Enhanced Air Conditioner . . . . . 7
- 3.4 Variable-Speed AC with Heat Pipes or Desiccant Wheel . . . . . 12
- 3.5 AC with Gas-fired Desiccant Wheel . . . . . 12
- 3.6 AC with Kathabar Dehumidifier . . . . . 15

4.0 SOFTWARE MODEL DEVELOPMENT OF AC COMPONENTS . . . . . 18
- 4.1 Overview . . . . . 18
- 4.2 ORNL Heat Pump . . . . . 18
- 4.3 DOE-2 Packaged AC . . . . . 19
- 4.4 Variable-Speed AC . . . . . 22
- 4.5 Heat Pipes . . . . . 24
- 4.6 Desiccant Wheel for DEAC . . . . . 29
- 4.7 Forward-Curved Fan . . . . . 35
- 4.8 Variable-Speed Fan . . . . . 35
- 4.9 Gas-fired Desiccant Wheel . . . . . 35
- 4.10 Kathabar Liquid-Desiccant Dehumidifier . . . . . 37
- 4.11 Assembling Components into Systems . . . . . 38

5.0 PERFORMANCE COMPARISON OF ELECTRIC AC SYSTEMS . . . . . 40
5.1 Overview . . . . . 40
5.2 Cooling Capacity Variation with SHR . . . . . 40
5.3 EER Variation with SHR . . . . . 43

6.0 BUILDING ENERGY ANALYSIS SOFTWARE: FSEC 2.2 . . . . . 46
6.1 Heat and Moisture Transfer Models . . . . . 46
6.2 EMPD Approach . . . . . 47
6.3 Conduction Transfer Function (CTF) Method . . . . . 47
6.4 Buildings and Mechanical Systems Design Parameters Simulated with FSEC 2.2 . . . . . 47
6.5 AC Systems and Components Control Options . . . . . 47
6.5.1 SSAC with/without Heat Pipes or a Desiccant Wheel and/or Reheat . . . . . 49
6.5.2 VSAC with/without Heat Pipes or a Desiccant Wheel and/or Reheat . . . . . 49
6.5.3 Gas-fired Hybrid Systems . . . . . 50

7.0 DESCRIPTION OF BUILDINGS CHARACTERISTICS . . . . . 52
7.1 Residence . . . . . 52
7.2 Office Building . . . . . 52
7.3 Fast Food Restaurant . . . . . 56

8.0 BUILDINGS LOADS ESTIMATION AND AC SYSTEMS SIZING . . . . . 58
8.1 Ideal Loads . . . . . 58
8.2 Load Sensible Heat Ratio . . . . . 66
8.3 Applicability of AC Types to Buildings . . . . . 66
8.4 Sizing of AC Systems . . . . . 70
8.4.1 Supply Air Flow Rate . . . . . 70
8.4.2 Single-Speed AC and Air-Reheat Systems . . . . . 71
8.4.3 Single-Speed AC with Heat Pipes or a Desiccant Wheel . . . . . 71
8.4.4 Variable-Speed AC with Heat Pipes or a Desiccant Wheel . . . . . 71
8.4.5 Gas-fired Hybrid Systems . . . . . 72

9.0 RESIDENCE AIR CONDITIONING ENERGY ANALYSIS . . . . . 73
9.1 Introduction . . . . . 73
9.2 Simulation Basis . . . . . 74
9.2.1 Size of AC Systems . . . . . 74
9.2.2 Capital Costs of AC Systems . . . . . 75
9.3 Comparison Measure: Relative Merit Index . . . . . 75
9.4 Simulation Results . . . . . 76
9.5 Residential Applicability of DEAC . . . . . 84

9.5.1 Humidity Control by Existing "Average" Air Conditioner . . . . . 84
9.5.2 Humidity Control by "Current" Air Conditioner (State-of-the-Art) . . . . . 86
9.5.3 Comparative Energy Use of the DEAC and Current AC Units . . . 86
9.6 Potential for Energy Savings with the DEAC system in a Residence . . . . . 91
9.7 Projected Nationwide Energy Savings with DEAC for Residential Sector . . . . . 91
9.8 Conclusion . . . . . 93

10.0 FAST FOOD RESTAURANT AIR CONDITIONING ENERGY ANALYSIS . . 96
10.1 Introduction . . . . . 96
10.2 Simulation Basis . . . . . 97
10.2.1 Size of AC Systems . . . . . 97
10.2.2 Capital Costs of AC Systems . . . . . 97
10.3 Simulation Results . . . . . 97
10.4 Conclusion . . . . . 103

11.0 OFFICE BUILDING AIR CONDITIONING ENERGY ANALYSIS . . . . . 104
11.1 Introduction . . . . . 104
11.2 Simulation Basis . . . . . 104
11.2.1 Size of AC Systems . . . . . 105
11.2.2 Capital Costs of AC Systems . . . . . 106
11.3 Simulation Results . . . . . 106
11.4 Conclusion . . . . . 110

## APPENDICES

APPENDIX A - Comparison of Results from FSEC 2.2 and ESPRE 2.1 for a Residence . . . . . A-1

APPENDIX B - Evaluation of Run Around Belt Configuration for DEAC System . . . B-1

APPENDIX C - TMY Weather Data . . . . . C-1

APPENDIX D - Air Conditioning Equipment Sizes for Restaurants . . . . . D-1

APPENDIX E - Electric and Gas AC System Component Costs . . . . . E-1

APPENDIX F - Gas and Electric Rates . . . . . F-1

APPENDIX G - Residence, Office and Fast Food Restaurant Building Schedules . . . G-1

## LIST OF FIGURES


Figure ES1 - Primary Region of Residential Applicability of the DEAC . . . . . xvii
Figure 1.1 - Schematic Diagram of DEAC . . . . . 3
Figure 3.1 - Psychrometric Process for Conventional AC . . . . . 8
Figure 3.2 - Psychrometric Process for Air-Reheat System . . . . . 9
Figure 3.3 - Psychrometric Process for Heat Pipe-Assisted AC . . . . . 10
Figure 3.4 - Psychrometric Process for DEAC . . . . . 11
Figure 3.5 - Schematic Diagram of Gas-fired Hybrid System . . . . . 13
Figure 3.6 - Psychrometric Process for Gas-fired Hybrid System . . . . . 14
Figure 3.7 - Schematic Diagram of AC with Kathabar Dehumidifier . . . . . 16
Figure 3.8 - Psychrometric Process for AC with Kathabar Dehumidifier . . . . . 17
Figure 4.1 - ORNL AC: Capacity Versus Temperature and Humidity . . . . . 20
Figure 4.2 - ORNL AC: EER Versus Temperature and Humidity . . . . . 21
Figure 4.3 - DOE-2 AC: EER and SHR Contours . . . . . 23
Figure 4.4 - Variable Speed AC: Capacity as a Function of Compressor Speed . . . . . 25
Figure 4.5 - Variable Speed AC: EER as a Function of SHR . . . . . 26
Figure 4.6 - Heat Pipe Effectiveness as a Function of Face Velocity and Number of Rows . . . . . 28
Figure 4.7 - DEAC Psychrometric Process State Points Predicted as a Function of Wheel Depth ($N_{tu}$) . . . . . 31
Figure 4.8 - DEAC Sensible Heat Ratio as a Function of Desiccant Wheel Depth ($N_{tu}$) . . . . . 32
Figure 4.9 - DEAC: Sensible Heat Ratio as a Function of Desiccant Wheel Rotational Speed . . . . . 33
Figure 4.10 - Impact of Air Bypass on DEAC Capacity, EER and SHR . . . . . 34
Figure 4.11 - Variable Speed Fan Power as a Function of Air Flow Rate . . . . . 36
Figure 5.1 - Comparison of AC Options: Total Capacity as a Function of SHR . . . . . 41
Figure 5.2 - Comparison of AC Options: Efficiency as a Function of SHR . . . . . 44
Figure 6.1 - Schematic of EMPD Approach . . . . . 48
Figure 7.1 - Schematic of Residence . . . . . 53
Figure 8.1 - Sensible and Latent Loads for a Residence . . . . . 59
Figure 8.2 - Sensible and Latent Loads for a Medium Office Building . . . . . 60
Figure 8.3 - Sensible and Latent Loads for a Fast Food Restaurant . . . . . 61
Figure 8.4 - Design Peak Loads: Residence . . . . . 62
Figure 8.5 - Design Peak Loads: Medium Office Building . . . . . 63
Figure 8.6 - Design Peak Loads: Fast Food Restaurant (Dining) . . . . . 64
Figure 8.7 - Design Peak Loads: Fast Food Restaurant (Kitchen) . . . . . 65
Figure 8.8 - SHR Histogram for Miami Residence . . . . . 67
Figure 8.9 - Monthly Load-SHR Profile for Miami Residence . . . . . 68
Figure 8.10 - Monthly Load-SHR Profile for Miami Office Building . . . . . 69
Figure 9.1 - Miami Residence: AC Systems Performance . . . . . 77
Figure 9.2 - Indoor Temperature and Humidity Profile in the Modeled Miami Residence . . . . . 78

Figure 9.3 - SHR of Heat Pipe-Assisted AC and DEAC as a Function of Return Air RH . . . . . 80
Figure 9.4 - St.Louis Residence: AC Systems Performance . . . . . 81
Figure 9.5 - Washington Residence: AC Systems Performance . . . . . 82
Figure 9.6 - Residence: AC Systems Relative Merit Index . . . . . 83
Figure 9.7 - Correlation Between CDD and LEH: Selected Cities for Analysis . . . . . 85
Figure 9.8 - Residence: Hours Indoor RH > 60% (Air Conditioners in Use (1984-1991)) . . . . . 87
Figure 9.9 - Residence: Hours Indoor RH > 60% (DEAC and "Current" AC) . . . . . 88
Figure 9.10 - Annual Air Conditioning Energy use (DEAC, "Average" and "Current" AC) . . . . . 89
Figure 9.11 - DEAC's Relative Merit Index in Ten U.S. Cities . . . . . 90
Figure 9.12 - Miami Residence: Comparison of Energy Use and Comfort for Current AC and DEAC . . . . . 92
Figure 9.13 - Primary Region of Residenrtial Applicability of the DEAC . . . . . 94
Figure 10.1 - Miami Restaurant: AC Systems Performance . . . . . 99
Figure 10.2 - St.Louis Restaurant: AC Systems Performance . . . . . 100
Figure 10.3 - Washington Restaurant: AC Systems Performance . . . . . 101
Figure 10.4 - Restaurant: AC Systems Relative Merit Index . . . . . 102
Figure 11.1 - Miami Office: AC Systems Performance . . . . . 107
Figure 11.2 - St.Louis Office: AC Systems Performance . . . . . 108
Figure 11.3 - Washington Office: AC Systems Performance . . . . . 109
Figure 11.4 - Office: AC Systems Net Present Value . . . . . 111

## LIST OF TABLES

TABLE ES1 . . . . . xiii
TABLE 4.1 - Basic AC Systems and Components . . . . . 18
TABLE 4.2 - Range of conditions specified for Single-Speed AC . . . . . 19
TABLE 4.3 - Nominal performance of Single-Speed AC . . . . . 19
TABLE 4.4 - Range of Conditions Specified for Variable-Speed AC . . . . . 22
TABLE 4.5 - Nominal Performance of Variable-Speed AC . . . . . 24
TABLE 4.6 - DEAC Simulation Model Parameters . . . . . 29
TABLE 4.7 - Silica Gel Desiccant Wheel: Range of Curve-fit Conditions . . . . . 37
TABLE 4.8 - Kathabar Dehumidifier: Range of Curve-fit Conditions . . . . . 38
TABLE 6.1 - Modes of operation for Gas-Fired Hybrid Systems . . . . . 50
TABLE 7.1 - Residence Characteristics . . . . . 54
TABLE 7.2 - Medium Size Office Building Characteristics . . . . . 55
TABLE 7.3 - Fast Food Restaurant Characteristics . . . . . 57
TABLE 8.1 - AC Systems Compared Using Seasonal Simulations . . . . . 70
TABLE 9.1 - Residence: AC Systems Compared . . . . . 73
TABLE 9.2 - Residence: AC Systems Size . . . . . 74
TABLE 9.3 - Capital Costs of Residential AC System Components . . . . . 75
TABLE 9.4 - Residential Applicability of DEAC: Demographic Information . . . . . 93
TABLE 10.1 - Fast Food Restaurant: AC Systems Compared . . . . . 96
TABLE 11.1 - Medium Office: AC Systems Compared . . . . . 105
TABLE 11.2 - Office: AC Systems Size . . . . . 106

## GLOSSARY

| | |
|---|---|
| Absolute Humidity | Absolute Moisture Content of Air. (lb of water/lb of dry air) |
| AC | Air Conditioner |
| ADP | Apparatus Dew Point. The average surface temperature of direct expansion air conditioner evaporator coil. |
| ARI | Air Conditioning and Refrigeration Institute. Manufacturers Association located in Arlington, VA. |
| ASHRAE | American Society of Heating, Refrigerating, and Air-Conditioning Engineers. Professional Society. Located in Atlanta, GA. |
| Capacity | Delivered, usable output of a system. Btu/h. |
| CPH | Cycles Per Hour. |
| DB | Dry Bulb Temperature. |
| Dehumidification | The removal of moisture from air. |
| Dew Point | The temperature at which water vapor begins to condense from air at a constant pressure. |
| DOE-2 | Building energy analysis software developed by the U.S. Department of Energy. |
| EER | Energy Efficiency Ratio. Total delivered Cooling Capacity per Watt of Electrical Input. (Btu/Wh) |
| Evaporator | An AC heat exchanger which cools and dehumidifies the incoming air. |
| FSEC | Florida Solar Energy Center. |
| HX | Heat Exchanger. |
| Latent capacity | The rate of dehumidification provided by an AC. (Btu/h) |

| | |
|---|---|
| Latent load | The rate of dehumidification (moisture removal) required to maintain the desired humidity level in a space. |
| $N_{tu}$ | Number of Transfer Units. A Measure of the rate at which heat and/or mass may be transferred in an heat exchanger system. |
| ORNL AC | A detailed air conditioner model developed at Oak Ridge National Laboratory (ORNL). |
| Relative Humidity | The percentage moisture content of the air compared to the moisture content of saturated air at the same pressure. |
| SEER | Seasonal Energy Efficiency Ratio. Average EER for the AC system over the cooling season (Btu/Wh). |
| Sensible Capacity | The rate of sensible cooling provided by a System (Btu/h). |
| Sensible Load | The rate of sensible cooling required to maintain the space temperature |
| Sensible Heat Ratio | The ratio of sensible capacity to the total capacity of an AC. (Dimensionless) |
| SSAC | Single-Speed Air Conditioner |
| Total Capacity | The sum of sensible and latent capacity. (Btu/h) |
| Total Load | The sum of sensible and latent load. (Btu/h) |
| Ventilation Mode | Mode of operation of a gas-fired desiccant system where process air comes from outdoors. |
| VSAC | Variable-Speed Air Conditioner |
| WB | Wet Bulb temperature |

# EXECUTIVE SUMMARY

## Focus

There is a growing recognition of the need to address humidity control for buildings in hot and humid climates. Some of the tangible benefits from improved indoor humidity control are: healthier environments, energy savings, demand reduction and improved productivity. The increase in ventilation rates to meet ASHRAE standard 62-89 for commercial buildings, and the potential for high humidity problems have further motivated the electric and gas industries to develop advanced dehumidification technologies.

A new concept, the desiccant enhanced air conditioner (DEAC) dehumidification option, appears to offer improved humidity control at lower costs. Based on the results of this study, an energy saving potential of 3.25 to 6.5 x $10^6$ MWatt hours per year exists for the estimated primary area of residential DEAC users. A comparative analysis of the DEAC system and its performance form the basis of this study.

## DEAC and Alternative AC Systems

The Desiccant Enhanced Air Conditioning (DEAC) concept was conceived and patented by C. Cromer (1, 2) of the Florida Solar Energy Center (FSEC). The DEAC uses a desiccant wheel or liquid desiccant to pre-cool and humidify return air before it enters a cooling coil. It then heats and dehumidifies the supply air after it leaves the coil. A unique feature of this process is that regeneration of the desiccant component is accomplished by the return air rather than by an external heat source. The result of the DEAC psychrometric process is increased dehumidification of the supply air.

One advantage of the desiccant approach over a sensible heat exchanger (e.g., heat pipes) is the relatively high wet bulb temperature of air entering the evaporator coil. This helps maintain high efficiency of the air conditioner while improving dehumidification.

Other air-conditioning (AC) systems currently available for improved dehumidification and considered in this study include:

a. Heat pipe-assisted air conditioners
b. Air-reheat systems (e.g., electric reheat)
c. Variable-speed air conditioners (VSAC)
d. Gas-fired solid and liquid desiccant systems.

## Goals of Study

The goals of this study were:

- Compare DEAC EER to alternative electric systems over a range of SHR values
- Compare DEAC performance to that of electric and gas dehumidification systems when operating in three building types (residence, office and fast food restaurant) in three climates (Miami, St.Louis and Washington D.C.)

- Identify the DEAC primary geographical region of applicability
- Estimate for the primary region of applicability, the potential residential energy saving of the DEAC relative to the current high-efficiency air conditioner for the case when both meet the same standard of comfort
- Provide an analysis of a run-around belt option as an alternative to the desiccant wheel for use in the DEAC.

This report summarizes the basis and results of the computer simulation studies performed to achieve the above goals.

Methodology

In this project, we studied the potential of the DEAC for use in commercial and residential applications by comparing it to a variety of alternative air conditioning (AC) systems. The approach was to compare the various AC systems by modeling their performance and integrating the results with the FSEC 2.2 (Florida Software for Enervironment (Energy and Environment) Computation, version 2.2) building simulation code (3). This method allowed the systems to be compared for a variety of building types and climates. The primary tasks performed were:

a. Develop component performance maps
b. Develop AC system modules
c. Compare performance of selected electric AC systems over a range of energy efficiency and sensible heat ratios
d. Develop building characteristics
e. Select appropriate baseline AC characteristics for each building application
f. Size AC systems for each building/climate matrix
g. Perform annual or seasonal simulations for AC system/building/climate matrix and interpret results.
h. Analyze run-around belt configuration for use in the DEAC

Comparison of Electric AC Systems for Performance

The first major task of the project was to develop simplified component performance models for all the AC sub-systems. Following the completion of software code development and testing of all the required AC systems for this study, a parametric study was conducted to compare the energy efficiency ratings of the DEAC, single-speed air conditioner (SSAC) with and without electric reheat and heat pipes, and variable-speed air conditioner (VSAC) over a range of SHR values (0.4 to 0.9).

The basic air conditioner model used in this comparison is the DOE-2 AC model (4). The assigned nominal energy efficiency ratio (EER) of the basic 3 ton AC model was 12.0 and the assigned sensible heat ratio (SHR) rating was 0.84 at American Refrigeration Institute (ARI) standard test conditions (Indoor; 80°F Dry Bulb, 67°F Wet Bulb: Outdoor; 95°F). A nominal air flow rate of 450 cfm/ton was specified at the above conditions.

The parametric study results showed that, to meet a typical residential load-SHR value of 0.7 using a DEAC with these AC characteristics, a wheel thickness of about 1" ($N_{tu}$: 0.69) is required. The reduction of the DEAC EER associated with an SHR of 0.7 was only 3 percent.

When the DEAC SHR was reduced to a low value of 0.425 using a 6 inch wheel thickness ($N_{tu}$: 4.1), the reductions in cooling capacity and EER were a moderate 3.5 and 12 percent respectively. Cooling capacity and EER reduction of the DEAC result from increased air-side pressure drop associated with the increased wheel thickness and the resulting increase in fan power.

The optimum desiccant wheel speed (the speed at which maximum mass transfer is attained) of the DEAC occurs in the range of 7.5 to 10 cycles per hour depending on the $N_{tu}$ of the wheel. We used a constant wheel speed of ten cycles per hour for the DEAC when we compared the performance of the above electric AC systems over a range of SHR values. However, the rotational speed of the wheel can be varied and, thereby, used as a control option to meet a limited range of building load-SHR values.

From the described comparison of the DEAC with the previously noted alternative electric AC systems we concluded that the DEAC was able to maintain higher EER values than any of the above electric AC systems while meeting the same total cooling load over a wide range of SHR values.

Buildings, Climates and AC Systems

To compare the DEAC performance with that of other state-of-the-art AC systems using seasonal simulations, we selected the following building applications and climates.

| **Buildings** | **Climates** |
|---|---|
| 1) 1500 sq.ft. residence | 1) Miami, FL |
| 2) 4000 sq.ft. fast food restaurant | 2) St.Louis, MO |
| 3) 10,000 sq.ft. office building | 3) Washington, DC |

The characteristics of these three buildings were developed based on extensive literature surveys of typical residential, and commercial buildings. TMY weather data for each of these climates were used for the analysis. Each building type was modeled in each climate area. However, the applicability of a given air-conditioning system to a given building/climate set was based on preliminary studies and engineering judgement. For example, electric air-reheat control was modeled with all of the electric AC systems for the fast food restaurant application in order to meet the low-SHR loads whereas it was not considered for the residential application because reheat is not used in residences. The air conditioning systems compared for each of the above building applications are listed in Table ES1.

**TABLE ES1**

| AC Systems Compared Using Seasonal Simulations | | | |
|---|---|---|---|
| **AC System** | **Residence** | **Office** | **Fast Food Restaurant** |
| Single Speed Air Conditioner (SSAC) | Yes | Yes[1] | Yes[1] |
| SSAC with Heat Pipes | Yes | Yes | Yes[1] |
| Desiccant Enhanced Air Conditioner (DEAC) | Yes | Yes | Yes[1] |
| Variable Speed Air Conditioner (VSAC) | Yes | Yes | Yes[1] |
| VSAC with Heat Pipes | Yes | No | Yes[1] |
| VSAC with Desiccant wheel | Yes | No | Yes[1] |
| SSAC with Gas-fired Desiccant Wheel | Yes | Yes | Yes |
| SSAC with liquid Kathabar Dehumidifier | No | Yes | Yes |

Note [1] Electric reheat was available for additional dehumidification.

Comparison Measure: Relative Merit Index

The purpose of an air conditioner is to provide a comfortable environment. The cost for providing such an environment is made up of the initial capital cost and the operating cost of the given systems. The market sets costs and, for all practical purposes, ASHRAE sets comfort boundaries.

Our goal was to identify which of the AC systems listed in the above Table provides an accepted level of comfort at the least cost. As the results in the simulations will show, all AC systems are not created equal! To quantify the differences we defined a measure called the "Relative Merit Index" to account for both comfort and cost.

The Relative Merit Index (RMI) is defined as:

$$RMI = \frac{C_{AAC} - C_{CAC}}{TC_{AAC} - TC_{CAC}} \qquad (1)$$

where

$C_{AAC}$ - Comfort hours provided by an "alternative" AC system over the equipment life
$C_{CAC}$ - Comfort hours provided by the "conventional" SSAC over the equipment life
$TC_{AAC}$ - Total Costs for an "alternative" AC System
$TC_{CAC}$ - Total Costs for the "conventional" SSAC

Similar concepts were used by Paul Scanlon (5) to compare several AC systems.

Comfort hours refer to the number of hours the zone conditions are maintained within the ASHRAE comfort zone. The total costs of the AC systems were determined as the sum of the capital cost and net present value (NPV) of the operating energy cost. The capital cost of the components were determined from various equipment vendors, FSEC's previous experience and TECOGEN's study of desiccant system component costing. For the NPV calculations, we used an equipment life of 10 years and a discount rate of 8 percent. Energy costs applicable for each climate zone (Appendix F) were used to determine the energy costs. Demand costs were not included in this study.

In the present case, the "alternative" air conditioners have a low sensible heat ratio relative to the "conventional" SSAC. The SSAC was taken as the baseline because it is the simplest and most widely used option for the applications considered.

Results of Buildings, Climates and AC Systems Simulations

Following the comparison study of electric AC systems described earlier in the summary, detailed seasonal simulations were performed for the air-conditioning equipment/building/climate matrix indicated in the preceding pages. We conducted annual simulations for Miami and seasonal simulations for Washington and St.Louis for all the building types. The difference is a result of the fact that Miami has a year-around air conditioning requirement whereas Washington and St.Louis have short cooling seasons ranging from April to October. The following conclusions were made based on the simulation results.

**Residence:**

When the residence AC systems are ranked using the Relative Merit Index criterion, the DEAC system provided the best combination of cost and comfort in all three cities.

**Office:**

Office buildings typically have relatively high (0.7 to 0.9) SHR loads and consequently do not require much dehumidification. Consequently, all the AC systems, including the SSAC, provided the required comfort for more than 90 percent of the cooling period for the modeled Miami office. As a result, the Relative Merit Index of the AC systems for the office building turned out to be a very small value (< 1.0). Hence, only the net present value (NPV) was used to rank the systems.

A high efficiency AC with a Kathabar dehumidifier had the highest ranking for the Miami office building. This system was simulated in the "mixed" mode. In this mode, the dehumidifier treats only ventilation air while the AC treats only recirculated air. The DEAC was the next best system for the Miami office building.

All of the AC systems could maintain comparable comfort conditions throughout the cooling season in Washington and St.Louis climates. The NPV values show that the SSAC is the best choice for these climates. Therefore, air conditioner dehumidification enhancement is not required for this building as modeled in Washington and St.louis. However, if a separate air conditioner was required to meet, for instance, conference room load requirements then the DEAC or some alternative low-SHR equipment would be viable options.

**Fast Food Restaurant:**

The fast food restaurant model treated the dining and kitchen areas as two independent zones. These two zones were conditioned with two independent AC systems. The combined (Dining and Kitchen AC systems) Relative Merit Index was used to rank the AC systems.

From the simulation study, it was concluded that the DEAC and the gas-fired hybrid system are the two best options for the fast food restaurant application. Based on the Relative Merit Index rating, the gas-fired hybrid system is the best option for the fast food restaurants in Miami and St.Louis. The DEAC with electric reheat is the best electric AC alternative for this application in these climates. The DEAC had the highest Relative Merit Index rating for the Washington fast food restaurant.

DEAC: Energy Saving Potential for Residences

In order to demonstrate the energy savings possible with the DEAC compared to a generic high-efficiency air conditioner (SEER of around 12.0) it is necessary to define a common playing field. One approach is the Relative Merit Index discussed above. Alternatively, we can take a more direct approach by forcing the competing air conditioning systems to maintain comfort conditions within the ASHRAE comfort zone. We did this in the same way that homeowners try to do it; by manipulating the thermostat. Doing so makes it possible to approach a realistic energy use comparison.

Simulations were initially run for the modeled residence in Miami using the high-efficiency air conditioner and then a DEAC system. Thermostat settings were changed as required each hour to remain in the comfort zone. These simulation results showed that, if comparable comfort conditions were maintained, the DEAC would use 10 percent less energy than the high-efficiency air conditioner. In addition lower humidity would be maintained which would result in healthier conditions (6).

Projected Nationwide Energy Savings with DEAC for Residential Sector

In order to quantify the potential energy saving resulting from use of the DEAC to maintain comfort conditions, it was first necessary to define its region of applicability. The area of DEAC residential applicability was selected based on residential simulation results for ten U.S. cities having a wide range of annual latent loads. These results suggest that the Southeast/Gulf coast and Hawaii regions represent the primary residential market area. The applicable geographical area is shown in Figure ES1.

The projected energy savings for the estimated area of residential DEAC users based on unit energy savings of five to ten percent are 3.25 to 6.5 x $10^6$ Mega Watt hours per year. When translated into primary energy (fuel) with a conversion efficiency of 33 percent, this equates to 9.8 to 19.7 x $10^6$ Mega Watt Hours or about 33 to 67 x $10^6$ Million BTU per year.

Run-around Belt Configuration of DEAC

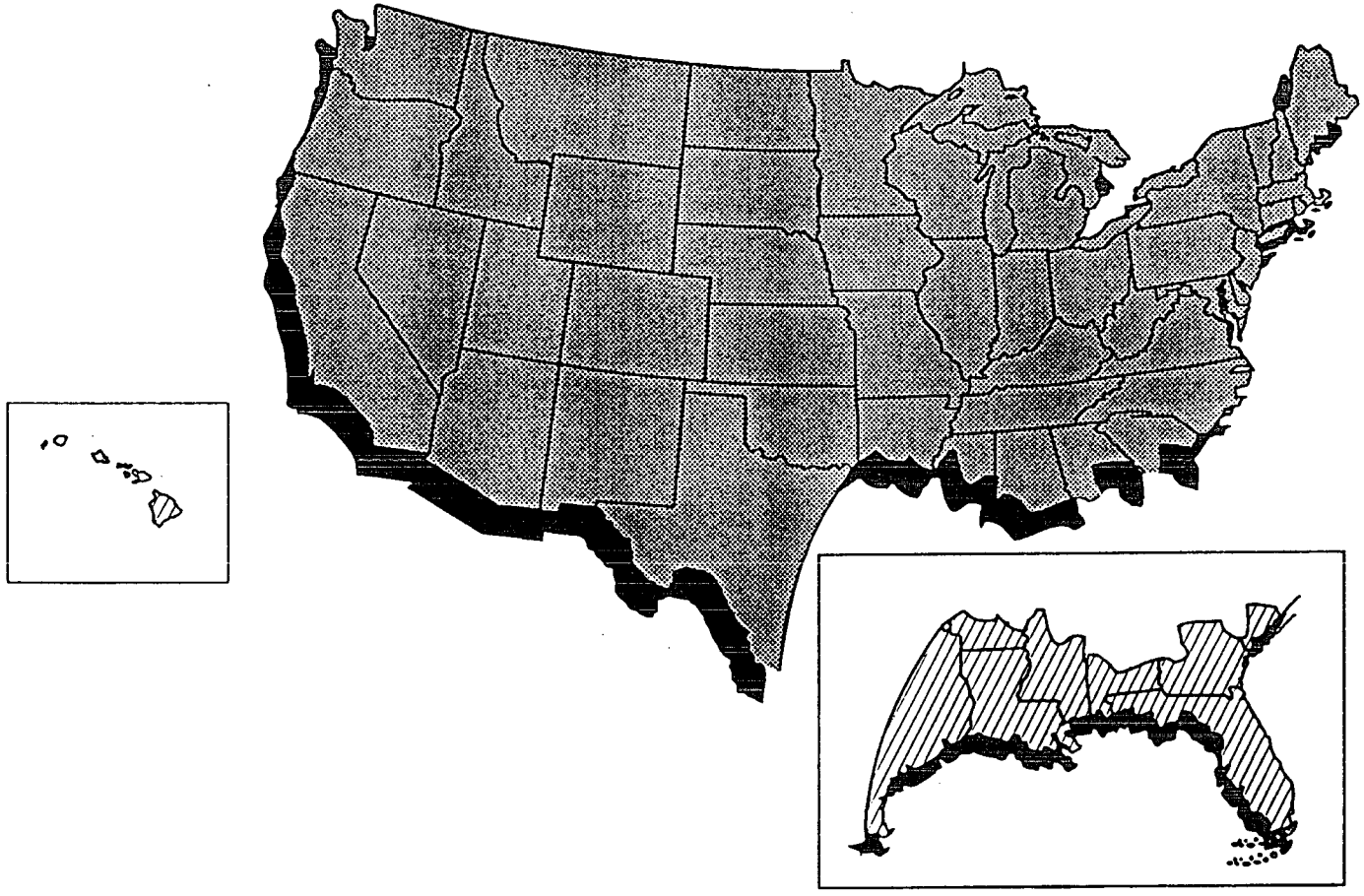
Throughout the main body of this study the configuration of the DEAC system encompassed a silica gel desiccant wheel. In practice, integration of the desiccant wheel into an air conditioner would typically require changing the conventional ducting design. The run-around belt configuration would allow for a simpler and more conventional ducting system. We also looked at the belt option because it is compatible with a number of hydrophilic low cost materials. Two such belt materials were modeled: activated carbon mat and jute.

The heat transfer and pressure drop characteristics of the deployed materials were determined. Primarily because of the increased pressure drop associated with the belts (relative to the wheel configuration), the EER of the belt model was approximately 12 percent lower than that of the wheel model for an SHR of 0.59. Although we did not do a cost/benefit analysis of the belts, their low cost may compensate for the reduced performance.

Conclusion

The parametric study and the seasonal simulation results indicate that the DEAC process is feasible and holds promise for maintaining a healthy and comfortable environment at a lower cost for residential and fast food restaurant applications. In addition, the DEAC can save energy compared to current high-efficiency air conditioners if both systems are required to maintain the ASHRAE recommended comfort levels.

Other potential DEAC applications include supermarkets, libraries, museums, computer room facilities and buildings where humidity control is essential or better humidity control would enhance productivity. Further investigation is required to identify the minimum cost/high performance solid/liquid desiccant materials, the life time of such high performance materials and additional energy efficiency improvements possible with the DEAC.



Applicable zones include south-east U.S. and Hawaii

Figure ES1 - Primary Region of Residential Applicability of the DEAC

References

ES 1. Cromer, C.J. 1988a. Dehumidification enhancement of air conditioners by desiccant moisture exchange. Report # FC-CR-CR-198-88 to U.S. Department of Energy.

ES 2. Cromer. C.J. 1988b. United States Patent Number 4,719,761

ES 3. Kerestecioglu, A., Swami, M., Brahma, P., Gu, L., Fairey, P., Chandra, S. "Users Manual: Florida Software for Enervironment (Energy and Environment) Computation", Florida Solar Energy center. August 1989.

ES 4. DOE. 1982. DOE-2 engineers manual. LBL-11353. Berkeley, CA: Lawrence Berkeley Laboratory.

ES 5. Paul Scanlon, P.E., Heating, Piping Air Conditioning: Economic Impact of HVAC System Selection, September 1992.

ES 6. Sterling, E.M., A. Arundel, and T.D. Sterling, 1985, "Criteria for human exposure to humidity in occupied buildings", ASHRAE Transactions 91(1B).

## 1.0 INTRODUCTION

Buildings in the United States used approximately 30 quadrillion Btu of energy in 1989 at a cost of 200 billion dollars. In addition, they accounted for 36 percent of the total U.S. consumption of primary energy (1)*. HVAC systems use substantial portions of this energy in an effort to provide healthy and comfortable building environments.

There is an increasing awareness that the control of humidity is an integral element in maintaining a healthy indoor environment. In the rapidly growing south-eastern United States, an area of relatively high humidity, maintaining a low humidity building environment can be difficult to accomplish in an energy efficient manner. There is a need for energy efficient cooling based dehumidification alternatives which will meet low Sensible Heat Ratio (SHR) building or process loads. Extensive research is being conducted by the Electric Power Research Institute (EPRI), Gas Research Institute (GRI) and Department of Energy (DOE) to develop cost effective dehumidification technologies.

Air-conditioning systems currently available to maintain indoor humidity levels include:

a. Single-speed air conditioners (SSAC)
b. Heat pipe-assisted air conditioners
c. Air-reheat systems
d. Thermal storage systems with cold air distribution
e. Dual path air-conditioning systems
f. Variable-speed air conditioners (VSAC)
g. Gas-fired solid and liquid desiccant systems.

A relatively recent concept, the desiccant enhanced air conditioner (DEAC) dehumidification option appears to offer potential for increased comfort at lower costs. The DEAC process has been patented by C. Cromer of the Florida Solar Energy Center (FSEC). A detailed study of the DEAC system and its comparison to alternative options form the basis of this report.

### 1.1 DEAC Process

The DEAC uses a desiccant wheel to pre-cool and humidify the return air before it enters a cooling coil. It then heats and dehumidifies the supply air after it leaves the coil. A unique feature of this process is that regeneration of the desiccant component is accomplished by the return air rather than an external heat source. The result of the DEAC psychrometric process is increased dehumidification of the supply air. This process can provide high dehumidification capacity while maintaining a relatively high energy efficiency ratio (EER). It also has the potential to maintain lower absolute humidity than any of the electric dehumidification concepts.

---

* Note: Number in parenthesis refer to references listed at the end of each Chapter

A schematic of the DEAC process is shown in Figure 1.1. Return air [1] flows through the regeneration side of the rotating desiccant wheel before it enters the cooling coil [2]. The air leaving the cooling coil [3] passes through the process side of the desiccant wheel [4] and exits via the supply fan [5] to the conditioned zone. For this configuration, the desiccant wheel would be enclosed in a cassette with seals between the regeneration and process sides. A small electric motor is required to rotate the wheel.

## 1.2 Goals of Study

The goals of this study were:

- Compare DEAC EER to alternative electric systems over a range of SHR values
- Compare DEAC performance to that of electric and gas dehumidification systems when operating in three building types (residence, office and fast food restaurant) in three climates (Miami, St.Louis and Washington D.C.)
- Identify the DEAC primary geographical region of applicability
- Estimate for the primary region of applicability, the potential residential energy saving of the DEAC relative to the current high-efficiency air conditioner for the case when both meet the same standard of comfort
- Provide an analysis of a run-around belt option as an alternative to the desiccant wheel for use in the DEAC.

The following Chapter discusses the approach used for this study.


Reference

1. Ren Anderson, NREL Building Energy Technology Program, Fourth Quarterly Report, FY 1992

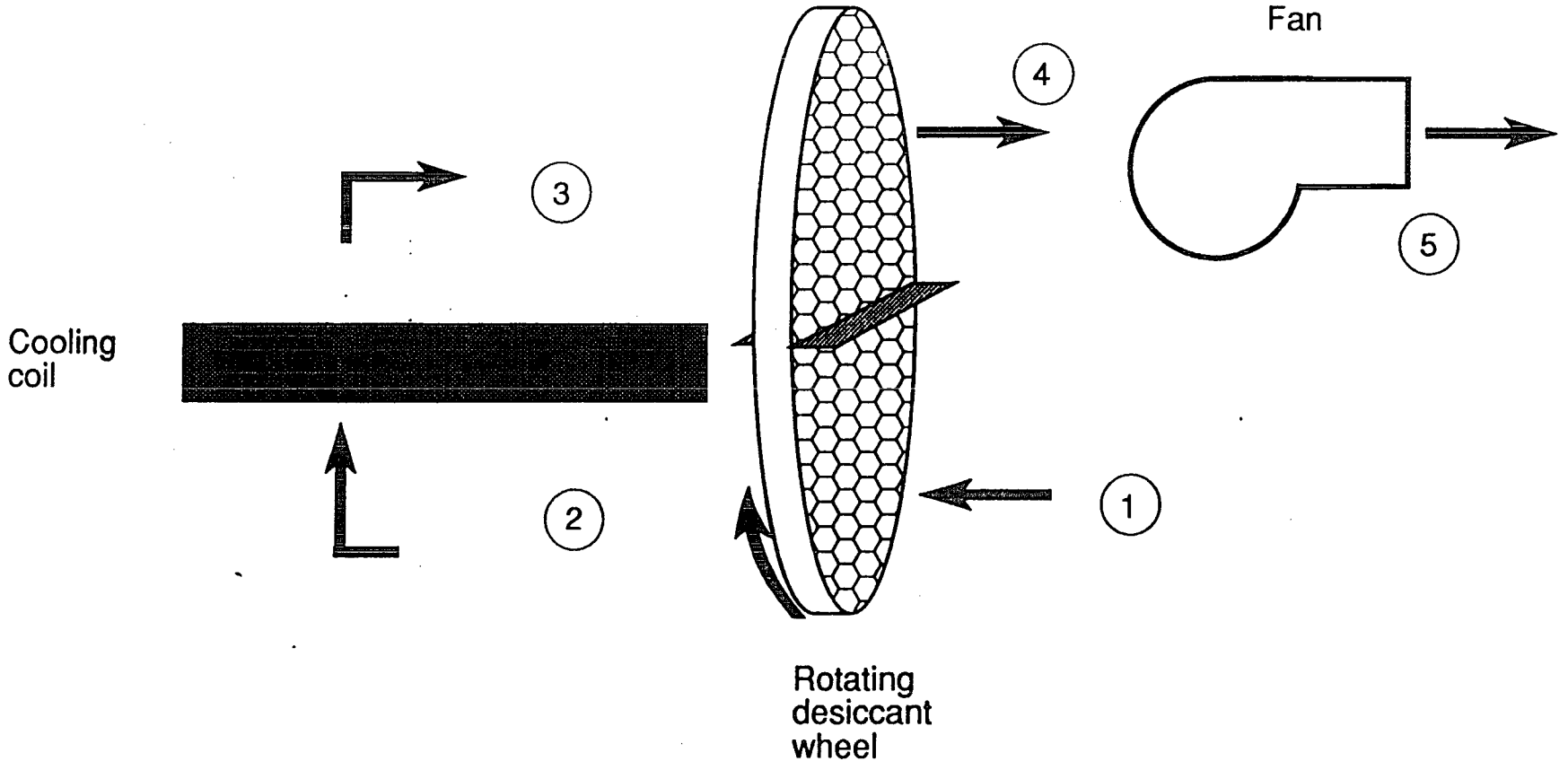



Figure 1.1 - Schematic Diagram of DEAC

## 2.0 APPROACH TO THE STUDY

### 2.1 Overview

The objective of this study was to quantify the potential of the DEAC for use in commercial and residential applications by comparing it to a variety of alternative air conditioning (AC) systems. In order to make this comparison, it was necessary to first model the various AC systems then and integrate these models with a building software program. We selected Florida Software for Energy and Environment Computation, version 2.2 (FSEC 2.2) as the building simulation code. The primary basis for the selection was the code's ability to treat moisture transport and storage. With the combination of the AC models and FSEC 2.2 code, it was possible to compare the various AC systems in a variety of building types and climates. The primary tasks performed were:

a. Develop component performance maps
b. Develop AC system modules
c. Compare performance of selected electric AC systems over a range of energy efficiency and sensible heat ratios
d. Develop building characteristics
e. Select appropriate baseline AC characteristics for each building application
f. Size AC systems for each building/climate matrix
g. Perform annual or seasonal simulations for AC system/building/climate matrix and interpret results.
h. Analyze run-around belt configuration for use in the DEAC (Appendix B)

Each of these tasks is described in more detail in the following sections.

### 2.2 Development of Component Performance Maps

Performance maps were developed either by running detailed analytical or numerical models over a wide range of operating conditions (e.g., Oak Ridge National Laboratories heat pump code (1)) or by curve fitting vendor performance data (e.g., heat pipe model (2)). In some cases, portions of existing software codes were used. Since the building program calls each component model a few thousand times during a seasonal simulation, curve-fits for some of the analytical and numerical models were also used to save computer run-time. The development of component models is described in detail in Chapter 4.

### 2.3 Development of AC System Modules

The AC systems compared (e.g., AC with heat pipes, DEAC, etc.) were combined into three major subroutines for simulation purposes. These subroutines model the performance of the following AC systems.

1. Single Speed Air Conditioner (SSAC) with/without heat pipes or desiccant wheel and/or reheat
2. Variable Speed Air Conditioner (VSAC) with/without heat pipes or desiccant wheel and/or reheat
3. Gas-fired solid and liquid desiccant systems

Each of the above subroutines was developed by integrating the performance maps of various components, including appropriate controls. Further details related to assembling these three modules are discussed in Section 6.5.

## 2.4 Parametric Comparison of Electric AC Systems

Following the completion of software code development and testing, a parametric study was conducted to compare the energy efficiency ratings of the DEAC, SSAC with and without electric reheat and heat pipes and VSAC over a range of SHR values (0.4 to 0.9). One of the major conclusions from this parametric study was that the DEAC would maintain higher EER values than any of the other systems while meeting the same total cooling load over a wide range of SHR values. This comparison study is discussed in detail in Chapter 5.

## 2.5 Development of Building Characteristics

A residence, a fast food restaurant and a medium size office building were the three building types for which the DEAC was compared with alternative air conditioning options. The characteristics of each building were developed based on extensive literature surveys of typical residential, and commercial buildings. The characteristics of the modeled buildings and the literature sources are discussed in Chapter 7.

## 2.6 AC System Component Sizing

To perform the annual simulations, it was necessary to first estimate AC equipment sizes. The component sizes were determined based on the results of simulation runs using a hypothetical or "ideal" air conditioner which perfectly matches both the sensible and the latent hourly loads. The ideal AC simulations were performed for each building/climate matrix. These ideal simulation results provided the basis for design loads and sensible heat ratio profiles required to size the AC systems. The sizing philosophy for each of the AC system components is discussed in Chapter 8.

## 2.7 Baseline AC Characteristics

Performance characteristics of alternative AC systems were compared with those of a typical single speed air conditioner (SSAC) for each building application. The baseline SSAC characteristics were specified based on the building load-SHR requirement. The load-SHR profiles were predicted by the ideal AC simulations.

The modeled residence had average annual load-SHR values ranging from 0.65 to 0.7 for the three climates. AC systems with SHR values in the range of 0.7 to 0.75 are readily available in the market. Hence, a medium-SHR air conditioner with an SHR of 0.73 at ARI rated conditions was selected as the baseline AC for residential application.

The medium size office building modeled had average load-SHR values ranging from 0.8 to 0.9 for the three climates. The relatively high SHR is due to high sensible loads resulting in part, from electronic equipment and lights used in the building. An air conditioner model with an SHR of 0.84 at ARI conditions was used in our study for this building application.

Fast food restaurants have high make up air requirements for odor control and to meet ventilation air requirements of a relatively high density of people. As a result, the latent cooling loads are higher than the sensible cooling loads when a fast food restaurant is located in a humid climate. For the restaurant, AC systems with desiccant components (DEAC and gas-fired hybrid) were modeled using a high-SHR (0.84) air conditioner and all the other AC systems were based on a medium-SHR (0.73) air conditioner.

The major portion of the latent loads is met by the desiccant wheel component for either of the AC/desiccant systems. With both of the AC/desiccant systems, the high-SHR air conditioner is used primarily to meet the sensible loads. Currently available high-SHR air conditioners typically have high efficiencies. Due to this advantage, the AC/desiccant systems with high-SHR air conditioners maintain higher EER values while providing the required dehumidification.

## 2.8 Seasonal Simulations

The three climates selected for this study were Miami, Washington and St.Louis. Miami represents a humid south-east coastal climate with year-around air-conditioning requirements. Washington represents a mid-Atlantic climate with a cooling season between April and October. St.Louis also has relatively shorter cooling season between April and October but the climate is more humid than that of Washington. Therefore, annual simulations for all the three building applications were carried out for Miami and seasonal simulations were carried out for Washington and St.Louis.

Preliminary simulations were first performed using the AC system component sizes estimated from the "ideal" AC procedure explained previously in Section 2.6. The component sizes were then revised based on simulation runs using a 98 percent design point. Following this, further simulations were run using the updated component sizes. The results of these simulations are presented and interpreted in Chapters 9, 10 and 11 for the residence, fast food restaurant and office building applications respectively.

References:

1. C. K. Rice, The ORNL modulating heat pump design tool - user's guide, June 1991.

2. Communications with QDT Supermarket Air-Conditioning Systems, QDT Ltd. Dallas.

## 3.0 AC SYSTEMS REVIEW

A variety of approaches to control building load sensible heat ratios (SHR) are presented in Sections 3.1 to 3.6. For each system discussed, the psychrometric path is described in detail in this section. This information is directly related to the development of model components discussed in Chapter 4.

### 3.1 Conventional Single-Speed AC

Conventional single-speed air conditioners (SSAC) are the most common systems in residences and commercial buildings. They are controlled by dry-bulb temperature sensing thermostats and are typically sized based on the sensible load. Dehumidification, a by-product of the sensible cooling process, is typically 15 to 25 percent of the total capacity. The psychrometric process for a conventional AC is shown in Figure 3.1. The addition of heat to the supply side air ("reheat") provides humidity control but at increased operating and/or capital costs. The psychrometric process for an air-reheat system is shown in Figure 3.2.

### 3.2 Single-Speed AC with Heat Pipes

Single-speed air conditioners have increased dehumidification capacity when equipped with heat pipes. Heat pipes pre-cool the return air before it contacts the evaporator and they heat the supply air which exits the cooling coil. Figure 3.3 shows typical psychrometric state points for heat pipe-assisted air conditioners. As shown in the figure, the return air dry bulb temperature is lowered from state 1 to state 2. The supply air dry bulb temperature is increased from state 3 to 4. The evaporator coil cooling now takes place between state points 2 and 3.

Alternate heat-exchanger systems, for instance run-around coils and plenum heat exchangers, result in similar psychrometric processes. Pre-cooling of the air before it reaches the cooling coil slightly reduces the ADP of the coil thereby increasing dehumidification capacity. Typically heat pipes are always turned "ON" and do not provide independent variable control of humidity. In theory (and occasionally in practice) it is possible to have step wise humidity control by "switching off" one or more rows of the heat pipes. Flexibility could also be achieved by using dampers to bypass the heat pipes to provide increased sensible and reduced latent capacity.

### 3.3 Desiccant Enhanced Air Conditioner

The DEAC system combines an air conditioner with a desiccant wheel to provide increased latent capacity. Figure 3.4 presents the DEAC psychrometric process and displays typical psychrometric state points when a desiccant wheel is used to better match the AC system SHR with that of the load. This improvement is accomplished by moisture and heat transfer in the regeneration and process air streams of the desiccant wheel.

Starting at state 1 of Figure 3.4, the return air is humidified and cooled as it flows through and regenerates the wheel desiccant material. These changes in the air properties result in a process which proceeds approximately along a line of constant enthalpy towards state 2. The

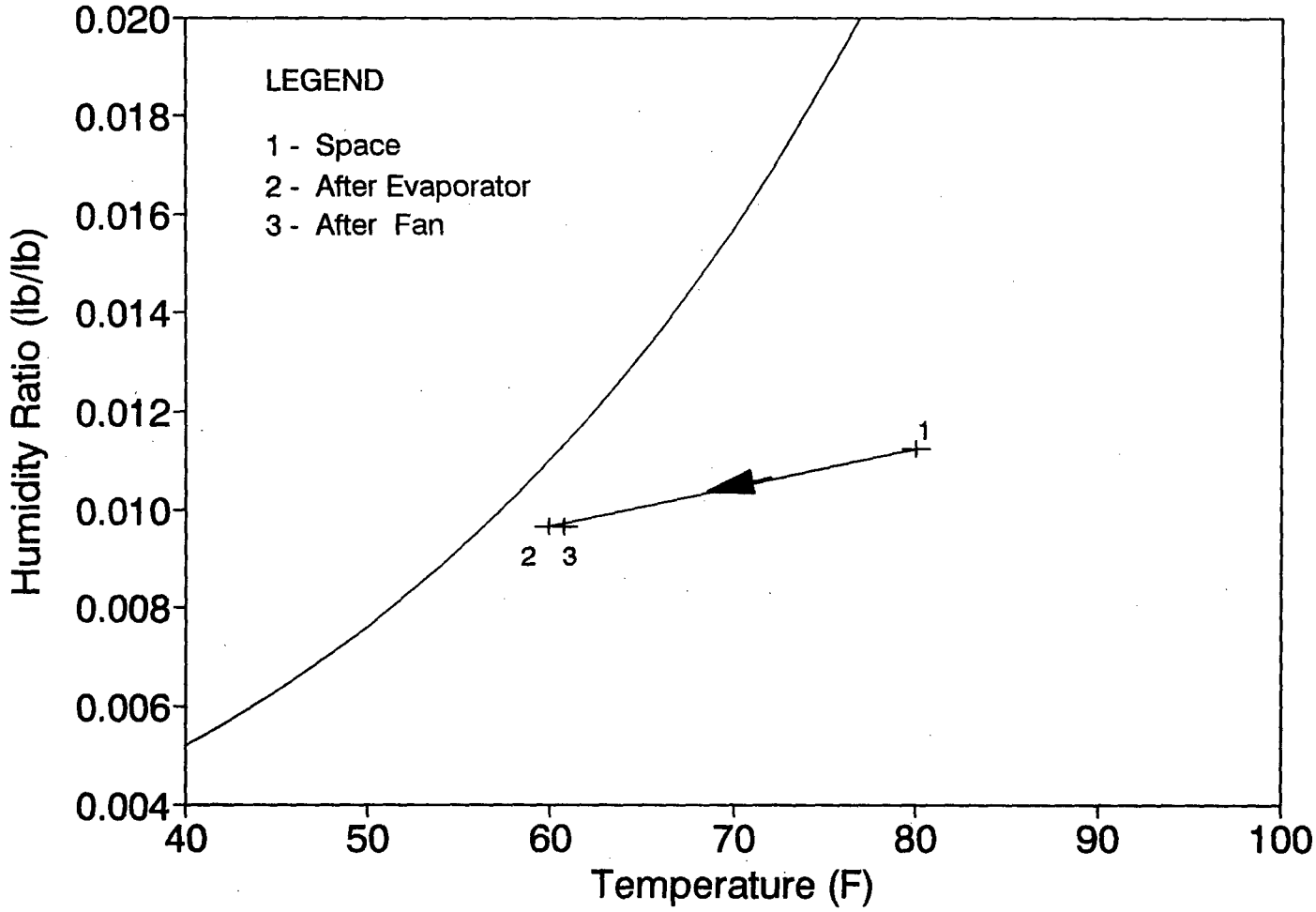



Figure 3.1 - Psychrometric Process for Conventional AC

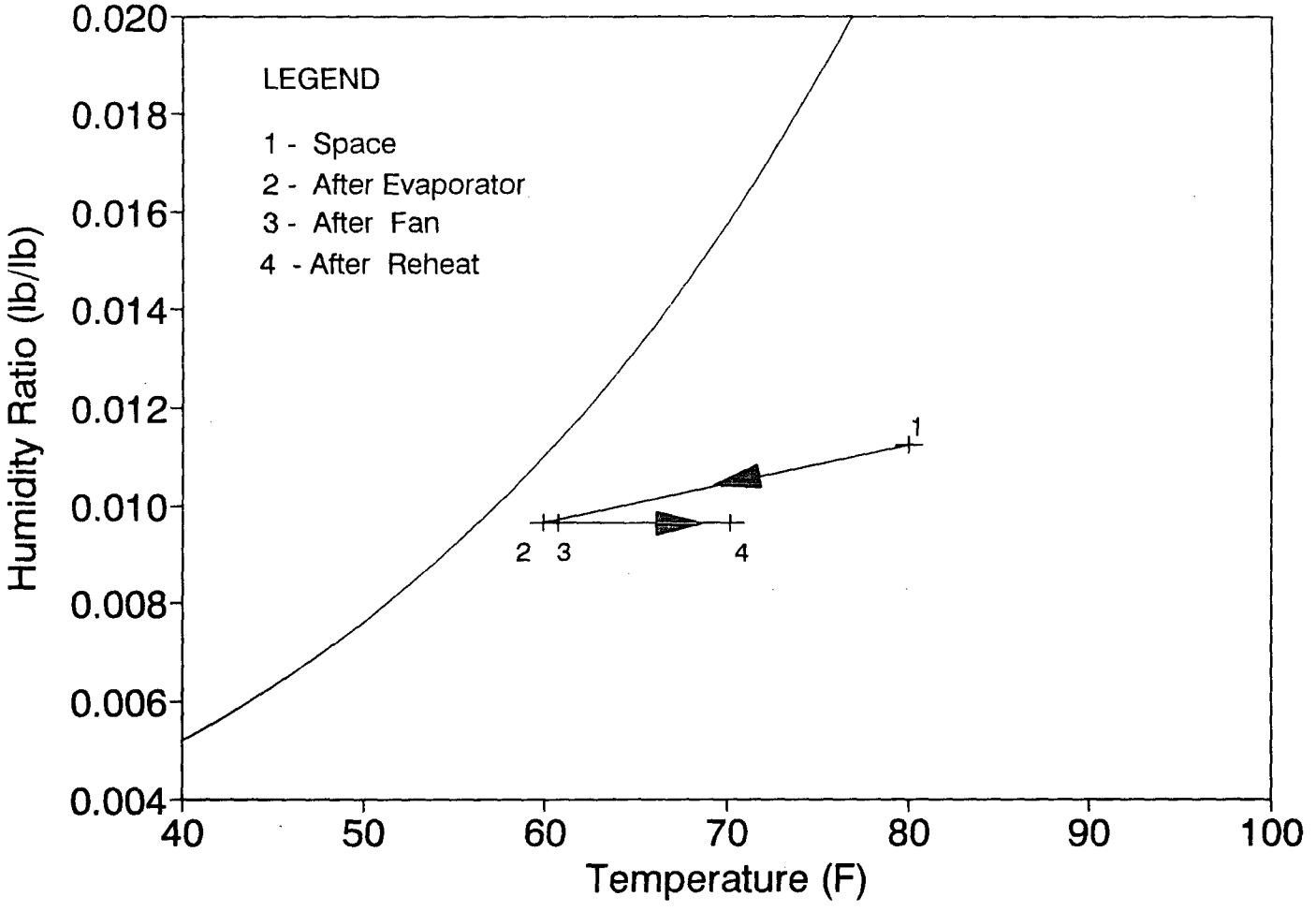



Figure 3.2 - Psychrometric Process for Air-Reheat System

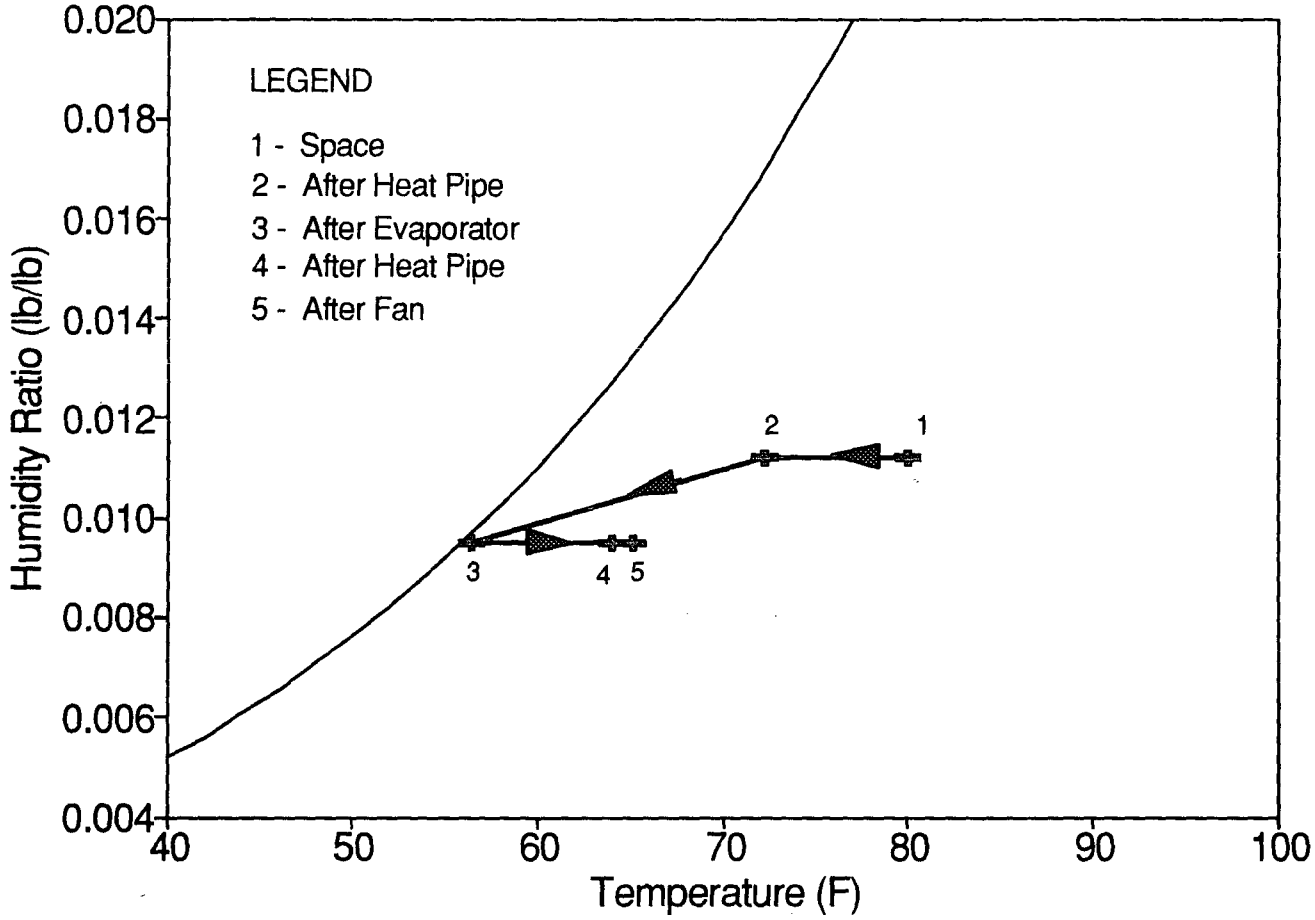



Figure 3.3 - Psychrometric Process for Heat Pipe-Assisted AC

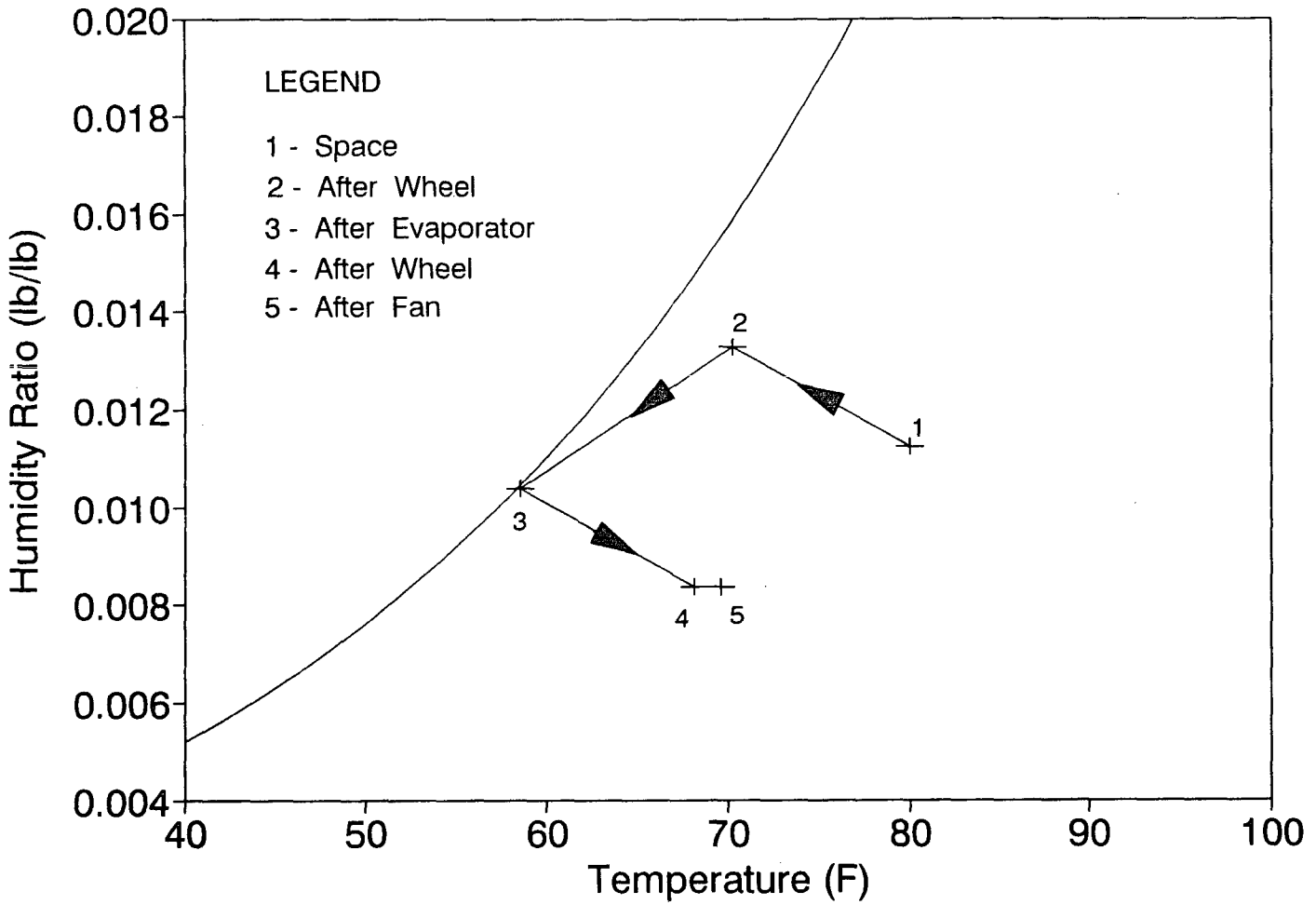



Figure 3.4 - Psychrometric Process for DEAC

return air then comes into contact with the cooling coil, process 2 to 3, where the entering decreased air temperature results in decreased sensible capacity of the coil and the higher humidity results in increased latent capacity. Stated another way, the potential for sensible heat transfer has been reduced while the potential for moisture transfer has been increased. The air exiting the coil at state 3 enters the process side of the desiccant wheel and is heated and dried as it passes through to state 4. The small increase in supply air sensible heat between states 4 and 5 is due to inefficiencies associated with the fan unit.

One advantage of the desiccant approach over a sensible heat exchanger is a small elevation of ADP resulting from the moisture transfer process. This allows maintaining a slightly higher efficiency of the air conditioner while providing better dehumidification.

### 3.4 Variable-Speed AC with Heat Pipes or Desiccant Wheel

Variable-speed air conditioners (VSAC) have become popular as the cost of variable speed motor controls and electronics have decreased. VSAC's can independently meet a broad range of sensible and latent loads by modulating compressor frequency, indoor and outdoor fan frequency, compressor inlet super heat, or condenser sub cooling. Seasonal energy efficiency ratios (SEER) of variable-speed air conditioners are typically higher than those of single-speed air conditioners because VSACs use lower compressor and fan speeds during part-load conditions.

To further increase VSAC dehumidification, heat pipes or desiccant wheels can be added. A VSAC with heat pipes can take advantage of the fact that as air flow rate over the evaporator decreases, heat pipe effectiveness increases. Similarly, when a desiccant wheel is added to a VSAC, reduction in air flow rates enhances the capability of the system to meet low SHR loads.

### 3.5 AC with Gas-fired Desiccant Wheel

This hybrid system is a version of a gas heat regenerated desiccant wheel operating with a sensible heat exchanger wheel and a high-efficiency and high-SHR air conditioner. The system schematic is shown in Figure 3.5. This system may be operated in either a "recirculation" or "ventilation" mode. In the recirculation mode, the indoor air is dried and heated by a desiccant wheel and cooled by regeneration air with the aid of a sensible heat exchanger. The warm, dry air at low dew point temperature is cooled by a high-efficiency air conditioner.

In the "ventilation mode" the ventilation air is dried and heated by a desiccant wheel and subsequently cooled by the AC. Compared to the recirculation mode this process is more efficient and requires a smaller dehumidifier system due to the higher humidity potential difference. The regeneration air comes from the conditioned zone in the "ventilation mode". The psychrometric process of this system is shown in Figure 3.6 for the recirculation mode.

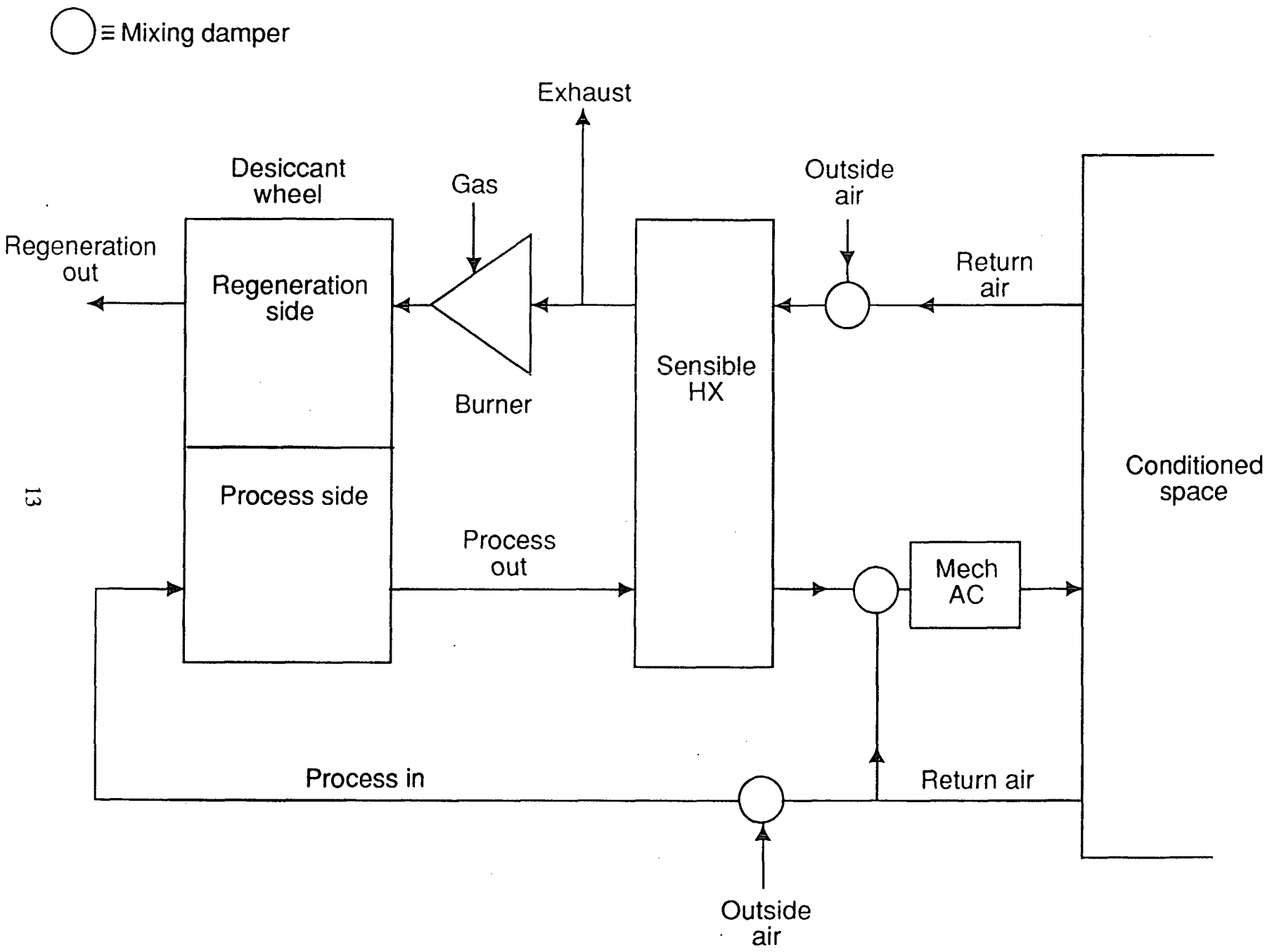



Figure 3.5 - Schematic Diagram of Gas-fired Hybrid System

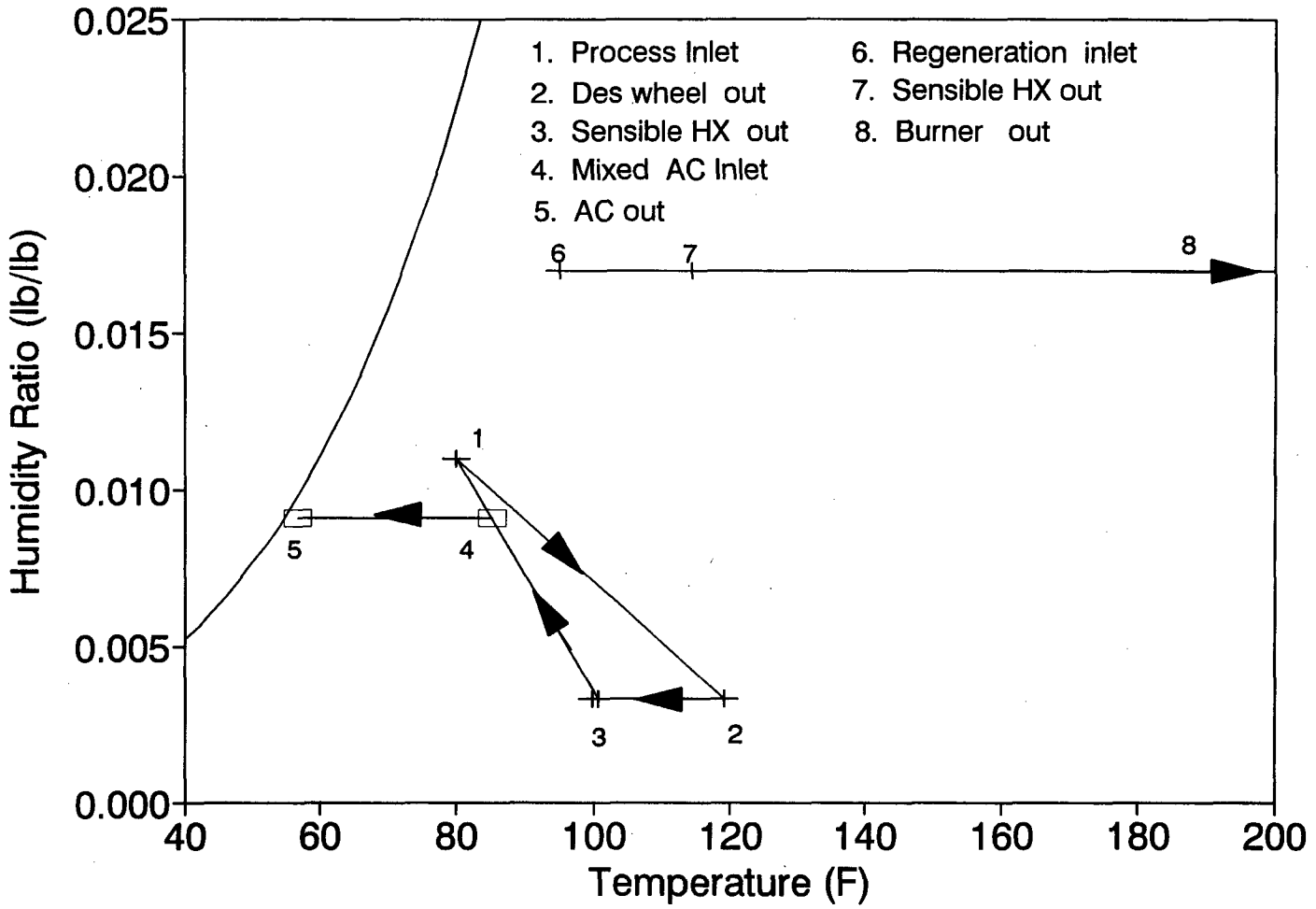



Figure 3.6 - Psychrometric Process for Gas-fired Hybrid System

### 3.6 AC with Kathabar Dehumidifier

This system is based on the concept used in the commercial Kathabar liquid-desiccant unit. Dehumidification occurs as the process air comes in contact with the strong liquid desiccant. The heat of absorption is removed by indirect evaporative coolers. The liquid desiccant is then regenerated by heating the liquid in a gas boiler and passing it over an outdoor packed bed tower. Relatively dry ambient air in contact with the liquid absorbs the moisture and the liquid-desiccant is regenerated. The process air is then mixed with the return air and cooled by a high-efficiency air conditioner.

This system is normally operated in the "mixed mode" where the process air and the regeneration air come from outdoors. A schematic of the AC with Kathabar dehumidifier is shown in Figure 3.7 and the psychrometric process is shown in Figure 3.8 for the "mixed mode".

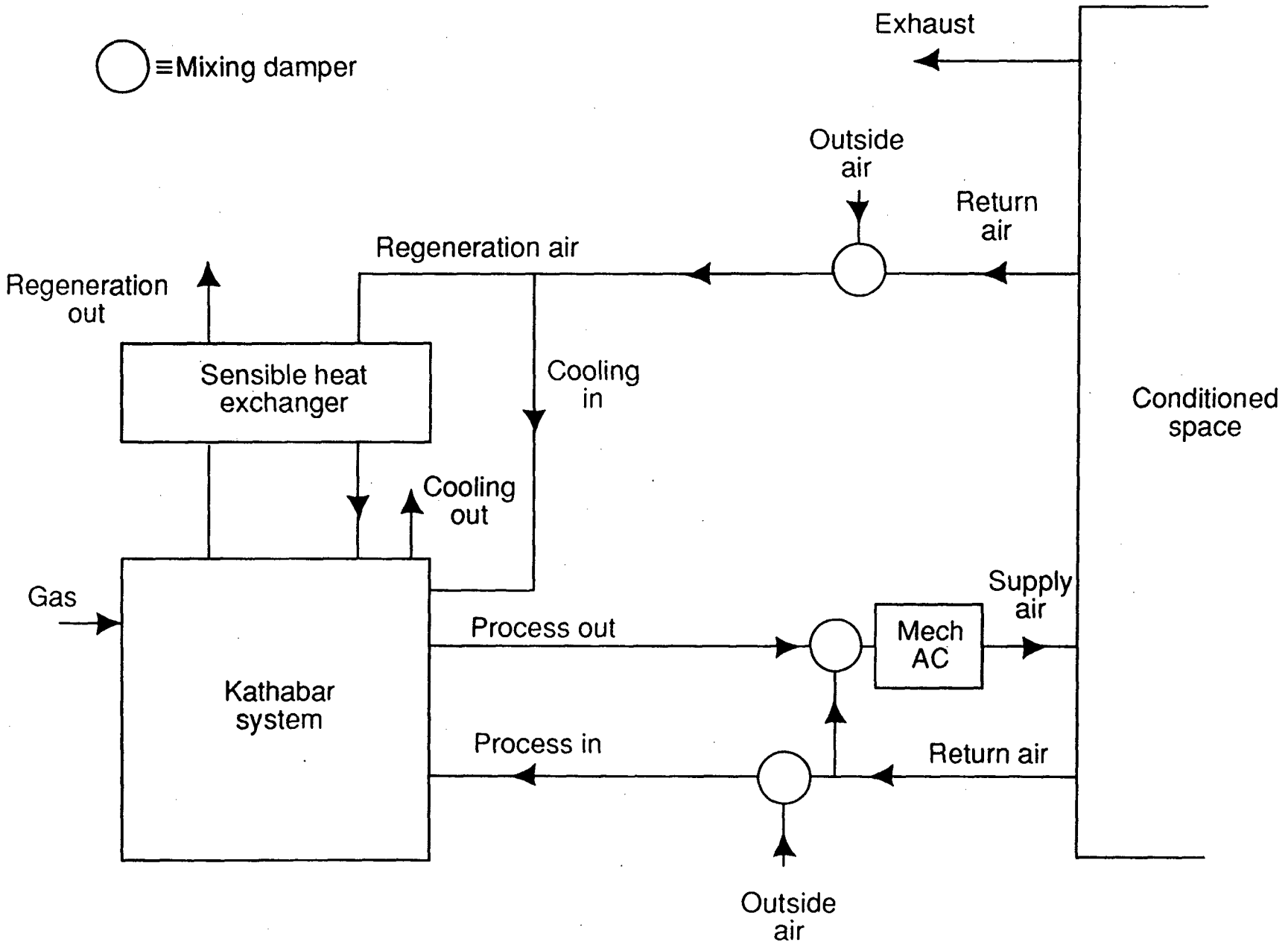



Figure 3.7 - Schematic Diagram of AC with Kathabar Dehumidifier

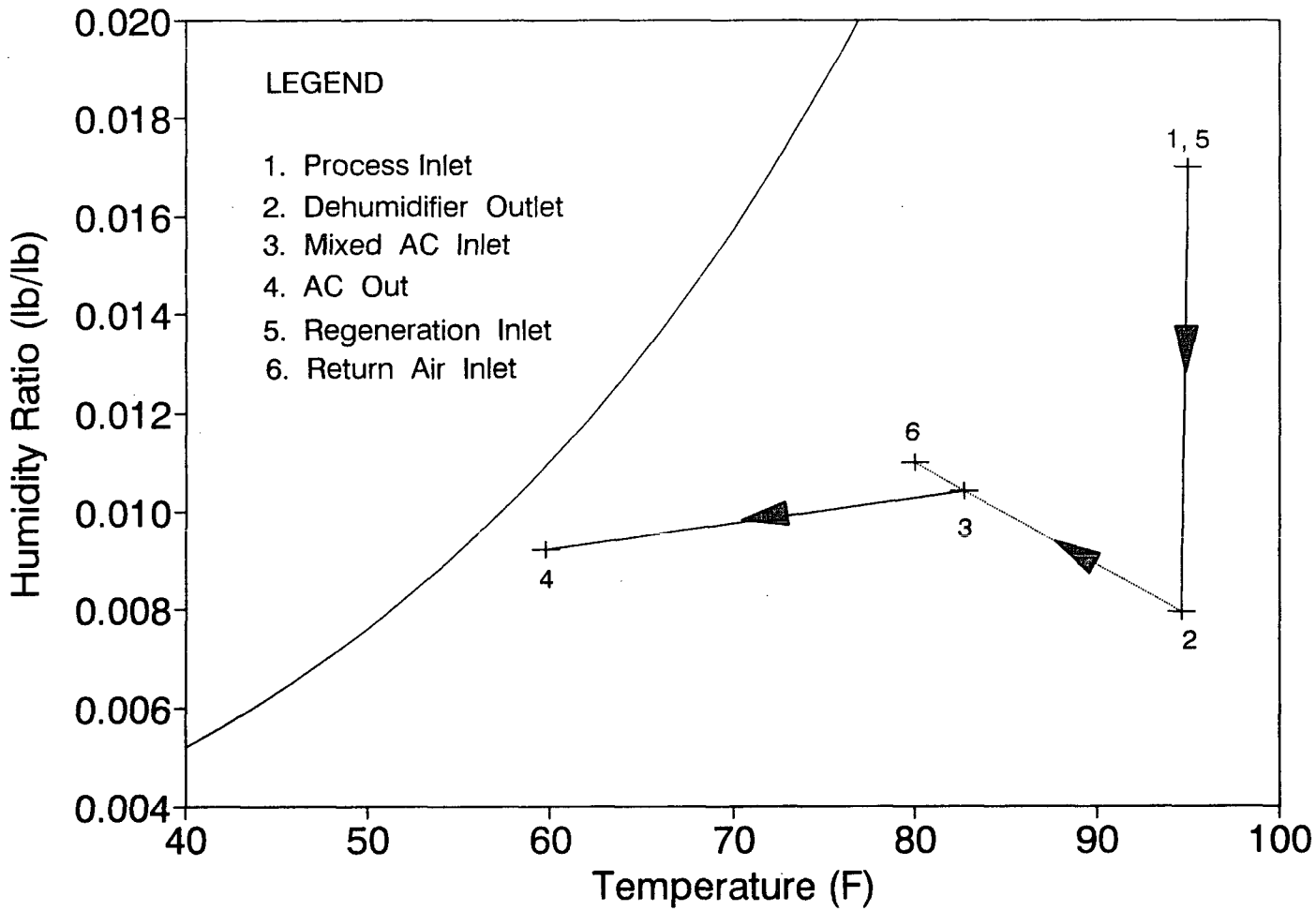



Figure 3.8 - Psychrometric Process for AC with Kathabar Dehumidifier

## 4.0 SOFTWARE MODEL DEVELOPMENT OF AC COMPONENTS

### 4.1 Overview

This Chapter explains the methodology used in modeling the AC components and systems. Component performance maps were developed either by running detailed analytical or numerical models over a wide range of operating conditions or by curve-fitting vendor performance data. Table 4.1 lists the basic systems and components which make up the equipment matrix for this study.

**TABLE 4.1**

| Basic AC Systems and Components |
|---|
| Electric AC models:<br><br>1. Oak Ridge National Laboratories (ORNL) heat pump<br>2. DOE-2 packaged air conditioner<br>3. Variable-speed air conditioner<br>(Based on ORNL heat pump model) |
| Electric AC system component models:<br><br>1. Heat Pipes<br>2. Silica gel desiccant wheel<br>3. Forward-curved fan<br>4. Variable-speed fan |
| Gas-fired system component models:<br><br>1. Silica gel desiccant wheel<br>2. Kathabar liquid-desiccant dehumidifier |

### 4.2 ORNL Heat Pump

The ORNL heat pump model, originally developed for studying component design alternatives, was used in the present study to predict overall operational performance of air-to-air heat pumps. Miller (1) compared performance results predicted by the ORNL model (Mark III version) with experimental results over a range of compressor and supply-air fan speeds. The experimental results for coefficient of performance (COP) and capacity of the air conditioner were closely predicted by the ORNL model as noted in reference (1).

In our study, the ORNL Mark III model was used to develop curve-fits of a single-speed air conditioner over a wide range of operating conditions. The performance of an AC is determined from various design parameters (e.g., compressor type, heat exchanger size, expansion device type, air flow rates, indoor and outdoor conditions, etc.,). We used a

reciprocating compressor performance map of a 3 ton AC in our model. Refrigerant subcooling of 14°F was specified at the condenser outlet for this AC model to simulate expansion device performance. The range of environmental conditions used for developing the curve-fits are shown in Table 4.2.

**TABLE 4.2**

| Range of conditions specified for Single-Speed AC | |
|---|---|
| Outdoor temperature: | 76 to 110°F |
| Indoor temperature: | 55 to 95°F |
| Indoor relative humidity: | 5 to 95% RH |
| Air flow rates: | 300 to 600 cfm/ton |

The nominal performance of the ORNL AC is shown in Table 4.3.

**TABLE 4.3**

| Nominal performance of Single-Speed AC | |
|---|---|
| SEER Predicted: | 9.7 BTU/Wh |
| At ARI rated point[a]: | |
| Capacity: | 37.25 kBtu/h |
| SHR: | 0.73 |
| EER: | 9.30 Btu/Wh |
| [a] Outdoor: 95°F, Indoor: 80°F dry bulb and 67°F wet bulb, Air flow rate: 400 cfm/ton | |
| Note: The capacity, EER and SHR include supply air fan power | |

Figure 4.1 and 4.2 show respectively the cooling capacity, EER values as a function of indoor temperature and humidity at constant outdoor temperature and flow rate. These figures compare the curve-fits to the actual output from the ORNL heat pump model. The wet and dry coil conditions are also identified in the plot.

### 4.3 DOE-2 Packaged AC

The EER of the ORNL AC (9.3 Btu/Wh) is typical of older, but still operating, units. To simulate the performance of a high efficiency air conditioner with high SHR, another AC model was used. This model combines DOE-2 packaged air conditioner default curves for AC total capacity and power use with bypass-factor (BF) and apparatus dew point relationships (2). Air conditioner performance can be simulated with minimal information using this model; AC capacity, EER and SHR at rated conditions are the only parameters required to predict the

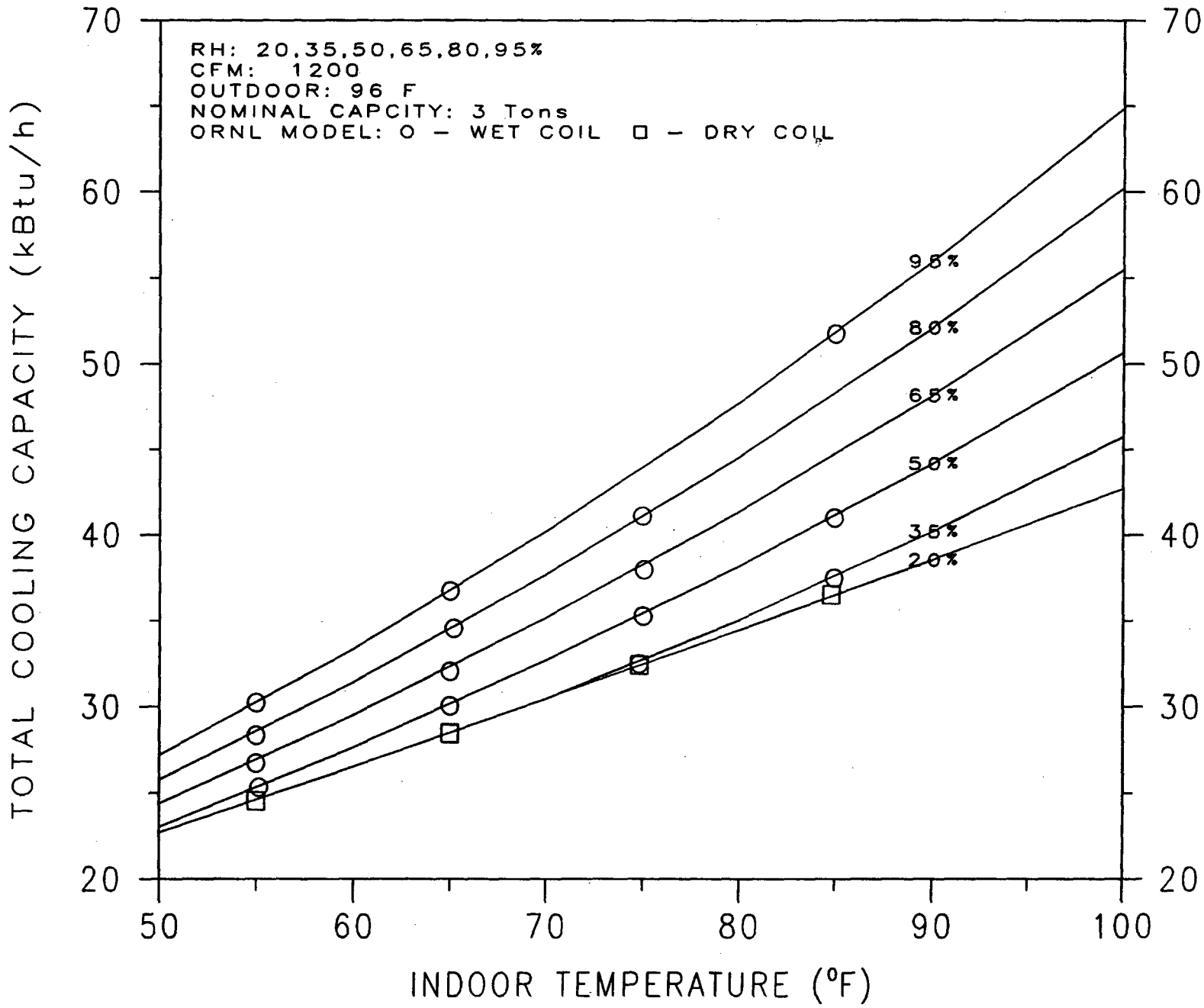



Figure 4.1 - ORNL AC: Capacity Versus Temperature and Humidity

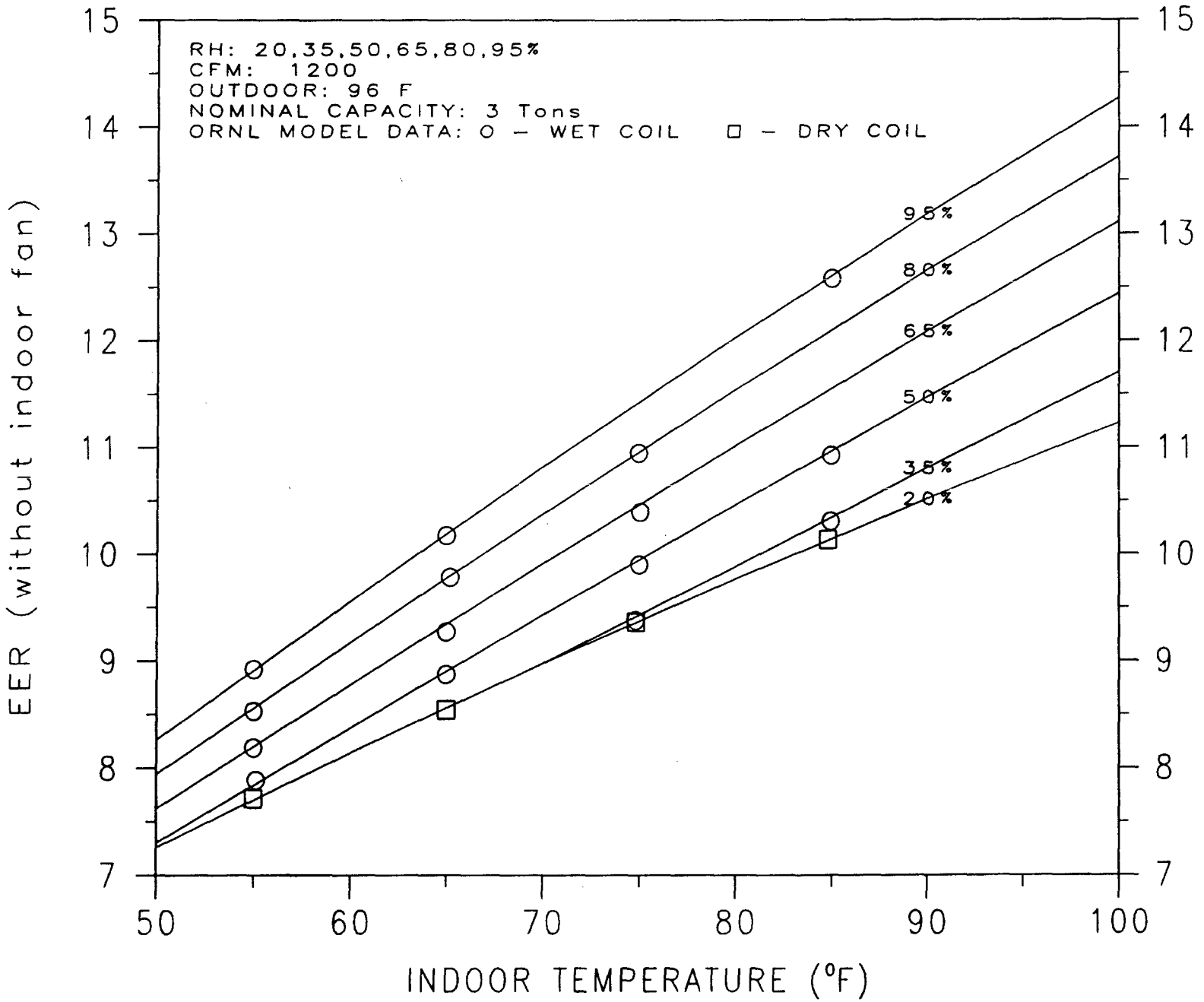



Figure 4.2 - ORNL AC: EER Versus Temperature and Humidity

performance at off-rated conditions. The performance map of the DOE-2 AC model is shown in Figure 4.3. Contours of constant SHR and EER for the AC unit are plotted on the psychrometric chart for a range of evaporator inlet temperatures and humidities.

## 4.4 Variable-Speed AC

The performance map of the variable-speed AC was developed based on the ORNL heat pump model by curve-fitting the performance of the air conditioner at five discrete compressor speeds and air flow rates (Table 4.4). The range of indoor and outdoor conditions used in the variable-speed AC curve-fits are also shown in Table 4.4. The performance of the VSAC at intermediate speeds was found by interpolation.

**TABLE 4.4**

**Range of Conditions Specified for Variable-Speed AC**

**Range of conditions:**

| | |
|---|---|
| Outdoor Temperature: | 76 to 110°F |
| Indoor Temperature: | 55 to 95°F |
| Indoor Relative humidity: | 5 to 95% |
| Air Flow rates: | 200 to 600 cfm/ton |

Compressor and outdoor fan speed: 30 to 90 Hz (step: 15 Hz)

**Comparison of curve-fit results with actual ORNL model output:**

| Dependant Variables | Average Error | Maximum Error |
|---|---|---|
| Sensible Capacity: | 1.2% | 7.5% |
| Latent Capacity: | 1.6% | 16.0% |
| Power | 0.5% | 7.8% |

Note:
The curve-fit was carried out with outdoor temperature, indoor dry bulb temperature, indoor wet bulb temperature and supply air flow rate as independent parameters for each compressor speed considered.

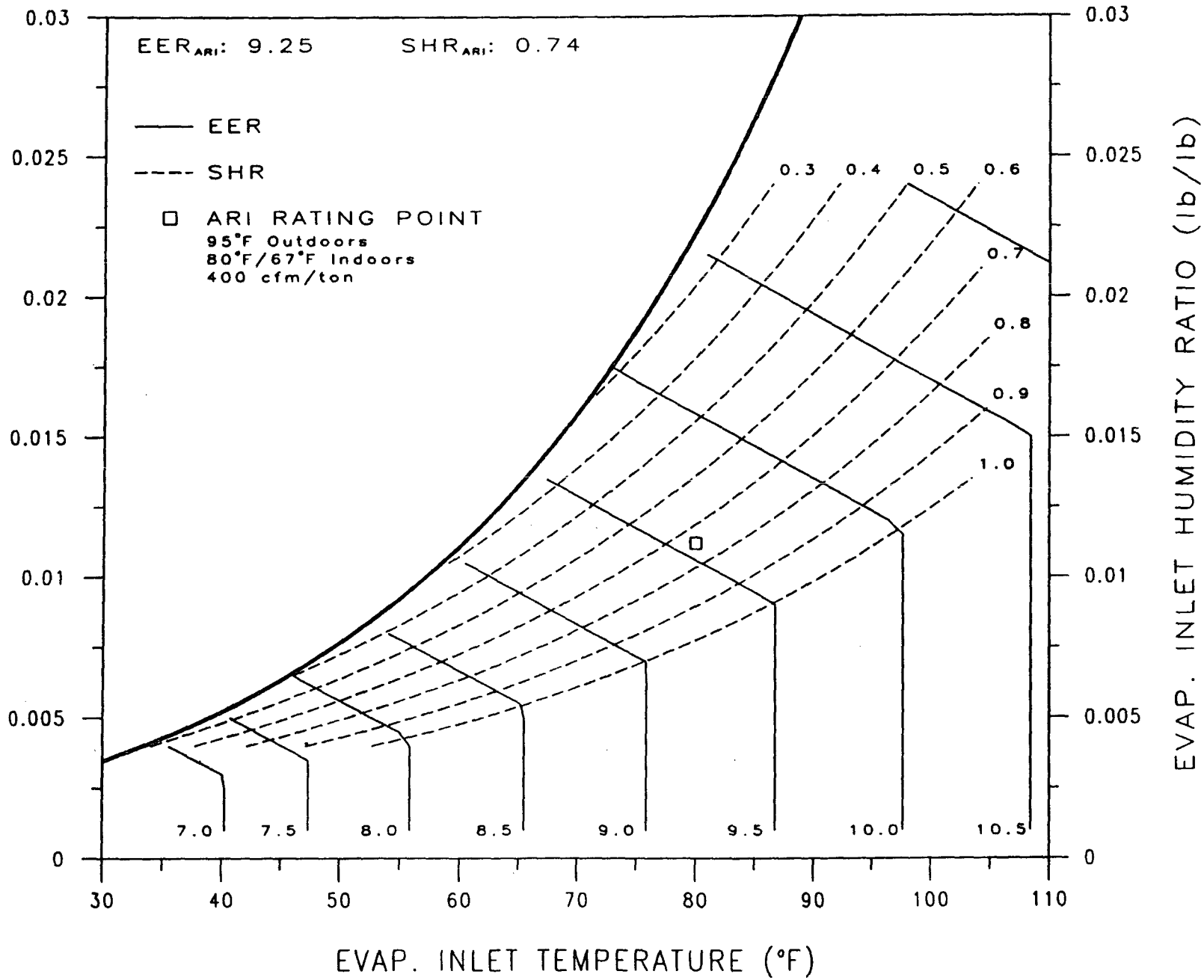



Figure 4.3 - DOE-2 AC: EER and SHR Contours

The nominal performance of the VSAC is shown in Table 4.5

**TABLE 4.5**

| Nominal Performance of Variable-Speed AC | |
|---|---|
| At ARI rating point[a]: (at 90 Hz) | |
| Capacity: | 36.5 kBtu/h |
| SHR: | 0.74 |
| EER: | 7.7 Btu/Wh |
| Flow rate: | 1200 CFM |
| SEER: | 13.3 Btu/Wh |
| [a] Outdoor: 95°F, Indoor: 80°F dry bulb and 67°F wet bulb | |

Figure 4.4 shows the cooling capacity variation with compressor speed and indoor temperature based on our curve-fits. The VSAC EER and SHR variations with varying compressor speed and air flow rate are shown in Figure 4.5. It may be noted from this figure that the efficiency of the air conditioner is a strong function of the compressor speed and the sensible heat ratio is a strong function of the air flow rate with a moderate dependance on compressor speed. The efficiency of the VSAC is higher at low compressor speeds due to the relative over-sizing of the heat exchangers for low speeds.

## 4.5 Heat Pipes

The heat pipe model was developed by curve-fitting heat pipe performance data from QDT, Ltd. (3) and using effectiveness - $N_{tu}$ relations from Kays and London (4). The heat transfer process between the supply and return air streams can be modeled using the $N_{tu}$ - effectiveness relationship of a counter flow heat exchanger. The effectiveness of a single row of heat pipe ($e_{row}$) for counter flow arrangement can be derived as:

$$e_{row} = \frac{N_{tu}}{1 + N_{tu}} \tag{4.1}$$

$N_{tu}$ is defined as:

$$N_{tu} = UA/mC_p \tag{4.2}$$

where
- U - overall heat transfer coefficient
- A - heat transfer surface area
- $C_p$ - specific heat of air
- m - mass flow rate of each air stream

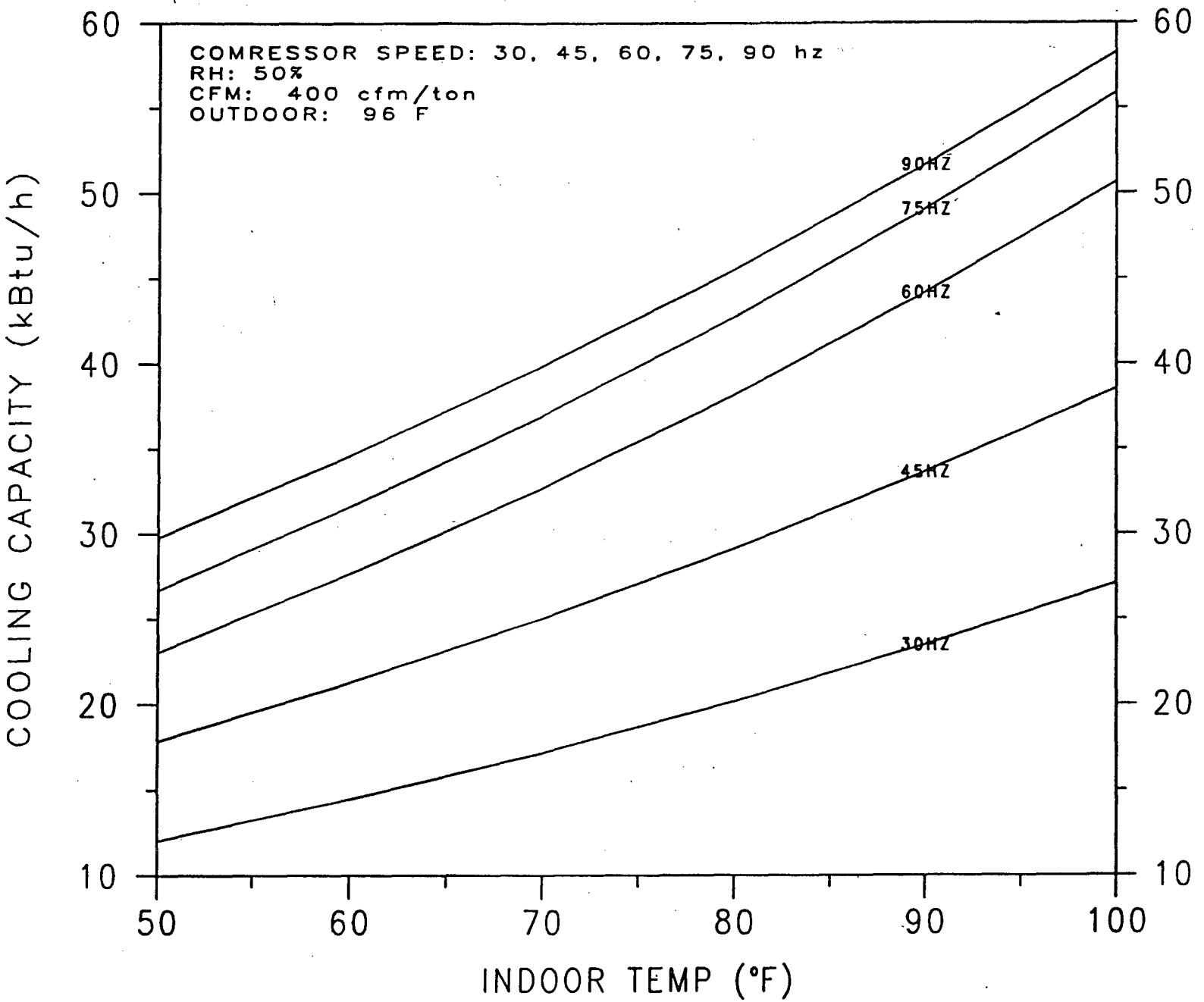



Figure 4.4 - Variable Speed AC: Capacity as a Function of Compressor Speed

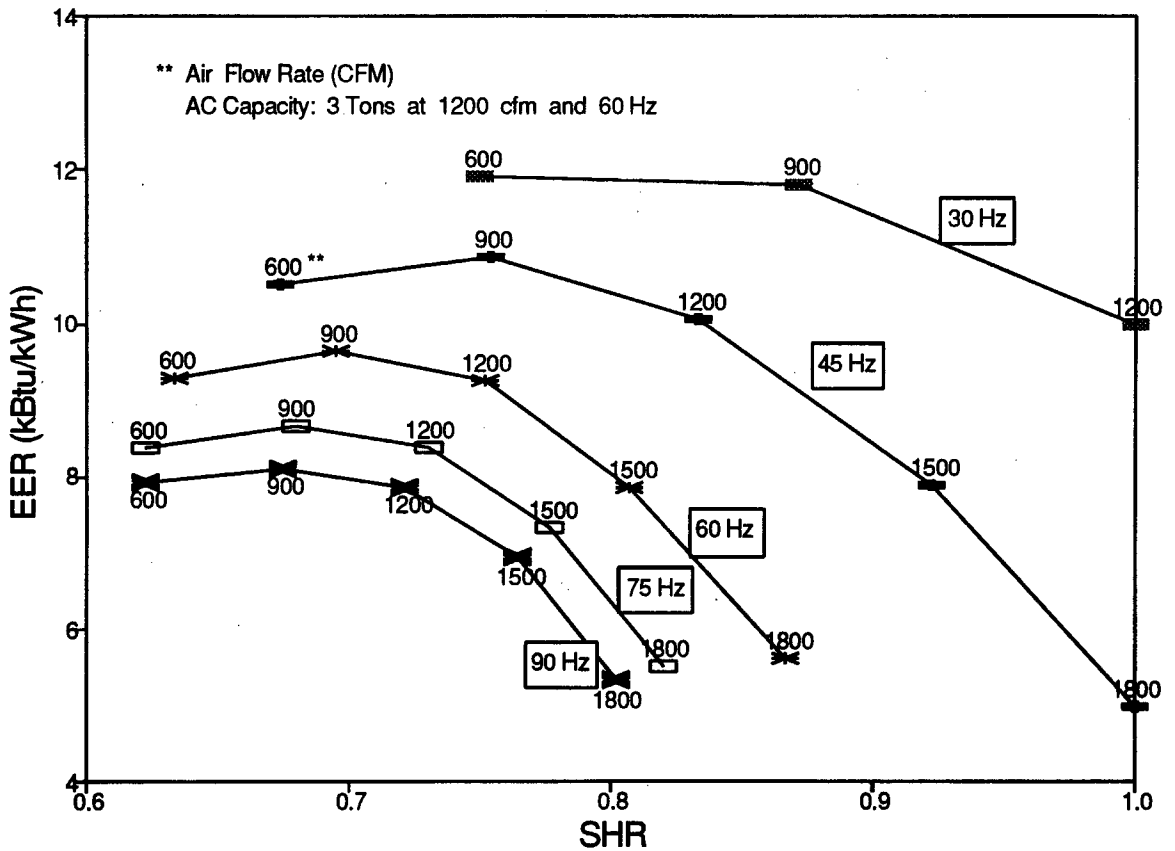



Figure 4.5 - Variable Speed AC: EER as a Function of SHR

U is defined as:

$$U = \frac{1}{(1/h + R_{HP} + 1/h)} \tag{4.3}$$

where h is the heat pipe outside film resistance and $R_{HP}$ is the internal resistance across the heat pipe. The heat transfer film coefficient depends on the face velocity over the heat pipe coils and the fin characteristics. The relationship between the face velocity and heat transfer film coefficient can be written as:

$$h = a_0 V^{(1-n)} \tag{4.4}$$

For multiple rows of a heat pipe heat exchanger, the total $N_{tu}$ available ($N_{tutot}$) for heat transfer would be:

$$N_{tutot} = N_{row} N_{tu} \tag{4.5}$$

where $N_{row}$ refers to the number of rows of heat pipes.

Hence, the effectiveness for multiple rows of heat pipes can be written as:

$$e = \frac{N_{row} N_{tu}}{(1 + N_{row} N_{tu})} \tag{4.6}$$

Combining equations 4.2, 4.3 and 4.4 led to a relationship for $N_{tu}$ of each fin geometry as a function of face velocity.

$$N_{tu} = \frac{a_1 V^{-n}}{(2 + a_2 V^{(1-n)})} \tag{4.7}$$

where n = coefficient from Kays and London (5) (n = 0.398 for plate fins)

Constants $a_1$ and $a_2$ were determined by curve-fitting effectiveness (e) against face velocity (V) and the number of rows ($N_{row}$) from manufacturers data. Figure 4.6 shows the effectiveness for the QDT plate-fin type heat pipes.

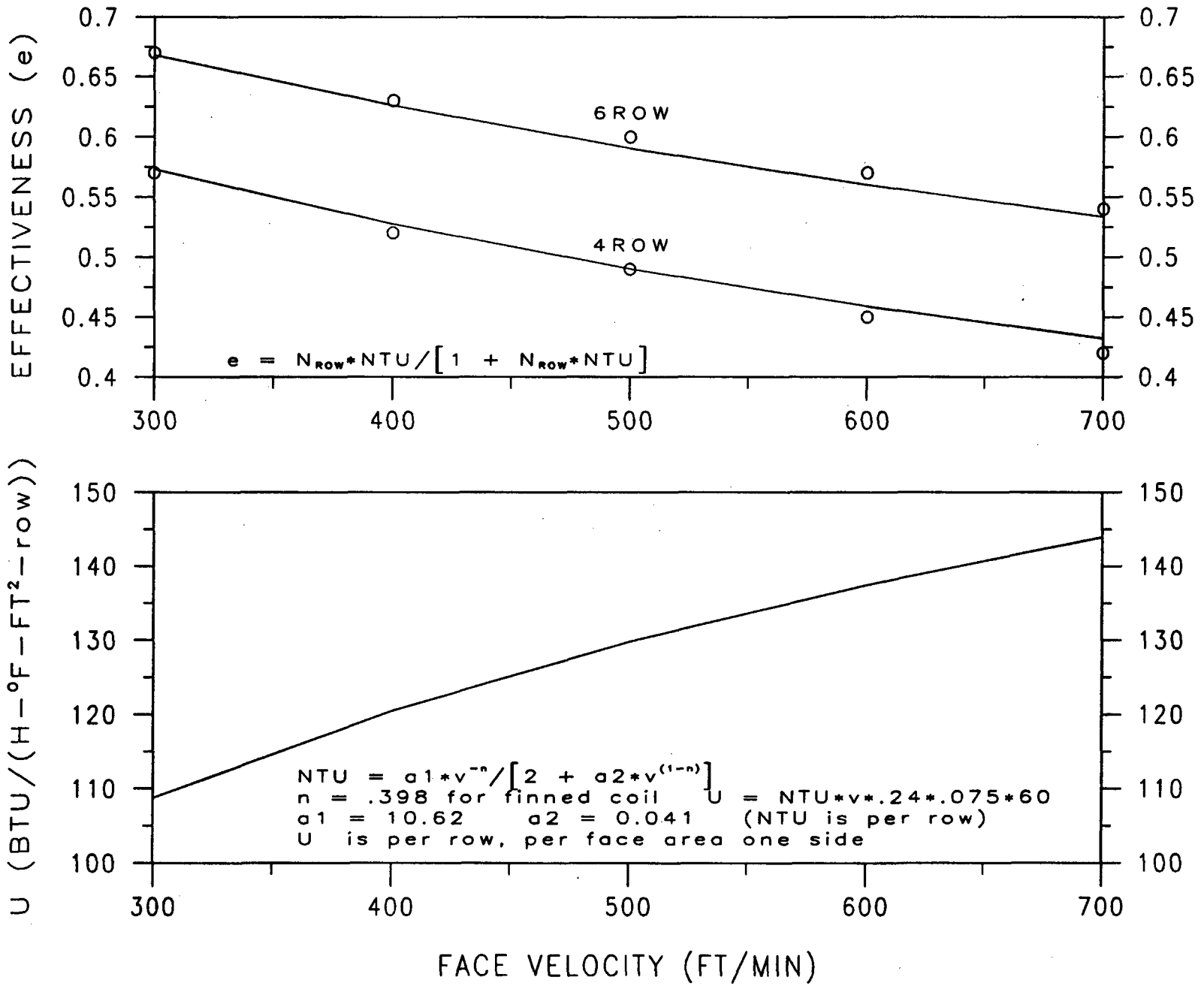



Figure 4.6 - Heat Pipe Effectiveness as a Function of Face Velocity and Number of Rows

## 4.6 Desiccant Wheel for DEAC

The performance curve-fits for the desiccant wheel were determined from the detailed desiccant simulation program developed by Kirk Collier of Enerscope (5). The periodic steady-state operation of the desiccant wheel was predicted by following a slice of the wheel until the change in outlet states from one rotation to the next and the overall mass balance closure meet a defined convergence criteria. This procedure essentially follows the transient response of the wheel from an initial assumed matrix profile to that at periodic steady state. The wheel modeled is representative of those currently available.

The desiccant wheel specification and the range of operating conditions used for the desiccant wheel performance curve-fits are listed in Table 4.6.

**TABLE 4.6**

| DEAC Simulation Model Parameters | |
|---|---|
| **Desiccant Wheel:** | |
| Desiccant bed: | Silica Gel on aluminum |
| Desiccant mass fraction: | 0.83 |
| Wheel composite density: | 70.5 lb/ft$^3$ |
| Single channel hydraulic diameter: | 0.05" |
| Wheel specific heat: | 0.338 Btu/(lb.°F) |
| Lewis number: | 1.0 |
| Nusselt number: | 2.47 |
| Depth: | 0.25" to 6" |
| Rotational speed range: | 5 to 30 cycles/hour |
| Optimum wheel speed: | 10 cycles/hour |
| Face velocity range: (supply and return) | 250 - 800 ft/min |
| **Regeneration inlet conditions: (Return air)** | |
| Temperature: | 68 to 84°F |
| Absolute Humidity: | 0.008 to 0.012 |
| **Adsorption inlet conditions: (AC supply air)** | |
| Temperature: | 48 to 60°F |
| Relative Humidity: | 80 to 90% |

The DEAC performance was simulated by integrating the DOE-2 packaged AC model and the detailed desiccant simulation program developed by Enerscope. Figure 4.7 shows the predictions for the DEAC performance as a function of number of transfer units ($N_{tu}$) of the desiccant wheel. If air flow rates, rotational speed and heat transfer rates are constant, the $N_{tu}$ of a desiccant wheel is increased by increasing its surface area exposed to the air streams. Increasing the surface area of the wheel can be accomplished by increasing its diameter, or by making the wheel thicker, or by adding more desiccant materials to the matrix. In our study, the $N_{tu}$ of the wheel was increased by increasing the wheel depth. The results shown in Figure 4.7 are for a wheel depth range of 0.25 to 8 inches.

As can be seen from Figure 4.7 dehumidification of supply air increases with an increase in depth of the desiccant wheel. As the wheel revolves and the desiccant is regenerated by the return air, an increase in dehumidification results in increased moisture in the return air thereby moving the return air closer to the saturation line. The moisture transport within the DEAC desiccant wheel is driven by the difference in relative humidity between the return and supply air streams. In this case, the supply air outlet state reached the return air inlet relative humidity level with a desiccant wheel of approximately 8" depth. Any further increase in wheel depth will not, with the specified conditions, produce a significant increase in dehumidification.

Figure 4.8 shows the impact of $N_{tu}$ on the SHR of the DEAC over a range of return air temperatures. The SHR of the air conditioner decreases sharply in the range of 0 to 2 in. depth of the wheel. This implies that a relatively small $N_{tu}$ and, therefore low cost wheel may provide adequate dehumidification for the stated conditions.

The results presented in Figure 4.7 and 4.8 were based on a constant desiccant wheel speed of ten cycles per hour. However, the rotational speed of the wheel can be varied and, thereby, used as a control option to meet a limited range of building load-SHR values. Figure 4.9 shows the impact on SHR of varying the desiccant wheel rotational speed. The nominal optimum is the lowest wheel speed at which maximum mass transfer can be attained. This optimum speed occurs in the range of 7.5 to 12.5 cycles per hour for the DEAC.

The impact of air leakage (bypass) through gaps between the desiccant wheel cassette seals and the desiccant wheel was investigated. The basic assumption was that air would bypass from both edges of the wheel (i.e., air entering and leaving the wheel) when the seals were worn. The DEAC capacity, EER, and SHR variation as a function of desiccant wheel air leakage is plotted in Figure 4.10. The results show that the loss in capacity and EER for a maximum 10 percent of air leakage (5 percent from both edges of the wheel) for a DEAC system would be 2.3 and 1.6 percent respectively. The DEAC SHR variation as a function of leakage shows that, a 3 percent SHR reduction results from a maximum of 10 percent air leakage. This reduction in SHR is due to the reduced air flow over the cooling coil when the leakage occurs.

Although the above analysis treats only the case of a desiccant wheel other configurations, such as a run around belt, are possible. Appendix B presents an analysis of a run around belt and presents results for two alternative desiccant materials.

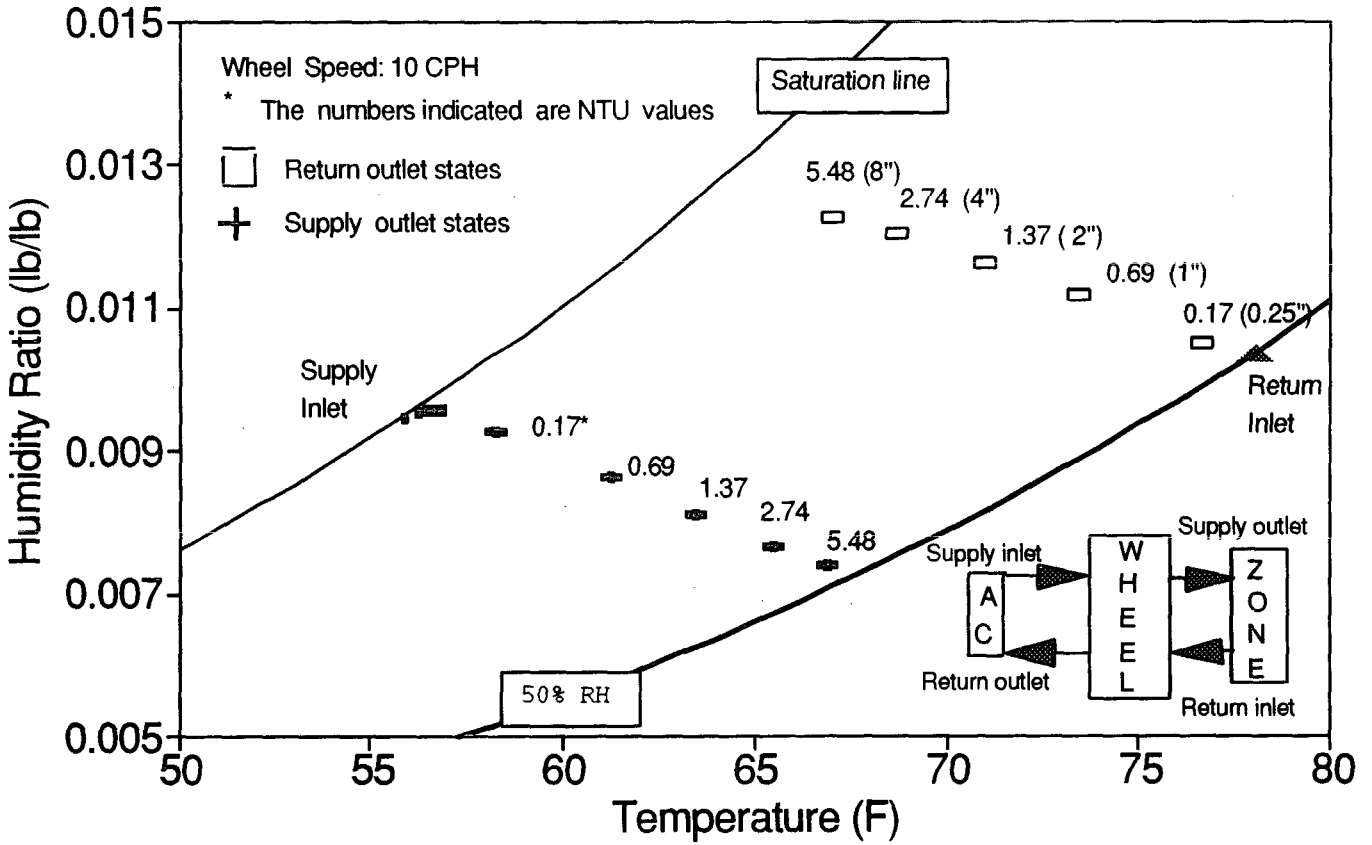



Figure 4.7 - DEAC Psychrometric Process State Points Predicted as a Function of Wheel Depth (Ntu)

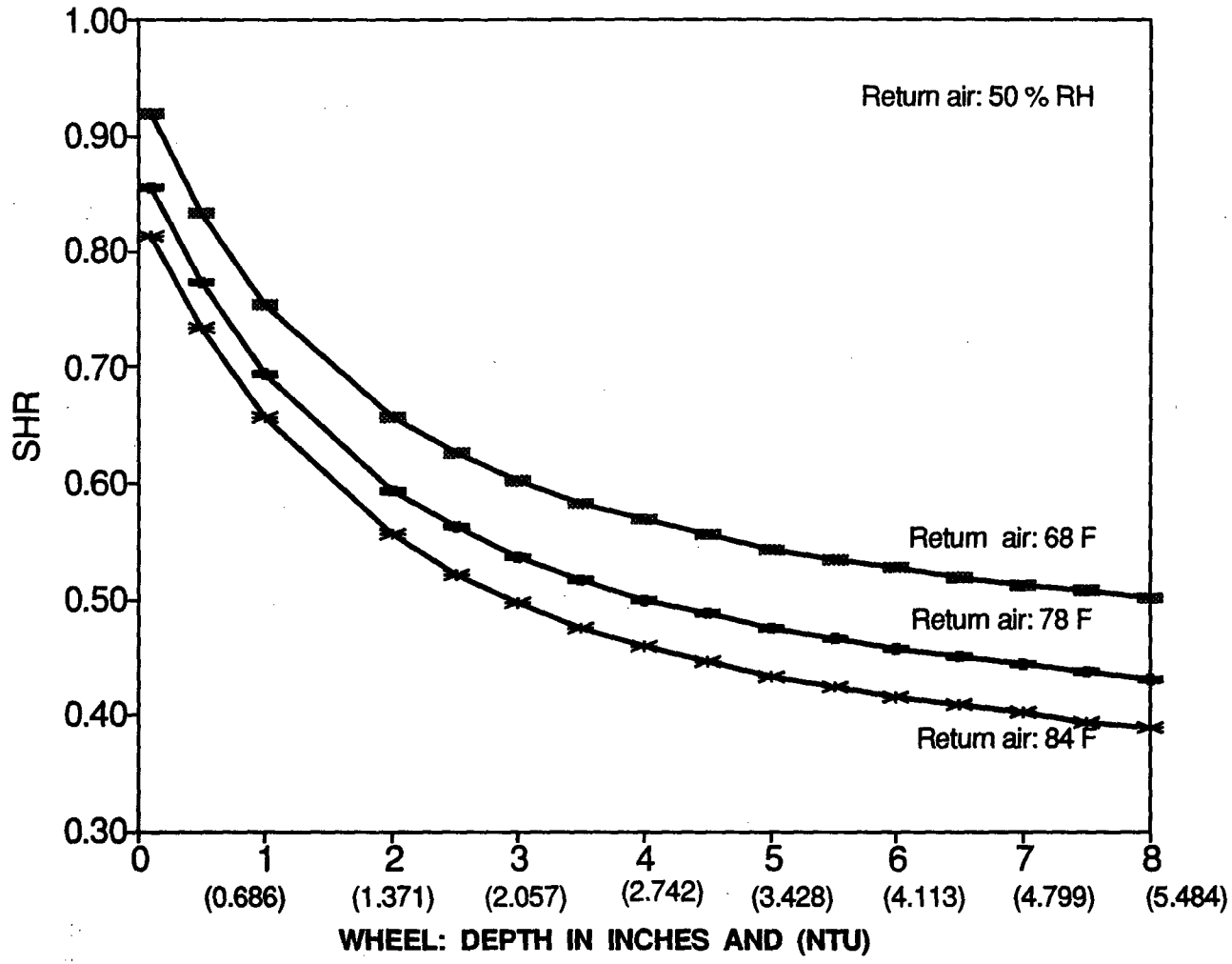



Figure 4.8 - DEAC Sensible Heat Ratio as a Function of Desiccant Wheel Depth (Ntu)

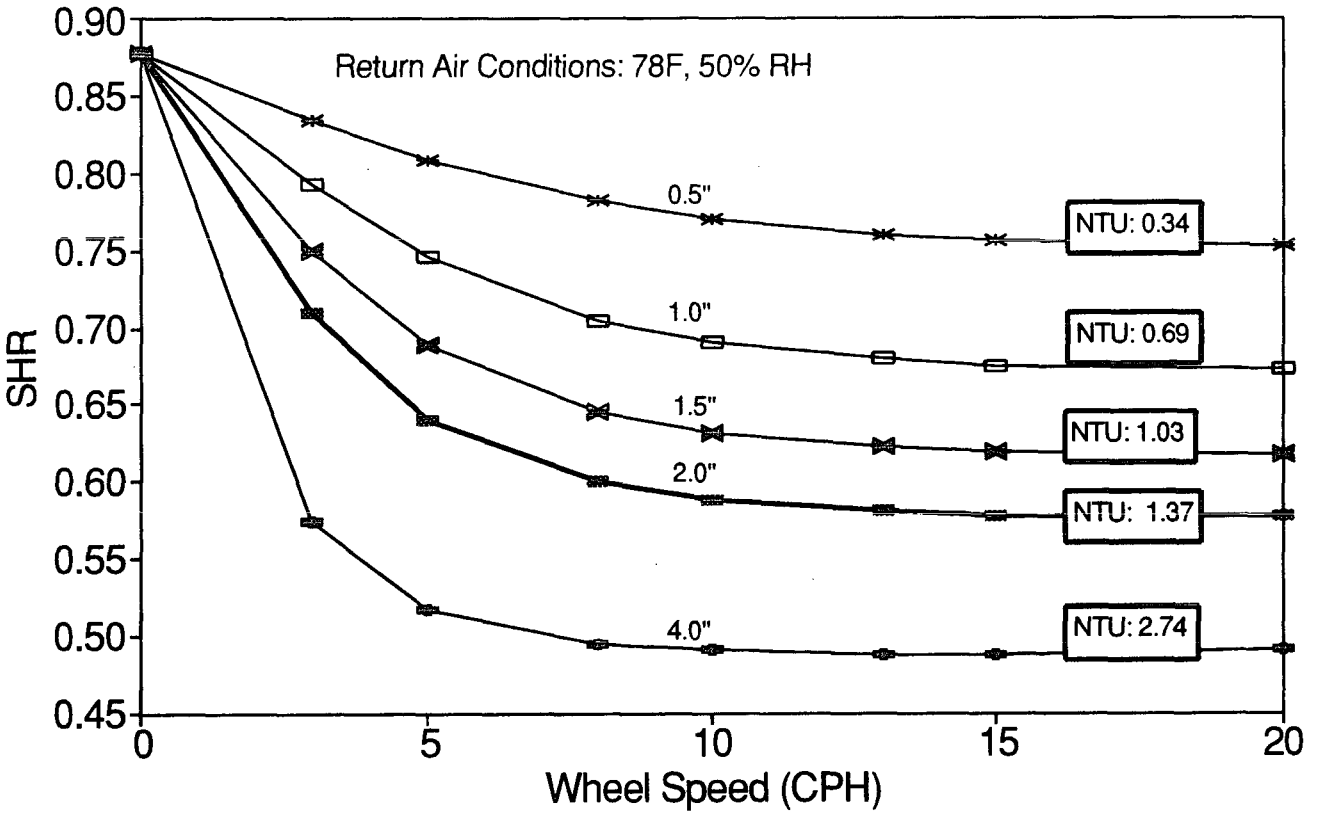



Figure 4.9 - DEAC: SHR as a Function of Desiccant Wheel Rotational Speed

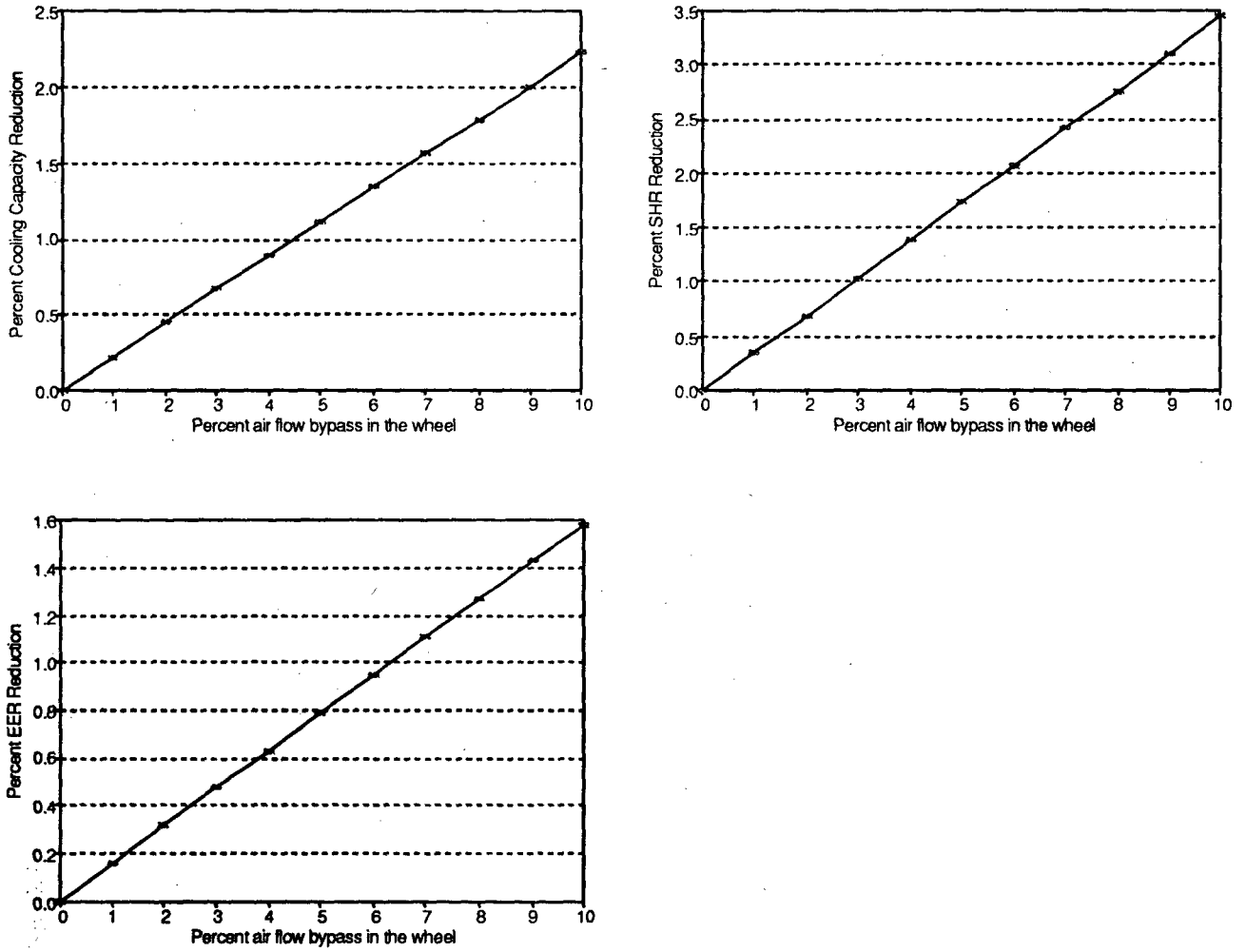



Figure 4.10 Impact of Air Bypass on DEAC Capacity, EER and SHR

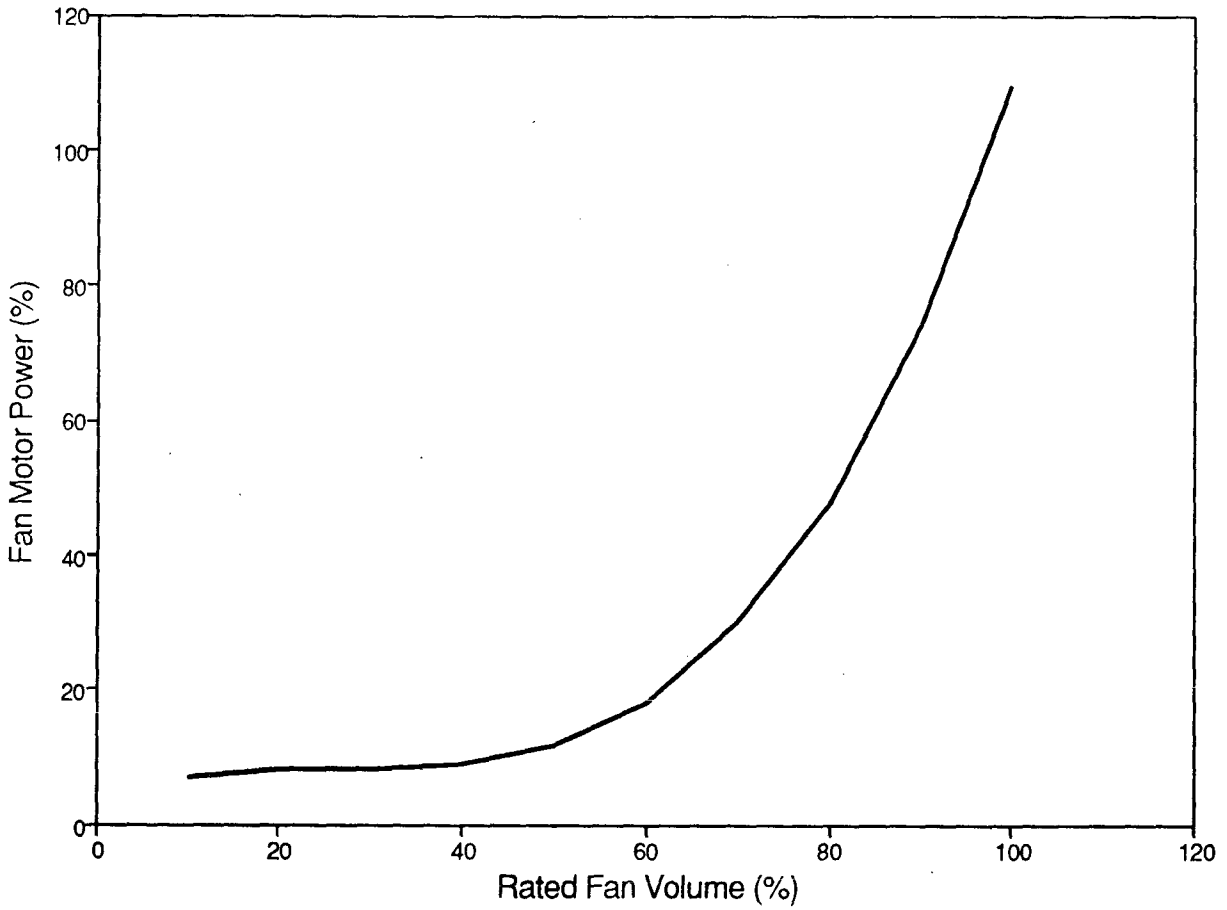



Figure 4.11 - Variable Speed Fan Power as a Function of Air Flow-Rate

## 4.7 Forward-Curved Fan

Forward-curved supply air fans are commonly used primarily for low-pressure HVAC applications, such as residential units and package air conditioners (6). A generic model of this fan type was used in this study. Static and motor efficiency are the input parameters. The fan power consumption may be calculated from equation 4.8.

$$P_{fan} = \frac{0.12\ Q\ \Delta P}{\eta_f\ \eta_m} \tag{4.8}$$

where:

| | |
|---|---|
| $P_{fan}$ | - Fan motor power, W |
| $Q$ | - Air flow rate, cfm |
| $\Delta P$ | - Pressure drop in $H_2O$ |
| $\eta_f$ | - Fan static efficiency (0-1) |
| $\eta_m$ | - Fan motor efficiency (0-1) |

Typical fan static efficiencies used in our seasonal simulations ranged from 0.45 to 0.55 and motor efficiencies ranged from 0.70 to 0.85.

The air flow rate may change with the addition of components such as heat pipes or desiccant wheel in an air stream due to the increased pressure drop. The forward-curved fan model used in this study allows systems to "ride" the fan curve (i.e., simulates the variation in air flow rates) as the pressure drop changes.

## 4.8 Variable-Speed Fan

A variable-speed fan model was developed from commercially available performance data (7). Fan speeds were calculated over a range of air flow rates and static pressures using a curve-fit developed from a manufacturer's data (7). As with any fan, the power consumption of a variable speed fan depends on both the fan speed and the static pressure it needs to develop. Hence, another curve-fit relationship was developed to predict the fan power consumption as a function of fan speed and static pressure. The percent rated fan power as a function of the percent rated air flow is shown in Figure 4.11 for the variable speed fan model.

## 4.9 Gas-fired Desiccant Wheel

The silica gel desiccant wheel performance curve-fits used in our simulation of gas-fired dehumidification systems were based on the detailed simulation model developed by Kirk Collier of Enerscope (5) and discussed in Section 4.6. The range of conditions used for developing the curve-fits are shown in Table 4.7.

The optimum ratio of the process area to the total face area of the wheel (wheel split) was determined from the product of wheel capacity and wheel efficiency (COP). Wheel capacity is defined as the ratio of latent capacity provided by the wheel to the total surface area of the wheel. Wheel efficiency (COP) is defined as the ratio of latent capacity produced by the wheel to the regeneration heat input.

Process air flow rate, desiccant wheel speed and wheel depth were varied to determine an optimum wheel split. Initially, an optimum wheel speed (which provided maximum dehumidification) was identified corresponding to each of the process area fractions considered (ranged from 0.5 to 1.0). Wheel capacity and efficiency were determined for each process area fraction at the corresponding optimum wheel speed. Following this, the product of wheel capacity and wheel efficiency was calculated at each of the process area fractions. Subsequently, the optimum process area was determined using these product values.

The optimum wheel split for silica gel desiccant wheel was found to be 65 and 75 percent at regeneration temperatures of 149°F and 284°F respectively.

**TABLE 4.7**

| Silica Gel Desiccant Wheel: Range of Curve-fit Conditions | |
|---|---|
| Regeneration air inlet: | |
| Temperature: | 149°F, 284°F |
| Humidity: | 0.012 to 0.018 lb/lb |
| Process air inlet: | |
| Temperature: | 68 to 104°F |
| Humidity: | 0.008 to 0.024 lb/lb |
| Face velocity: | 285 to 408 ft/min |
| Wheel depth: | 2.75 to 11 in. |

## 4.10 Kathabar Liquid-Desiccant Dehumidifier

A performance map for the Kathabar liquid-desiccant dehumidifier model was developed based on the performance data used in DOE-2 building energy analysis software for modeling a 12 ton Kathabar system. The range of operating conditions for the modeled Kathabar dehumidifier is shown in Table 4.8.

**TABLE 4.8**

| Kathabar Dehumidifier: Range of Curve-fit Conditions | |
|---|---|
| Regeneration air inlet: | |
| Temperature: | 65 to 105°F |
| Humidity: | 0.008 to 0.020 lb/lb |
| Process air inlet: | |
| Temperature: | 65 to 105°F |
| Humidity: | 0.008 to 0.020 lb/lb |
| Flow rate: | 2500 to 4500 cfm |

## 4.11 Assembling Components into Systems

The components were assembled through performance subroutines for the AC systems. The performance subroutines were developed to simulate the performance of the following systems.

1. SSAC with heat pipes and/or reheat
2. DEAC with/without reheat
3. VSAC with heat pipes and/or reheat
4. VSAC with desiccant wheel and or reheat
5. AC with gas-fired desiccant wheel
6. AC with Kathabar liquid-desiccant dehumidifier

The details and control logic used to integrate the performance subroutines of the AC systems into the building loads simulation module are discussed in Chapter 6.

References


1. Miller W.A. 1988 "Laboratory capacity Modulation Experiments, Analyses and Validation" Proceedings of the 2nd DOE/ORNL Heat Pump Conference: Research and development on Heat Pumps on Space Conditioning Applications, CONF-8804100, Washington,D.C., April, pp.7-21.

2. Henderson H.I., K. Rengarajan, D.B. Shirey, III, "The impact of comfort control on air conditioner energy use in humid climates", ASHRAE Transactions, 1992, V.98, Pt 2.

3. Communications with QDT Supermarket Air-Conditioning Systems, QDT Ltd. Dallas.

4. Kays, W.M. and London, A.L. Compact Heat Exchangers, Third Edition, Mcgraw-Hill Book Co., New York, 1984.

5. Collier R.K. 1988. DCSSET Simulation Code for Desiccant Performance. (Unpublished)

6. ASHRAE Handbook, 1992, HVAC Systems and Equipment

7. Variable speed air handlers literature, TRANE XV 1500 variable speed systems.

# 5.0 PERFORMANCE COMPARISON OF ELECTRIC AC SYSTEMS

## 5.1 Overview

Chapter 4.0 has provided background on the development of the basic air-conditioning systems and components to be used in this simulation study. This Chapter compares the EER and SHR performance of the DEAC relative to the following electric AC options: single speed AC with electric reheat or heat pipes and a variable-speed AC with variable-speed indoor fan system. The overall focus of this comparison is on the ability of each system to maintain a high EER over a wide range of SHR values. The results of this study were presented at the 1993 ASHRAE Winter Annual Meeting (1).

## 5.2 Cooling Capacity Variation with SHR

Performance simulations of the electric AC systems noted above were conducted to compare the merits of the DEAC with other electric AC options. Each of the equipment options are discussed in the following paragraphs.

The basic air conditioner model used for this performance comparison is the DOE-2 AC model described in Section 4.3. The assigned nominal EER of the basic 3 ton AC model was 12.0 and the assigned SHR rating was 0.84 at ARI conditions (Indoor; 80°F Dry Bulb, 67°F Wet Bulb: Outdoor; 95°F). A nominal air flow rate of 450 cfm/ton was specified at the above conditions.

Reheating the supply air to lower the SHR of an air conditioner has been common practice for commercial buildings. In air-reheat systems, a lower SHR is achieved by overcooling using an oversized AC unit and then reheating the supply air with a heat source. If the heat source is electric reheat, the reduction in sensible cooling capacity of the AC is equivalent to the amount of electric reheat added. Hence, the total capacity of air-reheat systems decreases with increasing reheat. This trend is shown in Figure 5.1 for a range of electric reheat values from 0 to 4 kW.

Heat pipes, described previously in Sections 3.2 and 4.5, protrude into the indoor air stream before and after the evaporator coil and, as a result, increase the pressure drop of the air flow. This in turn increases the supply fan heating effect which, slightly reduces the system capacity. In addition, the apparatus dew point (ADP), or effective cooling coil temperature, is slightly reduced with the heat pipe system. This reduction in ADP also contributes to a reduction in total capacity. The total capacity reduction trend for a heat pipe-assisted AC is shown in Figure 5.1. The symbols on the heat pipe curve refer to the SHR values that could be achieved with 0 to 6 rows of 14 fins per inch, commercially available heat pipes. The effectiveness values of the modeled heat pipes ranged from 0 to 0.6. Figure 5.1 also shows that, for the same SHR, total cooling capacity reduction due to the addition of heat pipes to the AC was greater than that due to the addition of the desiccant wheel to the AC.

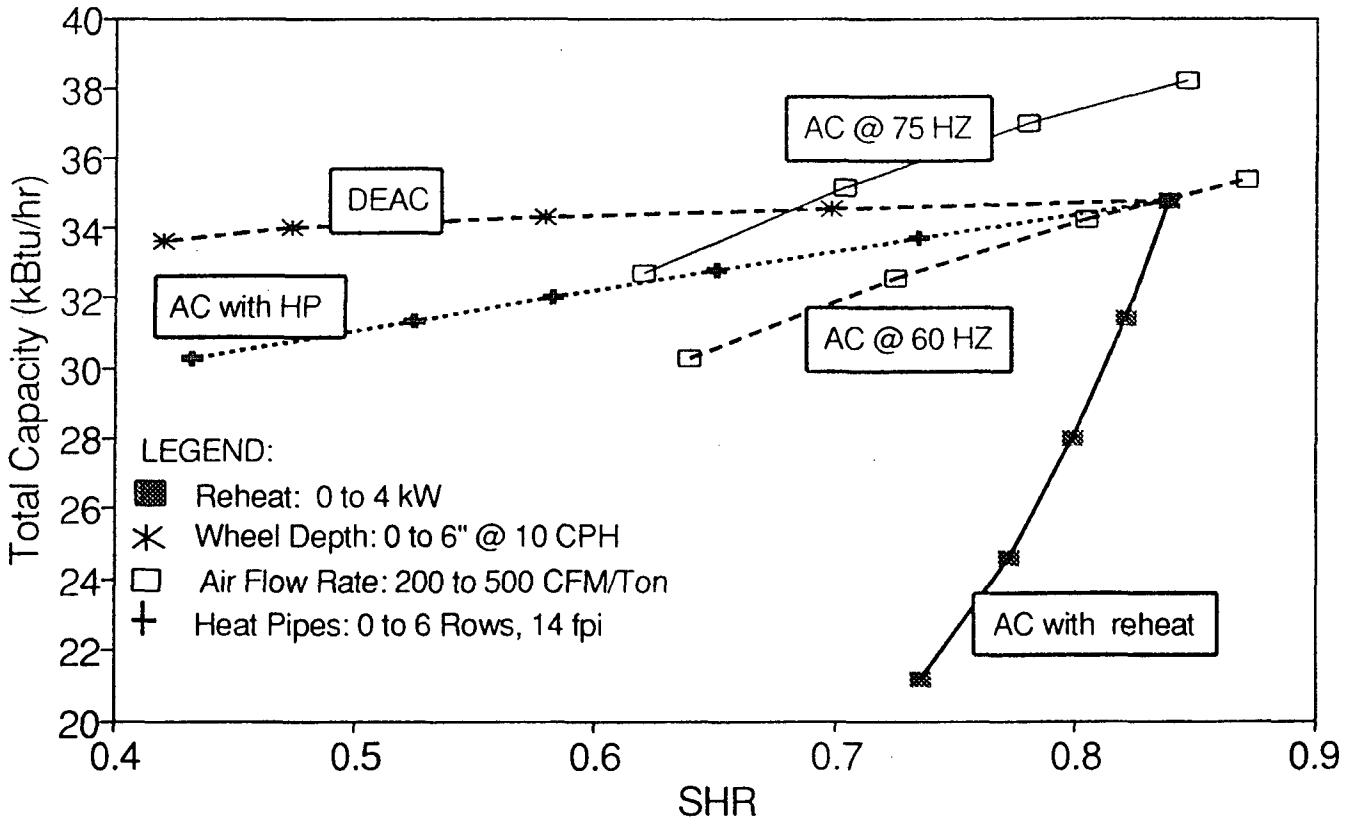



Figure 5.1 - Comparison of AC Options: Total Capacity as a Function of SHR

One of the advantages of the DEAC approach over a sensible heat exchanger for achieving lower SHR values is the relatively higher apparatus dew point. The capacity reduction of the DEAC system as wheel depth is increased is due to the additional air-side pressure drop through the wheel. As shown previously (Figure 4.8), the desiccant wheel depth required to achieve a nominal SHR of 0.7 is less than 1 inch. The pressure drop for this wheel depth at a nominal face velocity of 450 fpm is approximately 10 percent of the total air-side pressure drop for the DEAC system . The corresponding air side pressure drop for heat pipes is approximately 18 percent. The DEAC capacity curve in Figure 5.1 shows a relatively small reduction in total capacity with decreasing SHR compared to the alternative systems. The desiccant wheel depth was varied between 0 and 6 inches for this comparison.

For a variable-speed air conditioner (VSAC), lower SHR values can be achieved by lowering indoor fan speed and thereby reducing air flow over the coil. The impact of varying air flow rate on the air conditioner's total capacity was initially studied using the DOE-2 AC model at a compressor speed of 60 Hz. The air flow rate was varied in steps of 100 cfm from 200 to 500 cfm/nominal ton of air conditioning. The nominal capacity of the AC was 3 tons at an air flow rate of 450 cfm/ton as indicated earlier. As the air flow rate was reduced, the latent capacity of the AC increased while the sensible heat capacity decreased. The overall result was a reduction in total capacity of the AC with reduction in air flow. This is shown in Figure 5.1 for the curve labeled "AC @ 60 Hz".

However, one of the benefits of the VSAC with its variable compressor speed and a variable indoor fan speed is that the capacity can be modulated. To take advantage of these VSAC characteristics we made a second set of comparison runs. This second set focused on a single SHR of 0.7, a value typical of residences. By this approach, we substantially reduced the computer runs to a comparison of the DEAC and VSAC systems at a single SHR and a single capacity.

The capacity of the AC at a 60 Hz compressor speed and an SHR of 0.7 was lower than the DEAC as shown in Figure 5.1. To match the DEAC capacity at this SHR, a higher compressor speed for the AC was required. The DOE-2 AC model is limited to a single compressor speed of 60 Hz. A different compressor speed which matched the DEAC capacity at SHR of 0.7 was predicted using, the ORNL variable speed heat pump model previously described in Section 4.4.

The ORNL heat pump performance map was used to develop a set of parametric curves for total cooling capacity and EER as a function of SHR for a range of compressor speeds and air flow rates over the evaporator coils. The air flow rate was varied from 200 to 500 cfm/nominal ton of air conditioning. Using the ORNL heat pump performance maps, and knowing the capacity of the DOE-2 AC at 60 Hz compressor speed and an SHR of 0.7, it was determined that, at a compressor speed of approximately 75 Hz, the AC would be able to meet the desired SHR of 0.7 and closely match the DEAC cooling capacity. The results of this intermediate step matching process are presented in Figure 5.1.

## 5.3 EER Variation with SHR

A number of the steps used to develop the capacity change with SHR in the previous section are similar to those used here to map EER as a function of SHR.

Figure 5.2 shows the EER performance curve for the AC at 75 Hz compressor speed. The approach used to develop this curve was to first construct the EER curve as a function of SHR using the DOE-2 model with a 60 Hz compressor. The EER parametric curves from the ORNL model were then used to predict the offset of the previously generated DOE-2 EER values at 60 Hz to the values at 75 Hz. The air flow rate range was 200 to 500 cfm/nominal ton. The new curve is labeled "AC @ 75 Hz".

As shown in Figure 5.2, up to a certain limit, the EER of the AC at 75 Hz compressor speed increases with the reduction in air flow rate. This is due to the reduced variable-speed fan power consumption. It then decreases as we decrease the air flow rate below 300 cfm/ton. This variation in EER is also a function of power consumption characteristics of the variable-speed fan. For typical heat pumps, approximately 40% of full air flow is the lower limit beyond which there is a rapid decrease in total capacity and EER (2). In this study, the air flow was limited to 44% of the nominal air flow (200 cfm/ton) and hence, only a small decrease in EER at reduced air flow is shown. Our model indicated that, at 75 Hz compressor speed, reducing the air flow rate beyond 200 cfm/ton would result in DX coil freeze up. For the stated conditions, the EER for the AC at 75 Hz compressor speed is, in all cases, lower than that of the DEAC.

Our VSAC performance map, which was developed based on the ORNL MARK III software code did not include the motor inverter drive efficiency variation with the compressor speed. The new version of the ORNL modulating heat pump model (which was not available at the time our analysis) has this capability (3). The EER of the VSAC predicted using our curve-fits will be slightly higher (within 10 percent) than the values predicted by the new ORNL modulating heat pump model. If we include the impact of VSAC motor drive efficiency variation, the EER of the VSAC at 75 Hz will be lower than the values shown in Figure 5.2.

The addition of heat pipes to an air conditioner results in two impacts on EER. First, the increase in pressure drop caused by the heat pipe coils increases the supply fan power consumption. Second, as a result of the pre-cooling of return air by the heat pipes, the ADP of the machine is slightly reduced resulting in a small drop in efficiency. Both of these factors contribute to the heat pipe performance curve of Figure 5.2 being lower than that of the DEAC. The system curve of EER as a function of SHR was generated for a range of 0 to 6 rows of heat pipes.

In the air-reheat system, the total EER of the AC system was calculated based on the total power use of the compressor, supply fan, outdoor fan, and reheat electricity used. The variation of EER with SHR for an AC with electric reheat ranging from 0 to 4 kW is shown in Figure 5.2. The EER of the electric reheat option drops off rapidly making this, as expected, the poorest performer.

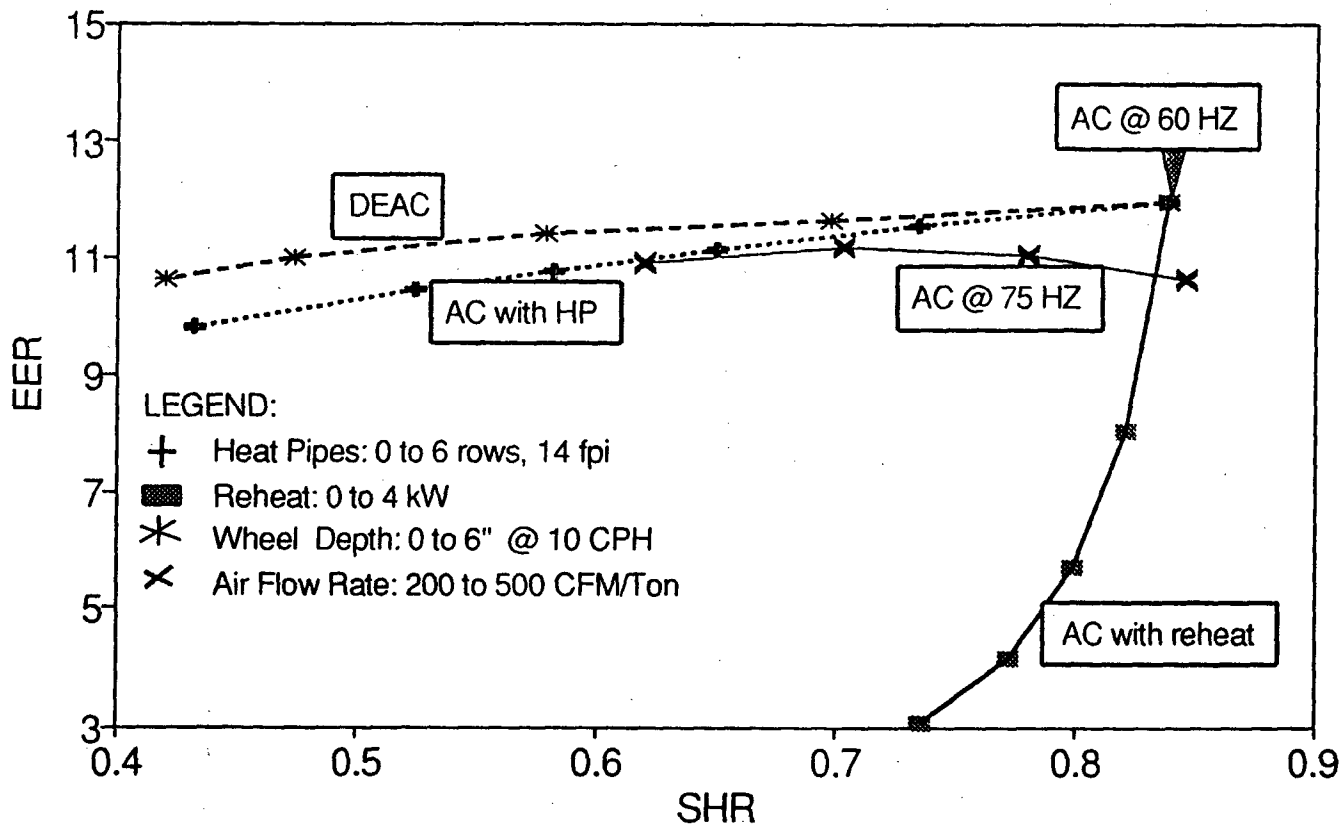



Figure 5.2 - Comparison of AC Options: Efficiency as a Function of SHR

The EER of the DEAC system is a function of return air conditions and wheel $N_{tu}$ (e.g., wheel depth). The return air conditions for all AC systems were maintained at 80°F dry bulb temperature and 67°F wet bulb temperature. The desiccant wheel was maintained at an optimum speed of 10 cycles per hour for the EER comparison study. The decrease of the DEAC system EER is only due to the fan power increase associated with the increased wheel depths. Figure 5.2 shows that the DEAC system achieves an SHR reduction from 0.84 to 0.425 with only 12% decrease in EER. The effect of air leakage (bypass) through the gaps between the seals and the desiccant wheel was not considered for the EER comparison study. However, as indicated in Section 4.6, the reduction in DEAC EER would be only 1.6 percent for a maximum air leakage of 10 percent through the seals of the wheel.

Reference:


1. Nimmo, B.G., R.K. Collier and K. Rengarajan, 1993, DEAC: Desiccant Enhancement of Cooling-Based Dehumidification, ASHRAE Transactions, V.99, Pt. 1.

2. Mac Arthur, J.W., and E.W. Grald, 1988, Optimal Comfort Control for Variable-Speed Heat Pumps, ASHRAE Transactions 94(2).

3. The ORNL modulating heat pump design tool - user's guide C.K. Rice June 1991.

## 6.0 BUILDING ENERGY ANALYSIS SOFTWARE: FSEC 2.2

There are a variety of commercial software packages available for building energy analysis (e.g. DOE-2.1D, TRANSYS, BLAST, ESPRE 2.1, Micro-Axcess, HAP, TRACE ULTRA, etc.). We used FSEC 2.2, a building simulator developed at the Florida Solar Energy Center, to compare the DEAC performance within buildings with that of other AC equipment options.

Accurate modeling of combined heat and moisture transfer in buildings is important to predict indoor conditions, loads, comfort levels and performance of air conditioning equipment. AC systems condition the space for only a few hours in a day (10 to 14 hours) for certain building applications (e.g., office, fast food restaurants). If moisture sorption/desorption is not modeled for such building types, seasonal and instantaneous latent cooling loads may be under-estimated (1,2). The FSEC software has the unique capability to model moisture adsorption/desorption of building materials as well as heat transfer.

For a comparison of results from FSEC 2.2 with the results from a software that does not model the moisture adsorption/desorption characteristics of materials, we selected ESPRE 2.1 (EPRI Simplified Program for Residential Energy). ESPRE 2.1, an integrated microcomputer program, was designed to simulate hourly loads and energy use for residential applications and is widely used by utilities. ESPRE uses a lumped moisture mass approximation to model the moisture storage in buildings. Residential simulations were performed using both FSEC and ESPRE for three different climates. The variations in indoor humidity levels and latent loads which result from the above differences in moisture modeling are discussed in Appendix A.

The algorithms used in the FSEC software to predict the building loads and, control options modeled for the AC systems and components are discussed in the following sections.

### 6.1 Heat and Moisture Transfer Models

Heat and moisture transport models are the basis of any building energy analysis software. FSEC 2.2 can model moisture interactions by either a lumped moisture model referred to as the Effective Moisture Penetration Depth (EMPD) model (1) or by a distributed moisture transport model using evaporation and condensation theory models (3). The distributed moisture transport model uses Finite Element Method (FEM). Since, the distributed moisture transport model is computantionaly intensive and not required for the average hourly simulations, we used the EMPD model. The philosophy behind this EMPD model is explained in Section 6.2.

Conduction Transfer Function (CTF) method (4,5) and finite element method (FEM) are the two options available in FSEC 2.2 to model the building heat transfer. Both of these methods predict similar results when the diffusivity of materials is assumed to be a constant. The CTF method is outlined in Section 6.3.

EMPD approach of moisture modeling can be used with both CTF and FEM methods. In our simulations, the CTF and EMPD methods were used for modeling heat and moisture transport respectively because they use relatively less computer CPU time.

## 6.2 EMPD Approach

The EMPD approach assumes that all moisture interactions take place within 0.04 to 0.2 inches of the surface layer and very little moisture is transported through the building materials (1). The EMPD method is appropriate when moisture sorption-desorption cycles in building materials takes place within the order of a few hours due to cycling or scheduled operation of the air conditioner.

An illustration picturing the EMPD concept is shown in Figure 6.1. The moisture transport potential for each interacting surface is determined based on the wall surface temperature and the vapor pressure potential between the surface and the surrounding air. The EMPD method reduces the computational time for simulations, but retains the accuracy needed for predicting latent loads and the zone moisture content at each hour.

## 6.3 Conduction Transfer Function (CTF) Method

This method may be used as an option in FSEC 2.2 to predict the surface temperature of a multi-layer wall. It is not related to nor does it use ASHRAE transfer function coefficients. The CTF method uses Laplace Transforms to transform the heat conduction equation with respect to time. The resulting second order ordinary differential equation is solved analytically with the specified boundary conditions. These boundary conditions are: ground temperature, ambient conditions, convective heat transfer coefficients, solar flux, night sky radiation and heat of sorption. Following this, an inverse Laplace Transform is applied to obtain the exact solution to the heat conduction equation. The CTF method treats thermal diffusivity as a constant. This is a reasonable assumption for the temperature range experienced by building materials.

## 6.4 Buildings and Mechanical Systems Design Parameters Simulated with FSEC 2.2

Results from building energy analysis software are dependent in part on the extent to which details can be modeled. FSEC 2.2 capabilities used in this study include:

a. Simulation of residential and commercial building characteristics
b. Simulation of part-load performance characteristics of the AC systems
c. Modeling of moisture evaporation into the space from evaporator coils
d. Modeling of condensation on heat pipes
e. Simulation of comfort control options (Effective Temperature (ET*))

## 6.5 AC Systems and Components Control Options

The performance subroutines of the air-conditioning components (for example: AC with heat pipes) were assembled and integrated with the building loads software using appropriate control logic. The AC system models compared in this study and listed previously were:

a. SSAC with or without heat pipes or desiccant wheel and/or reheat
b. VSAC with or without heat pipes or desiccant wheel and/or reheat
c. Gas-fired solid and liquid desiccant systems

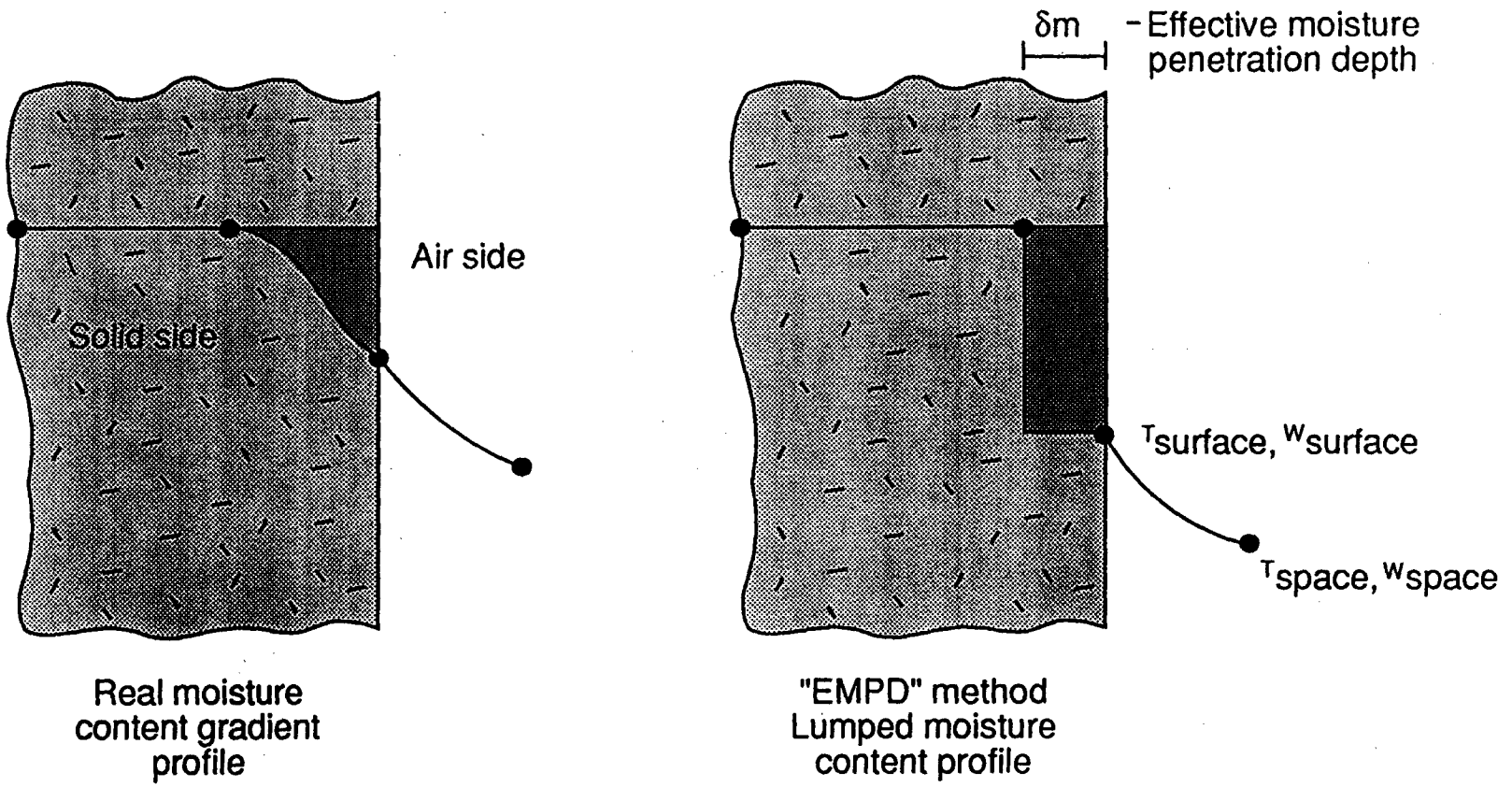



Figure 6.1 - Schematic of EMPD Approach

The control options modeled for these AC systems and components are discussed in the following sections.

### 6.5.1 SSAC with/without Heat Pipes or a Desiccant Wheel and/or Reheat

Building AC systems are typically controlled by either a dry bulb sensing thermostat or a combination of thermostat and humidity sensing humidistat. These control options were modeled as follows. For the AC systems running only on thermostat control, the run-time is calculated based on the sensible load to be met. In this case, latent capacity may be looked at as only a byproduct of the cooling effect. Under humidistat control, the system run-time can be increased to meet the latent load and the excess sensible cooling can be offset using reheat.

Component control options used in our simulation are:

a. "Constant" or "Auto" mode supply fan operation.
b. Desiccant wheel speed control

The heat pipes and desiccant wheel can be locked "ON" to cycle with the air conditioner or independently turned "ON/OFF". In addition, the active number of rows of heat pipes can be varied for each simulation time step. All results presented in the study were obtained with the heat pipes and the desiccant wheel locked "ON" with the air conditioner (SSAC or VSAC) cycle.

### 6.5.2 VSAC with/without Heat Pipes or a Desiccant Wheel and/or Reheat

The control sequence modeled for the VSAC is based on data from commercially available electronic motor controls which modulate the compressor and indoor fan speeds. The control algorithm described below was developed based on an iterative technique to match the compressor and indoor fan speeds with the building total loads and load-SHR values.

Initially, the AC is turned "ON" at a compressor speed of 30 Hz with a minimum required air flow rate. This minimum required air flow rate is determined based on either the ventilation air requirement or the air flow rate required to avoid coil freeze up. The indoor fan power is calculated based on the supply air flow rate and total system pressure drop.

The AC run-time is calculated when the VSAC has just enough sensible capacity to meet the sensible load at the minimum compressor and fan speeds. If the sensible capacity is not sufficient, then the required compressor and supply fan speeds are determined as follows. The compressor speed is estimated based on the ratio of the difference between two successive compressor speeds and the corresponding sensible capacities. The equation for estimating the new compressor speed can be written as:

$$hz_{new} = hz - ((hz - hz_{old}) \times (qsens_{hz} - qsens_l) / (qsens_{hz} - qsens_{hzold}) \quad \textbf{(6.1)}$$

where hz and $hz_{old}$ are the compressor speeds in the last two successive iterations and $hz_{new}$ is the required compressor speed for meeting the sensible load ($qsens_l$). In this case, the supply fan speed is increased proportionately to the compressor speed. Since, the AC capacity

variation with compressor speed is non-linear, an iterative process is carried out until the required convergence criteria is met. The compressor speed may be varied between 30 and 90 Hz.

Once, the required compressor speed to meet the sensible loads is determined, the latent capacity and the latent load to be met are compared. If the latent capacity is not sufficient then the supply fan speed is reduced, if possible, until the AC meets the required latent load. During this process, the compressor speed is increased simultaneously to meet the sensible load. The minimum required air flow rate is updated at each of the compressor speeds to avoid coil freeze up.

If the compressor and fan speeds could not be adjusted to meet the existing low-SHR load then, the availability of a reheat source is checked. If a reheat system is available, the compressor and fan speeds are initially increased to raise the total cooling capacity. Following this, the indoor fan speed is reduced with a smaller step size to meet the latent load. The excess sensible cooling is offset using reheat.

The performance of the VSAC with heat pipes or desiccant wheel and/or reheat can be simulated using this module. As noted earlier, for this study, the dehumidification enhancement components (desiccant wheel, heat pipes) were always locked "ON" with the variable speed air conditioner.

### 6.5.3 Gas-fired Hybrid Systems

One of the advantages of gas-fired hybrid systems (both solid and liquid desiccant systems) is that they can meet the sensible and the latent loads independently (i.e., air conditioner and dehumidifier can have different run-times for the same hour). The configurations that can be simulated with FSEC 2.2 are listed in Table 6.1. The hybrid-systems can be operated in recirculation, ventilation or mixed mode. The air conditioner operation can be in series or in parallel with the dehumidifier. When the AC is in series with the dehumidifier, the return air is mixed with the process air from the dehumidifier and cooled. In the parallel operation mode, the process air from the dehumidifier is mixed with the supply air from the AC.

**TABLE 6.1**

| Modes of operation for Gas-Fired Hybrid Systems | | | |
|---|---|---|---|
| **No** | **Description** | **Process air from** | **Regeneration air from** |
| 1 | Recirculation mode | zone | outdoor |
| 2 | Ventilation mode | outdoor | zone |
| 3 | Mixed mode | outdoor | outdoor |

References


1. Kerestecioglu, A., Swami, M., Kamel, A., Theoretical and Computational Investigation of Simultaneous Heat and Moisture Transfer in Buildings: Effective Penetration Depth" Theory. Presented at ASHRAE Winter Meeting 1990.

2. Gorken Melikian,1990. Modeling for indoor air quality control, United Technologies Research Center, Presented at EPRI Ventilation Workshop, EPRI CU-6972.

3. Kerestecioglu, A., GU. L., Theoretical and Computational Investigation of Simultaneous Heat and Moisture Transfer in Buildings: Evaporation and Condensation Theory" Presented at ASHRAE Winter Meeting 1990.

4. Stephenson and Mitalas, 1972, "Calculation of Heat Conduction Transfer for Mulit-Layer Slabs" ASHRAE Transactions, Vol 78, pp. 117-126.

5. D.C. Hittle, 1981, "Calculation of Building Heating and Cooling Loads Using the Frequency Response of Multi-Layered Slabs" U.S. Army CERL Report, CERL-TM-E-169.

## 7.0 DESCRIPTION OF BUILDINGS CHARACTERISTICS

To compare the performance of the DEAC with the other air conditioning options, the following three building applications were selected.

1. Single family residence (1500 $ft^2$)
2. Medium size office building (10000 $ft^2$) and
3. Fast food restaurant (4000 $ft^2$)

The modeled characteristics of the these three buildings are discussed in this section.

### 7.1 Residence

The residence model used in this study is a 1500 $ft^2$ house with slab-on-grade and thermal coupling to the earth. The residence was treated as having two zones; an attic and the conditioned space. Construction and internal load details are presented in Table 7.1. A schematic of the modeled residence is shown in Figure 7.1. Thermal and moisture capacitance effects of typical furniture, beds and other miscellaneous equipment in the residence were included in the model. The internal gain and the infiltration schedules for the residence modeled are presented in Appendix G.

### 7.2 Office Building

The medium size office building model is a 10,000 square foot slab-on-grade, single story building. The construction and internal load details are presented in Table 7.2.

The recent trend towards using extensive electronic equipment in modern offices has increased the sensible heat generation. Nevertheless, office equipment loads have been consistently under-estimated according to reference (1). A recent study (2) on office equipment loads showed that peak equipment gain of 2.5 $W/ft^2$ is a nominal value for a modern office building where desk top computers, laser printers and copiers are widely used. Other literature surveys (3,4) showed that, office equipment load levels range from 0.75 to 2.0 $W/ft^2$.

Electrical lighting load is another important factor required to estimate the AC loads for office buildings. Most of the studies and the metered data suggest a peak value of 1.7 to 2.0 $W/ft^2$ for the office lighting load (3,8). However, with the high-efficiency ballistic lights, the load can be reduced as low as 1.4 $W/ft^2$.

The total equipment and lighting loads assumed in our study was 3.3 $W/ft^2$. This value is representative of the average internal gains of modern office buildings which range from 2.75 $W/ft^2$ to 3.6 $W/ft^2$. (3)

Equipment, people, ventilation, lighting, and temperature and humidity set points schedules modeled in our simulations are listed in Appendix G. These schedules are based on the ASHRAE standard IES 90.1 (5).

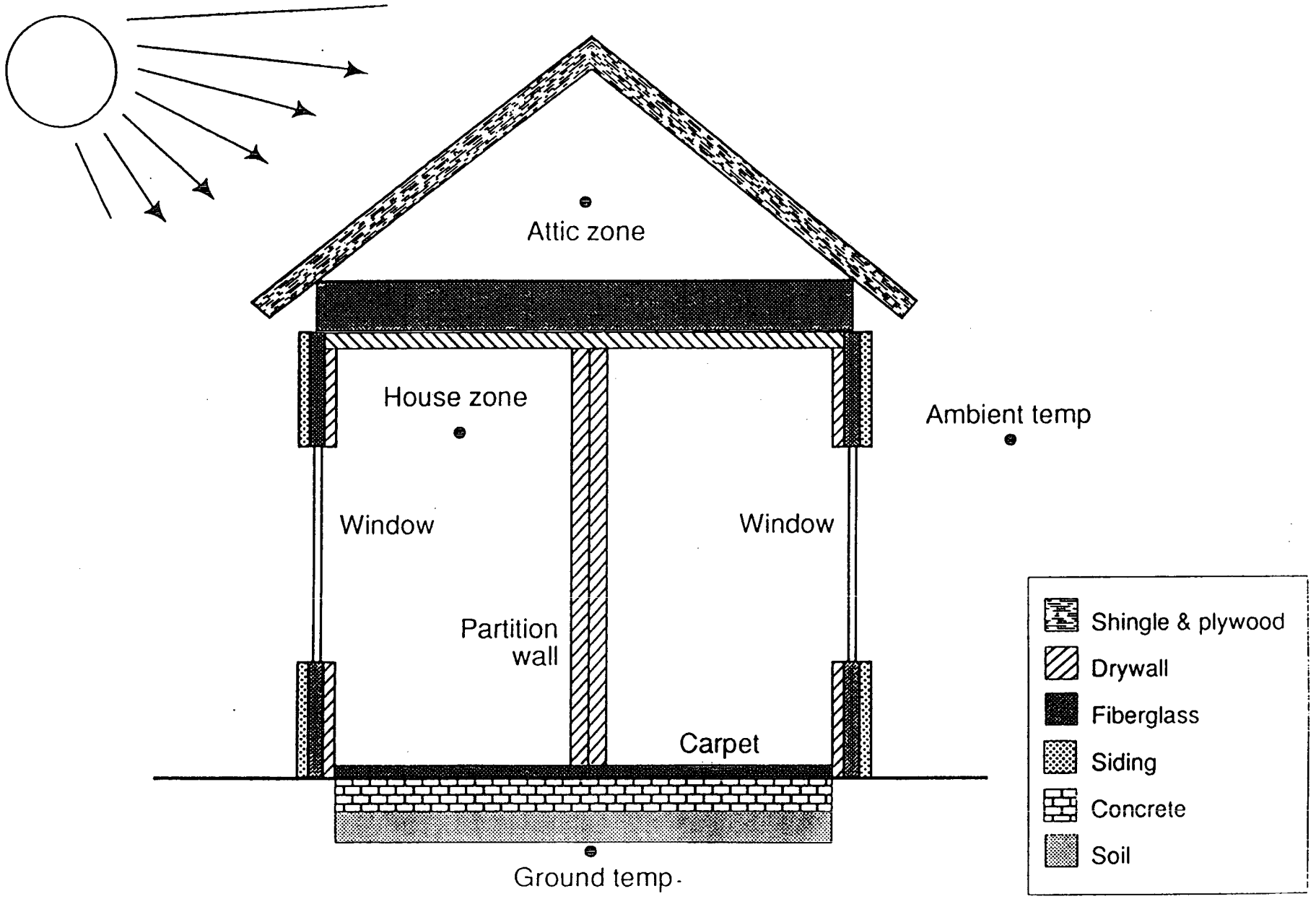



Figure 7.1 - Schematic of Residence

**TABLE 7.1**

| Residence Characteristics | |
|---|---|
| Geometry: | |
| Conditioned Floor Area ($ft^2$) | 1500 |
| Dimensions | 50 ft x 30 ft |
| Number of Stories | 1 |
| Number of Zones | 2 |
| Ceiling Height (ft) | 8 |
| Construction: | |
| Floor | Carpet over 4" concrete slab |
| Walls | R-11 (0.5" drywall, 3.25" fiber glass and 0.5" Masonite Wall) |
| Partition Walls | 1" drywall |
| Ceiling | 0.5" drywall |
| Roof | R-24 (6" fiberglass insulation, 0.5" shingle and 0.5" plywood) |
| Percent of Glass in Wall: | 17.5% equally distributed |
| Glazing Type | Single Pane |
| Internal and External Building Loads: | |
| Number of Occupants (Peak) | 3 |
| Heat Gain from Occupancy | |
| Sensible (Btu/hr per person) | 230 |
| Latent (Btu/hr per person) | 190 |
| Indoor Lighting + Equipment ($W/ft^2$) | 0.75 (SHR: 0.77) |
| Infiltration (air changes/hour) | 0.54 average, 0.72 peak |
| Operating Conditions: | |
| Cooling Temperature Set point | 78°F |
| Humidity Set point | 50% RH |

## TABLE 7.2

| Medium Size Office Building Characteristics | |
|---|---|
| Geometry: | |
| Conditioned Floor Area ($ft^2$) | 10,000 |
| Dimensions | 100 ft x 100 ft |
| Number of Stories | 1 |
| Number of Zones | 1 |
| Ceiling Height (ft) | 9 |
| Partition Walls | 7500 $ft^2$ |
| Construction: | |
| Floor | Carpet over 4" concrete slab |
| Walls | R-11 (0.5" drywall, 3.25" fiberglass and 0.5" Masonite Wall) |
| Ceiling | 0.5" drywall |
| Roof | R-24 (6" fiberglass insulation and built-up roof) |
| Partition | Fabric Cover (estimated equivalent properties of carpet). |
| Percent of Glass in Wall: | 20% equally distributed |
| Glazing Type | Single Pane |
| Internal and External Building Loads: | |
| Number of Occupants (Peak) | 60 |
| Heat Gain from Occupancy | |
| Sensible (Btu/hr per person) | 250 |
| Latent (Btu/hr per person) | 200 |
| Indoor Lighting + Equipment ($W/ft^2$) | 3.3 (SHR: 1.0) |
| Ventilation | 20 cfm/person(6) |
| Infiltration (air changes/hour) | 0.228 (when AC is "off") |
| Operating Conditions: | |
| Cooling Temperature Set point | 76°F |
| Humidity Set point | 50% RH (occupied)<br>60% RH (unoccupied) |

### 7.3 Fast Food Restaurant

The fast food restaurant model used in our study is based on the extensive research carried out by the Electric Power Research Institute (EPRI) (7) in characterizing modern electric restaurants. The construction details and the internal loads used for our model restaurant are presented in Table 7.3.

The weekday and weekend schedules of equipment, occupancy, lighting, and temperature and humidity set points modeled for the fast food restaurant are shown in Appendix G.

References


1. R.G. Pratt., "Errors in Audit Predictions of Commercial Lighting and Equipment Loads and their impacts on Heating and Cooling Estimates", ASHRAE Transactions, 1990, PP. 994-1004.

2. Christopher K.Wilkins, Risto Kosonen and Tuomas Laine," An analysis of office equipment load factors", ASHRAE Journal, September 1991.

3. R.S. Briggs, D.B. Crawley and J.S. Schliesing. 1992. "Energy Requirements for office buildings" Volume 1

4. ASHRAE Handbook - Fundamentals, 1989, I-P edition.

5. ASHRAE standard IES 90.1, 1989, "Energy Efficient Design of New Buildings Except New Low-Rise Residential Buildings".

6. ASHRAE Standard 62-1989, "Ventilation for Acceptable Indoor Air Quality".

7. The Model Electric Restaurant Volume:1 - EPRI CU 6702 Prepared by Architectural Energy Corporation Boulder, Colorado and The Pennsylvania State University, Pennsylvania.

8. T.Alereza, Dohrmann D.R., Martinez M., and Mort D., "End-Use Metered data for commercial buildings", ASHRAE Transactions, 1990, PP. 1004 - 1010.

## TABLE 7.3

| Fast Food Restaurant Characteristics | | |
|---|---|---|
| Geometry: | | |
| Conditioned Floor Area ($ft^2$) | 4,000 | |
| Dimensions | 80 ft x 50 ft | |
| Number of Stories | 1 | |
| Number of Zones | 2 | |
| Ceiling Height (ft) | 10 | |
| Construction: | | |
| Floor | Tiles over 4" concrete slab | |
| Walls | R-11 (4" brick wall, 3.25" fiberglass and 0.5" Gypsum) | |
| Ceiling | 0.5" drywall | |
| Roof | R-11 (1/2" plywood, 4" fiberglass insulation and built-up roof) | |
| Percent of Glass in Wall: | North Wall: 35% South Wall: 32%<br>East Wall: 0% West Wall: 45% | |
| Glazing Type | Single Pane | |
| | **Dining** | **Kitchen** |
| Internal and External Building Loads: | | |
| Number of Occupants (Peak) | 40 | 6 |
| Heat Gain from Occupancy | | |
| Sensible (Btu/hr per person) | 275 | 275 |
| Latent (Btu/hr per person) | 275 | 275 |
| Indoor Lighting ($W/ft^2$) | 1.09 | 2.4 |
| Equipment Load (W) | 0.0 | 9600 (Peak) |
| Equipment SHR | | 0.6 |
| Ventilation (cfm) | 4000 | 2000 |
| Infiltration (air changes/hour) | 0.216 | 0.216 |
| Operating Conditions: | | |
| Cooling Temperature Set point | 75°F | 80°F |
| Heating Set point | 72°F | 72°F |
| Humidity Set point | 50% RH | 50% RH (Occupied) |

## 8.0 BUILDINGS LOADS ESTIMATION AND AC SYSTEMS SIZING

In this Chapter, we describe how each of the AC systems modeled for this study was sized for the various buildings and climates. In summary, the preliminary building loads for each climate were determined using a hypothetical air-conditioner (an "Ideal" AC) which perfectly matches both the sensible and the latent hourly loads. Peak hourly sensible and latent loads for each of the building/climate matrices were determined using these ideal AC performance results.

HVAC systems are typically sized to meet the building loads when the outdoor temperature is at the 98th percentile. In accordance, our AC systems were sized to meet the sensible and latent loads that occurred during 98 percent of the cooling season. It should be noted that in all cases, once the required size of the AC system was determined based on the above, the nearest standard sizes of the AC units currently available were selected and used in the simulations.

### 8.1 Ideal Loads

The annual cooling loads (sensible + latent) determined from the "ideal" AC simulations are referred to as ideal loads in the following discussions. In each case, simulations were performed for the whole year to identify the cooling season for each climate and building type. TMY weather data for each of these climates were used for the analysis. Monthly average weather conditions of Miami, Washington and St.Louis are presented in Appendix C.

Figures 8.1, 8.2 and 8.3 show the ideal annual sensible and latent cooling loads for the modeled residence, medium size office building and fast food restaurant respectively. The fast food restaurant is treated as having independent dining and kitchen loads. The combined load is shown in Figure 8.3.

We earlier indicated that the actual AC systems were sized to meet the sensible and latent loads occurring below the 98th percentile levels of the cooling hours. These "98 percentile level loads" are referred to as "design" loads in the following discussions.

The design sensible and latent loads for the residence are shown in Figure 8.4 for the three climates modeled. The design loads for Washington and St.Louis residences were comparable and therefore, the same size residential AC system was used in the simulations of these two climates.

Figure 8.5 shows the design loads for the medium office building. The office building internal load was dominated by sensible load. The design loads for all the three climates were close to each other. In this case, we used the same size AC systems for all three climates.

As noted above, dining and kitchen zones of the fast food restaurant were modeled as two independent zones controlled by two independent AC systems. The design loads for the dining and kitchen zones are shown in Figures 8.6 and 8.7. Fast food restaurant loads are typically weather driven due to the large amount of outside air entering the zones. Different

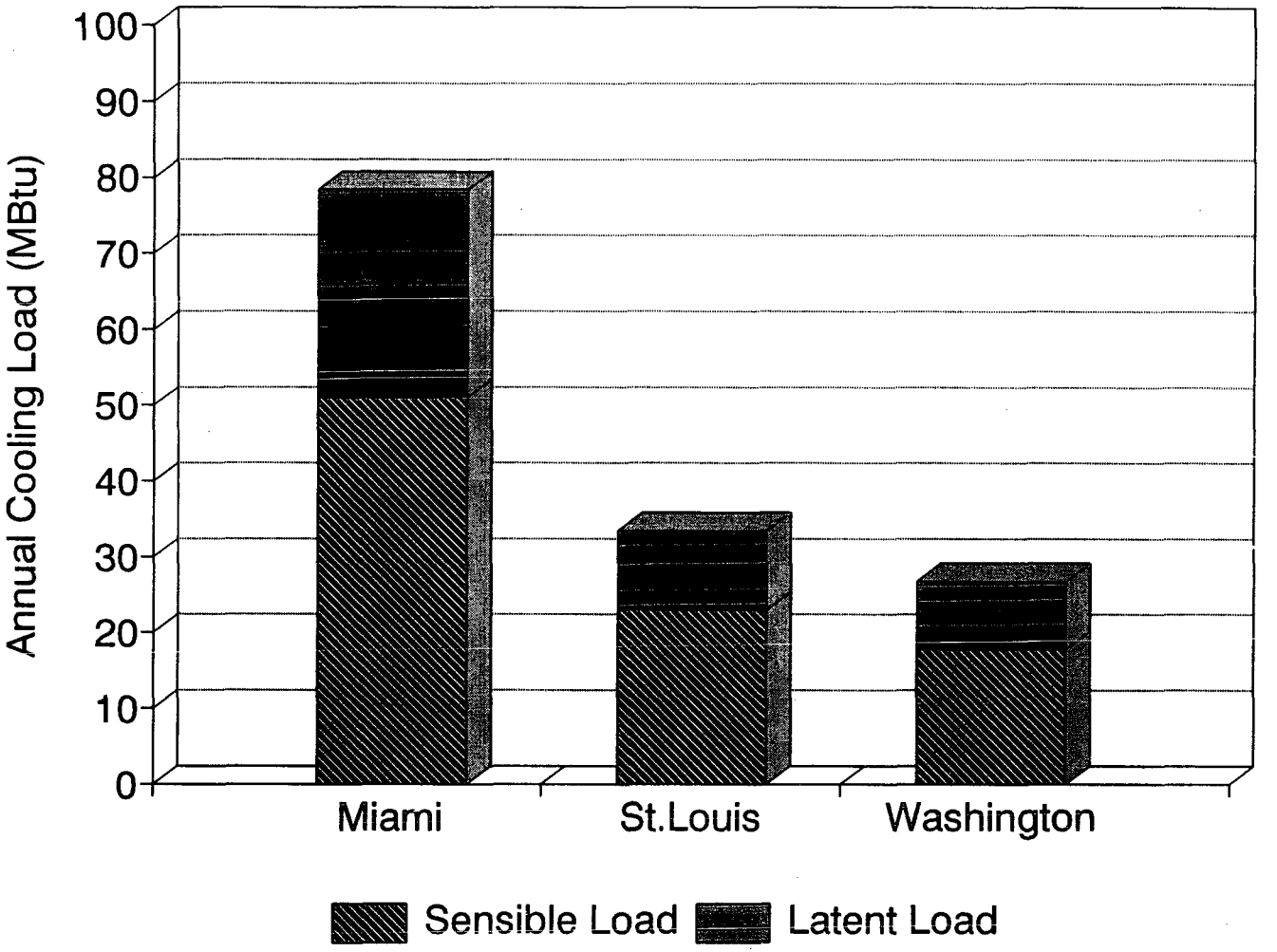



Figure 8.1 - Sensible and Latent Loads for a Residence

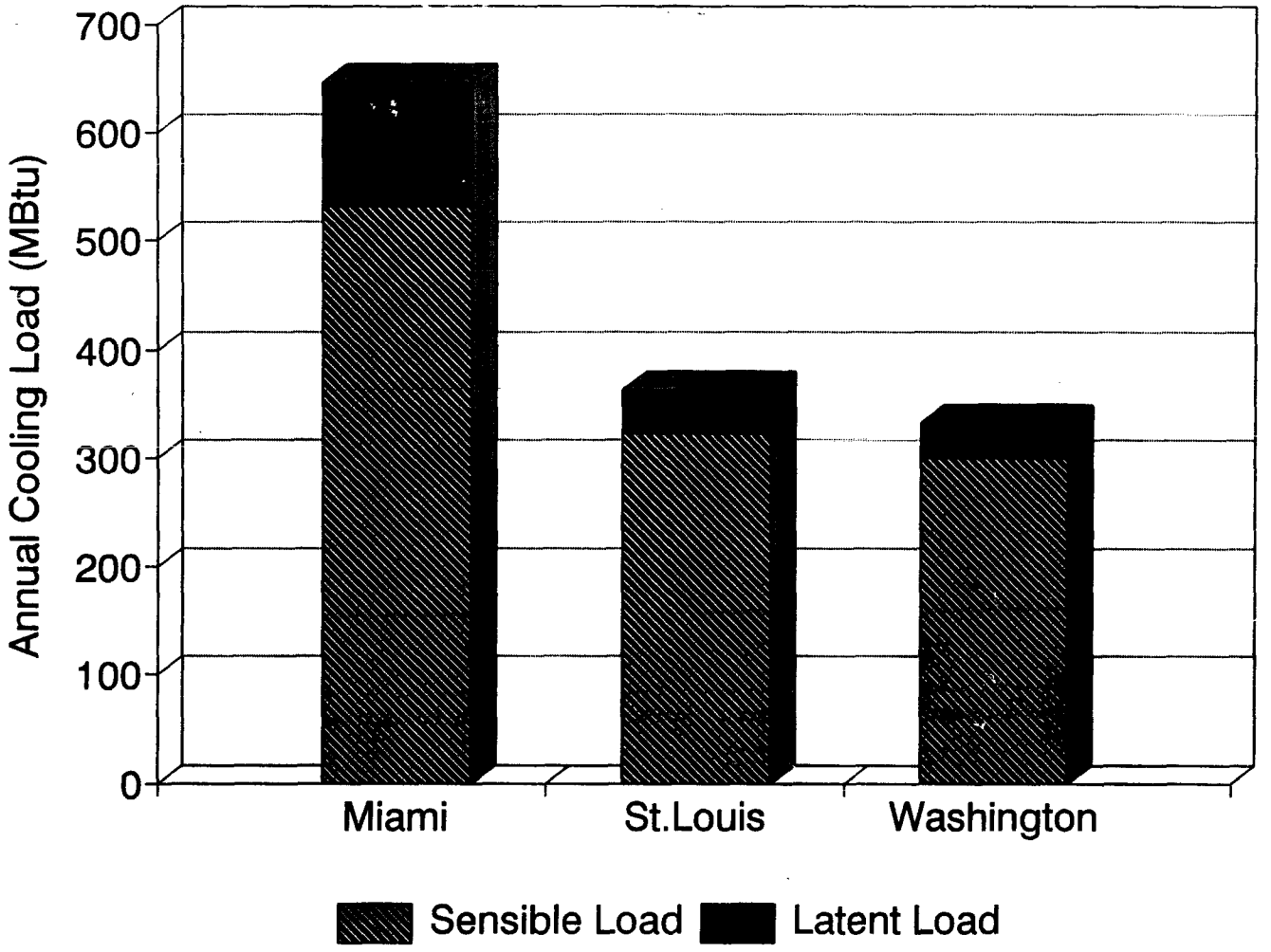



Figure 8.2 - Sensible and Latent Loads for a Medium Office Building

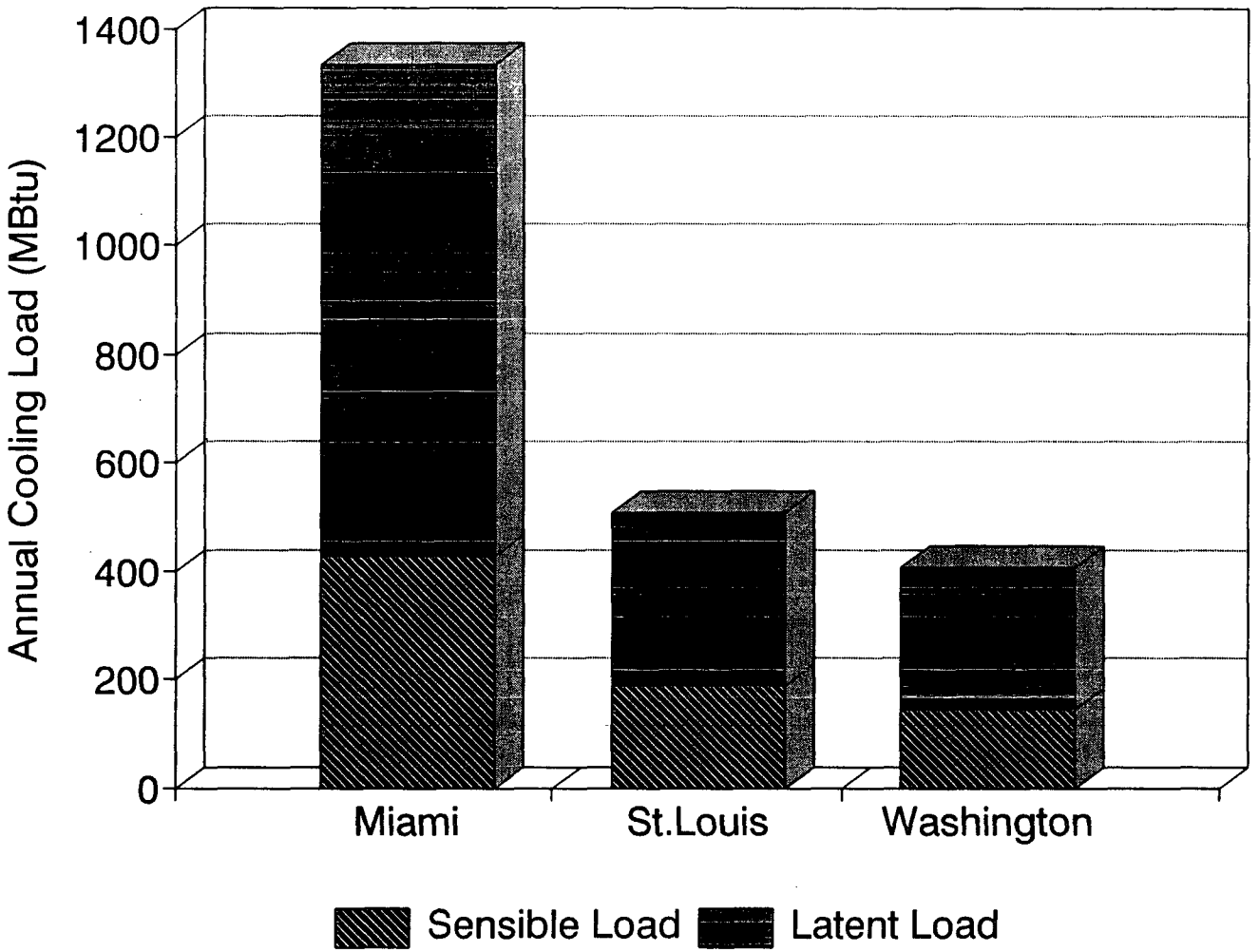



Figure 8.3 - Sensible and Latent Loads for a Fast Food Restaurant

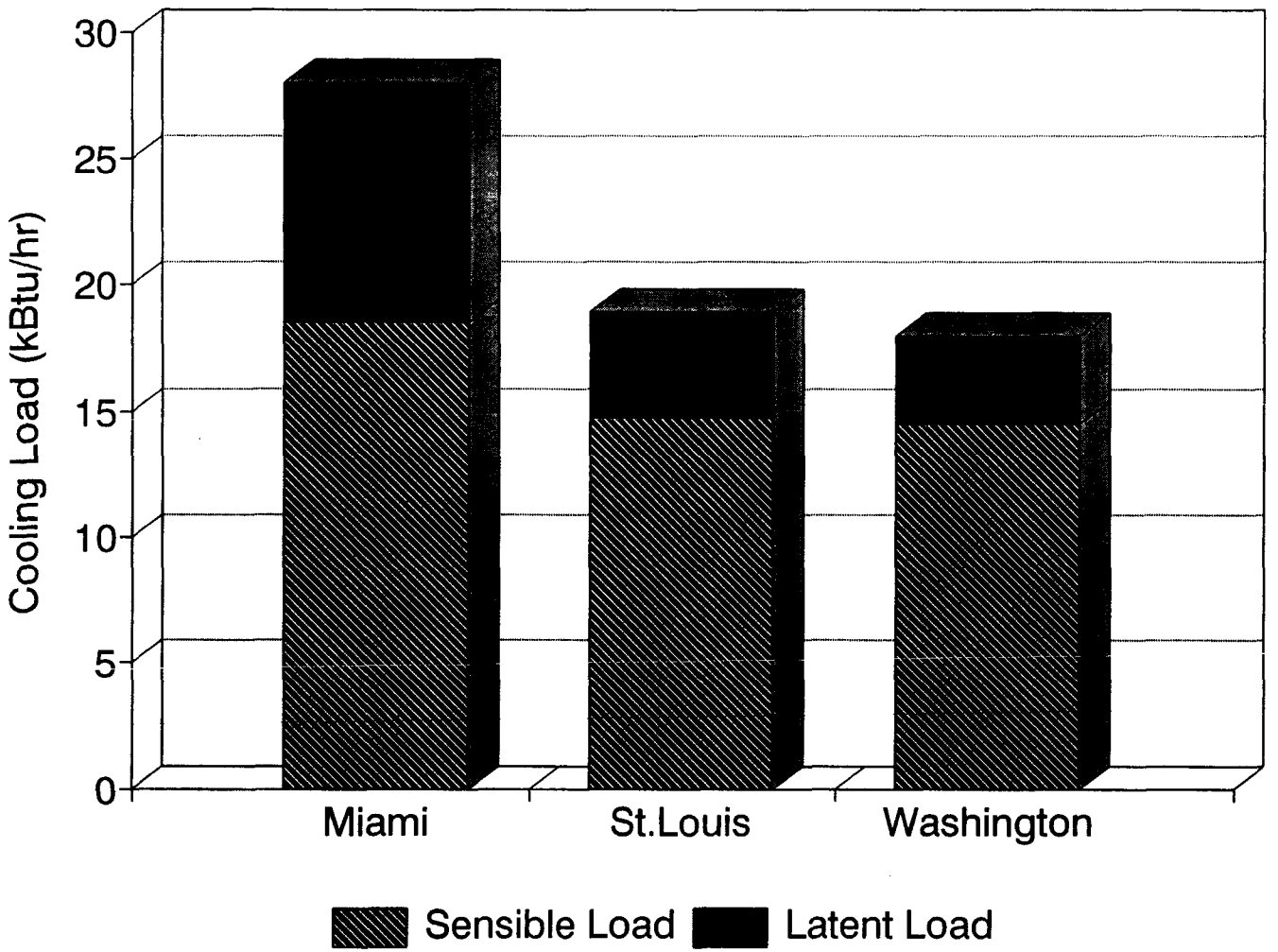



Figure 8.4 - Design Peak Loads: Residence

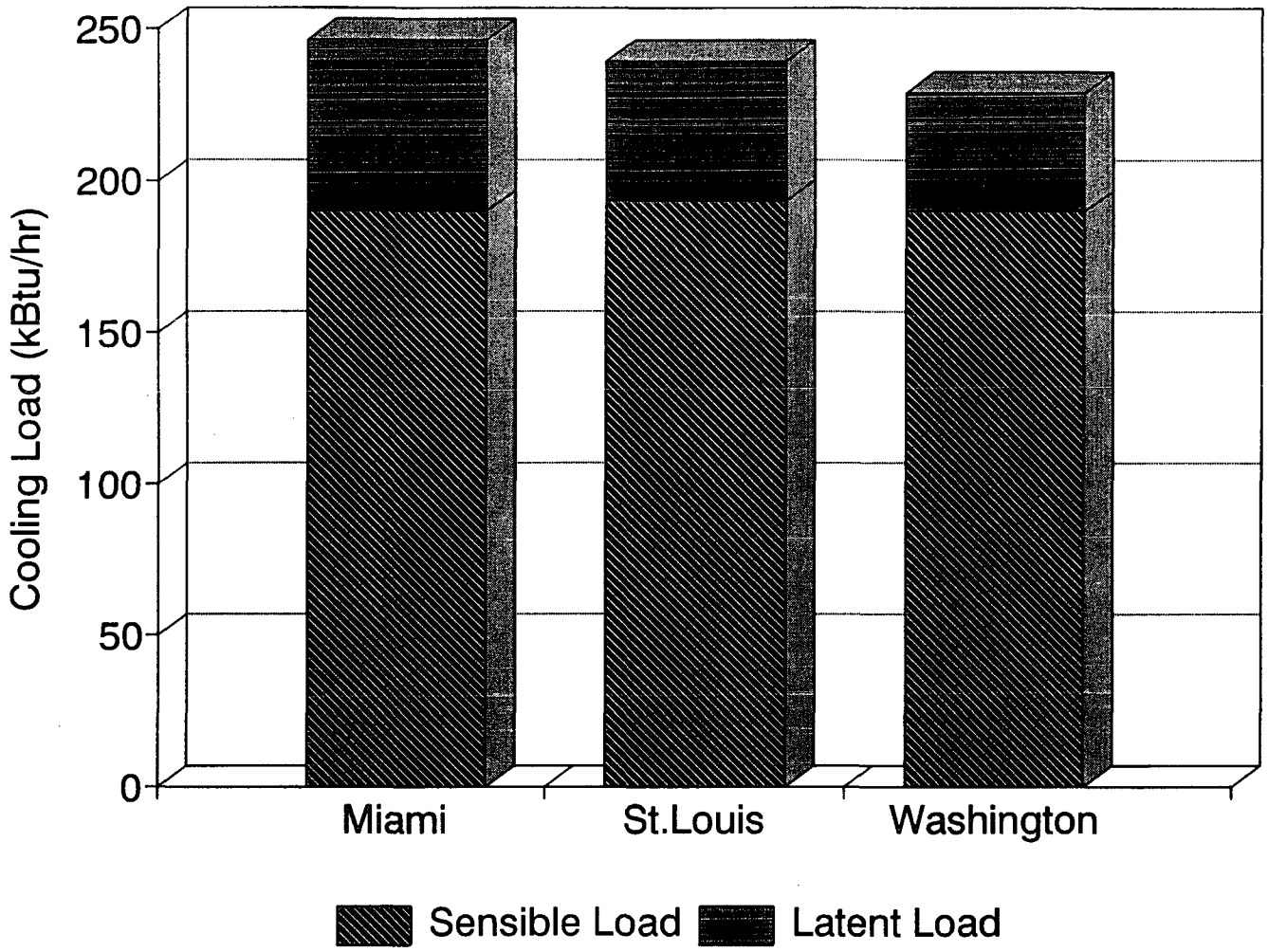



Figure 8.5 - Design Peak Loads: Medium Office Building

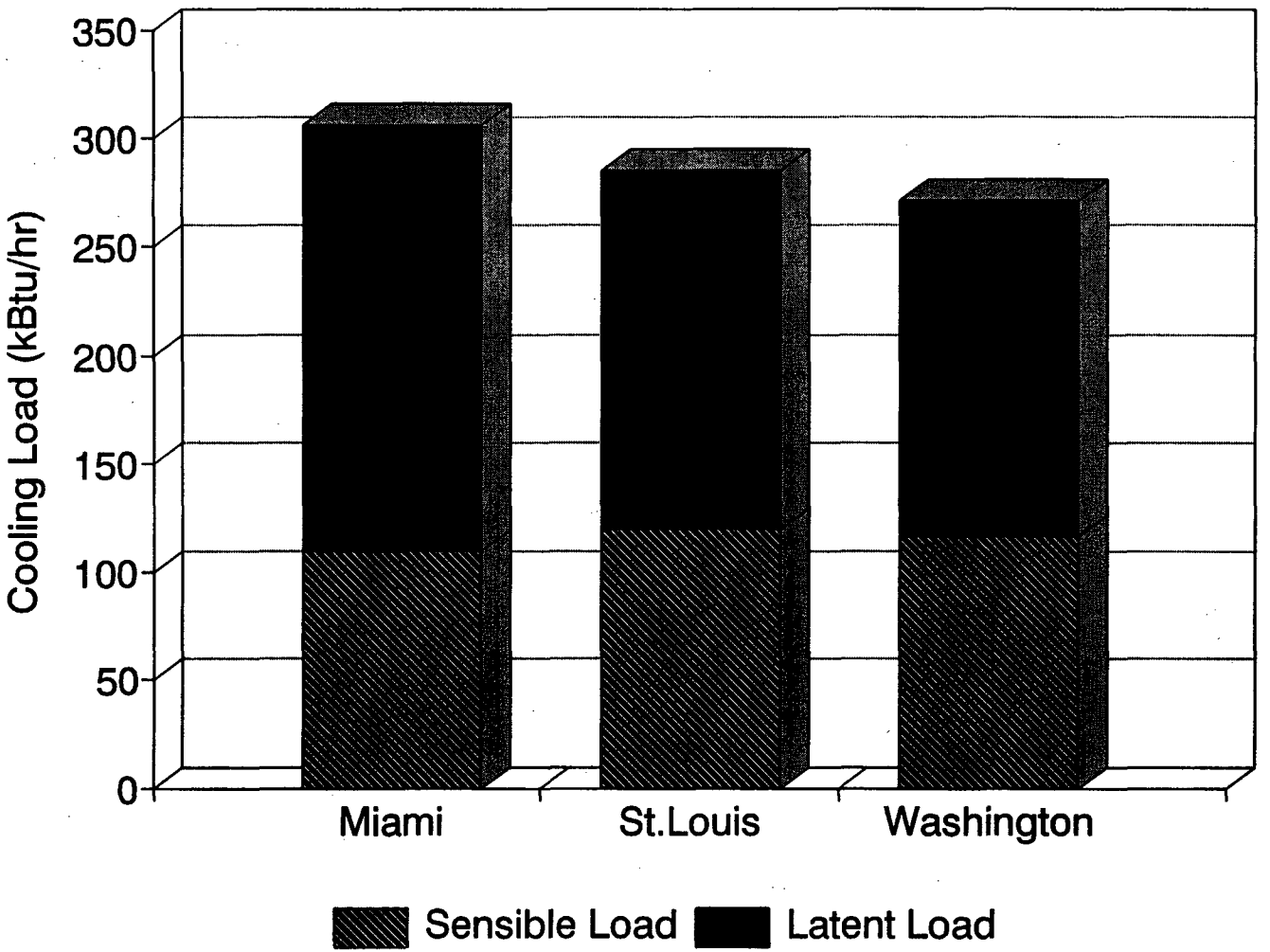



Figure 8.6 - Design Peak Loads: Fast Food Restaurant (Dining)

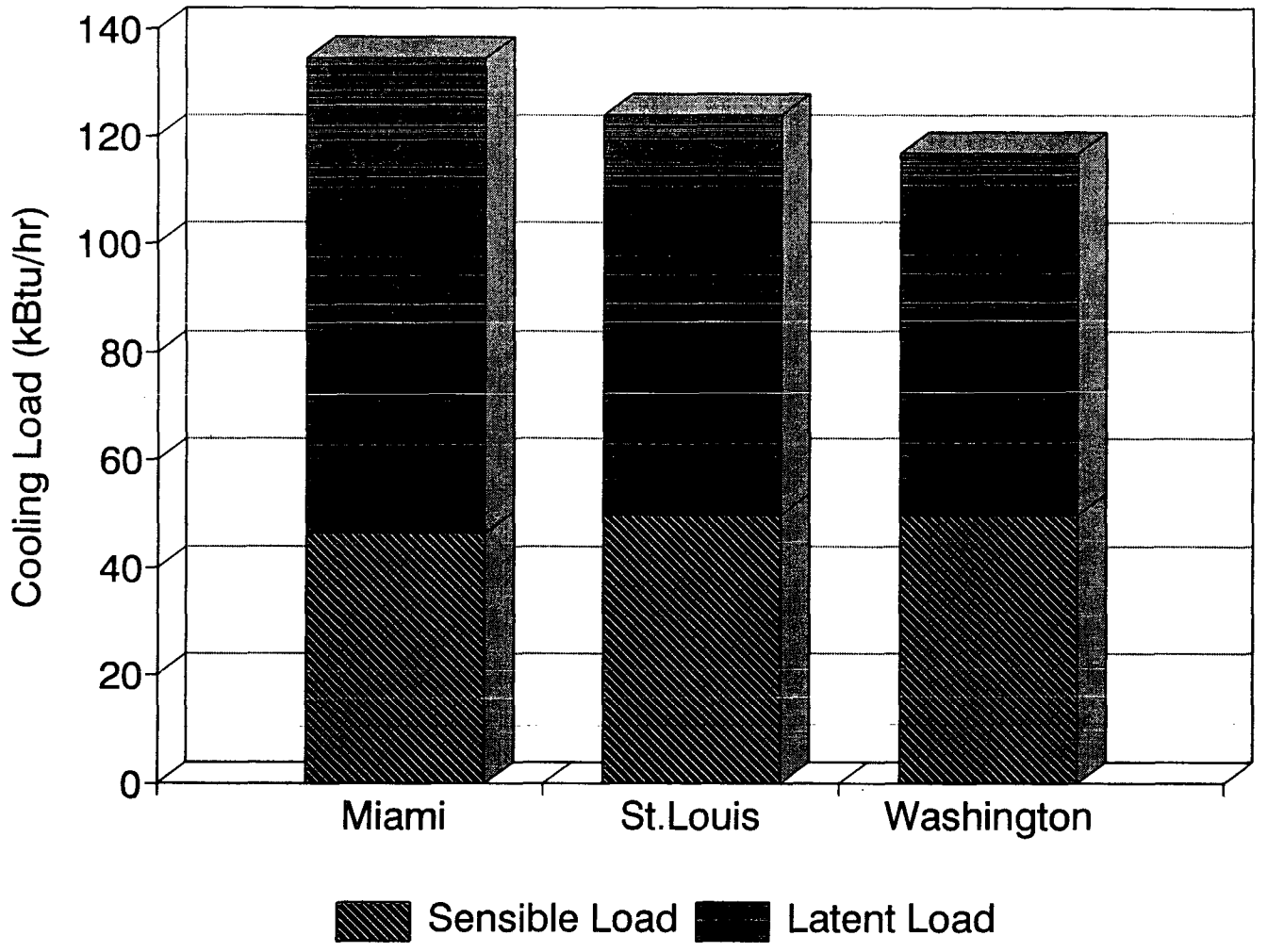



Figure 8.7 - Design Peak Loads: Fast Food Restaurant (Kitchen)

sizes of AC systems were required for the three climates due to the large differences in building design loads for this application.

## 8.2 Load Sensible Heat Ratio

In addition to determining the design sensible and latent loads used for sizing the AC systems, the load-SHR variations over the year were also taken into account. Detailed histograms were developed to study the load-SHR profile of the buildings. The modeled Miami residence load-SHR histogram is shown in Figure 8.8. It can be noted from Figure 8.8 that the load-SHR ranges from 0.4 to 0.9 for the modeled Miami residence over the cooling season. Average monthly load-SHR variations for the Miami residence and office building are shown in Figures 8.9 and 8.10 respectively.

AC systems with air reheat (e.g., electric reheat) have the capability to match essentially any combination of sensible and latent loads. However, most other AC systems (e.g., AC with heat pipes) do not have this limitless flexibility and therefore are not capable of meeting extremely low SHR values which sometimes occur during the cooling season. Consequently, for design purposes it is necessary to select some reasonable cut off point beyond which the AC system will not meet the latent load. Somewhat arbitrarily, we selected this point such that the hourly load-SHR will be met 80 percent of the time during the cooling season.

## 8.3 Applicability of AC Types to Buildings

Each building type was modeled in each climate area. However, the application of a given air-conditioning system to a given building/climate set was based on engineering judgement. For example, electric air-reheat control was considered with all electric AC systems for the fast food restaurant application to meet the low-SHR loads whereas it was not considered for the residential application because reheat is not used in residences. The air conditioning systems compared for each of the above building applications are listed in Table 8.1.

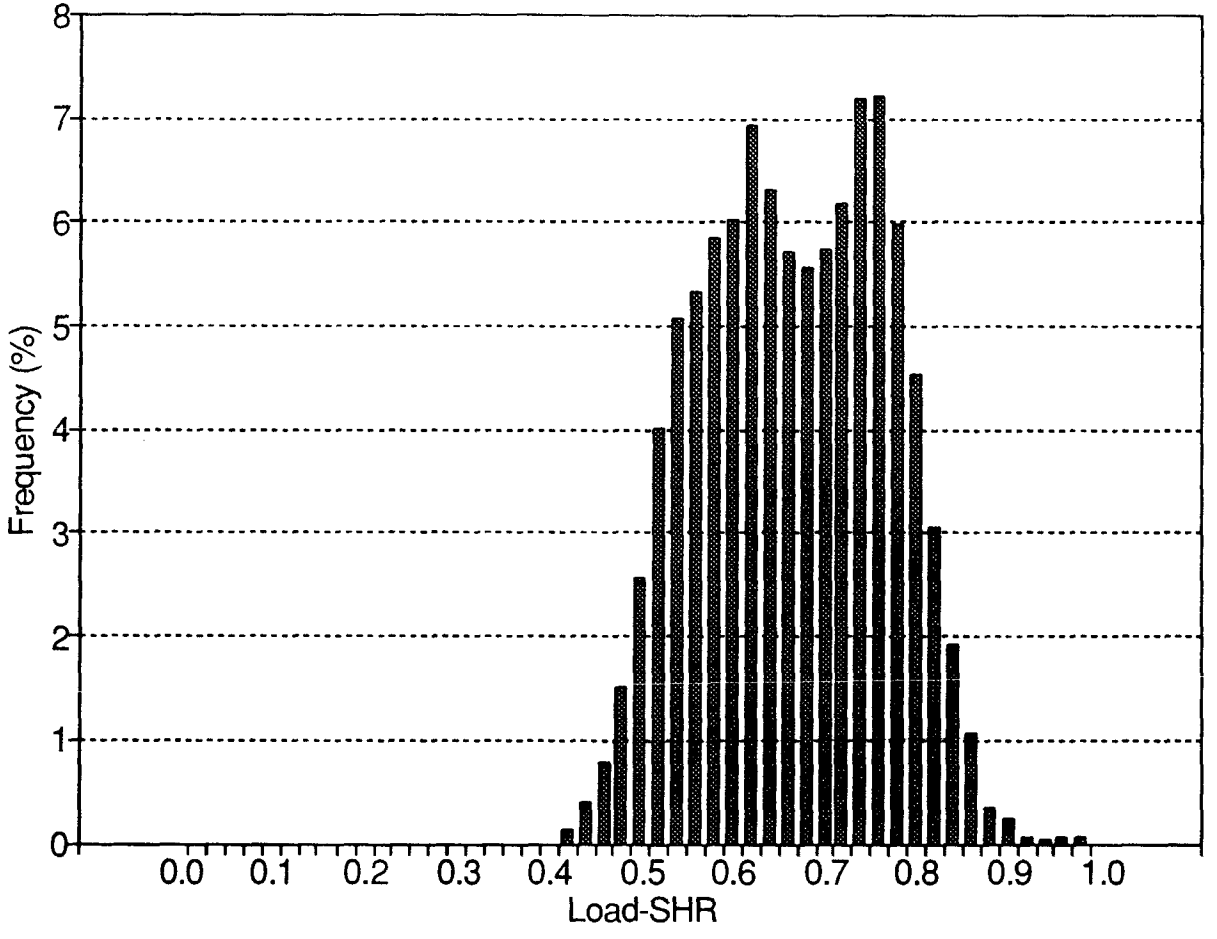



Figure 8.8 SHR Histogram for Miami Residence

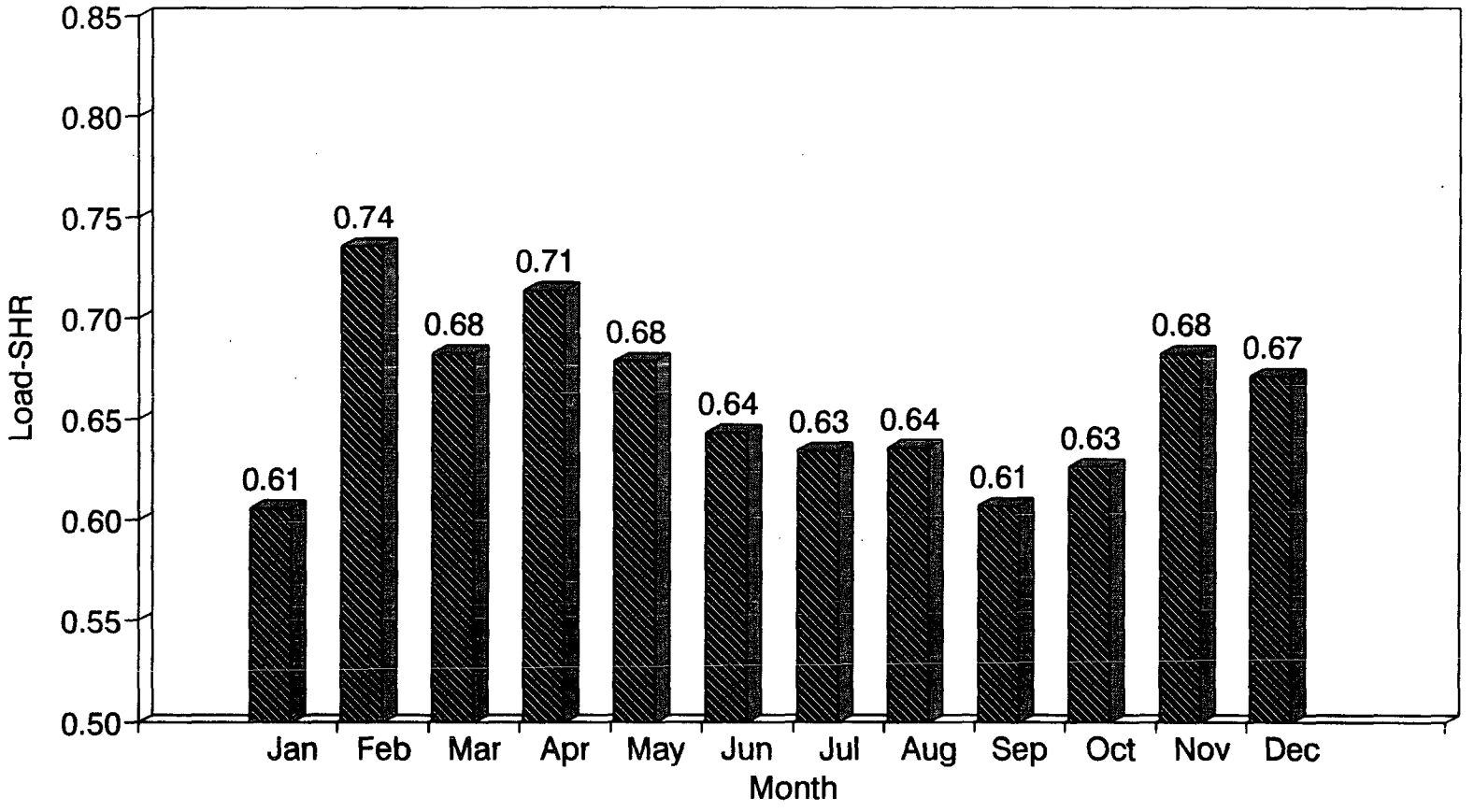



Figure 8.9 - Monthly Load-SHR Profile for Miami Residence

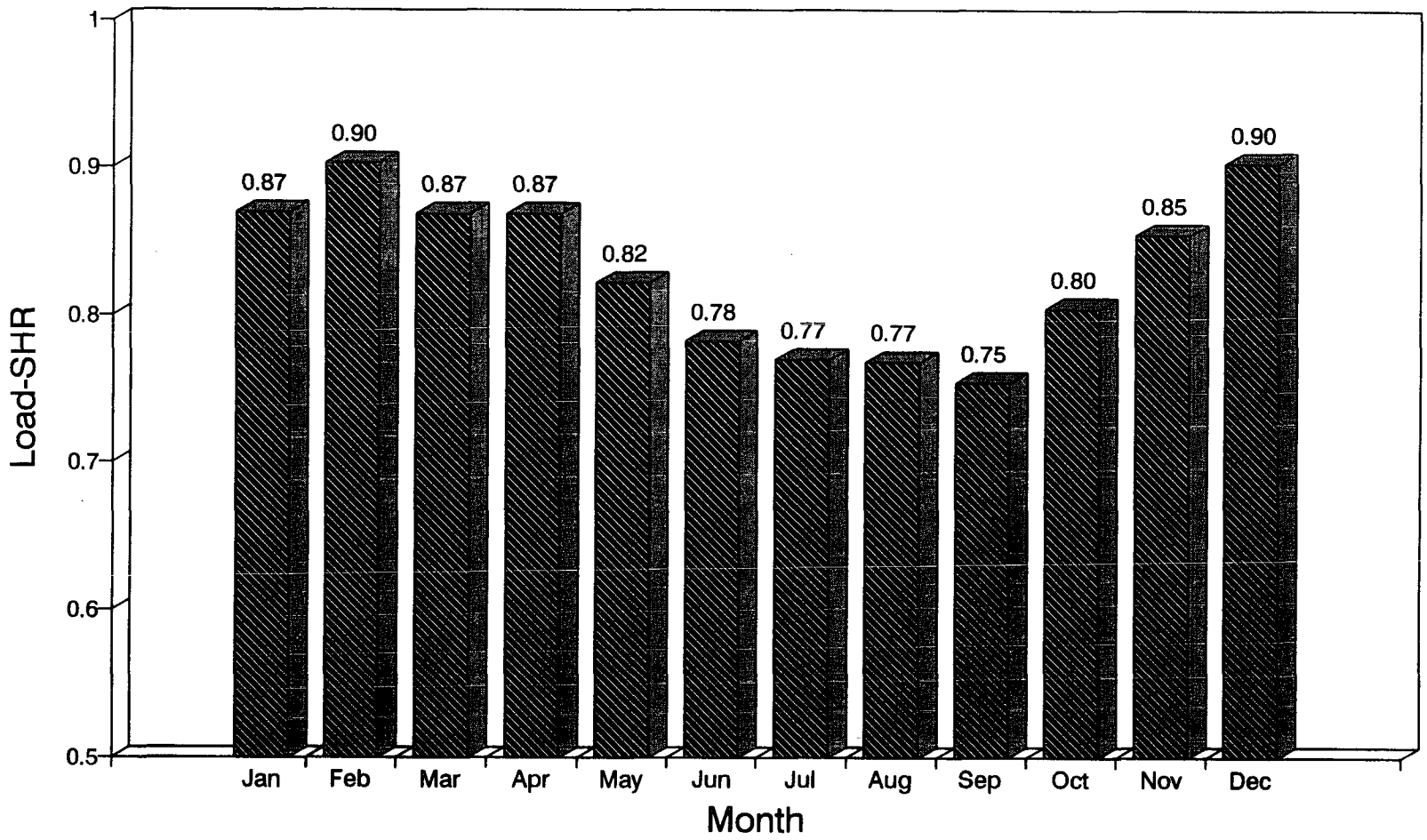



Figure 8.10 - Monthly Load-SHR Profile for Miami Office Building

TABLE 8.1

| AC Systems Compared Using Seasonal Simulations | | | |
|---|---|---|---|
| **AC System** | **Residence** | **Office** | **Fast Food Restaurant** |
| Single Speed Air Conditioner (SSAC) | Yes | Yes[1] | Yes[1] |
| SSAC with Heat Pipes | Yes | Yes | Yes[1] |
| Desiccant Enhanced Air Conditioner (DEAC) | Yes | Yes | Yes[1] |
| Variable Speed Air Conditioner (VSAC) | Yes | Yes | Yes[1] |
| VSAC with Heat Pipes | Yes | No | Yes[1] |
| VSAC with Desiccant wheel | Yes | No | Yes[1] |
| SSAC with Gas-fired Desiccant Wheel | Yes | Yes | Yes |
| SSAC with liquid Kathabar Dehumidifier | No | Yes | Yes |

Note [1] Electric reheat was available for additional dehumidification.

## 8.4 Sizing of AC Systems

As would be expected, the AC systems compared (Table 8.1) did not have the same capability to meet sensible and latent loads. Hence, each of the systems had to be individually sized based on its potential to meet the loads. Details of the sizing are provided in the following Sections.

### 8.4.1 Supply Air Flow Rate

For a typical air conditioner, the total capacity and SHR increase with an increase in air flow rate over the cooling coil. On the other hand, at low air flow rates, the total capacity decreases but the moisture removal capacity increases. Hence, based on the building loads and type of AC system, an optimum air flow rate was used to minimize capital costs and maximize the dehumidification potential of the system.

Typical supply air flow rates over the evaporator coil range from 300 to 400 cfm per ton for single speed air conditioners. A nominal supply air flow rate of 400 cfm per ton was used for the residence and the office building AC systems.

The average load-SHR value of the fast food restaurant was on the order of 0.4. To better meet this low load-SHR with a single speed electric AC system, a lower air flow rate of 300 cfm per ton was used for the fast food restaurant. At this same air flow rate, simulation results for the heat pipe-assisted AC system showed a few hours of DX coil freezing. As a result, a nominal air flow rate of 400 cfm per ton had to be maintained for the heat pipe-assisted AC. Gas-fired hybrid systems have the capability to meet the sensible and the latent loads independently. Hence, a nominal supply air flow rate of 400 cfm per ton was used over the AC cooling coil for the hybrid system.

An appropriate duct pressure drop was determined for each building type. The supply air flow rate varies with the type of the AC system used and it's size for each building type. In our simulations, the building duct pressure drop was varied for each of the AC systems based on the supply air flow rates. This allowed a common duct size to be used for any AC option.

### 8.4.2 Single-Speed AC and Air-Reheat Systems

Conventional single-speed air conditioners operate on thermostat control. Hence, they were sized to meet only the design sensible loads for all the buildings. The air-reheat systems were sized to meet both the sensible and latent design loads.

### 8.4.3 Single-Speed AC with Heat Pipes or a Desiccant Wheel

Single-speed air conditioners have increased dehumidification capacity when equipped with heat pipes or a desiccant wheel. However, unlike an air-reheat system, they do not have the capability to meet all combinations of sensible and latent loads that occur over the cooling season. Hence, these systems were sized to meet the 98 percent occurrence sensible load and 80 percent occurrence load-SHR value for the residence and the office building. To meet the low-SHR loads occurring in fast food restaurants, reheat control was added to these systems. As a result, restaurant AC systems could be sized to meet both sensible and latent design loads.

### 8.4.4 Variable-Speed AC with Heat Pipes or a Desiccant Wheel

Variable-speed air conditioners meet both sensible and latent loads by modulating compressor and supply fan speeds. Variable speed units are rated by manufacturers at a variety of compressor speeds. For example, the Hydrotech 2000 variable speed units are rated at compressor speeds of 4400 and 5400 RPM for 2 ton and 3 ton units respectively (1). In our study, variable-speed air conditioners were initially sized to meet the design sensible loads at a compressor speed of 5400 RPM (90 Hz).

Based on reference (2), variable speed units typically operate at speeds of 50 to 70 Hz in the cooling mode. Our initial simulations indicated that, VSAC systems (VSAC with or without heat pipes or a desiccant wheel) were operating much of the time at a less efficient speed (e.g., greater than 70 Hz). As a result, we increased the sizes of the VSAC systems to operate in more energy efficient mode.

### 8.4.5 Gas-fired Hybrid Systems

Gas-fired hybrid systems have the capability to meet sensible and latent loads independently. These systems are typically controlled by both thermostat and humidistat. Consequently, it was possible to size these units to meet both the sensible and latent design loads.

Reference


1. Product data, Carrier Hydrotech 2000 - Advanced Technology Heat Pump System.

2. John Kesserling and Arvo Lannus, "Field Testing of the Hydrotech 2000 Heat Pump", EPRI Journal, December ,1991.

## 9.0 RESIDENCE AIR CONDITIONING ENERGY ANALYSIS

Previous Chapters have presented the background material and assumptions for a comparative study of air conditioner performance in three types of buildings in three climates. In this Chapter, the results of the residence simulations are presented in terms of energy use, capital costs and comfort. A single criteria is presented which brings these three issues together into a merit index.

Based on the relatively high performance of the DEAC in the initial three cities (Miami, St.Louis and Washington D.C.,), additional simulations were run to determine the DEAC region of highest applicability and the potential for DEAC energy savings in that region.

### 9.1 Introduction

Household air conditioning energy use in the U.S. ranges from 5 to 17 million Btu per year (1). Air conditioning energy use represents 5 percent of the total household energy use at the national level (2), but it is a much higher percentage in the hot and humid southeastern regions. High humidity problems are prevalent in this area of rapid residential growth. In Florida, for example, approximately 50 percent of the energy costs in summer months go for creating comfortable indoor conditions. The above data provide substantial incentive to seek out approaches which would more effectively control and maintain healthy and comfortable residential indoor environments. The DEAC is proposed as one such approach.

The parametric study results of Chapter 5 showed that the DEAC could maintain a higher EER over a wide range of SHR values compared to other electric AC options. In order to identify the potential benefits (e.g., energy saving, additional comfort, etc.) of the DEAC over the cooling season, its performance was compared with other AC options for a modeled residence using computer simulations. The air conditioning systems compared for the residence are listed in Table 9.1. The characteristics of the modeled residence were described earlier in Section 7.1. The results of these seasonal simulations are discussed in Sections 9.4 to 9.7.

**TABLE 9.1**

| Residence: AC Systems Compared |
|---|
| 1. Conventional single-speed air conditioner (SSAC) |
| 2. SSAC with Heat pipes |
| 3. DEAC |
| 4. Variable-speed air conditioner (VSAC) |
| 5. VSAC with heat pipes |
| 6. VSAC with desiccant wheel |
| 7. SSAC with gas-fired desiccant wheel |

## 9.2 Simulation Basis

Residential cooling equipment has been primarily controlled by temperature-sensing thermostats. Recent technologies, such as the VSAC, includes humidity sensors to enhance control of indoor comfort. In our simulations, single speed AC systems were controlled by a thermostat while the VSAC system had a humidistat to control fan speed and a thermostat to control the compressor. The desiccant wheel gas-fired hybrid system also had both humidistat and thermostat controls.

The results of the ideal load simulation for the modeled residence indicated an average annual load-SHR value ranging from 0.65 to 0.68 for the three climates. Residential AC systems with SHR values in the range of 0.7 to 0.75 are readily available in the market. The ORNL air conditioner model described earlier in Section 4.2 has an SHR of 0.73 at ARI rated conditions (indoor drybulb: 80°F, indoor wetbulb: 67°F, Outdoor drybulb: 95°F) and used as the baseline AC for the residential simulations. Electric reheat is not normally employed for humidity control in residences and therefore it was not included as an option in Table 9.1.

Seasonal performance simulations were conducted for each of the AC systems (Table 9.1) using TMY weather data for the appropriate city. The results of these simulations are discussed in Section 9.4.

### 9.2.1 Size of AC Systems

The residential air conditioning systems were sized using the procedure described in Chapter 8. The AC system sizes used for the residential simulations are listed in Table 9.2.

**TABLE 9.2**

| Residence: AC Systems Size | | |
|---|---|---|
| Description | AC Capacity (Tons) Miami | AC Capacity (Tons) Washington and St.Louis |
| Single Speed AC | 2.0 | 2.0 |
| Single Speed AC with Heat pipes (3 Rows, 14 fpi) | 2.5 | 2.0 |
| DEAC (Wheel depth: 2.1") | 2.5 | 2.0 |
| VSAC (@90 Hz) | 2.0 | 2.0 |
| VSAC with Heat pipes (2 Rows, 14 fpi) | 2.5 | 2.0 |
| VSAC with Desiccant wheel (Wheel depth: 1.5") | 2.5 | 2.0 |

Each of the AC systems was ranked based on capital costs, energy use and the number of hours in a cooling season during which they could maintain the space conditions in the ASHRAE comfort zone. The number of hours in the ASHRAE comfort zone were obtained from the simulation results.

### 9.2.2 Capital Costs of AC Systems

Capital costs of each AC system's components were compiled from various equipment vendors, FSEC's previous costing experience and TECOGEN's study of desiccant systems component costing (3). The capital cost of the AC components applicable for the residential systems are listed in Table 9.3.

**TABLE 9.3**

| Capital Costs of Residential AC System Components | |
|---|---|
| **Description of Components** | **Capital Cost** |
| Single-speed air conditioner | 500 $/Nominal Ton |
| Variable-speed air conditioner | 1500 $/Nominal Ton |
| Heat Pipes:<br>3 rows, 14 fpi and total face area: 2 $ft^2$ | 300 $/Nominal Ton |
| Desiccant Wheel:<br>Silica Gel bed, Depth: 2.1" and total face area: 2 $ft^2$ and the housing for the wheel | 150 $/Nominal Ton |
| Gas-fired desiccant wheel system:<br>Silica gel wheel, heat exchanger, burner, blower, dampers and cabinets | 1600 $/sq.ft of face area of the wheel |

Note: Ducting, electrical and controls costs were assumed to be the same for all the AC systems in the analysis and were not included.

## 9.3 Comparison Measure: Relative Merit Index

The purpose of an air conditioner is to provide a comfortable environment. The cost for providing such an environment is made up of the initial capital cost and the operating cost of the given systems. The market sets costs and, for all practical purposes, ASHRAE sets comfort boundaries.

Our goal is to identify which of the AC systems described in Table 9.2 provides the most comfort at the least cost. As the results in the simulations will show, all AC systems are not created equal!. To quantify the differences we define a measure called the "Relative Merit Index" to account for both comfort and cost.

The Relative Merit Index (RMI) is defined as:

$$RMI = \frac{C_{AAC} - C_{CAC}}{TC_{AAC} - TC_{CAC}} \tag{9.1}$$

where

$C_{AAC}$ - Comfort hours provided by an "alternative" AC system over the equipment life
$C_{CAC}$ - Comfort hours provided by the "conventional" SSAC over the equipment life
$TC_{AAC}$ - Total Costs for an "alternative" AC System
$TC_{CAC}$ - Total Costs for the "conventional" SSAC

Comfort hours refer to the number of hours the zone conditions are maintained within the ASHRAE comfort zone. The total costs of the AC systems were determined as the sum of the capital cost and net present value (NPV) of the operating cost. For the NPV calculations, we used an equipment life of 10 years and a discount rate of 8 percent. Energy costs applicable for each climate zone (Appendix F) were used to determine the energy costs. Demand costs are not applicable to residences.

In the present case, the "alternative" air conditioners have a lower sensible heat ratio than the conventional SSAC. The SSAC was selected as the baseline because it is the most commonly used AC for residences. As indicated earlier, our modeled SSAC baseline unit had an SHR of 0.73 at rated ARI conditions (Indoor: drybulb 80°F, Wetbulb 67°F, Outdoor: drybulb 95°F).

The Relative Merit Index can provide insight into the advantages or disadvantages of low-SHR AC systems for any building application or climate and, it is a point of departure for more sophisticated design criteria (4).

### 9.4 Simulation Results

Figure 9.1 shows the capital costs, annual air conditioning energy use, percent comfort hours provided and Seasonal Energy Efficiency Rating (SEER) of the AC systems compared for the Miami residence. The percent comfort hours were calculated with respect to the total number of hours in the cooling season (January to December for Miami).

The SSAC met the comfort conditions for approximately 60 percent of the cooling season for the Miami residence. Figure 9.2 shows the monthly average temperature and humidity conditions maintained by the SSAC for this residence. The simulations predict a steady increase in relative humidity levels during the cooler evenings and early morning periods. During this time of the day, an SSAC would have low run-times or not be running at all and therefore would be unable to meet the latent load. The advantage of the low-SHR AC systems

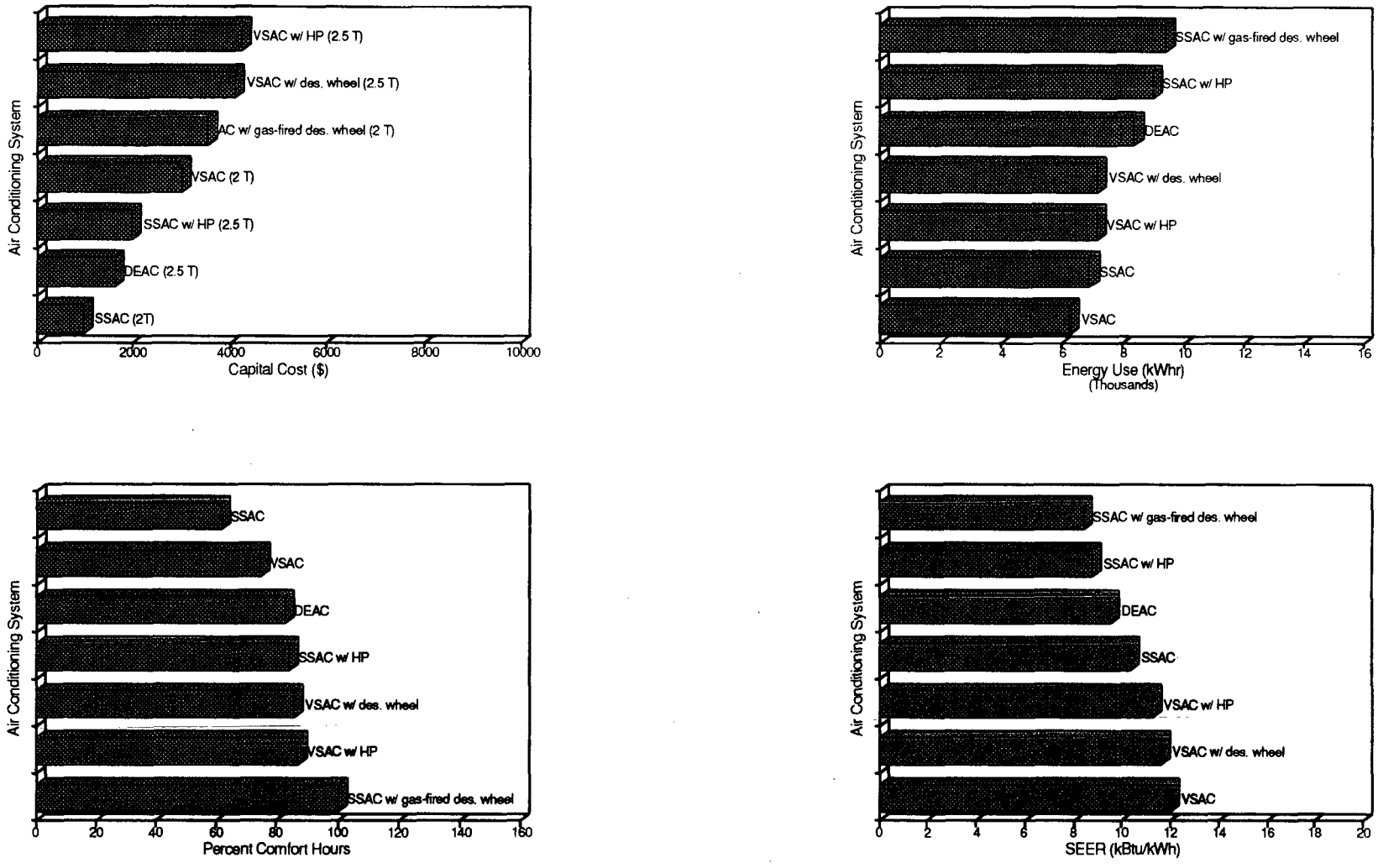



Figure 9.1 - Miami Residence: AC Systems Performance

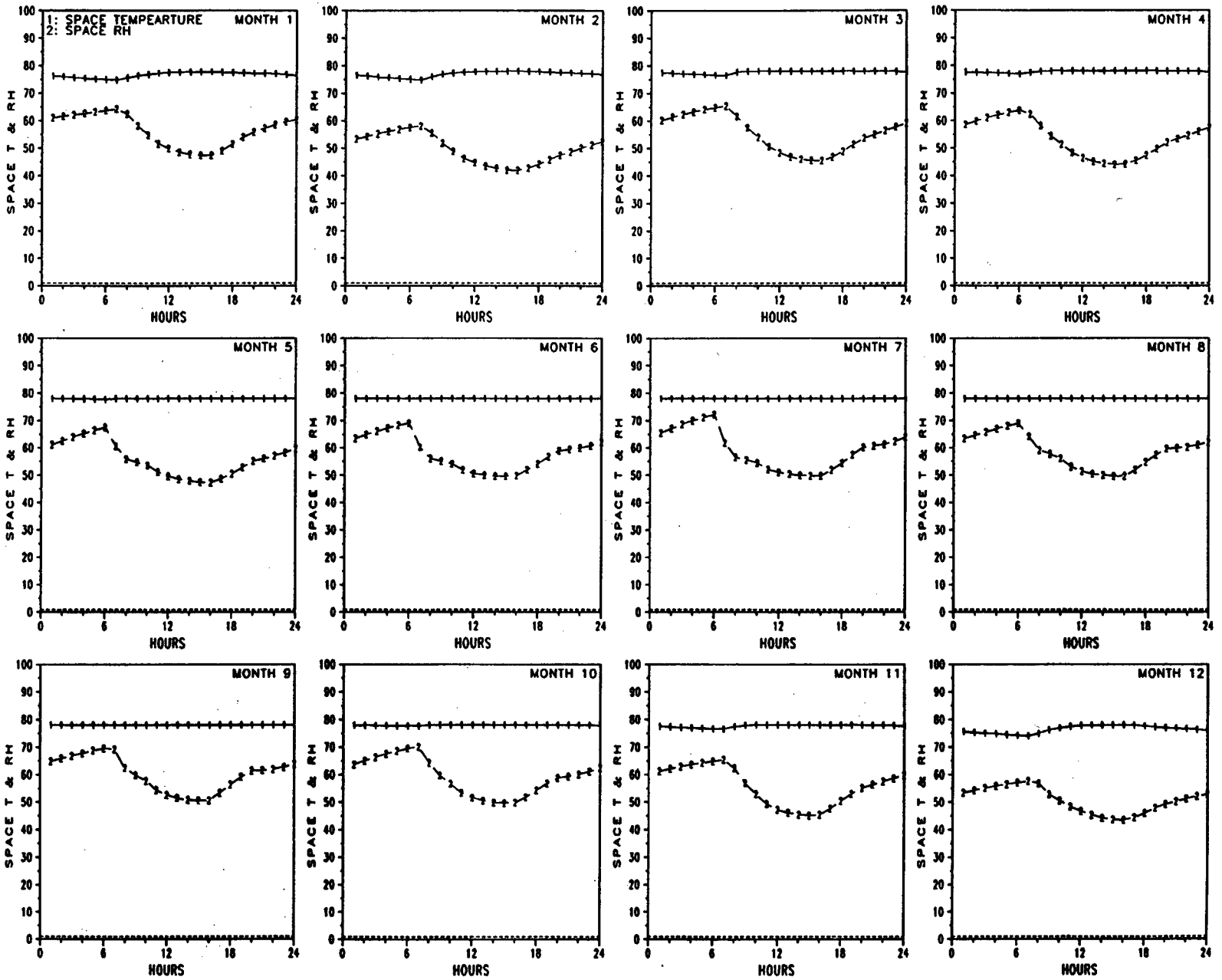



Figure 9.2 - Indoor Temperature and Humidity Profile in the Modeled Miami Residence

(e.g. SSAC with a desiccant wheel or heat pipes, VSAC with a desiccant wheel or heat pipes) is that they are able to meet the latent loads for a longer period of time and therefore will reduce humidity levels.

The VSAC often operates at low compressor speeds to meet low sensible cooling loads. As shown in Chapter 4 (Figure 4.5), as the compressor speed decreases, the SHR of the VSAC increases for the same supply air flow rate. A "low" fan speed for a VSAC unit corresponds to a supply air flow rate of around 200 cfm/nominal ton (5). Initially we used this air flow rate as the lowest value in the simulations. At this flow rate, the simulation results showed that the VSAC could maintain comfort conditions for only a few more hours than the SSAC.

Subsequently, we further reduced the supply air flow rates and carried out one more set of simulations. The supply air flow rate was reduced at each compressor speed until the supply air temperature leaving the cooling coil was approximately 40°F (lower temperature would freeze up the coil). The simulation results showed that the VSAC could provide approximately 10 percent more comfort hours than the SSAC at these low air flow rates. However, since the seasonal energy efficiency of the VSAC decreases as the air flow rate is lowered below 200 cfm/nominal ton, the SEER of the VSAC running at these low flow rates and shown in Figure 9.1 is lower than the value that one would expect for these units. Typically VSAC units have nominal SEER values around 14.0.

The heat pipe-assisted AC provided a few more hours of comfort than the DEAC for the Miami residence. At high indoor relative humidity conditions, the heat pipe-assisted AC meets a lower load-SHR than the DEAC, even though the SHR of both the systems may be the same at a particular design condition (52 percent RH) as shown in Figure 9.3. Hence, the comfort hours provided by the DEAC were less than that of the heat pipe-assisted AC.

The gas-fired desiccant system, which was controlled by both a thermostat and a humidistat met comfort conditions for more than 95 percent of the cooling season as shown in Figure 9.1. The other AC systems (except SSAC and VSAC) met the comfort conditions for only 80 percent of the cooling season.

The SSAC has a higher SEER than the DEAC but provides 20 percent less comfort hours. The VSAC has the highest SEER due to its low compressor speed operation and better part-load efficiency during many hours of the cooling season. However, the initial cost of the VSAC is relatively high (1500 $/ton compared to 500$/ton of SSAC). In addition, it does not provide as many hours of comfort as the DEAC does.

Similar trends were observed for the energy use, SEER and comfort provided by the AC systems for Washington and St.Louis residences. The comparison of the AC systems for St.Louis and Washington residences are shown in Figures 9.4 and 9.5 respectively.

Figure 9.6 shows the Relative Merit Index (RMI) of the AC systems for the three climates. The RMI ranking for the systems shows that the DEAC can provide better comfort at a lower total cost in each of the three climates. Even though the heat pipe-assisted variable-speed air conditioner and the gas-fired hybrid systems provided more hours of comfort than the DEAC, their initial costs are higher and hence they have a lower Relative Merit Index rating.

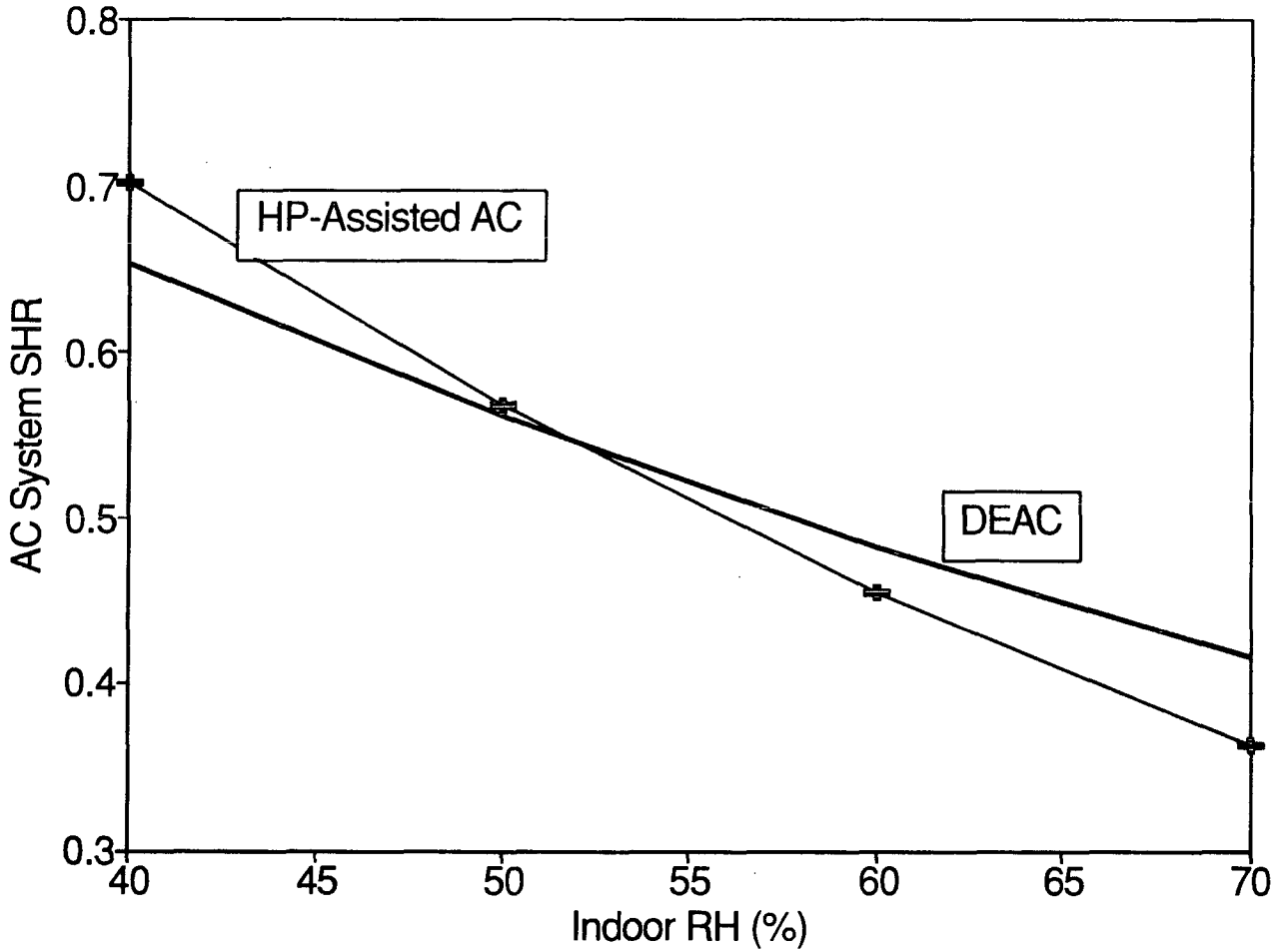



Figure 9.3 - SHR of Heat Pipe-Assisted AC and DEAC as a Function of Return Air RH

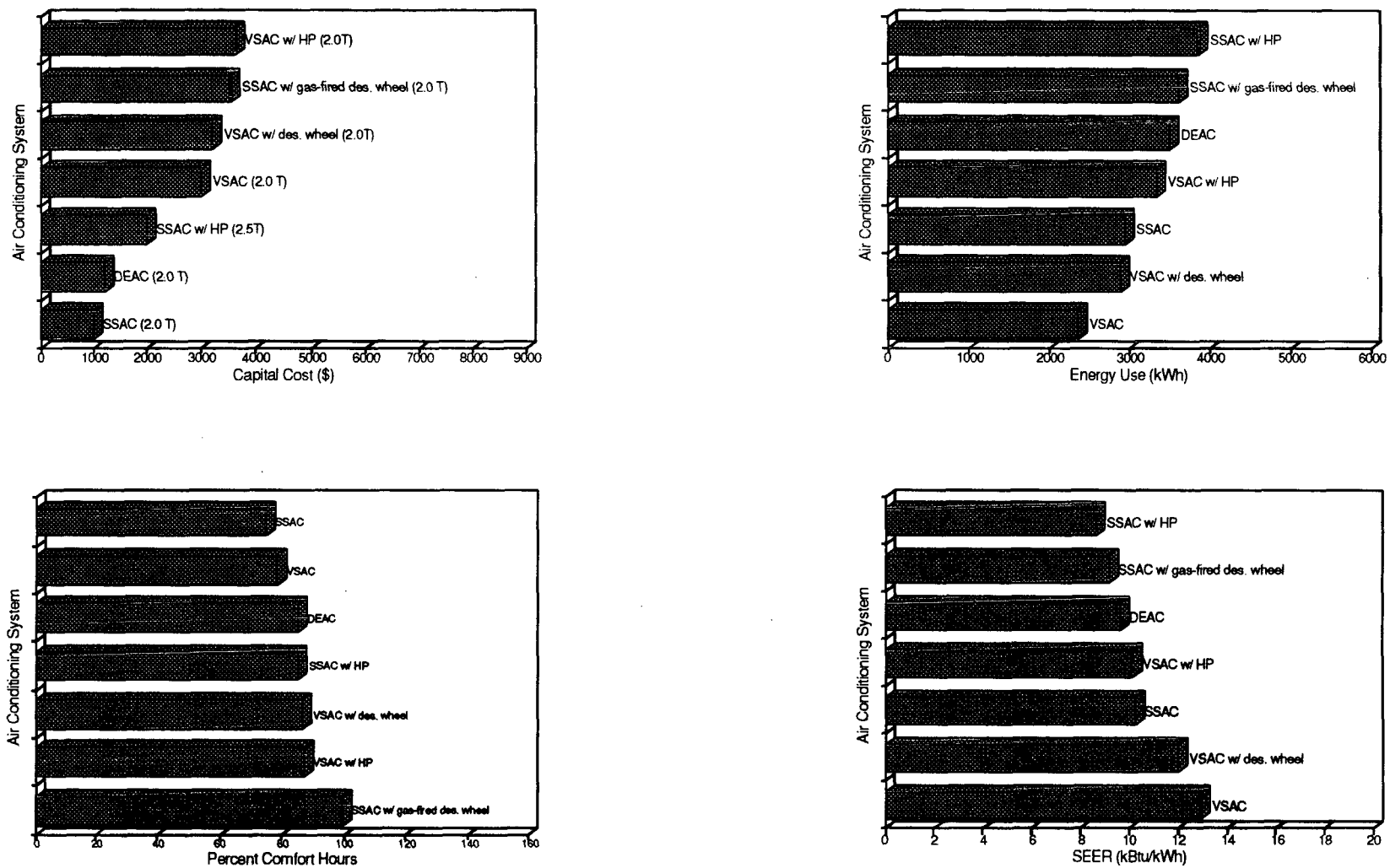



Figure 9.4 - St. Louis Residence: AC Systems Performance

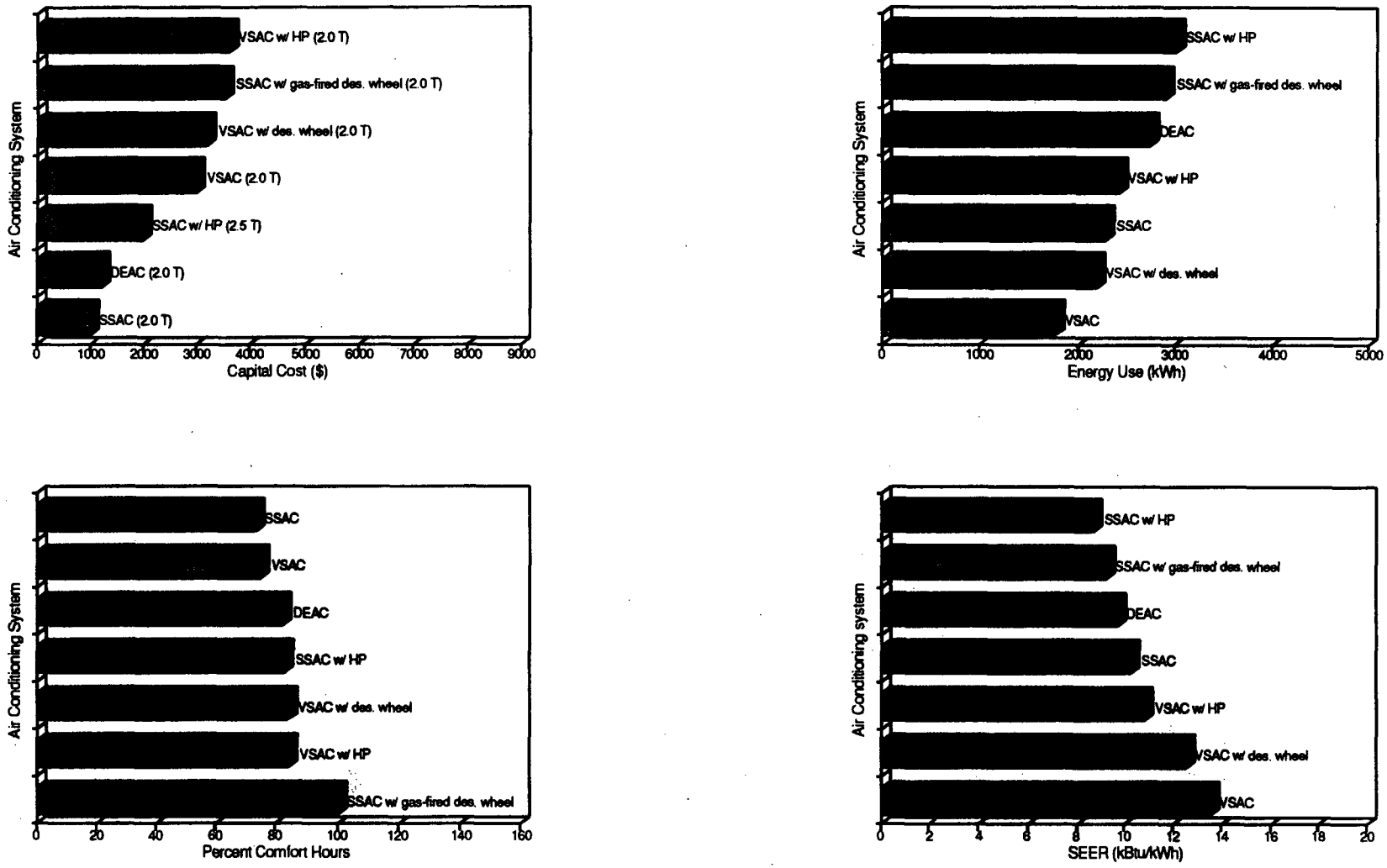
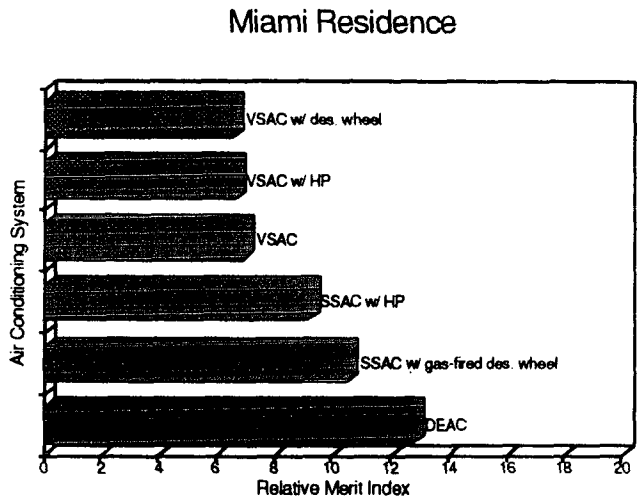
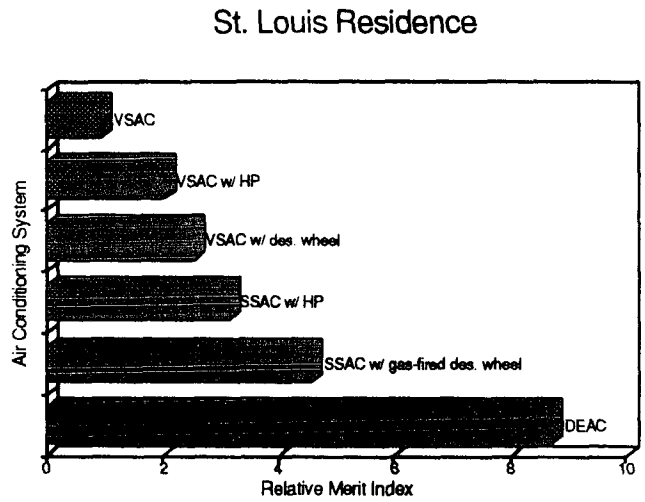



Figure 9.5 - Washington Residence: AC Systems Performance



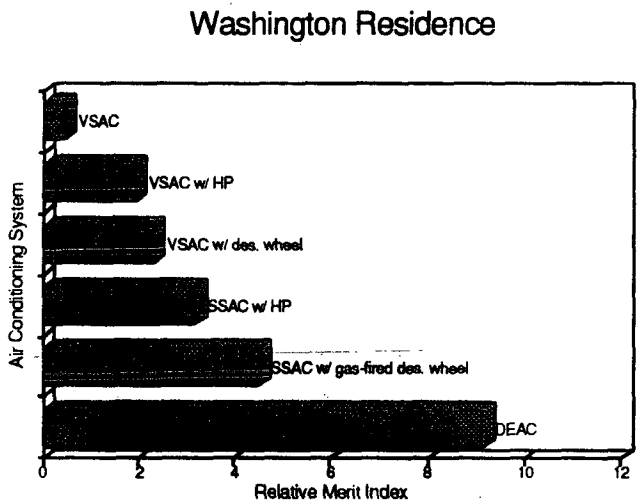




Figure 9.6 - Residence: AC Systems Relative Merit Index

In addition to the above studies, we performed another set of simulations to identify the energy saving potential of the DEAC system compared to a current high-efficiency air conditioner. These details are discussed in Sections 9.5 and 9.6.

## 9.5 Residential Applicability of DEAC

In this part of the study, we concentrated on identifying the energy saving potential of the DEAC system compared to a current high-efficiency air conditioner. To evaluate the energy saving potential of the DEAC, the geographical region in the continental U.S. where the DEAC would be applicable was identified using computer simulations.

A paper by Anderson et al. (6) of Lawrence Berkeley Laboratories (LBL) was used to initiate this task. The LBL work characterized population centers based on climate parameters which related to building energy use. By agglomerating geographic areas having a common range of climatic parameters (for example, cooling degree days) it was possible to rationally segregate areas of like climate type.

The LBL characterizing climate parameter used in our work was the number of latent enthalpy hours. The number of latent enthalpy hours is a measure of the amount of moisture that must be removed from outdoor air to bring it to 77°F and 60% relative humidity. LBL found that, in general, the higher the number of latent enthalpy hours, the higher the number of cooling degree days. The reverse is not true. This correlation suggests that, since the DEAC is a cooling based dehumidification system, it will be applicable in geographical regions with relatively high latent loads.

Our estimates of the number of hours per year during which a typical residence would have indoor relative humidity exceeding 60 percent was based on simulation runs for ten cities with latent enthalpy hours ranging from 700 to 28,000 hours. Figure 9.7 is a plot of the cooling degree days as a function of latent enthalpy hours for each of the ten cities identified.

For each of the ten climates, we simulated the performance of, first, an air conditioner of average age and then a current air conditioner. Both operated in the 1500 square foot residence described in Section 7.1. Based on the results of the hour-by-hour simulation, the total time that the house relative humidity exceeded 60 percent was determined.

The software code AC model for this part of the study was developed based on the DOE-2 packaged unit subroutine. The advantage of this model is that rated EER and SHR values for the air conditioner can be specified for this model and the performance at off rated conditions can be predicted. A residence thermostat set point of 78°F was maintained unless noted otherwise. All AC units (average (old), current and DEAC without the wheel) were sized to meet the sensible loads for each city selected.

### 9.5.1 Humidity Control by Existing "Average" Air Conditioner

The energy efficiency of air conditioners is a function of age of the unit. The average age of U.S. central air conditioners in residences is between 8 and 9 years (2). The "average" air conditioner modeled for this study had a SEER of 9.0 and a sensible heat ratio of 0.76 (7).

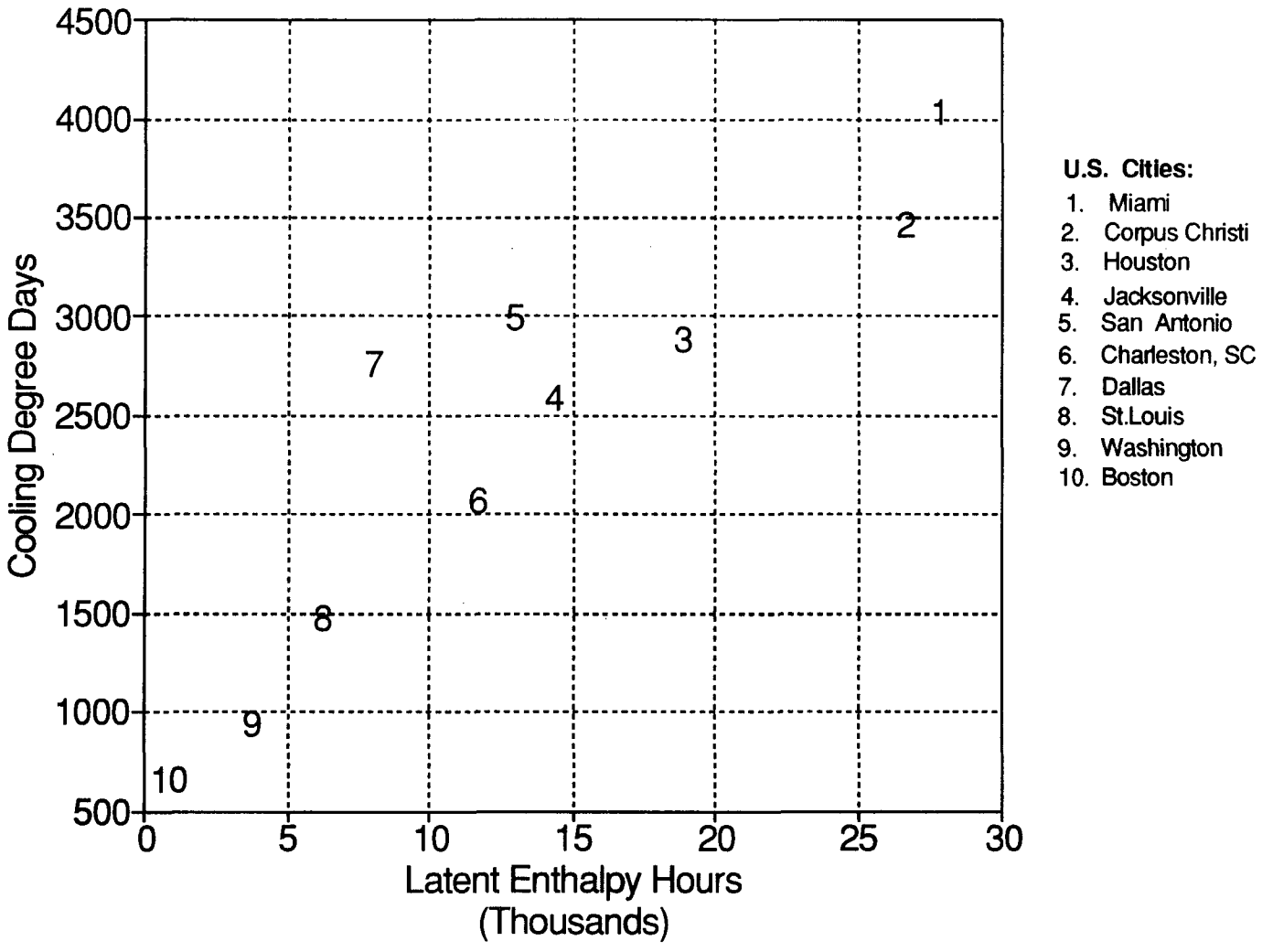



Figure 9.7 - Correlation Between CDD and LEH: Selected Cities for Analysis

Based on the above data for the "average" air conditioner, the simulation results showed that the number of hours the indoor humidity level exceeded 60 percent RH in the ten representative cities ranged from about 400 to 3200. The specific number of hours for each of the ten cities is presented in Figure 9.8.

### 9.5.2 Humidity Control by "Current" Air Conditioner (State-of-the-Art)

New vapor compression air-conditioning units typically have relatively high energy efficiency ratios and high sensible heat ratios. The latter leads to a reduced latent capacity. The "current" AC unit modeled for this case was assigned a relatively high SEER of 12 and an SHR of 0.84. The same SEER and SHR were assigned to the AC used with the DEAC system.

Simulation results predicting the number of hours indoor relative humidity levels exceed 60 percent with the "current" air conditioner are shown in Figure 9.9. For comparison, the number of hours with over 60 percent RH that would result if a DEAC was used are also presented in Figure 9.9. For this part of the study, the desiccant wheel speed was controlled with a humidistat to tailor the system SHR. The DEAC advantage in ability to meet latent load is clear.

### 9.5.3 Comparative Energy Use of the DEAC and Current AC Units

At this point, energy savings relative to comfort experienced are presented based on performance comparisons of the "average" and "current" air conditioners with the DEAC. Figure 9.10 compares the annual air conditioning energy use for the three cases.

The current air conditioner has a higher efficiency than the average existing air conditioner and hence consumes less energy. The current AC also consumes less energy than the DEAC. This is because, first, the DEAC sensible capacity has been reduced and, as a result, the run time increases to satisfy the thermostat setting. Second, DEAC fan power requirements increase to handle the additional pressure drop of the DEAC desiccant wheel.

The DEAC Relative Merit Index, based on using the "current" air conditioner as the RMI baseline system, is shown in Figure 9.11. The index shows that the DEAC RMI is 20 or more in those cities which have more than 2,000 cooling degree days (CDD) and 10,000 latent enthalpy hours (LEH) (refer also to Figure 9.7). The DEAC uses 8 to 10.7 percent more energy than the current air conditioner and it provides 15 to 45 percent more comfort in these cities. These figures indicate that the percentage increase in comfort is significantly greater than the percentage increase in energy use. Clearly, there are trade-offs associated with providing a more healthy, comfortable, productive environment.

It must be kept in mind that the "Current" AC used to construct Figure 9.11 has different SEER and SHR ratings than the SSAC used to compare various alternative AC systems in Section 9.4.

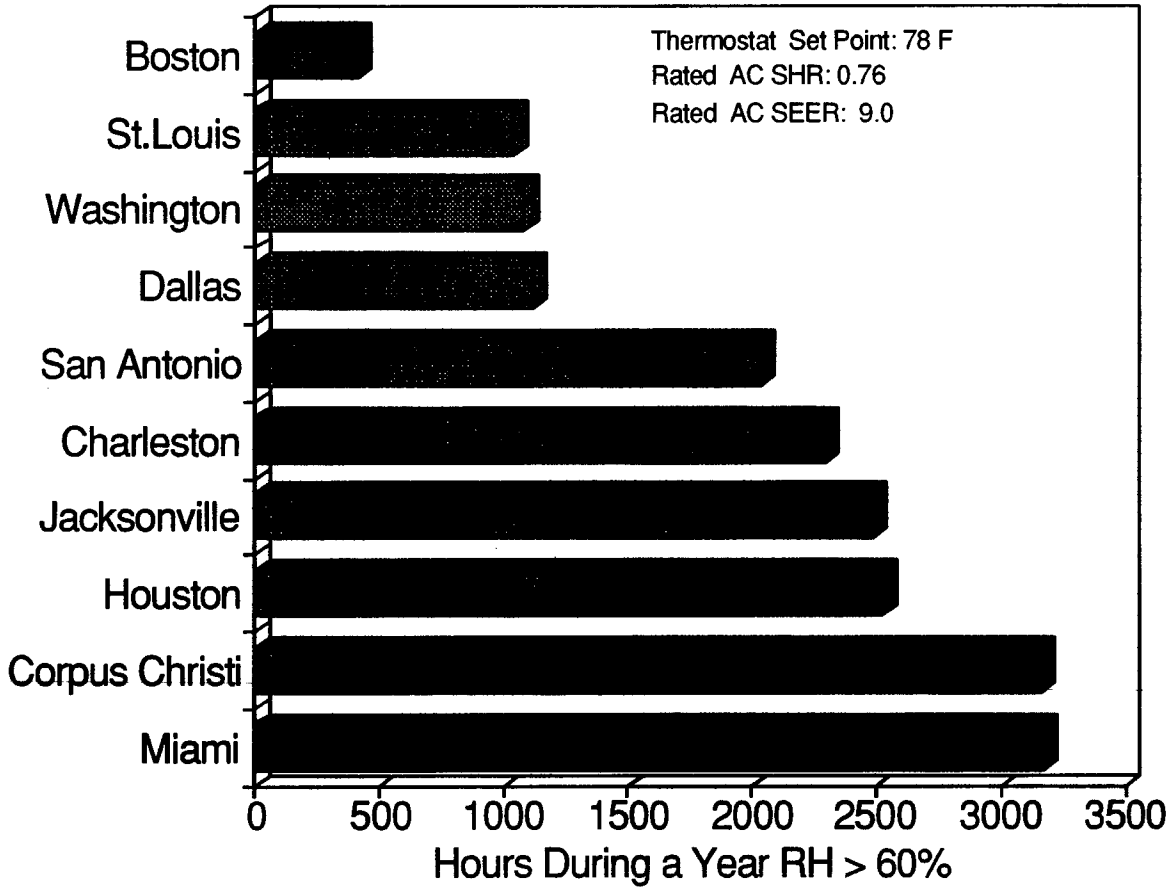



Figure 9.8 - Residence: Hours Indoor RH > 60% (Air Conditioners in Use (1984-1991))

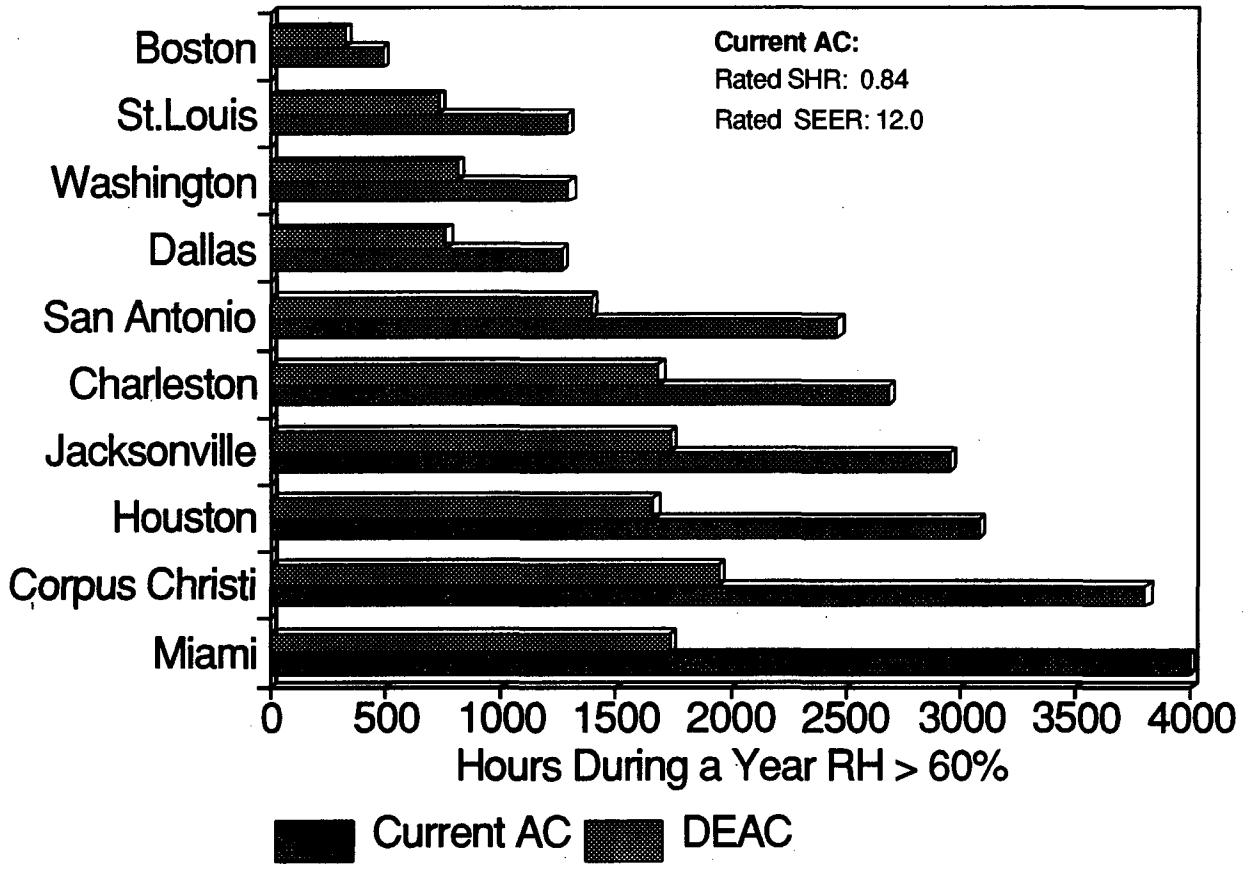



Figure 9.9 - Residence: Hours Indoor RH > 60% (DEAC and "Current" AC)

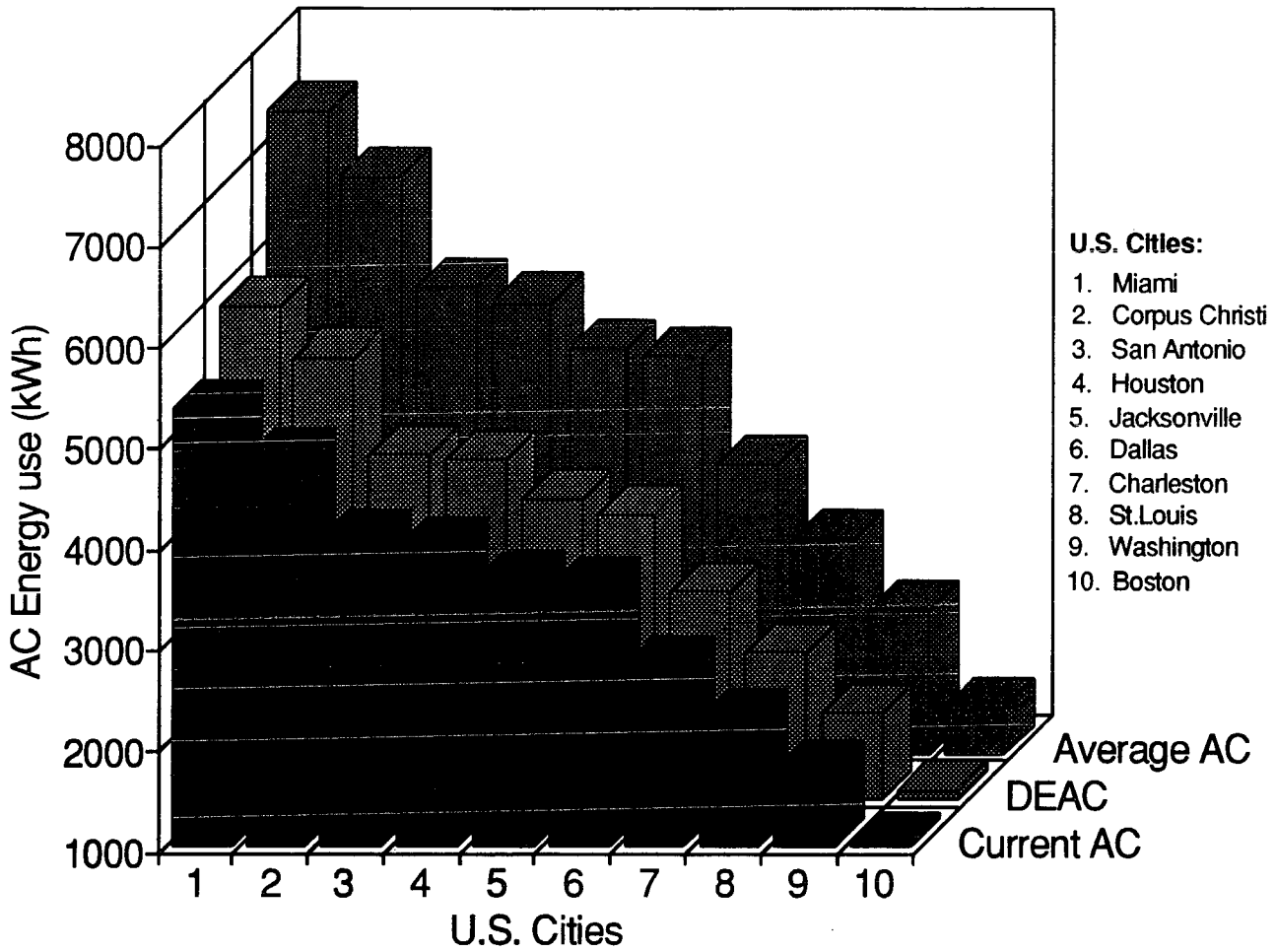



Figure 9.10 - Annual AC Energy Use (DEAC, "Average" and "Current" AC)

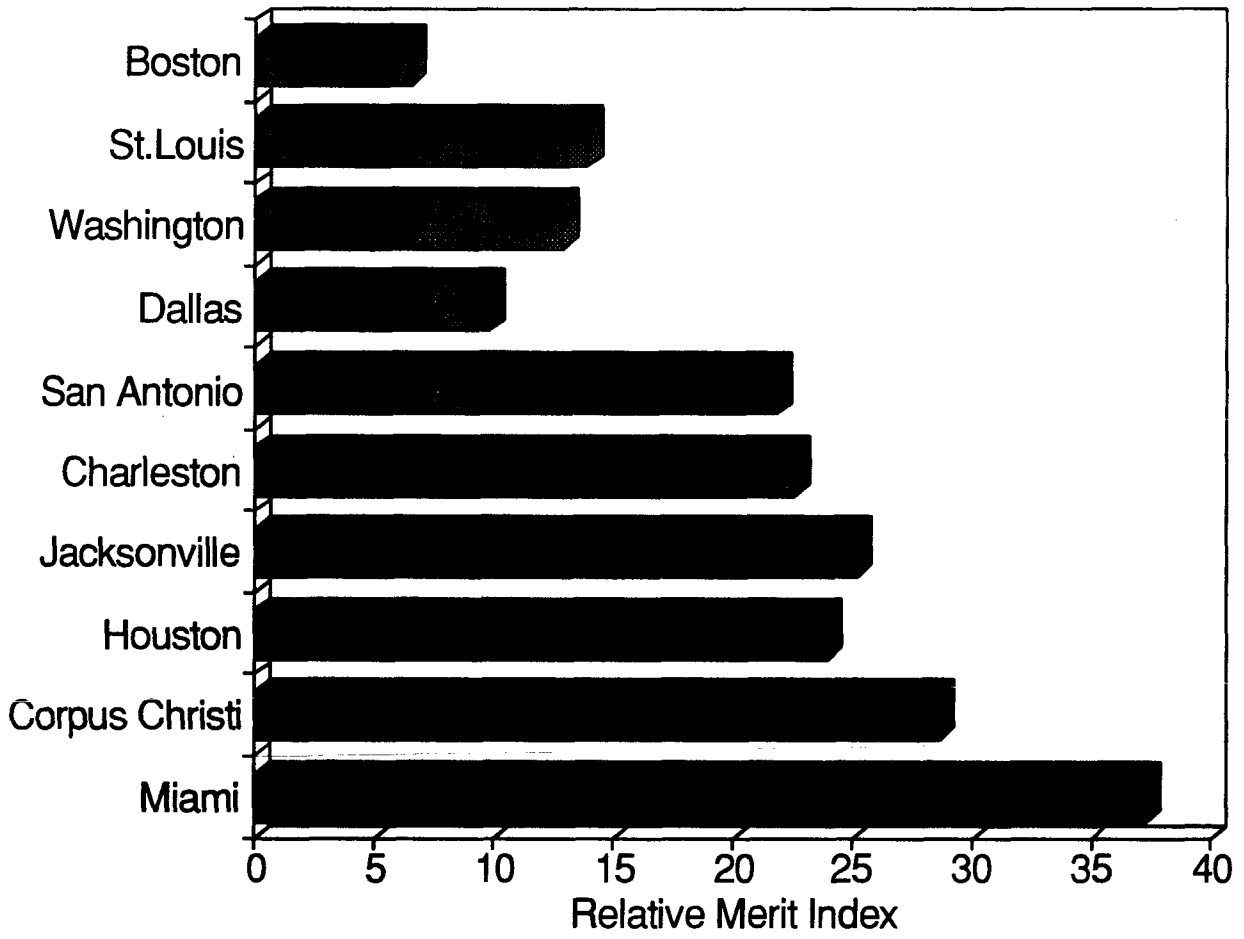



Figure 9.11 - DEAC's Relative Merit Index in Ten U.S. Cities

## 9.6 Potential for Energy Savings with the DEAC system in a Residence

In order to demonstrate the energy savings possible with the DEAC it is necessary to define a common playing field. One approach is the Relative Merit Index discussed above. Alternatively, we can take a more direct approach by forcing the competing air conditioners to maintain comfort conditions within the ASHRAE comfort zone. We do this in the same way that homeowners try to do it; by manipulating the thermostat. Doing so makes it possible to approach a realistic energy use comparison.

Simulations were run for the model residence in Miami using first the current air conditioner and then a DEAC system. Thermostat settings were changed as required each hour (depending on level of comfort). This is equivalent to moving along a line of constant comfort (ET*). When the thermostat, and therefore the relative humidity, was reset each hour to produce what was effectively the ET* line of 78°F, and the absolute indoor humidity was conservatively maintained just within the upper boundary of the ASHRAE comfort zone, the annual **average** space temperature was 75.85°F with the "current" AC and 77.22°F with the DEAC. There were many hours when the current AC unit, in order to stay in the comfort zone, required thermostat settings lower than the **average** setting.

Energy use and comfort maintained by the current AC system are compared to the same factors for the DEAC in Figure 9.12. For the above simulation runs, the DEAC consumed 10 percent less energy than the current air conditioner to maintain comparable comfort conditions. It should be noted that the above comparison was an ideal case in which the current AC thermostat was checked and/or adjusted each hour. More realistically, thermostats are reset once or twice a day to compensate for high humidity conditions. This type of reset would likely increase the current energy use depending on the time of reset and consequently result in greater relative energy savings for the DEAC than the projected 10 percent.

## 9.7 Projected Nationwide Energy Savings with DEAC for Residential Sector

The projected energy savings for the estimated area of residential DEAC users based on unit energy savings of five to ten percent are 3.25 to 6.5 x $10^6$ Mega Watt hours per year. When translated into primary energy (fuel) with a conversion efficiency of 33 percent, this equates to 9.8 to 19.7 x $10^6$ Mega Watt Hours or about 33 to 67 x $10^6$ Million BTU per year. The baseline data to reach this result is provided in Table 9.4.

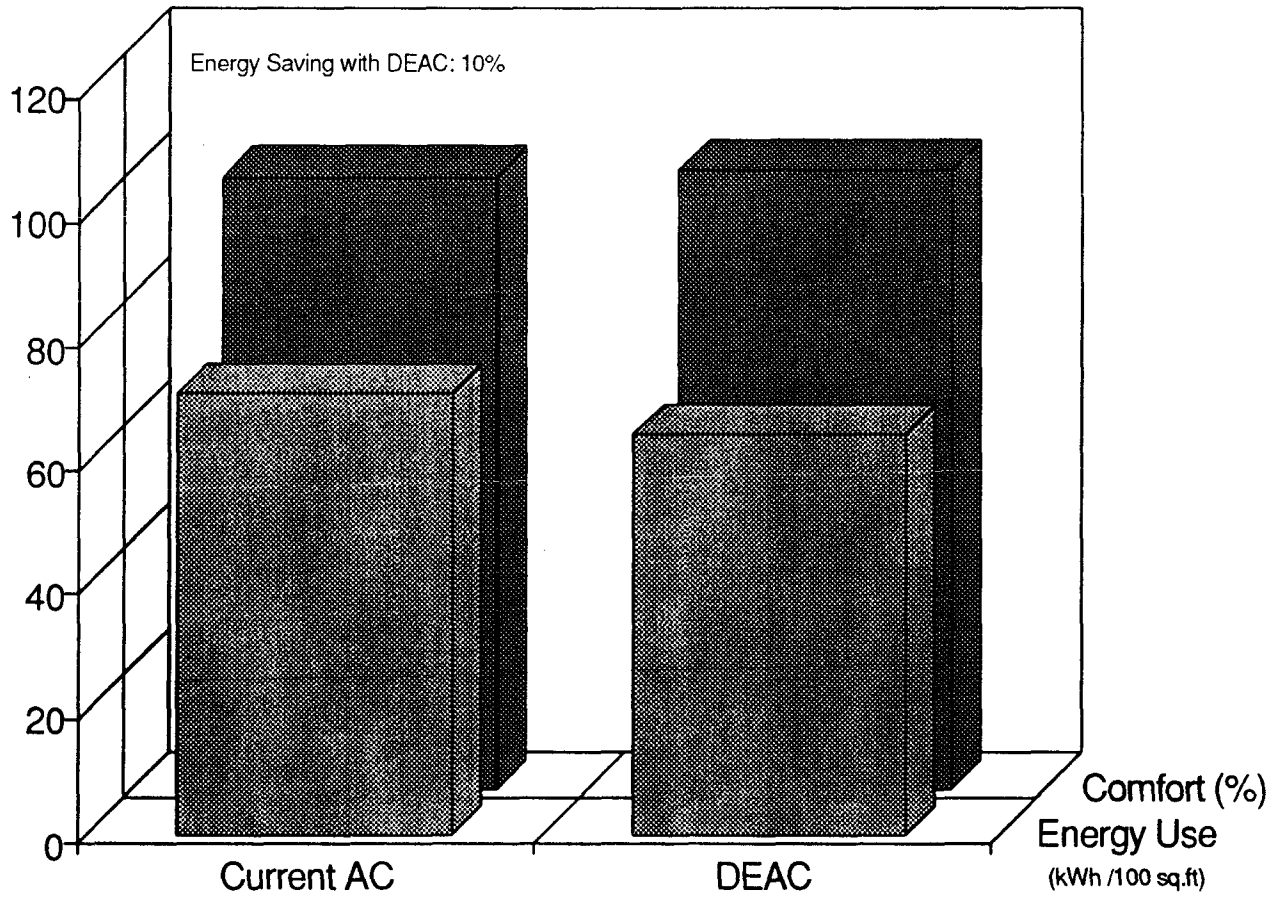



Figure 9.12 - Miami Residence: Comparison of Energy Use and Comfort for Current AC and DEAC

TABLE 9.4

| Residential Applicability of DEAC: Demographic Information | |
|---|---|
| a. | Approximate total number of households which have more than 2000 cooling degree days and high (> 10,000) latent enthalpy hours - 14.7 Million (2) |
| b. | Percentage of households air conditioned - 87% (Southern region of U.S.) |
| c. | Average air conditioning energy use/household - 17.3 MBtu (1) |
| d. | Energy savings with the DEAC - 5 to 10% to maintain comparable comfort conditions |
| e. | Total energy savings with 5 to 10% reduction in energy use with the DEAC 3.25-6.5 x $10^6$ MWh/Year |
| | Total primary energy use - **0.033 to 0.067 QUADS** |

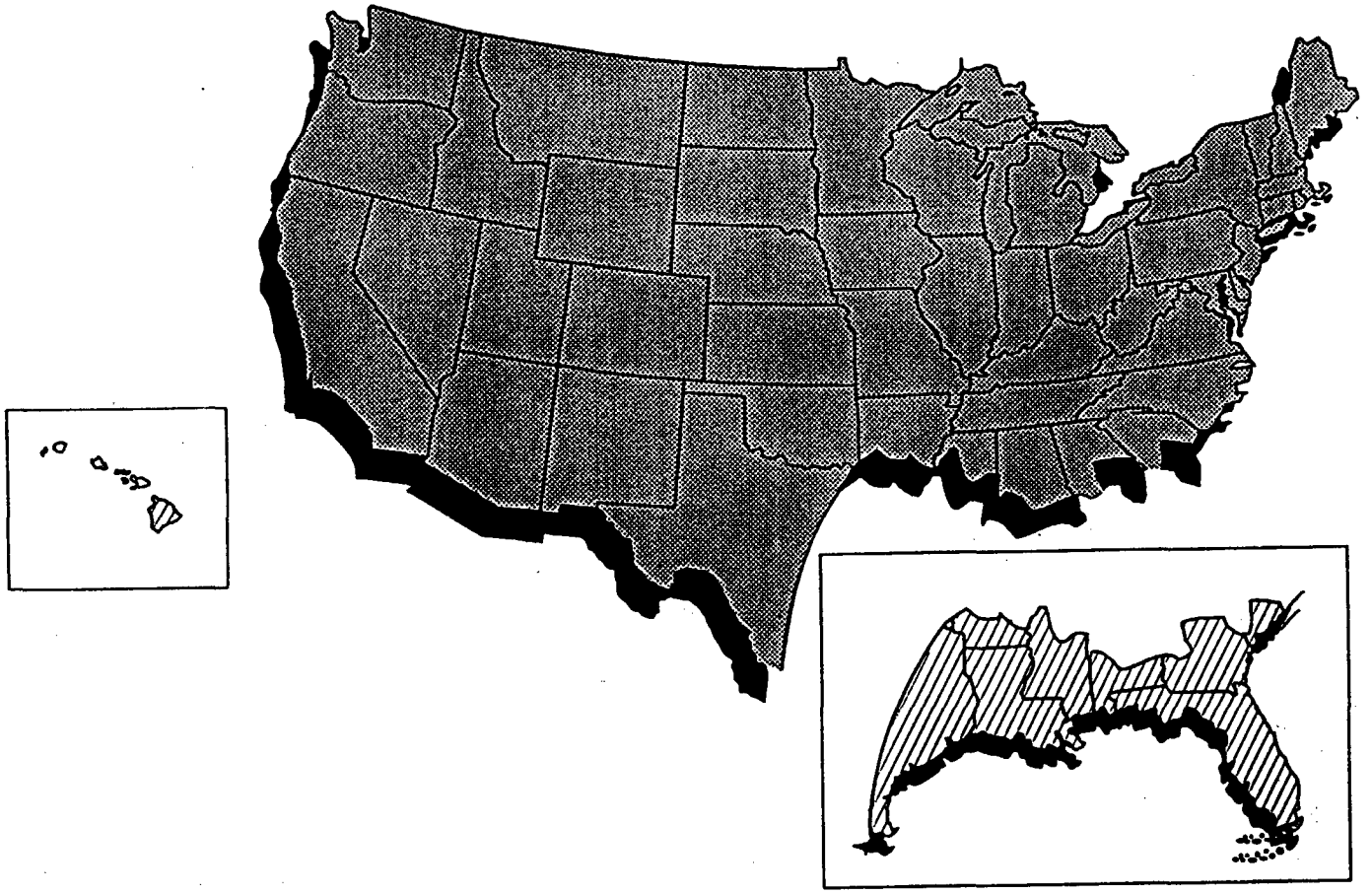
The area of DEAC residential applicability was selected based on the Relative Merit Index. These results indicate that the Southeast/Gulf coast and Hawaii regions represent the initial residential market area. Figure 9.13 shows the area boundary.

## 9.8 Conclusion

This Chapter addressed the merits of the desiccant enhanced air conditioning system (DEAC) for residential applications. The DEAC was found to be a viable electric AC option for enhancing the comfort in a residence. The energy use, comfort provided by the systems and capital costs were compared for several AC options for Miami, St.Louis and Washington and the DEAC showed superior performance.

Further studies showed that, when used in a residence, the DEAC system could save energy compared to the current high-efficiency AC. For comparable comfort, the DEAC provides up to ten percent energy savings for the south east/Gulf coast area.

If the SHR of the building load is high, a current state-of-the-art high-efficiency air conditioner is appropriate. However, if the load-SHR is low, the DEAC can provide high efficiency cooling combined with improved dehumidification. Improved humidity control often has secondary benefits such as increased productivity and increased equipment life.



▨ Applicable zones include south-east U.S. and Hawaii

Figure 9.13 - Primary Region of Residential Applicability of the DEAC

References


1. Household Energy Consumption and Expenditures, 1987, Part 1: National Data, Prepared by Energy Information Administration, U.S. Department of Energy, Washington.

2. Housing Characteristics 1990, Prepared by Energy Information Administration, U.S. Department of Energy, Washington.

3. Advanced Desiccant Cooling System Development: Manufacturing costs and applications Economics, Prepared by Tecogen Inc. for Gas Research Institute (GRI-87/0043.2).

4. Paul Scanlon, P.E., Heating, Piping Air Conditioning: Economic Impact of HVAC System Selection, September 1992.

5. Product data, Carrier Hydrotech 2000 - Advanced Technology Heat Pump System.

6. Brandt Anderson, William L. Carroll and Marlo R. Martin, 1984, "Aggregation of U.S. Population Centers Using Climate Parameters Related to Building Energy Use", Passive Research and Development Group,m Lawrence Berkeley Laboratories, Berkeley, CA 94270.

7. Telephone communication with American Refrigeration Institute (ARI)

## 10.0 FAST FOOD RESTAURANT AIR CONDITIONING ENERGY ANALYSIS

Seasonal simulations were performed for the modeled fast food restaurant (characteristics of the restaurant are described in Section 7.3) for Miami, St.Louis and Washington climates. Results of the simulations are presented in terms of energy use, comfort, capital costs, seasonal energy efficiency rating and overall Relative Merit Index rating.

### 10.1 Introduction

Restaurants are among the most intensive building energy users. Typical energy use rates for restaurants are approximately 400 kWh/ft$^2$/day. Thirty five to forty percent of this energy goes into creating comfortable environments in the restaurants (1).

Cooling season humidity control in fast food restaurants having high internal and external loads have been recognized as a significant problem. In order to meet the dining ventilation and kitchen exhaust requirements of fast food restaurants, high ventilation rates (as high as 60 percent of supply air) are used. If the air conditioners currently in use have poor dehumidification capacity, the result can be reduced comfort control, damage to sensitive electronic equipment, increased mold and mildew growth, and adverse effects on food quality (2).

To circumvent the discomfort due to high indoor humidity levels, restaurant air conditioners are currently sized to meet a lower dry bulb temperature. This increases the run-time of the air conditioner resulting in higher energy use. In addition, the lower space temperatures in the restaurants create an uncomfortable environments.

In this Chapter, we compare the dehumidification performance of the DEAC system to alternative AC system options (Table 10.1) for a fast food restaurant.

**TABLE 10.1**

| Fast Food Restaurant: AC Systems Compared |
|---|
| 1. Single-speed AC (SSAC) |
| 2. SSAC with heat pipes and reheat |
| 3. Desiccant enhanced AC (DEAC) with reheat |
| 4. SSAC with reheat |
| 5. Variable-speed air conditioner (VSAC) |
| 6. VSAC with heat pipes and reheat |
| 7. VSAC with a desiccant wheel and reheat |
| 8. VSAC with reheat |
| 9. AC with gas-fired desiccant wheel |
| 10. AC with Kathabar dehumidifier |

## 10.2 Simulation Basis

A typical fast food restaurant is conditioned by two roof top units servicing the Kitchen and three roof-top units servicing the dining area (2). The roof-top units have electric resistance heating and direct expansion cooling, and are on individual thermostats. Humidity control is most often accomplished by temperature reset (i.e., adjustment of temperature set point).

In our simulations, we used both thermostat and humidistat control to maintain the space conditions depending upon the system. The dining and kitchen areas were treated as two independent zones. These two zones were conditioned by two independent AC systems of the same type. The option of routing the conditioned dining air through the kitchen was not studied in this analysis.

The desiccant based AC systems (DEAC and gas-fired hybrid) were modeled using a high-SHR (0.84) air conditioner. The other AC systems were based on a medium-SHR (0.73) air conditioner. The major portion of the latent loads were met by the desiccant wheel in the desiccant based systems. The high-SHR air conditioners are used primarily to meet the sensible loads. In general, the desiccant based systems with high-SHR air conditioners maintain a high EER, while providing the required dehumidification.

The performance of the gas-fired hybrid system was simulated in the "ventilation" mode for this building application. In this mode the ventilation air is dried and heated by the dehumidifier and subsequently cooled by the air conditioner. The return air is heated by the gas flame and used for regeneration purpose. The gas-fired systems are highly efficient in this mode of operation and requires a smaller dehumidifier system due to the high humidity potential difference.

Seasonal performance simulations were conducted for each of the AC systems (Table 10.1) using TMY weather data for the appropriate city. The results of these simulations are discussed in Section 10.3.

### 10.2.1 Size of AC Systems

The sizes of the AC systems were determined using the "ideal" air conditioner performance simulations for each of the climates as discussed in Chapter 8.0. The AC system sizes used for the dining and kitchen zones for the three climates are shown in Appendix D.

### 10.2.2 Capital Costs of AC Systems

The AC system component costs estimated for this application are listed in Appendix E. As indicated earlier, component costs were estimated based on vendor quotes, FSEC's previous costing experience and Tecogen's study of desiccant components costing (3).

## 10.3 Simulation Results

To rank the performance of the DEAC and the alternative AC systems for the fast food restaurant application, energy use, capital cost, SEER and comfort were compared. Figures 10.1, 10.2 and 10.3 show the performance of the AC systems for Miami, St.Louis and Washington fast food restaurants.

The SSAC and the VSAC were not able to fully meet the latent cooling loads in any of the three cities and, as a result, they maintained the comfort conditions for only to 40 to 50 percent of the cooling period. All of the other AC systems were able to maintain comfort throughout the cooling season (i.e., the space conditions were in the ASHRAE comfort zone for about 98 percent of the cooling period). In the case of SSAC with heat pipes or desiccant wheel and VSAC with heat pipes or desiccant wheel, the use of electric reheat made it possible to reach these levels of comfort.

In order to compare the energy use of electric system with that of gas-fired hybrid systems we assumed a typical power plant operating efficiency of 33 percent.

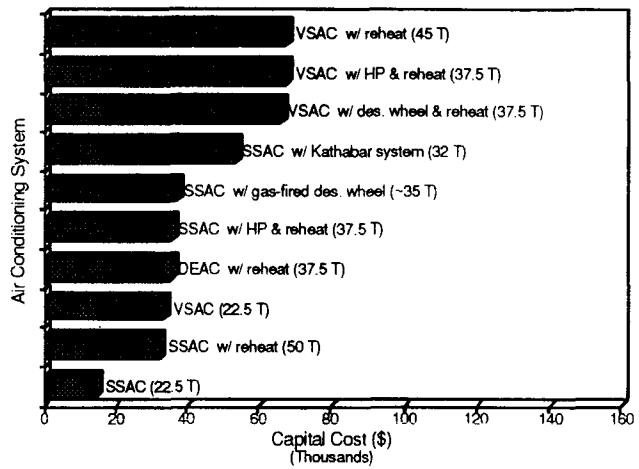
The simulation results for the fast food restaurant in all three climates show that the high efficiency AC with a Kathabar dehumidifier system uses the lowest energy while still maintaining comfort.

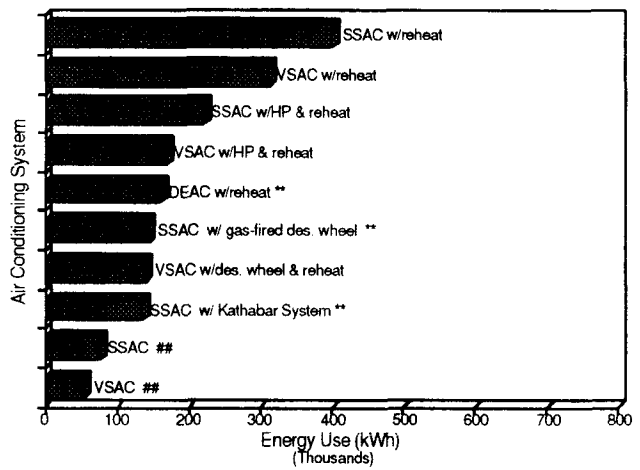
The SEER ranking of the fast food restaurant AC systems are shown for Miami, St.Louis and Washington climates in Figures 10.1, 10.2 and 10.3 respectively.

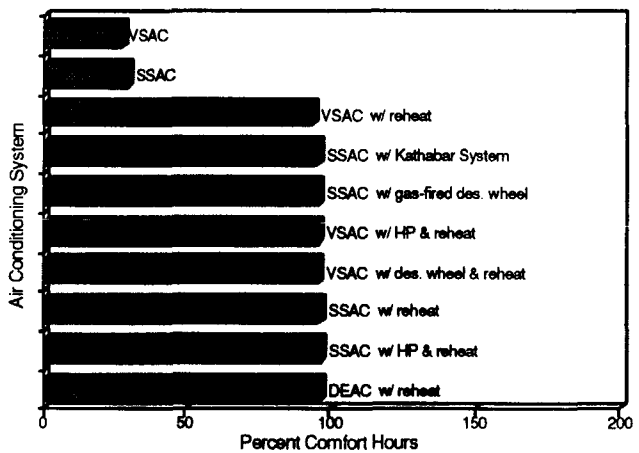
The VSAC with a desiccant wheel had the highest SEER in the Miami fast food restaurant. The gas-fired hybrid systems (both solid and liquid desiccant systems) also maintained high SEER values in this climate. The SEER of the heat pipe-assisted air conditioner was lower than that of the DEAC and the SEER of the air-reheat system was the lowest, as expected, in all the three climates.

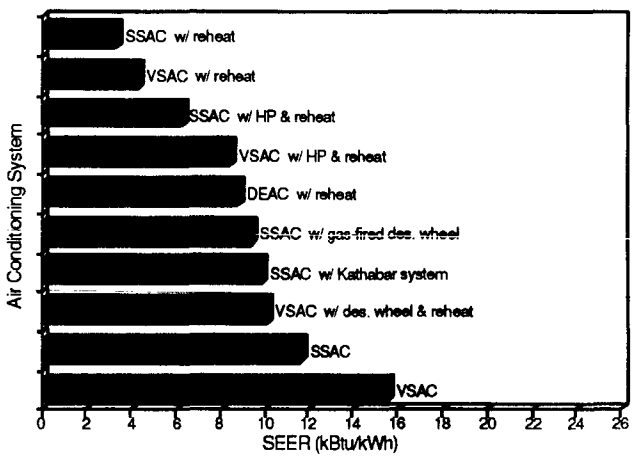
It can be noted from Figures 10.1, 10.2 and 10.3 that the heat pipe-assisted AC uses more energy than the DEAC. Basically, the DEAC is more efficient than the heat pipe-assisted AC as shown in the comparison study of electric AC systems (Section 5.3). In addition, the supply air flow rate for the heat pipe assisted-air conditioner had to be maintained higher than that of the DEAC for this application to avoid cooling coil freeze up.

As indicated earlier, the dining and kitchen areas of the fast food restaurant were modeled as two independent zones. These two zones were conditioned by two independent AC systems. The combined (Dining and Kitchen AC systems) Relative Merit Index is shown in Figure 10.4. The Relative Merit Index ranking shows that the desiccant wheel gas-fired hybrid system is the best AC option for the Miami fast food restaurant.



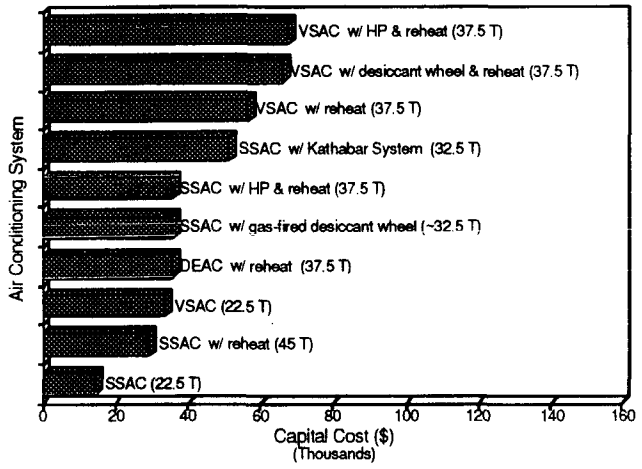



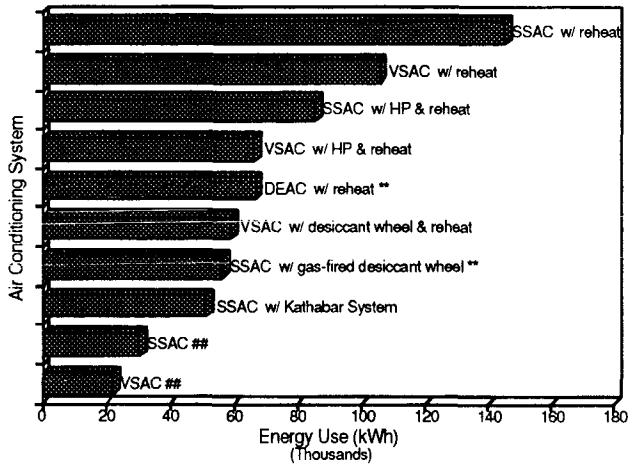

** Systems with high EER (12) and high SHR (0.84) air conditioner

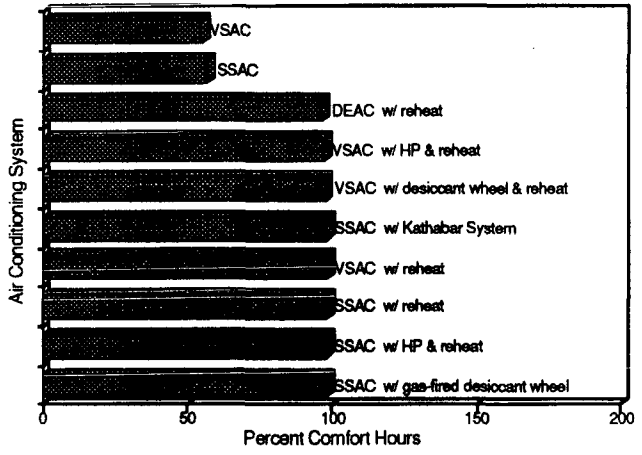
## did not meet latent loads

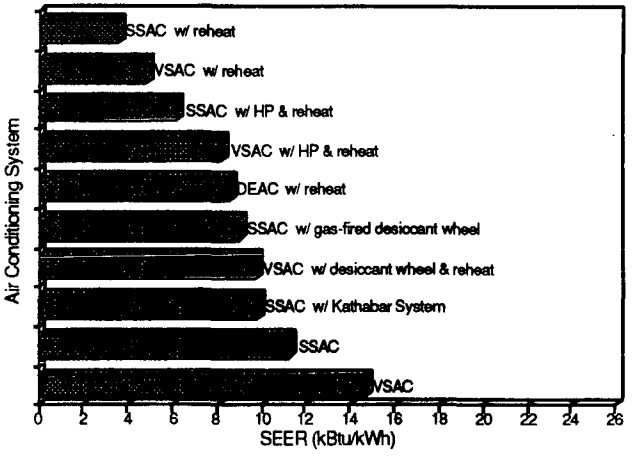
SSAC: Rated EER: 9.3, Rated SHR: 0.73

Figure 10.1 - Miami Restaurant: AC Systems Performance



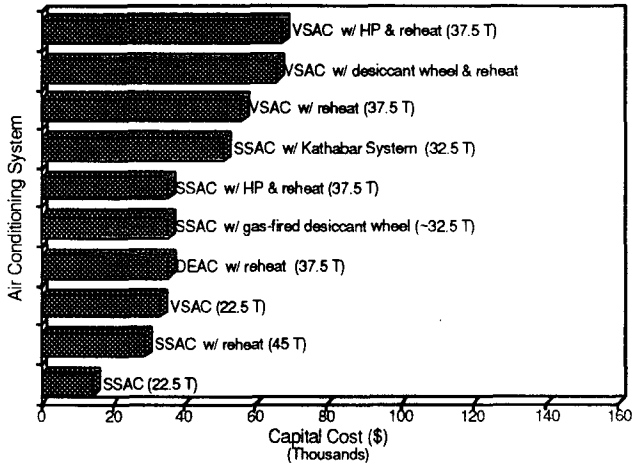



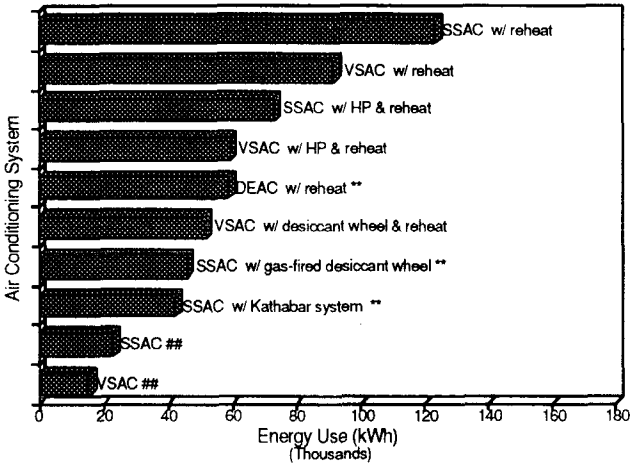

** Systems with high EER (12) and high SHR (0.84) air conditioner

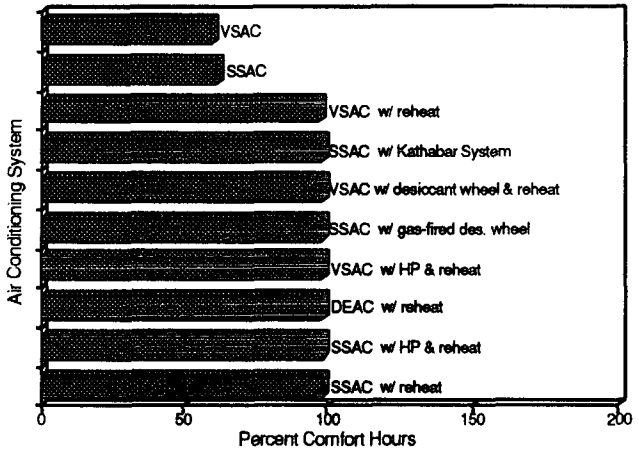
## did not meet latent loads

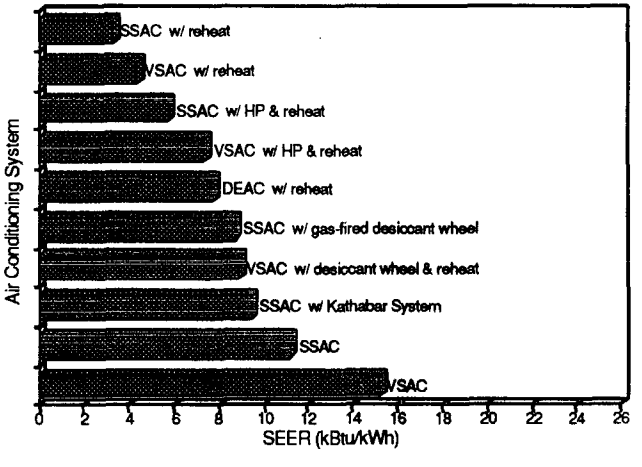
SSAC: Rated EER: 9.3, Rated SHR: 0.73

Figure 10.2 - St. Louis Restaurant: AC Systems Performance



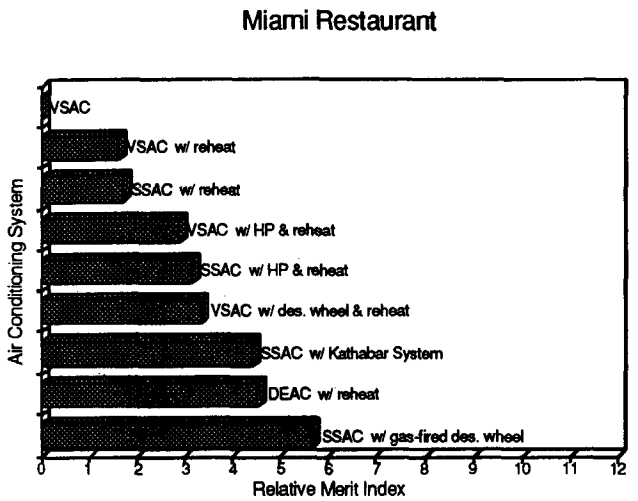



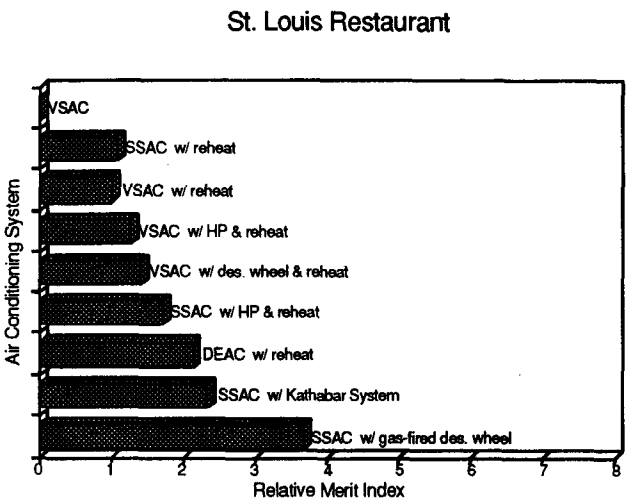

** Systems with high EER (12) and high SHR (0.84) air conditioner

## did not meet latent loads

SSAC: Rated EER: 9.3, Rated SHR: 0.73

Figure 10.3 - Washington Restaurant: AC Systems Performance



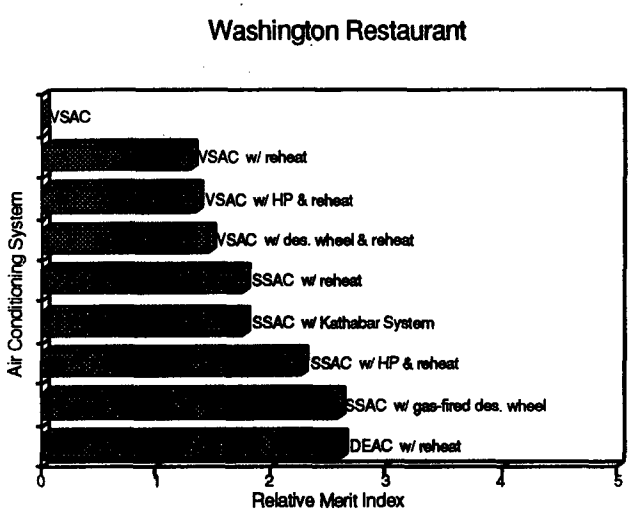




Figure 10.4 - Restaurant: AC Systems Relative Merit Index

Fast food restaurants require large amounts of outside air. The gas-fired hybrid system can condition large quantities of outside air with high operating efficiency in the "ventilation" mode as explained in Section 10.2. In this mode, the gas-fired hybrid system requires lower AC tonnage than the DEAC. For example, for the Miami fast food restaurant, the gas-fired hybrid system used a 25 ton air conditioner whereas the DEAC required a 37.5 ton air conditioner. This was also the case for St.louis and Washington.

In addition to the performance characteristics of AC systems, local gas to electric price ratios also played a role in ranking the AC systems for this application. The gas to electric price ratio is moderate (8.31) in Miami. Consequently, the gas-fired hybrid system performed better than the DEAC in the Miami fast food restaurant. The DEAC with reheat is the best electric AC alternative. Even though the Kathabar system had the lowest operating cost, the Relative Merit Index of this system was not higher than that of the DEAC in Miami. This is due to the Kathabar system's high initial cost and the gas to electric price ratio in Miami.

The gas to electric price ratio is low in St.Louis (4.71) and high in Washington (12.31). The DEAC had the highest Relative Merit Index in Washington. The Relative Merit Index of both the Kathabar dehumidifier hybrid system and the desiccant wheel gas-fired hybrid system were higher than that of the DEAC in St.Louis.

## 10.4 Conclusion

The DEAC and the "alternative" AC options available were compared for the modeled fast food restaurant in Miami, Washington and St.Louis. The simulation results show that the DEAC is the only electric AC system that can compete with the performance of the gas-fired hybrid systems commonly used for low-SHR building applications. The DEAC proved to be the best AC option for the Washington fast food restaurant. In the other two cities (Miami and St.Louis) the desiccant wheel gas-fired hybrid system had a higher Relative Merit Index ranking than the DEAC. In addition to the performance characteristics of the AC systems, the gas to electric price ratio also played a role in their ranking for this application.

References

1. Frey D.J., P.A. Oatman, 1989. "The Model Electric Restaurant, Restaurant Subsystem Analysis and Evaluation", EPRI CU 6702 Volume 1 and 2.

2. T.J. Marciniak, R.N. Koopman, D.R. Kosar, "Gas-fired Desiccant Dehumidification System in a Quick-Service Restaurant", ASHRAE Transactions: Symposia, 1991, P 657-666.

3. Advanced Desiccant Cooling System Development: Manufacturing Costs and Applications Economics, Prepared by Tecogen, Inc. for Gas Research Institute, (GRI-8710043.2).

## 11.0 OFFICE BUILDING AIR CONDITIONING ENERGY ANALYSIS

In this Chapter we discuss the results of the seasonal performance simulations conducted to compare the DEAC with the alternative AC systems for a 10,000 square foot office building modeled for this study.

### 11.1 Introduction

Energy use in commercial buildings has received considerable attention over the past thirty years. The 1989 DOE survey on commercial buildings (1) shows an expenditure level of 70.8 billion dollars on energy for commercial buildings. Office buildings consume 21 percent of all the energy used in commercial buildings (1).

In the past twenty years the office building air conditioning loads were primarily internal high SHR loads rather than external weather loads. However, this may change with the advent of ASHRAE standard 62-89 which requires 20 cfm/person of ventilation air for office buildings compared to the earlier 5 cfm/person. The increase in latent cooling loads in humid climates could pose a problem in areas such as conference rooms where the density of people is sometimes very high. On the other hand, increased use of electronic equipment in modern offices has increased the internal sensible load. Hence, additional dehumidification capacity may only be required in some areas of the office building such as conference rooms noted as above.

To asses the applicability of the DEAC to office building applications, the characteristics of a typical 10,000 sq.ft office building were modeled. The building envelope characteristics, operating schedules and internal load generation details were previously discussed in Section 7.2.

### 11.2 Simulation Basis

The office building loads were determined by running simulations using the "ideal" AC. This process is described in some detail in Chapter 8.0. Estimates of the sensible and latent cooling loads and also the building load-SHR profile were obtained from the "ideal" AC simulations. The average load-SHR values ranged from 0.8 to 0.9 for the three climates simulated. As previously noted, this high load-SHR is a result of the high sensible load of electronic equipment and lights used in the office buildings.

Since the average load-SHR values for the modeled office building were high, we used a baseline single-speed air conditioner model with an SHR of 0.84 at ARI rated conditions for this application. The other AC systems compared for the office building are listed in Table 11.1.

**TABLE 11.1**

| Medium Office: AC Systems Compared |
|---|
| 1. Conventional single-speed air conditioner (SSAC)<br>2. SSAC with Heat pipes<br>3. DEAC<br>4. SSAC with reheat<br>5. Variable-speed air conditioner (VSAC)<br>6. SSAC with gas-fired desiccant wheel<br>7. SSAC with Kathabar dehumidifier |

Commercial office building AC systems are typically controlled by thermostat only. We simulated thermostat control for the SSAC, Heat pipe-assisted SSAC and DEAC. For the VSAC, the variable speed fan was under humidistat control and the compressor was under thermostat control. The air-reheat systems and gas-fired systems were controlled by both humidistat and thermostat.

The gas-fired desiccant wheel system was operated in its conventional "recirculation mode". In this mode, the process air comes from the conditioned space and the regeneration air comes from outdoors. Also, for this case, the AC was in series with the dehumidifier.

The Kathabar system was operated in its conventional "mixed mode" for the office building. In this mode, the outside air is treated by the dehumidifier and the air conditioner is in parallel with the dehumidifier.

### 11.2.1 Size of AC Systems

The "ideal" AC simulation results showed that the design sensible and latent loads (98 percentile loads) were close to each other for all three cities. The sizes of the AC systems for all the three climates turned out to be the same. The sizes of the AC systems used in our simulations are listed in Table 11.2.

TABLE 11.2

| Office: AC Systems Size | | |
|---|---|---|
| AC System | AC Nominal Tons | Comments |
| SSAC | 25 | ---------- |
| SSAC with Heat Pipes | 30 | Heat Pipes: 2 rows, 14 fpi and face area: 60 $ft^2$ |
| DEAC | 28 | Wheel Depth: 1.5" and face area: 56 $ft^2$ |
| SSAC with reheat | 30 | Reheat: 25 kW |
| VSAC | 25 | Capacity at 90 Hz |
| SSAC with gas-fired desiccant wheel | 25 | Regen Temp: 284°F, Wheel face area: 6.6 $ft^2$ |
| SSAC with Kathabar Dehumidifier | 25 | Kathabar size: 2 latent tons |

### 11.2.2 Capital Costs of AC Systems

The capital costs of the air conditioning systems were estimated using the component costs listed in Appendix E. The VSAC unit size used in our office building simulation (25 ton) is not currently available in the market. To estimate the capital cost of this unit, we used the capital cost data of residential units currently available in the market (i.e 1500 $/ton).

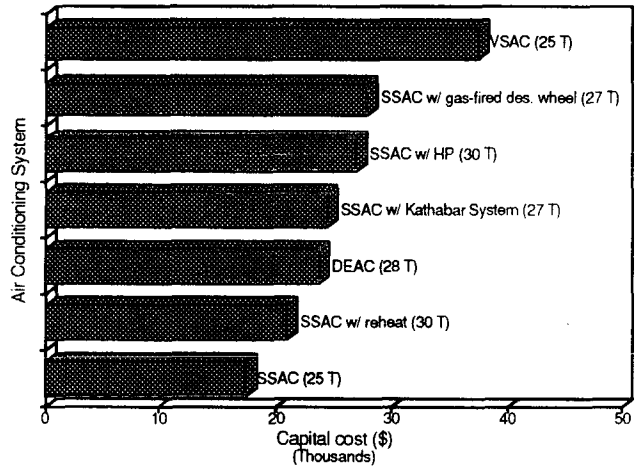
## 11.3 Simulation Results

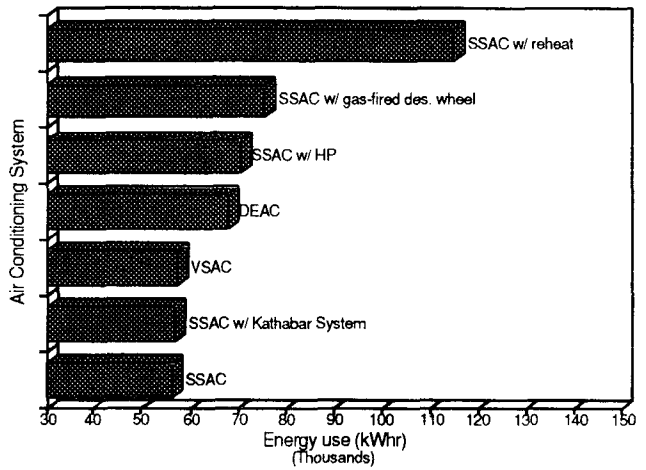
Seasonal simulation cases were run for the model office building using a matrix of TMY data for Miami, St.Louis and Washington and the air conditioning systems listed in Table 11.1.

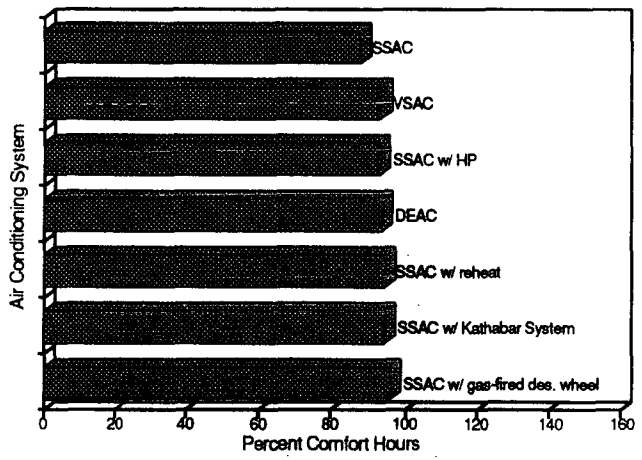
Figures 11.1, 11.2, and 11.3 compare, for each of the cities and AC systems, the AC capital costs, level of comfort provided, seasonal energy efficiency ratio (SEER), and energy use.

Because the sensible load is so dominant in office buildings, there is relatively little difference among the comfort levels provided by the various AC systems for Miami and, as would be expected, even smaller variation in the less humid climates of St.Louis and Washington.

The comfort provided by the air-reheat systems and gas-fired hybrid systems were higher than that of the DEAC because each was controlled by a humidistat in addition to a thermostat. Except for these two cases, the DEAC provided the maximum comfort.



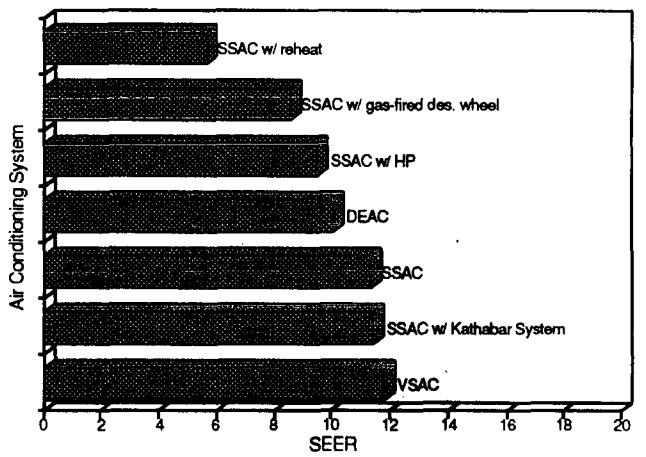
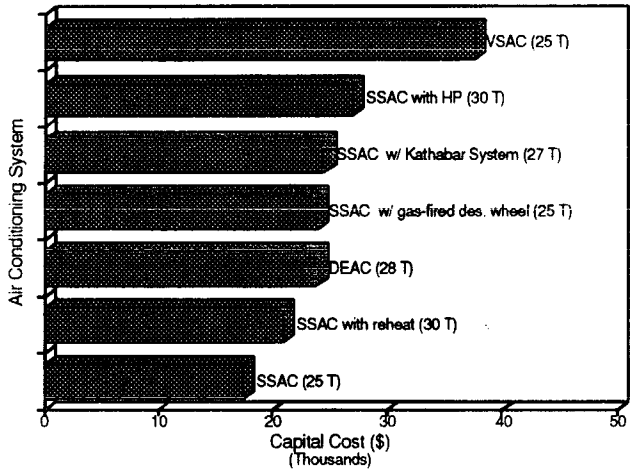
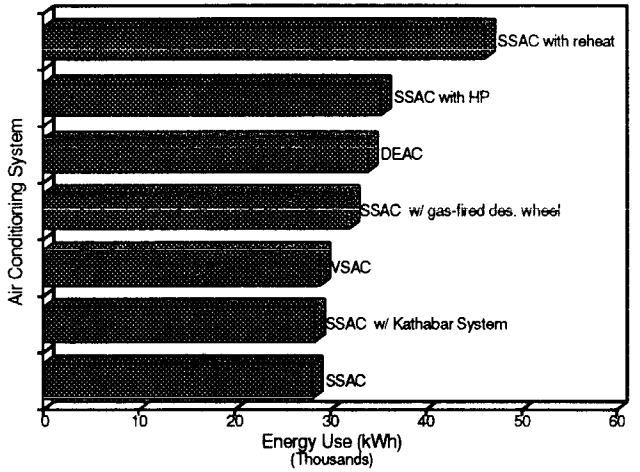
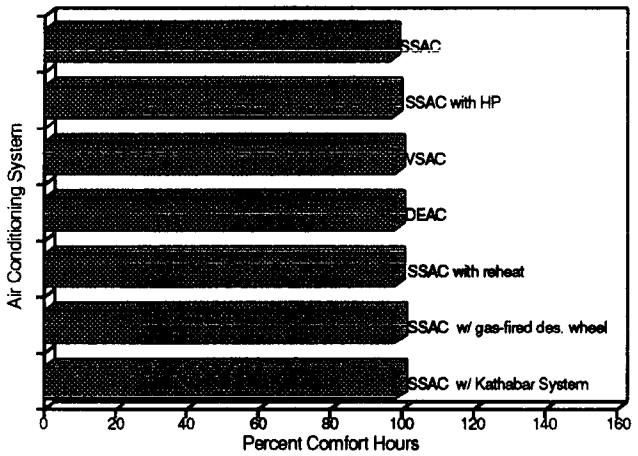




Figure 11.1 - Miami Office: AC Systems Performance



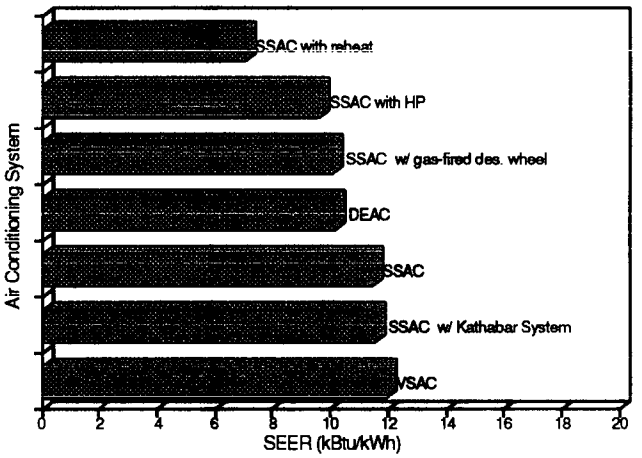
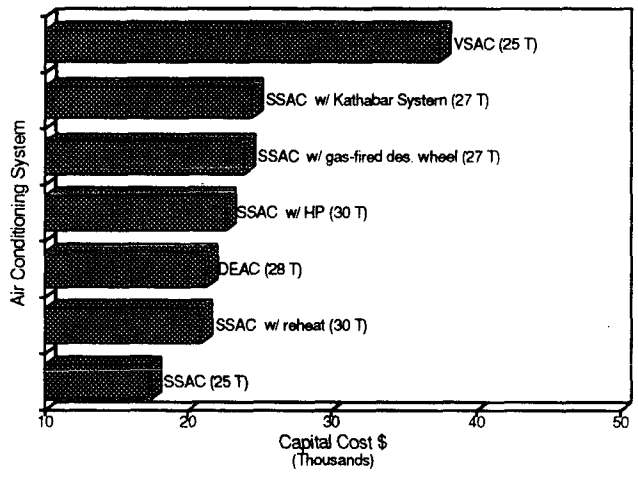
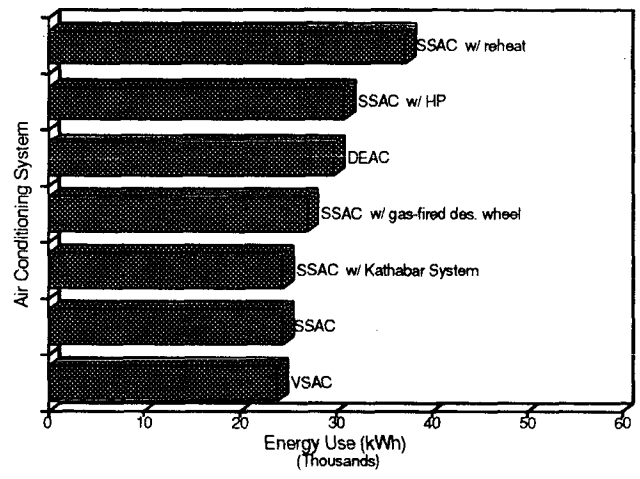
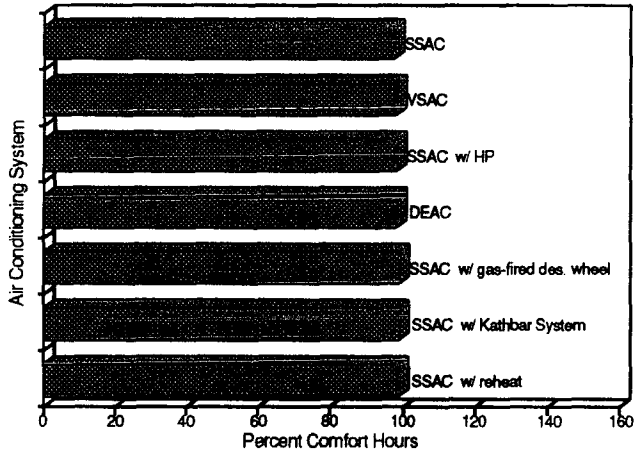




Figure 11.2 - St. Louis Office: AC Systems Performance



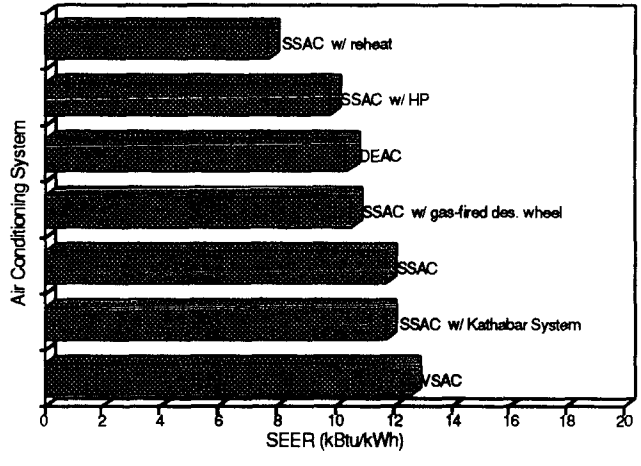




Figure 11.3 - Washington Office: AC Systems Performance

Figure 11.1 shows that the VSAC is among the lowest energy users and maintains comfortable conditions for a few more hours than the baseline SSAC for the Miami office. The VSAC has the advantage of variable speed fan control to reduce the air flow rate over the coil and thereby increase the dehumidification fraction. Hence, it was able to meet the latent loads for a few more hours than the SSAC.

The gas-fired desiccant system used more energy than the DEAC for the Miami office building. However, in the other two cities the equivalent electrical energy use of the desiccant wheel gas-fired system was less than that of the DEAC. This is due to the fact that the annual ventilation load for the office building in Miami is higher than the ventilation load in the other two cities. To meet the high ventilation loads, the gas-fired system uses more gas in Miami and the equivalent energy use in kWh is higher than that used by the DEAC for this climate.

Since all the AC systems, including the SSAC provided the required comfort for more than 90 percent of the cooling period, the Relative Merit Index of the systems was very small (< 1.0). As a result, only the Net Present Values (NPV) of the systems were used to rank them. The results (Figure 11.4) show that the NPV of the high efficiency AC with Kathabar dehumidifier is the best choice for the Miami office building. As indicated earlier in Section 11.2, this system was simulated in the "mixed" mode. In this mode, the dehumidifier treats only the ventilation air. The DEAC was the next best system for the Miami office building.

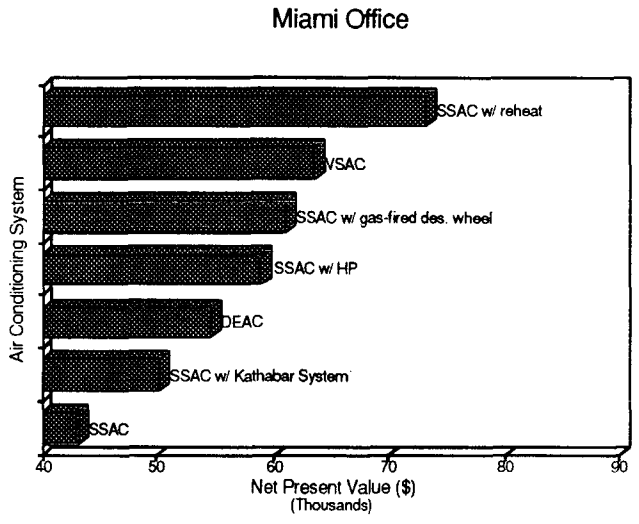
All of the AC systems modeled for this case could maintain comparable comfort conditions throughout the cooling season in Washington and St.Louis climates. The NPV values show that the SSAC is the best choice for these climates. Hence, dehumidification enhancement of the conventional air conditioner is not really required for these cities. However, if a separate air conditioner was required to meet, for instance, the conference room load requirements, then the DEAC would be a viable option.

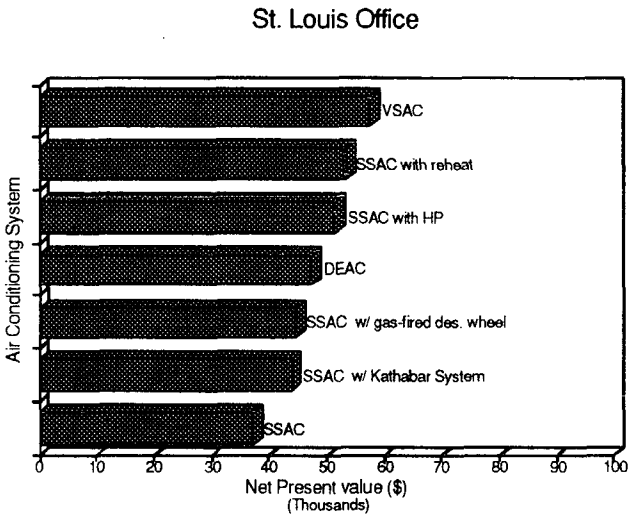
## 11.4 Conclusion

The commercial office buildings air conditioning loads are primarily internal loads rather than external weather loads. The modeled office building hourly load-SHR values were high enough to be met by a single speed AC for 90 percent of the cooling period in all the three climates. A high-efficiency AC with a Kathabar dehumidifier system had the highest ranking for the Miami office building. Dehumidification enhancement of the conventional single speed air conditioner is not really required for this building application in Washington and St.louis climates.

Reference:

1. Commercial buildings energy consumption and expenditures survey, DOE, 1989, DOE/EIA-0318 (89)



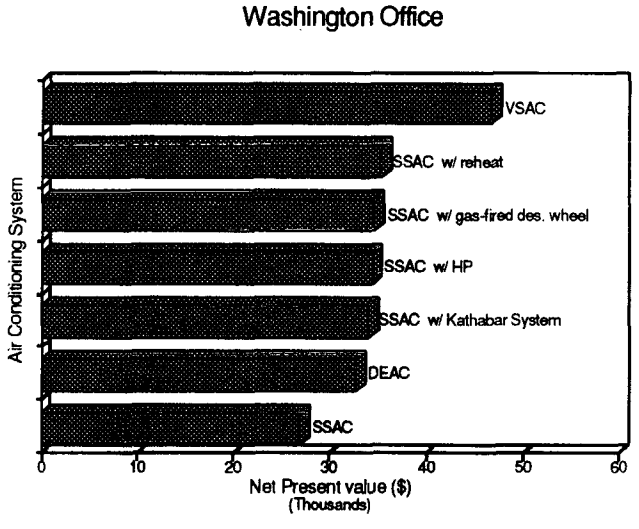




Figure 11.4 - Office: AC Systems Net Present Value

# APPENDIX A

## COMPARISON OF RESULTS FROM FSEC 2.2 AND ESPRE 2.1 FOR A RESIDENCE

### Overview

In the introductory sections of this report (Chapter 6.0), a rationale for the use of the software "FSEC" was presented and, up to this point, all the numerical results have been generated using the FSEC code (1) enhanced by the equipment models that have been added to it and resulting in version 2.2.

In this appendix we present a limited comparison of FSEC with an alternative code ESPRE (EPRI Simplified Program for Residential Energy), a well known residential simulation model (2). There are, in fact, a variety of residential modeling software packages commercially available. Based on previous studies (3), we selected ESPRE. None of the alternative packages, including ESPRE, have the capability to deal directly with moisture sorption/desorption by materials in a residence. This is the feature that makes the FSEC software unique. ESPRE 2.1, an integrated microcomputer program, was designed to simulate hourly loads and energy use patterns. The software is quite user friendly.

The goal of this task was to compare residential energy use results from FSEC 2.2 to the results of modeling using ESPRE 2.1 for the case of identical residences, air conditioning equipment and climates.

The following parameters were predicted by FSEC and then ESPRE for a 1500 sq.ft. modeled residence. The characteristics of the residence modeled are described in Table A-1.

1. Annual sensible and latent loads met by the air conditioner.
2. Air conditioner (compressor and fans) energy use.
3. Monthly average indoor temperature and humidity conditions.
4. Number of hours/year that the indoor RH level was greater than 60%

As noted previously, the major difference between the two software packages for this specific task was in modeling the moisture transport among the building materials and the conditioned space to predict the indoor humidity.

The FSEC software can predict indoor humidity levels using either a distributed or a lumped moisture transport model. The lumped moisture transport model, the Effective Moisture Penetration Depth (EMPD) concept was discussed in Chapter 6. ESPRE, on the other hand, assumes that the capacity of moisture storage is ten times the room air mass (4). This "lumped" moisture storage model is also available as an option in the FSEC software. To compare the two software packages on an equal basis, the results from the lumped moisture storage model of FSEC was compared with the results from ESPRE. Following this, we performed simulations using the EMPD approach for the FSEC residence. These results were

also compared with the ESPRE results to identify the differences between the two moisture modeling approaches.

Since treatment of moisture is one of the more important issues of the comparison, three weather conditions (Miami, Tallahassee and Atlanta), representing a wide range of annual dehumidification loads, were selected for the study.

Some of the major software differences between the two programs are discussed below. Following that, simulation results are presented and discussed.

**Software Differences**

Primary software modeling differences include the following:

1. Simulation of the FSEC building envelope characteristics requires values for thermal properties of wall and roof components whereas a combined "U" value and capacitance is used for ESPRE.

2. Hourly internal heat gain, infiltration and ventilation schedules can be specified for FSEC whereas the smallest schedule time period for ESPRE is two hours.

3. The ESPRE user has to specify the variations in air conditioner capacity and compressor power as a function of indoor wet bulb and outdoor dry bulb temperature for the air conditioner model. FSEC uses air conditioner performance curve fits developed from either the Oak Ridge National Laboratory heat pump model (5) or the DOE-2 packaged unit default air conditioner performance curves (6). FSEC calculates the fan power based on the air flow rate and total pressure drop in the system, whereas fan power consumption is specified at rated conditions for ESPRE.

For the purposes of comparison, the differences in air conditioner models noted above were eliminated by incorporating the ESPRE AC model in the FSEC software.

Comparable wall properties and identical internal load schedules were used for all simulation cases in both the FSEC and ESPRE. Descriptive information for the residence is provided in Table A-1.

## TABLE A-1

| Residence Description | |
|---|---|
| Overall | |
| Construction: | Slab-on-grade, single story |
| Floor Area: | 1500 $ft^2$ |
| Glazing: | 21% of wall area equally distributed |
| Insulation | |
| Walls: | R-11 (drywall,fiberglass,siding) |
| Ceiling: | R-24 (drywall, fiberglass) |
| Infiltration and Internal Gains | |
| Internal gains: | 1130 W (77% sensible) |
| ACH: | 0.54 average |
| People | |
| Occupancy: | 2 |
| Gain: | 263 W (55% sensible) |
| Cooling/Heating Set Points: | 78°F/70°F |

Schedules (Fraction of maximum):

| | Hour | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| **Infiltration:** | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 | .65 |
| **Equipment and Lighting:** | .3 | .3 | .3 | .3 | .38 | .38 | .43 | .43 | .43 | .43 | .44 | .44 | .24 | .24 | .48 | .48 | .67 | .67 | .67 | .67 | .67 | .67 | .3 | .3 |

To account for thermal capacitance of furniture, beds, etc., ESPRE uses an internal lumped thermal mass model with an appropriate air-thermal mass convective coupling (0.2 BTU/hr-ft$^2$-F). "Light", "medium" and "heavy" thermal mass options are available to characterize the thermal capacitance of furniture, beds, etc., in ESPRE. We chose "light" thermal mass (2 BTU/hr-F-ft$^2$) to model this residence. The thermal mass properties expressed above are based on floor area of the zone.

As previously noted for moisture, FSEC software also has an option to model the room capacitance using a lumped thermal mass approach. This capacitance has to be specified in terms of "room air mass" for the FSEC model. The above ESPRE thermal mass (2 BTU/hr-F-ft$^2$) is approximately equivalent to 15 times the room air mass.

In addition to room thermal mass, ESPRE uses an overall thermal capacitance approach to model the heat storage characteristics of walls. We selected "light" wall with a thermal mass of 2 BTU/hr-F-ft$^2$ for the ESPRE residence model. As mentioned earlier, the FSEC software calculates the equivalent capacitance of a wall from the individual capacitance of the components. This overall capacitance was 1.5 BTU/hr-F-ft$^2$ for each wall for the FSEC model residence.

**Comparison of Results**

**Case 1: Predictions using Lumped Moisture Mass Model for Both FSEC and ESPRE**

As noted in page A-1 ESPRE models moisture mass as having ten times the actual building air mass. For consistency in Case 1, we modeled the moisture mass for the FSEC software using the same approach. Table A-2 compares the results predicted for both FSEC and ESPRE. The differences in loads and energy use between the two software codes are included in Table A-2.

## TABLE A-2

**Comparison of FSEC 2.2 and ESPRE 2.1: Annual Results**
**(Predictions Using Lumped Moisture Mass Model for both FSEC and ESPRE)**

| City | AC Sensible load (kBtu) | AC Latent load (kBtu) | Compr-essor Energy (kWh) | Supply Fan Energy (kWh) | Annual Average Space T (F) | Annual Average Space RH (%) |
|---|---|---|---|---|---|---|
| Miami (E) | 53133 | 18969 | 5906 | 526 | 77.2 | 55.4 |
| Miami (F) | 53618 | 19197 | 6366 | 536 | 77.7 | 55.5 |
| Percent Variation | -0.9% | 1.2% | -7.7% | -1.9% | -0.6% | -0.2% |
| Tallahassee (E) | 35136 | 11512 | 3832 | 342 | 75.2 | 52.6 |
| Tallahassee (F) | 35019 | 11630 | 4118 | 348 | 76.5 | 51.5 |
| Percent Variation | 0.3% | -1.0% | -7.4% | -1.8% | -1.7% | -0.2% |
| Atlanta (E) | 25022 | 7200 | 2786 | 232 | 74.1 | 48.0 |
| Atlanta (F) | 24466 | 7182 | 2837 | 238 | 75.5 | 47.0 |
| Percent Variation | 2.2% | 0.2% | -1.8% | -2.5% | -1.3% | 2.0% |
| Average (absolute) percent variation | 1.1% | 0.7% | 5.6% | 2.0% | 1.2% | 0.8% |

[1] Note: E - ESPRE 2.1, F - FSEC 2.2

Annual sensible and latent cooling loads predicted by FSEC and ESPRE match very well (average within 2%). Also, the average space conditions predicted by each of the software codes match well. There is a larger difference in prediction of compressor energy use for all the cities. These differences are the result of the fact that the FSEC model includes the air conditioning energy use for meeting the supply air fan heat and ESPRE does not. A nominal air flow rate of 400 cfm/ton was used in FSEC simulations to calculate this fan heat. Table A-3 shows the number of hours of indoor conditions with high humidity (RH > 60%) predicted by each of the two software programs.

**TABLE A-3**

| City | Hours RH > 60% | Hours RH > 70% | Hours RH > 80% |
|---|---|---|---|
| Miami (E) | 2457 | 200 | 12 |
| Miami (F) | 2210 | 42 | 0 |
| Tallahassee (E) | 2370 | 431 | 44 |
| Tallahassee (F) | 1653 | 70 | 0 |
| Atlanta (E) | 1755 | 337 | 34 |
| Atlanta (F) | 958 | 45 | 0 |

**Case 2: FSEC model with EMPD approach**

In this case, instead of a lumped moisture mass to model the moisture capacitance effects, the moisture adsorption/desorption characteristics of walls, carpet and furniture were modeled using an EMPD value of 2 mm (0.08"). The residence carpeted area was assumed to be 1500 square feet. In addition, the equivalent of 1100 square feet of partition wall board was used to simulate the convective coupling of the air-thermal mass and moisture adsorption/desorption characteristics of bed and furniture in this case. The results from these FSEC simulation runs are compared with the ESPRE simulation results of Case 1 in Table A-4.

## TABLE A-4

**Comparison of ESPRE and FSEC 2.2: Annual Results (FSEC model with EMPD approach)**

| City | AC Sensible load (kBtu) | AC Latent load (kBtu) | Compr-essor Energy (kWh) | Supply Fan Energy (kWh) | Annual Average Space T (F) | Annual Average Space RH (%) |
|---|---|---|---|---|---|---|
| Miami (E) | 53133 | 18969 | 5906 | 526 | 77.2 | 55.4 |
| Miami (F) | 50377 | 16702 | 5984 | 501 | 77.5 | 57.5 |
| Percent Variation | 5.1% | 12.0% | -1.3% | 4.8% | -0.4% | -3.8% |
| Tallahassee (E) | 35136 | 11512 | 3832 | 342 | 75.2 | 52.6 |
| Tallahassee (F) | 32547 | 10048 | 3832 | 322 | 75.9 | 53.6 |
| Percent Variation | 7.3% | 12.7% | 0.0% | 5.8% | -0.9% | -1.9% |
| Atlanta (E) | 25022 | 7200 | 2786 | 232 | 74.1 | 48.0 |
| Atlanta (F) | 22615 | 6176 | 2623 | 219 | 74.8 | 48.8 |
| Percent Variation | 9.6% | 14.2% | 5.8% | -5.6% | -0.9% | -1.6% |
| Average (absolute) percent variation | 7.3% | 13.0% | 2.4% | 5.4% | 0.7% | 2.4% |

As shown in Table A-4, the FSEC model using the EMPD approach predicted higher average indoor relative humidity values than those shown in Case 1 Table A-2 for all three cities.

The FSEC model with EMPD approach also predicted lower latent loads than ESPRE for the three cities. This is due to the increased indoor humidity levels predicted by FSEC which result in reduced infiltration latent loads. Table A-5 shows the number of hours of indoor conditions with high humidity (RH > 60%) predicted by each of the two software programs.

**TABLE A-5**

| City | Hours RH > 60% | Hours RH > 70% | Hours RH > 80% |
|---|---|---|---|
| Miami (E) | 2457 | 200 | 12 |
| Miami (F) | 3247 | 681 | 3 |
| Tallahassee (E) | 2370 | 431 | 44 |
| Tallahassee (F) | 2384 | 825 | 3 |
| Atlanta (E) | 1755 | 337 | 34 |
| Atlanta (F) | 1520 | 458 | 21 |

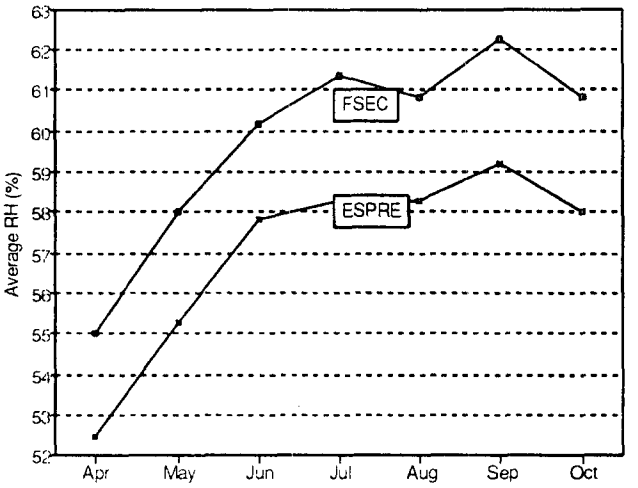
Monthly average humidity values as predicted by FSEC with EMPD approach and ESPRE were presented in Figure A.1 for all the three cities for the summer months (April - October). As shown in Figure A.1 the ESPRE model predicts RH values within 3% of the values predicted by the FSEC EMPD model in all three cities.

**Summary**

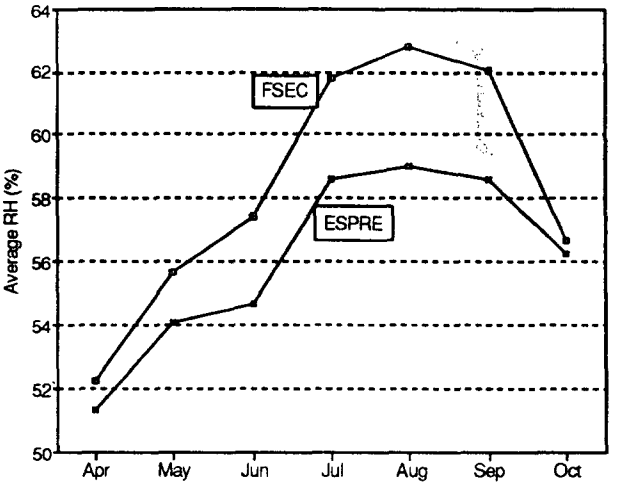
The AC loads, energy use and indoor conditions were predicted for a modeled residence in Miami, Atlanta and Tallahassee using FSEC 2.2 with a simplified moisture capacitance approach similar to that used in ESPRE. The results were compared with the corresponding results from ESPRE 2.1 simulations. The average variation in prediction of sensible and latent loads was within two percent. The average variation in AC energy use was within 6 percent. The relatively large difference in AC energy use can be attributed to the fact that the FSEC model air conditioning model accounts for fan heat removal by the AC and ESPRE does not.

The annual latent loads removed by the AC when FSEC incorporated EMPD were consistently lower (12 to 15%) than the values predicted by the ESPRE model. This is mainly due to the difference in the indoor RH levels predicted by the two software codes.

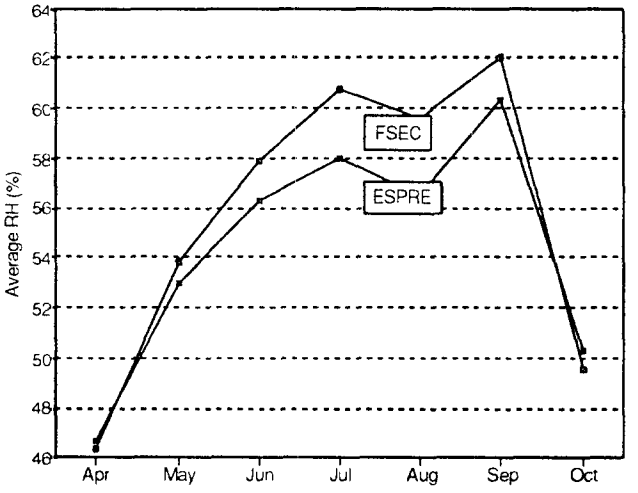
The FSEC software predicts higher indoor humidity levels using the EMPD approach. Accurate prediction of humidity levels is important in humid climates to analyze the advantages of various AC dehumidification enhancements. Hence, it is advantageous to use a modeling program such as FSEC which can take advantage of data for the moisture adsorption and desorption potential of various materials.




Miami




Tallahasse




Atlanta

Figure A.1 - Comparison of Predicted Indoor RH Using FSEC and ESPRE

References


1. Keresteciogiu, A., Swami, M., Brahma, P., Gu, L., Fairey, P., Chandra, S. "Users Manual: Florida Software for Enervironment (Energy and Environment) Computation", Florida Solar Energy center. August 1989.

2. ESPRE User's Manual, Prepared by Arthur D. Little, Inc. Cambridge, Massachusetts

3. "Evaluation of Existing Programs for Simulation of Residential Buildings Energy Use", EPRI report (EA-2575) and GRI report (81-0047), 1982.

4. R.L. Merriam, "ESPRE 2.1 Engineering Model" Simplified Energy Analysis Methods for Residential Buildings prepared for EPRI, Aug 1992.

5. The ORNL modulating heat pump design tool - user's guide C.K. Rice June 1991.

6. DOE. 1982. DOE-2 engineers manual. LBL-11353. Berkeley, CA: Lawrence Berkeley Laboratory.

## APPENDIX B

## EVALUATION OF RUN AROUND BELT CONFIGURATION FOR DEAC SYSTEM

### Objective

The objective of this task was to assess the viability of a "run around belt" configuration as an alternative to using a desiccant wheel for the DEAC system. Integration of a desiccant wheel in an air conditioner requires changing the conventional ducting design to allow the return air flow to be in a parallel but opposite flow direction to the supply duct air flow. The run-around belt, on the other hand, could literally surround the evaporator coil and as it rotates around the evaporator the belt material would alternately be absorbing moisture or being regenerated. This configuration would allow for a simpler and more conventional ducting system. In addition, we looked at the belt option because there are a number of materials which are hydrophilic and low cost. Two such belt materials were modeled in this study: activated carbon mat and jute.

### Materials

Fibrous activated carbon mat was previously used in a study of a direct solar regenerated dehumidifier by T. Ohigashi, et.al.(1). In addition to its dehumidifying characteristics, this material has been used commercially to absorb solvents from exhaust gases. It has a specific surface area in the range of 800-1000 $m^2/g$ (1) and a useful life of about 1.5 years (2). Jute is a low cost natural fiber hydrophilic textile material.

Other properties and configuration of the two materials used in the simulations are shown in Table B-1.

**TABLE B-1**

| Description | Activated Carbon | Jute |
|---|---|---|
| Bulk density | 500 $Kg/m^3$ | 1520 $Kg/m^3$ |
| Specific heat | 1109 J/kg.°K | 1339 J/kg.°K |
| Maximum water absorption content | 0.27 Kg/Kg of desiccant | 0.22 Kg/Kg of desiccant |
| Void fraction | 0.6 | 0.6 |
| Thickness of mat | 5, 10 mm (0.2", 0.4") | 5, 10 mm (0.2", 0.4") |

**Analysis**

Among the desirable characteristics for a dehumidifier desiccant material are:

a. High heat and mass transfer $N_{tu}$ values
b. A high ratio of heat and mass transfer to friction losses
c. Compact transfer surface (large heat transfer area for a given volume of the material)
d. Low cost
e. Long life and ease of maintenance

The following comments address some of these characteristics for the run-around belt configuration assuming carbon mat and jute as the desiccant materials.

The run-around belt heat transfer and pressure drop characteristics were determined assuming that both the carbon fiber and jute materials could be treated as matrix or packed bed configurations. Isotherm characteristics (1, 5) of the two selected materials were curve-fit and used to predict the adsorption and desorption characteristics of the two belt materials.

Since the fabric structure of both belt materials is similar to a randomly stacked woven matrix, the heat transfer coefficient was estimated using the experimental data published by Kays and London (3) for a woven-screen matrix.

The mass transfer process was modeled using an effective Lewis number which accounted for both convective resistance and solid-side resistance of the mass transfer diffusion process. For pure convection in an air/water system, the Lewis number is nearly unity. When the solid-side resistance to mass transfer increases, as it does in a packed bed configuration, the effective Lewis number also increases. In our cases, we assumed an effective Lewis number of 3 for the matrices based on studies by Barlow (4).

Pressure drop through the desiccant subsystem material is a factor of concern in a DEAC system. For a nominal air flow rate of 400 cfm/ton, a one inch thick silica gel wheel with laminar flow channels and two square feet of face area per ton of air conditioning will have an acceptable pressure drop (approximately 0.14 inch W.C).

On the other hand, matrix or packed bed geometries tend to have relatively high pressure drops. In order to maintain a realistic pressure drop with the run-around belts we reduced the face velocity through each side of the belt to reduce the pressure drop. We estimated a pressure drop of 0.65 inches of water column across a 0.4 inch thickness of the mat using our simulation model. The void fraction of the mat was 0.6 and the face velocity was maintained at 80 fpm for this simulation. The pressure drop value estimated is consistent with the experimental results published by T. Ohigashi, et.al.(1) for activated carbon fiber. Based on the above, the face velocity of 80 fpm was used for both the carbon and jute in all the desiccant belt simulations.

The rational of applying the reduced face velocity (80 fpm) to the jute material was based on the fact that both the materials were modeled with the same void fraction and this is the primary factor in determining pressure drop.

The desiccant portion of the modeling was carried out using a modified version of the simulation software code developed by Kirk Collier of Enerscope. The geometric characteristics of the woven screen matrix were accounted for in the modified version used to predict the heat and mass transfer coefficients and pressure drop values. For example, the desiccant particles were assumed to be cylindrical and uniformly spaced.

Using the above information, we ran simulations for the two selected materials; carbon fiber mat and jute. The results of the simulations are summarized in Table B-2.

**TABLE B-2**

| Description | Carbon Mat | Jute |
|---|---|---|
| AC: | | |
| Capacity | 2 T | 2 T |
| Rated EER | 12.0 kBtu/kW | 12.0 kBtu/kW |
| Rated SHR | 0.84 | 0.84 |
| Air Flow | 800 CFM | 800 CFM |
| Desiccant belt: | | |
| Thickness of mat | 0.2", 0.4" | 0.2", 0.4" |
| Face Area of mat | 10 sq.ft/side | 10 sq.ft/side |
| $N_{tu}$ | 0.421, 0.842 | 0.421, 0.842 |
| Rotational Speed | 10 CPH | 10 CPH |
| Estimated pressure drop across the mat including entrance and exit losses (Simulation) | 0.35", 0.65" | 0.35", 0.65" |
| DEAC: | | |
| EER,**SHR** ($N_{tu}$: 0.421) | 10.82, **0.691** | 10.8, **0.620** |
| EER,**SHR** ($N_{tu}$: 0.842 | 10.00, **0.588** | 9.96, **0.528** |

For the jute and the activated carbon belt, the results of the simulation (Table B-2) showed that, to meet an SHR of 0.588, the EER of the DEAC run-around belt system would be 11 to 12 percent lower than if the silica gel wheel configuration was used.

If the face area of the run-around belt increases to further lower the pressure drop, the benefit is offset by an increase in the capital cost of ducting to accommodate the expansion. On the other hand, both the duct and desiccant materials have relatively low cost (Table B-3).

**TABLE B-3**

| Cost of Desiccant Belt Materials | |
|---|---|
| 1. Activated Carbon Fiber Mat: | 0.50 $/sq.ft for 0.25" thick |
| 2. Jute Fiber: | 0.20 $/lb |

References:


1. OhiGashi, T., T. Kishimoto, Y. Imamura, M. Shimada, H. Komagata, Dehumidifying Capacity of Fibrous Activated Carbon Mat Regenerated by Direct Solar Radiation, a Paper presented at ASES annual meeting, 1983.

2. Encyclopedia of Chemical Technology, Vol 4.

3. Kays and London, "Compact Heat Exchangers", Third Edition, McGraw Hill Publication, 1984.

4. Barlow, R. S., An Assessment of Dehumidifier Geometries for Desiccant Cooling Systems, Solar Energy Research institute, 1983.

5. Byron D. Tapley, "Handbook of Engineering Fundamentals", Fourth Edition, Wiley-Interscience Publication

# APPENDIX C

## TMY WEATHER DATA

Miami, St.Louis, and Washington

Figure C.1 - Monthly Average Weather Data (Dry bulb, Dewpoint, RH): Miami

Figure C.2 - Monthly Average Weather Data (Solar Flux, Absolute Humidity): Miami

Figure C.3 - Monthly Average Weather Data (Dry bulb, Dewpoint, RH): St. Louis

Figure C.4 - Monthly Average Weather Data (Solar Flux, Absolute Humidity): St. Louis

Figure C.5 - Monthly Average Weather Data (Dry bulb, Dewpoint, RH): Washington

Figure C.6 - Monthly Average Weather Data (Solar Flux, Absolute Humidity): Washington

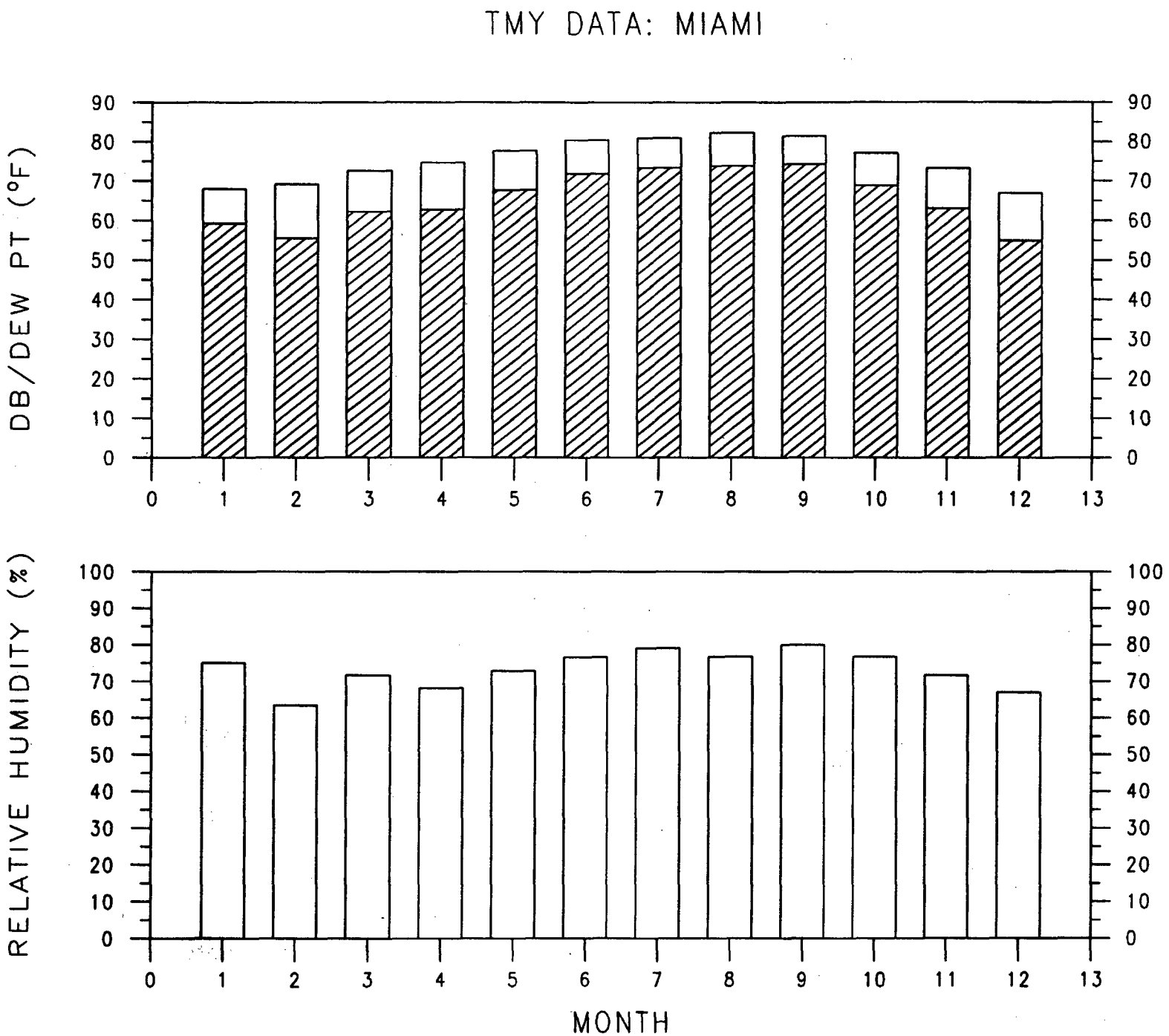



Figure C.1 - Monthly Average Weather Data (Dry bulb, Dewpoint, RH): Miami

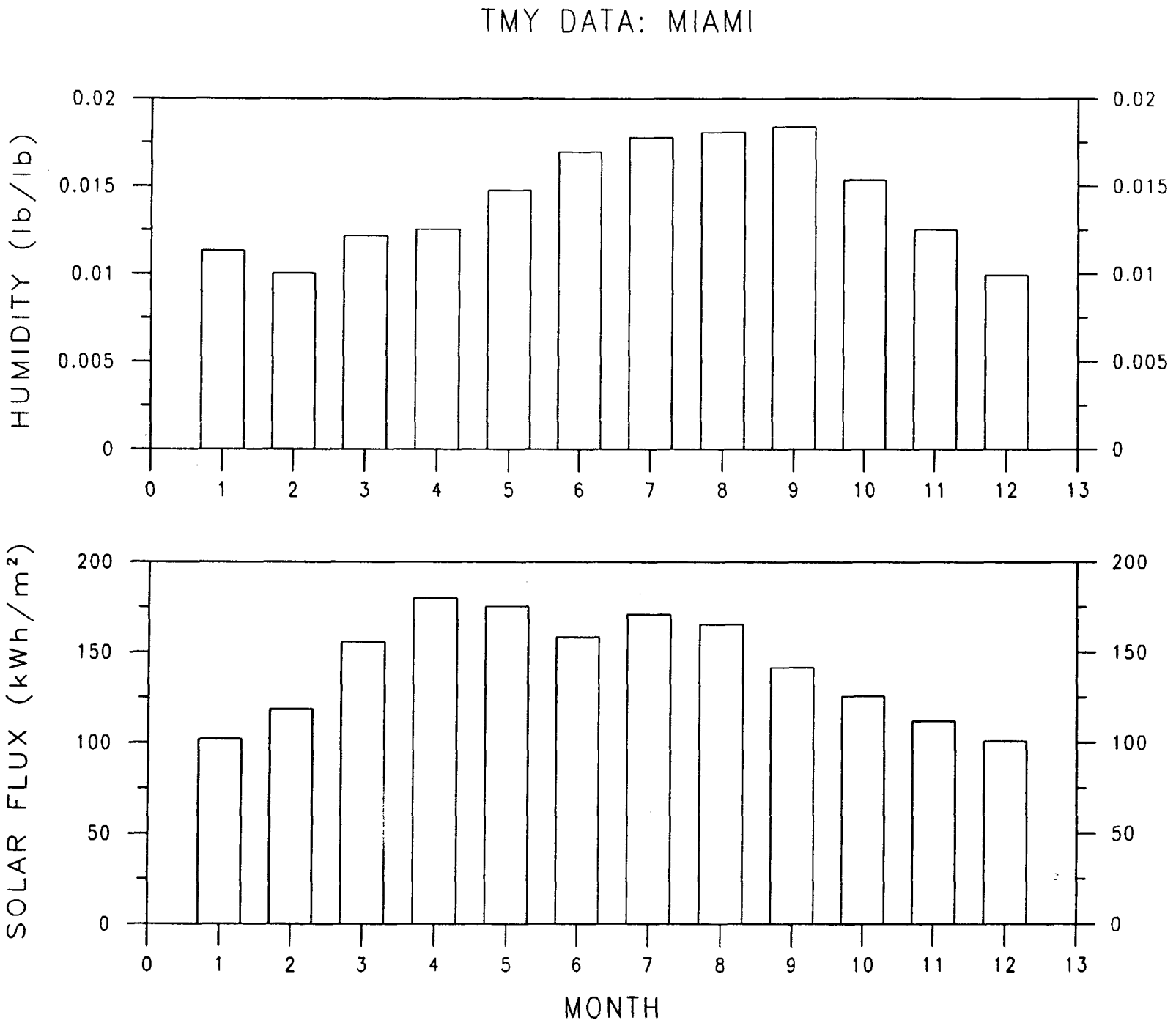



Figure C.2 - Monthly Average Weather Data (Solar Flux, Absolute Humidity): Miami

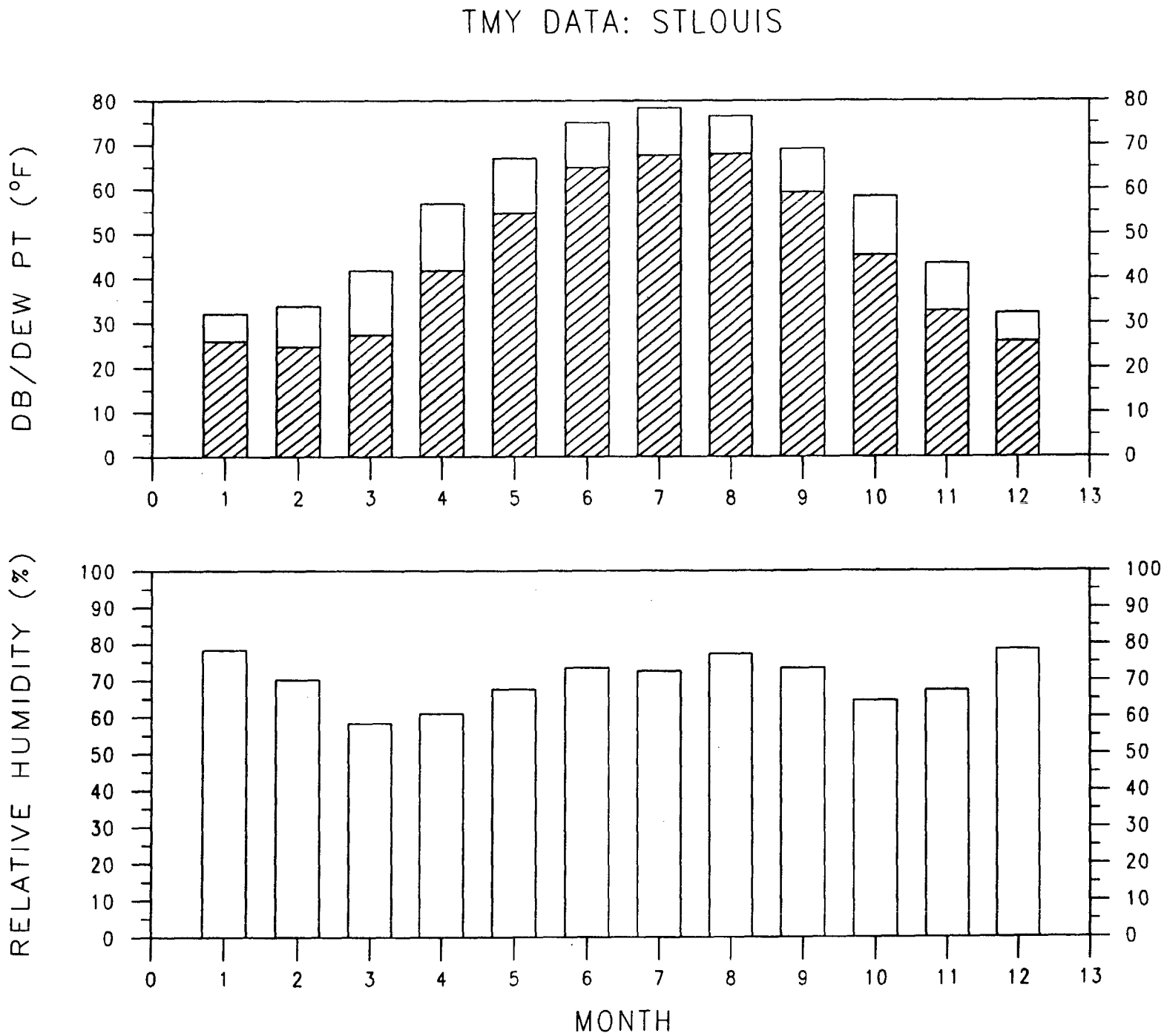



Figure C.3 - Monthly Average Weather Data (Dry bulb, Dewpoint, RH): St.Louis

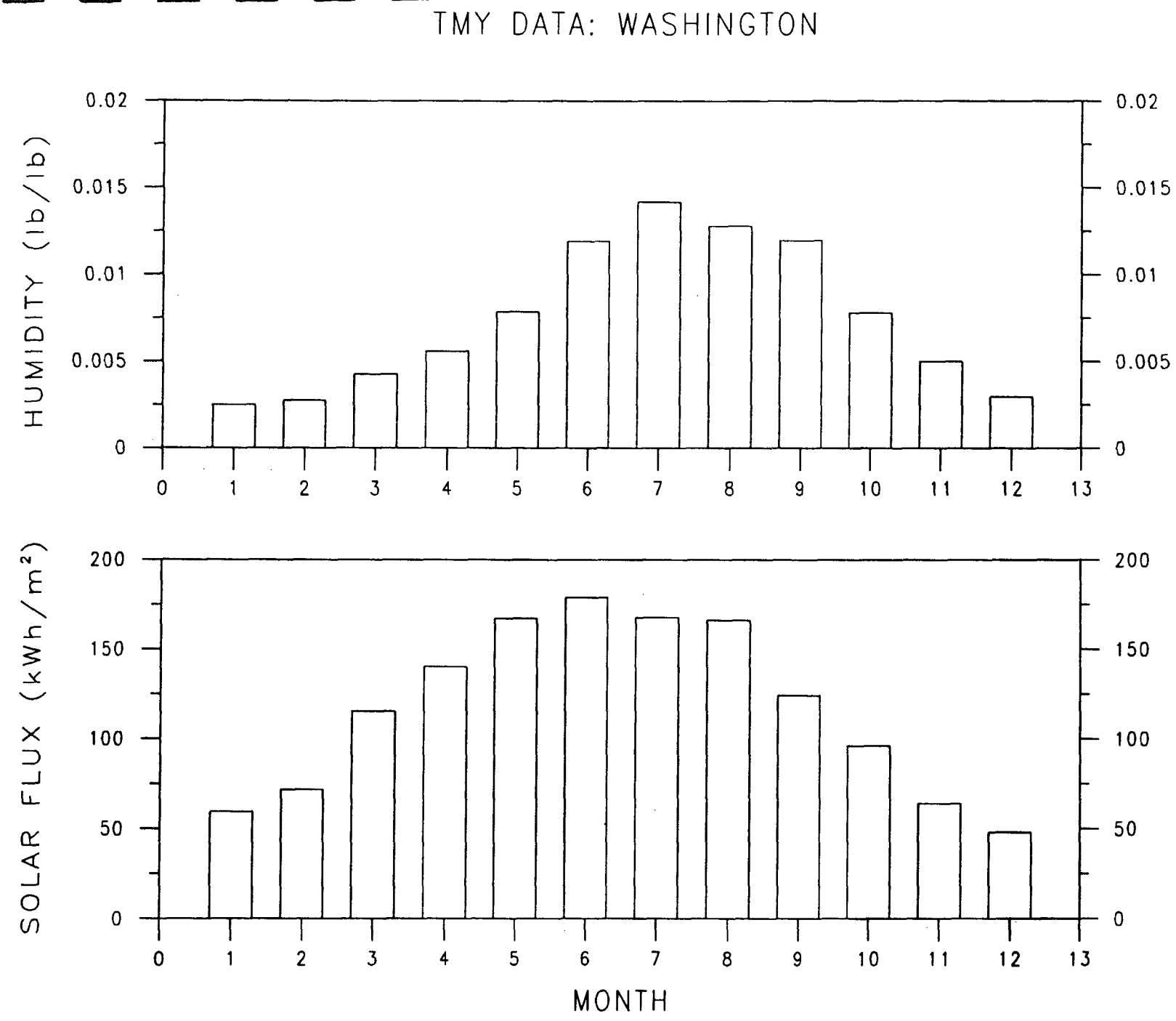



Figure C.4 - Monthly Average Weather Data (Solar Flux, Absolute Humidity): St.Louis

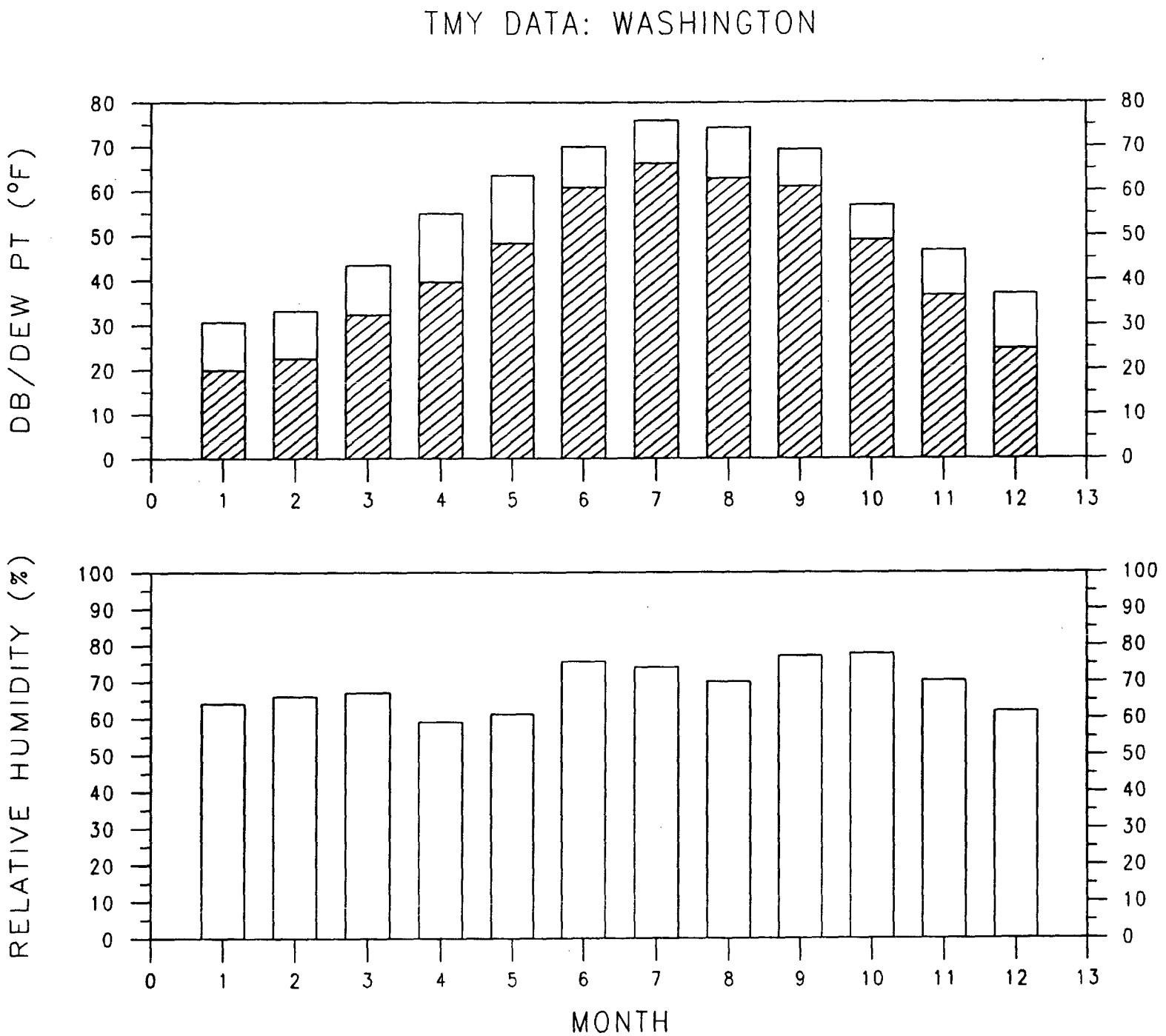



Figure C.5 - Monthly Average Weather Data (Dry bulb, Dewpoint, RH): Washington

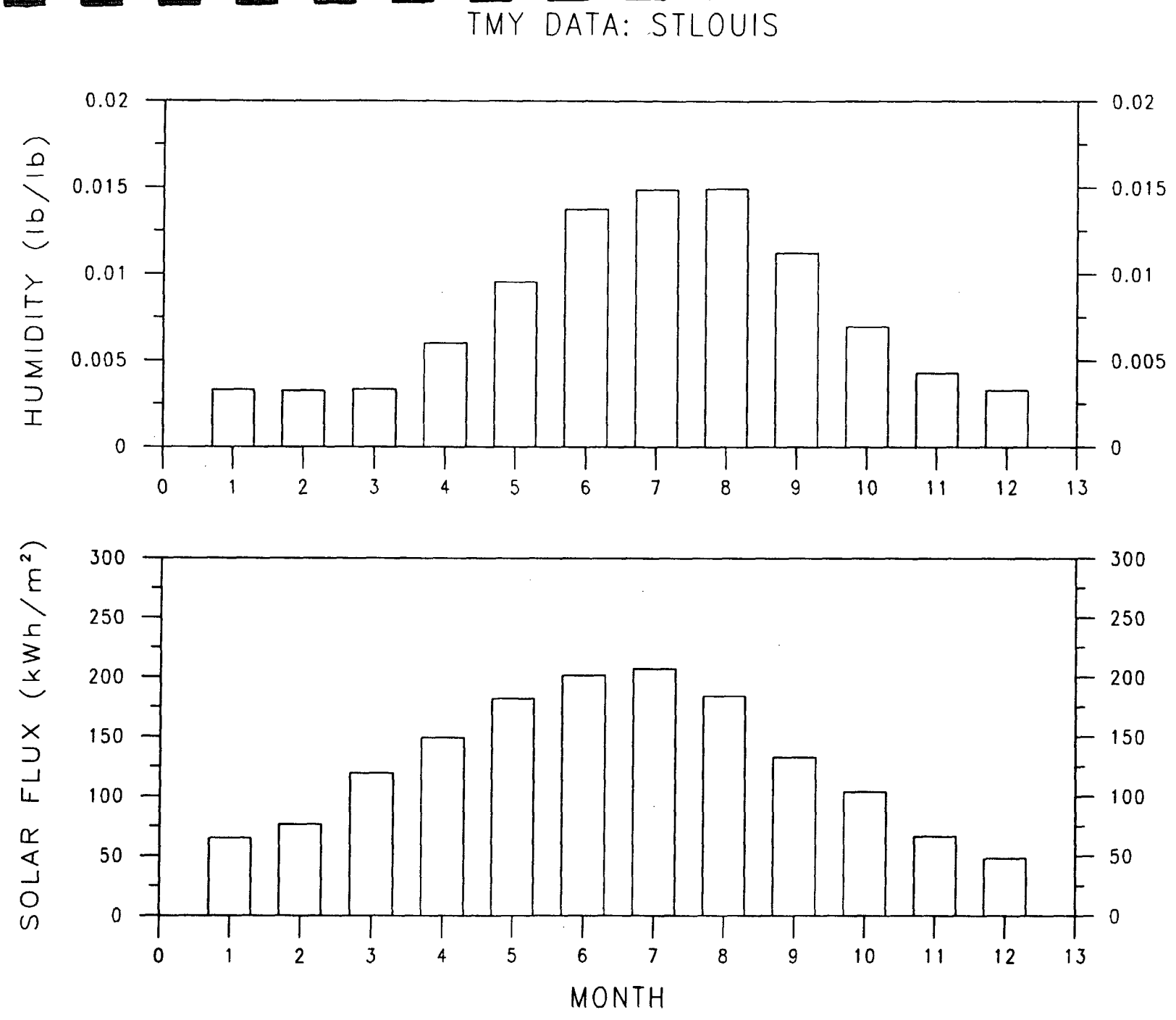



Figure C.6 - Monthly Average Weather Data (Solar Flux, Absolute Humidity): Washington

# APPENDIX D

## AIR CONDITIONING EQUIPMENT SIZES FOR RESTAURANTS

TABLE D-1 AC Sizes for Miami Restaurant (Dining)

TABLE D-2 AC Sizes for Miami Restaurant (Kitchen)

TABLE D-3 AC Sizes for Washington and St. Louis Restaurants (Dining)

TABLE D-4 AC Sizes for Washington and St. Louis Restaurants (Kitchen)

## TABLE D-1

### AC Sizes for Miami Restaurant (Dining)

| System Description | AC Nominal Tons | Components Description |
|---|---|---|
| SSAC | 15.0 | ---------- |
| SSAC with HP & reheat | 25.0 | HP: 3 rows, 14 fpi & face area/side: 25 sq. ft. Reheat: 50 kW |
| DEAC with reheat | 25.0 | Wheel Depth: 5"; Reheat: 50 kW |
| SSAC with reheat | 35.0 | Reheat: 50 kW |
| VSAC | 15.0 | ---------- |
| VSAC with HP & reheat | 25.0 | HP: 3 rows, 14 fpi & face area/side: 25 sq. ft. Reheat: 50 kW |
| VSAC with desiccant wheel & reheat | 25.0 | Wheel Depth: 5"; Reheat: 50 kW |
| VASC with reheat | 30.0 | Reheat: 50 kW |
| SSAC with gas-fired desiccant wheel | 15.0 | Wheel depth: 8.25", Total face area: 8.33 sq. ft., Regeneration temp: 284 F |
| SSAC with Kathabar system | 15.0 | Kathabar size (Nominal tons): 7, cfm/latent ton: 292 and Burner input 350 kBtu/hr |

## TABLE D-2

## AC Sizes for Miami Restaurant (Kitchen)

| System Description | AC Nominal Tons | Components Description |
|---|---|---|
| SSAC | 7.5 | ---------- |
| SSAC with HP & reheat | 12.5 | HP: 3 rows, 14 fpi & face area/side: 25 sq. ft. Reheat: 20 kW |
| DEAC with Reheat | 12.5 | Wheel Depth: 5"; Reheat: 20 kW |
| SSAC with reheat | 15.0 | Reheat: 20 kW |
| VSAC | 7.5 | ---------- |
| VSAC with HP & reheat | 12.5 | HP: 3 rows, 14 fpi & face area/side: 12.5 sq. ft. Reheat: 20 kW |
| VSAC with Desiccant wheel & reheat | 12.5 | Wheel Depth: 5"; Reheat: 20 kW |
| VASC with reheat | 15.0 | Reheat: 20 kW |
| SSAC with gas-fired desiccant wheel | 10.0 | Wheel depth: 8.25", Total face area: 4.0 sq. ft., Regen. temp: 284 F |
| SSAC with Kathabar system | 6.0 | Kathabar size (Nominal Tons): 4.00, Burner input 350 kBtu/hr |

## TABLE D-3

### AC Sizes for Washington and St. Louis Restaurants (Dining)

| System Description | AC Nominal Tons | Components Description |
|---|---|---|
| SSAC | 15.0 | ---------- |
| SSAC with HP & reheat | 25.0 | HP: 3 rows, 14 fpi & face area/side: 25 sq. ft. Reheat: 50 kW |
| DEAC with reheat | 25.0 | Wheel Depth: 5"; Reheat: 50 kW |
| SSAC with reheat | 30.0 | Reheat: 50 kW |
| VSAC | 15.0 | ---------- |
| VSAC with HP & reheat | 25.0 | HP: 3 rows, 14 fpi & face area/side: 25 sq. ft. Reheat: 50 kW |
| VSAC with desiccant wheel & reheat | 25.0 | Wheel Depth: 5"; Reheat: 50 kW |
| VASC with reheat | 25.0 | Reheat: 50 kW |
| SSAC with gas-fired desiccant wheel | 15.0 | Wheel depth: 8.25", Total face area: 8.33 sq. ft., Regen temp: 284 F |
| SSAC with Kathabar dehumidifier | 12.5 | Kathabar size (Nominal tons): 7.0, Burner input 350 kBtu/hr |

## TABLE D-4

### AC Sizes for Washington and St. Louis Restaurants (Kitchen)

| System Description | AC Nominal Tons | Components Description |
|---|---|---|
| SSAC | 7.5 | ---------- |
| SSAC with HP & reheat | 12.5 | HP: 3 rows, 14 fpi & face area/side: 12.5 sq. ft. Reheat: 20 kW |
| DEAC with reheat | 12.5 | Wheel Depth: 5"; Reheat: 20 kW |
| SSAC with reheat | 15.0 | Reheat: 20 kW |
| VSAC | 7.5 | ---------- |
| VSAC with HP & reheat | 12.5 | HP: 3 rows, 14 fpi & face area/side: 12.5 sq. ft. Reheat: 20 kW |
| VSAC with desiccant wheel & reheat | 12.5 | Wheel Depth: 5"; Reheat: 20 kW |
| VASC with reheat | 12.5 | Reheat: 20 kW |
| SSAC with gas-fired desiccant wheel | 7.5 | Wheel depth: 8.25", Total face area: 4.0 sq. ft., Regen. temp: 284 F |
| SSAC with Kathabar system | 6.0 | Kathabar size (Nominal Tons): 4, Burner input 350 kBtu/hr |

## APPENDIX E

## ELECTRIC AND GAS AC SYSTEM COMPONENT COSTS

| Description | Specification | Cost | Comments |
|---|---|---|---|
| Single Speed AC | EER: 9.25; SHR:0.73 | 500 $/Ton<br>650 $/Ton | Residential AC<br>Commercial AC |
| High-Efficiency SSAC | EER: 12.0; SHR: 0.84 | 550 $/Ton<br>700 $/Ton | Residential AC<br>Commercial AC |
| Heat Pipes | 3 rows, 14 fpi and face area: 2 $ft^2$ | 300 $ | |
| Desiccant wheel for DEAC (Including Housing) | Silica Gel wheel<br>Depth: 5" and<br>face area: 2 $ft^2$ | 250 $ | |
| Variable Speed AC | SEER: 13.30 | 1500 $/Ton | |
| Gas-fired desiccant wheel system | Silica gel wheel<br>Heat exchanger<br>Gas burner<br>Blowers, dampers and Cabinet | 1600 $ | per sq.ft of wheel face area |
| Kathabar dehumidifier | Heat Exchangers<br>Fan, Cabinet | 3500 $ | per latent ton |

## APPENDIX F

## GAS AND ELECTRIC RATES

| City | Electricity $/kWhr | Natural Gas $/Therm | Gas/Electric Ratio $/Therm ---------- $/kWh |
|---|---|---|---|
| Miami | 0.0675 | 0.5610 | 8.31 |
| Washington | 0.0968 < 6000 kWhr<br>0.0577 > 6000 kWhr | 0.7102 | 7.33<br>12.31 |
| St.Louis | 0.104 | 0.4887 | 4.70 |

# APPENDIX G

## RESIDENCE, OFFICE AND FAST FOOD RESTAURANT BUILDING SCHEDULES

TABLE G-1 Residence Schedules

TABLE G-2 Medium Size Office Building Schedules

TABLE G-3 Fast Food Restaurant Schedules

## TABLE G-1

### Residence Schedules

Schedules (Fraction of maximum):

| | Hour | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| Equipment and Lighting Loads: | | | | | | | | | | | | | | | | | | | | | | | | |
| Weekday | .6 | .6 | .6 | .6 | .6 | .6 | .6 | .9 | .9 | .6 | .6 | .6 | .6 | .6 | .6 | .9 | .9 | .9 | .9 | .6 | .6 | .6 | .6 | .6 |
| Saturday | .6 | .6 | .6 | .6 | .6 | .6 | .6 | .9 | .9 | .6 | .6 | .6 | .6 | .6 | .6 | .9 | .9 | .9 | .9 | .6 | .6 | .6 | .6 | .6 |
| Sunday | .6 | .6 | .6 | .6 | .6 | .6 | .6 | .9 | .9 | .6 | .6 | .6 | .6 | .6 | .6 | .9 | .9 | .9 | .9 | .6 | .6 | .6 | .6 | .6 |
| Infiltration: | | | | | | | | | | | | | | | | | | | | | | | | |
| Weekday | .3 | .3 | .3 | .3 | .33 | .43 | .43 | .43 | .56 | .29 | .56 | .32 | .24 | .24 | .48 | .48 | .7 | .7 | .77 | .77 | 1 | .43 | .29 | .3 |
| Saturday | .3 | .3 | .3 | .3 | .33 | .43 | .43 | .43 | .56 | .29 | .56 | .32 | .24 | .24 | .48 | .48 | .7 | .7 | .77 | .77 | 1 | .43 | .29 | .3 |
| Sunday | .3 | .3 | .3 | .3 | .33 | .43 | .43 | .43 | .56 | .29 | .56 | .32 | .24 | .24 | .48 | .48 | .7 | .7 | .77 | .77 | 1 | .43 | .29 | .3 |

## TABLE G-2

### Medium Size Office Building Schedules

Schedules (Fraction of maximum):

| | Hour | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| Occupancy and Ventilation: | | | | | | | | | | | | | | | | | | | | | | | | |
| Weekday | 0 | 0 | 0 | 0 | 0 | 0 | 0 | .1 | .2 | .95 | .95 | .95 | .95 | .45 | .95 | .95 | .95 | .95 | .3 | .1 | .1 | .1 | 0 | 0 |
| Saturday | 0 | 0 | 0 | 0 | 0 | 0 | 0 | .1 | .1 | .3 | .3 | .3 | .3 | .1 | .1 | .1 | .1 | .1 | 0 | 0 | 0 | 0 | 0 | 0 |
| Sunday | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Lighting: | | | | | | | | | | | | | | | | | | | | | | | | |
| Weekday | 0 | 0 | 0 | 0 | 0 | 0 | .1 | .3 | .3 | .9 | .9 | .9 | .9 | .8 | .9 | .9 | .9 | .9 | .3 | .3 | .2 | .2 | 0 | 0 |
| Saturday | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | .1 | .1 | .3 | .3 | .3 | .3 | .15 | .15 | .15 | .15 | 0 | 0 | 0 | 0 | 0 | 0 |
| Sunday | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Equipment: | | | | | | | | | | | | | | | | | | | | | | | | |
| Weekday | 0 | 0 | 0 | 0 | 0 | 0 | 0 | .10 | .20 | .95 | .95 | .95 | .95 | .45 | .95 | .95 | .95 | .95 | .30 | .10 | .10 | .10 | .0 | .0 |
| Saturday | 0 | 0 | 0 | 0 | 0 | 0 | 0 | .1 | .1 | .3 | .3 | .3 | .3 | .15 | .15 | .15 | .15 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Sunday | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Cooling Temperature Set Points (°F): | | | | | | | | | | | | | | | | | | | | | | | | |
| Weekday | 85 | 85 | 85 | 85 | 85 | 81 | 78 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 76 | 85 | 85 |
| Saturday | 85 | 85 | 85 | 85 | 85 | 81 | 78 | 76 | 76 | 76 | 76 | 76 | 76 | 78 | 81 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 |
| Sunday | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 | 85 |
| Humidity Set Points (°F): | | | | | | | | | | | | | | | | | | | | | | | | |
| Weekday | 60 | 60 | 60 | 60 | 60 | 60 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 60 | 60 |
| Saturday | 60 | 60 | 60 | 60 | 60 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 50 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 |
| Sunday | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 | 60 |

## TABLE G-3

### Fast Food Restaurant Schedules

Schedules (Fraction of maximum):

| | Hour | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| Occupancy | | | | | | | | | | | | | | | | | | | | | | | | |
| Weekday | 0 | 0 | 0 | 0 | 0 | .5 | .5 | .5 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 0 | 0 |
| Saturday | 0 | 0 | 0 | 0 | 0 | .5 | .5 | .5 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 0 | 0 |
| Sunday | 0 | 0 | 0 | 0 | 0 | .5 | .5 | .5 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 0 | 0 |
| Ventilation | | | | | | | | | | | | | | | | | | | | | | | | |
| Weekday | 0 | 0 | 0 | 0 | 0 | .5 | .5 | .5 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | .0 |
| Saturday | 0 | 0 | 0 | 0 | 0 | .5 | .5 | .5 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | .0 |
| Sunday | 0 | 0 | 0 | 0 | 0 | .5 | .5 | .5 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | .0 |
| Lighting: | | | | | | | | | | | | | | | | | | | | | | | | |
| Weekday | .16 | .16 | .16 | .16 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .69 | 1 | 1 | 1 | 1 | 1 | .16 | .16 |
| Saturday | .16 | .16 | .16 | .16 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .69 | 1 | 1 | 1 | 1 | 1 | .16 | .16 |
| Sunday | .16 | .16 | .16 | .16 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .38 | .69 | 1 | 1 | 1 | 1 | 1 | .16 | .16 |
| Equipment: | | | | | | | | | | | | | | | | | | | | | | | | |
| Weekday | 0 | 0 | 0 | 0 | 0 | .5 | .5 | .5 | .5 | 1 | 1 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .02 | .02 |
| Saturday | 0 | 0 | 0 | 0 | 0 | .5 | .5 | .5 | .5 | 1 | 1 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .02 | .02 |
| Sunday | 0 | 0 | 0 | 0 | 0 | .5 | .5 | .5 | .5 | 1 | 1 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .9 | .02 | .02 |
| Cooling Temperature Set Points - Dining (°F): | | | | | | | | | | | | | | | | | | | | | | | | |
| Weekday | 120 | 120 | 120 | 120 | 120 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 120 |
| Saturday | 120 | 120 | 120 | 120 | 120 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 120 |
| Sunday | 120 | 120 | 120 | 120 | 120 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 75 | 120 |
| Cooling Set Points - Kitchen (°F): | | | | | | | | | | | | | | | | | | | | | | | | |
| Weekday | 120 | 120 | 120 | 120 | 120 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 120 |
| Saturday | 120 | 120 | 120 | 120 | 120 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 120 |
| Sunday | 120 | 120 | 120 | 120 | 120 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 80 | 120 |