Desiccant Cooling Technology

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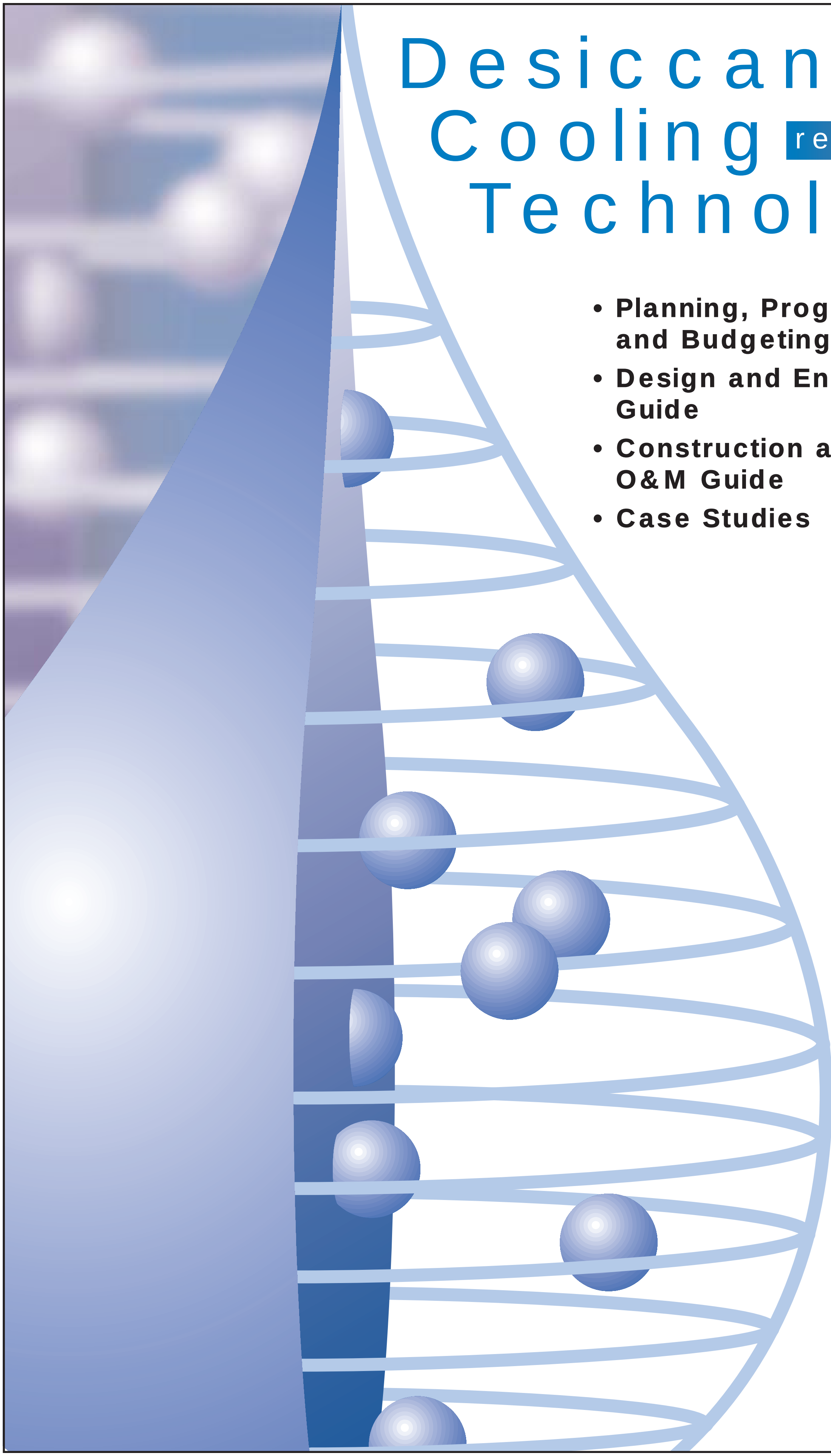
Desiccant Cooling Technology

resource guide

- Planning, Programming and Budgeting Guide
- Design and Engineering Guide
- Construction and O&M Guide
- Case Studies



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Desiccant Cooling Technology Resource Guide

Main Outline

Go to Navigation Instructions for this Resource Guide...
For Detailed Section Outlines, Click on Section Above...

EXECUTIVE SUMMARY

Purpose of this Resource Guide

This Resource Guide is an easy to use reference source for investigating, evaluating and installing desiccant cooling technologies. This document provides an overview of desiccant cooling technology and provides readers with reference materials for more in-depth analyses.

DoD and Desiccant Cooling

The U.S. Department of Defense (DoD) uses desiccant cooling technology to solve a variety of building comfort, quality and energy related issues. The evaluation and application of desiccants is not widely understood throughout DoD. Field tests of desiccants have been conducted at:

- Army Barracks
- Museum
- Hospital Operating Room
- Avionics Repair Laboratory
- Fast Food Restaurant
- Bowling Alley

Benefits of Controlling Humidity

Successful application of this technology is measured by:

- Occupant Comfort
- Net Energy Savings
- Source Emissions Reduction
- Building Air Quality Improvement
- Moisture Damage Control

When to Use Desiccant Cooling

Desiccant technologies should be considered when:

- Moisture levels are high
  - Latent/total cooling load ratio is >= 30%
  - High levels of outdoor air make-up required in building
  - High building occupancy

- Potential costs savings are significant
  - High electrical demand charges
  - Low natural gas rates
  - Low cost central steam available
  - Heat recovery options available

- Tight control over moisture levels is required
  - Hospital operating rooms
  - Avionics repair laboratories
  - Museums
  - Munitions storage

- Moisture is problematic to interior spaces such as:
  - Ice Arenas (fogging)
  - Hospitals (bacteria)
  - Hotels/Apartments (moisture damage)
  - Food Stores (freezer case moisture)

- Occupant comfort cannot be compromised

Navigating this Resource Guide

This Resource Guide is divided into four main sections plus this Executive Summary/Main Outline. The sections include:

I. Desiccant Planning, Programming and Budgeting Guide
II. Desiccant Design and Engineering Guide
III. Desiccant Construction and O&M Guide
IV. DoD Desiccant Systems Case Studies

In addition, there are several separate reference documents which are linked from within individual sections.

At the front of each section is a Section Outline that has a detailed table of contents and includes page numbers. All the section outline headings have been "linked" to their corresponding pages. To go to the desired section, simply click on the heading in the Section Outline (or use the bookmarks in the left column).

A few navigating tips...

Settings - Turn off the Open Cross-Document Links in Same Window setting (no check) in the Adobe Acrobat™ File/Preferences/General menu list. This will keep the Resource Guide document open when linking to referenced documents.

Back Tracking Your Steps - Using the Adobe Acrobat™ double arrow button, you can retrace the pages that you have previously viewed, even if you have navigated to other linked documents. Pressing "Ctrl" and "-" keys together will also step you back.

Going Back to Section Outlines - clicking on the side bar on the left will take you to the Section Outline page.

Going to Main Outline - Clicking on the side bar on the left when you are at the Section Outline page will take you to the Main Outline page for this Resource Guide.

Section Footers - At the bottom of each page, the footer tells you which section you are viewing within the Resource Guide.

Blue Type - Blue type generally indicates that the text is linked to a separate document or another location in the same document. Click to go to the referring entry.

Section Outline

I. Desiccant Planning, Programming and Budgeting Guide

I. Desiccant Planning, Programming and Budgeting Guide

A. Overview of Desiccant Cooling Technology

1. Introduction

Desiccant cooling technology provides a tool for controlling humidity (moisture) levels for conditioned air spaces. Desiccant systems work in conjunction with conventional air conditioning systems to dehumidify the air. Desiccant materials are those that attract moisture due to differences in vapor pressure. Most people are familiar with desiccants such as silica gel packages that are included with new electronics or textile products. Desiccants can be in the form of a solid or a liquid. People have identified types of desiccants that are appropriate as a component of commercial heating, ventilation and air conditioning (HVAC) systems. These desiccants have been selected based on their ability to hold large quantities of water, their ability to be reactivated, and cost.

In order to be effective, the desiccant must be capable of addressing the latent cooling load in a continuous process. In order to accomplish this, commercial desiccant systems consist of a process air path and a reactivation air path. The desiccant that is in the process air path has been prepared to have a lower vapor pressure than the air passing over it. Thus, the moisture in the air is transferred onto the desiccant material. As the desiccant vapor pressure increases due to the presence of the moisture that it has attracted, the desiccant material is transferred to a reactivation process. In the reactivation process, hot air is passed over the desiccant. The vapor pressure of the hot air is lower than the desiccant surface which forces the moisture to transfer from the desiccant surface into the hot air stream. The moist hot air is then exhausted from the system into the outdoor air. The desiccant material that has had the trapped moisture removed is now prepared to attract moisture as it is transferred back into the process air path. The dry process air leaving the desiccant is then passed over a conventional cooling coil which addresses the sensible cooling work required to meet the air specification of the conditioned space.

For more detailed information, please refer to the following references:

User Guide for Desiccant Dehumidification Technology, Facilities Engineering Applications Program (FEAP)

Two-Wheel Desiccant Dehumidification System, Federal Technology Alert

Applications Engineering Manual for Desiccant Systems, Chapter 1 - Introduction, American Gas Cooling Center, May 1996

American Gas Cooling Center's Web Site, www.agcc.org/documents/gascooling/basicsofdesiccants.

The Dehumidification Handbook, Second Edition, Chapter 3 - Methods of Dehumidification, Lewis Harriman, 1990

2. Benefits of Dehumidification with Desiccants

Increased Comfort	Independent control of humidity and temperature • Desiccant Unit Controls Humidity • Conventional Cooling System Controls Temperature
Lower Operating Costs	• Utilize Lower Cost Natural Gas for Re-generation • Conventional Cooling System Operates at a Higher Efficiency due to Higher Suction Temperatures
Lower Peak Electric Demand	Switch latent cooling to alternate energy sources • Natural Gas • Steam • Heat Recovery
Heat Recovery Options	Heat recovery sources • Engine Driven Chillers • Cogenerators • Condenser Heat • Steam Condensate
Dry Duct Systems	High humidity air and dust in ducting result in • Fungus Growth • Bacteria Growth Reduced indoor air quality
ASHRAE 62-89	The Standard addresses increased levels of outdoor air • Increase Total Cooling Load • Increase Latent Load Desiccant systems can directly address this prob-lem
CFC Free	Desiccant systems do not use CFC's for moisture removal
Improved Indoor Air Quality	• Appropriate Levels of Fresh Air • Reduced Levels of Air Borne Bacteria • Air Treatment Chemicals with Liquid Desiccants
Reduced Building Maintenance	Reduced building maintenance activities associ-ated with high humidity levels • Mold and mildew remediation • Corrosion • Replacement of wall coverings • Replacement of window coverings • Replacement of carpeting

2. Benefits of Dehumidification with Desiccants (cont.)

For more detailed information, please refer to the following references:

Applications Engineering Manual for Desiccant Systems, Chapter 8 - Evaluating Applications, American Gas Cooling Center, May 1996

American Gas Cooling Center's Web Site, www.agcc.org/documents/gascooling/basicsofdesiccants

User Guide for Desiccant Dehumidification Technology, Facilities Engineering Applications Program (FEAP)

Two-Wheel Desiccant Dehumidification System, Federal Technology Alert

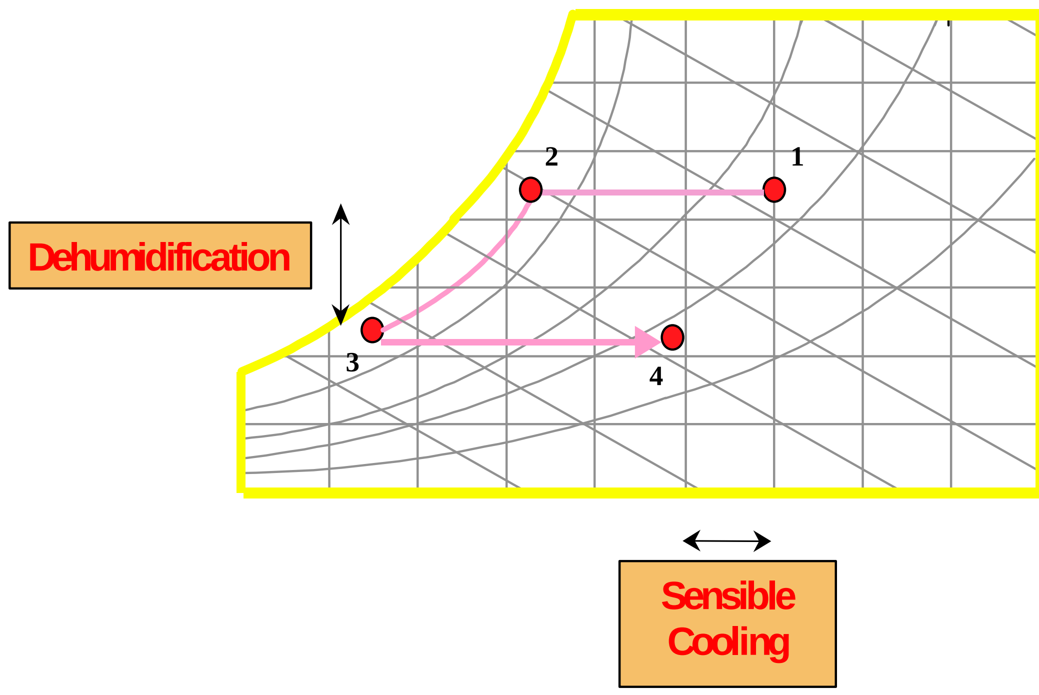
3. Alternatives to Desiccant Dehumidification

A conventional cooling system lowers the temperature of the air stream as the air passes over a cooling coil. Energy is removed from the air in the form of sensible cooling and latent cooling. Sensible cooling is simply the reduction of the dry bulb temperature of the air. Latent cooling is the removal of moisture from the air or dehumidification. Latent cooling takes place when the air is cooled below the air dew point. Cooling below the dew point causes the moisture in the air to condense and leave the air stream. The air that leaves the cooling coil under these conditions is near saturation. The air is then mixed or reheated to the desired supply air temperature. This process is illustrated on the psychrometric chart below.

State 1 to State 2: Air is cooled to the point of saturation

State 2 to State 3: Further cooling causes moisture to be condensed from the air as the temperature of the air continues to drop.

State 3 to State 4: The air is then mixed or passes through reheat to supply air at the desired temperature.

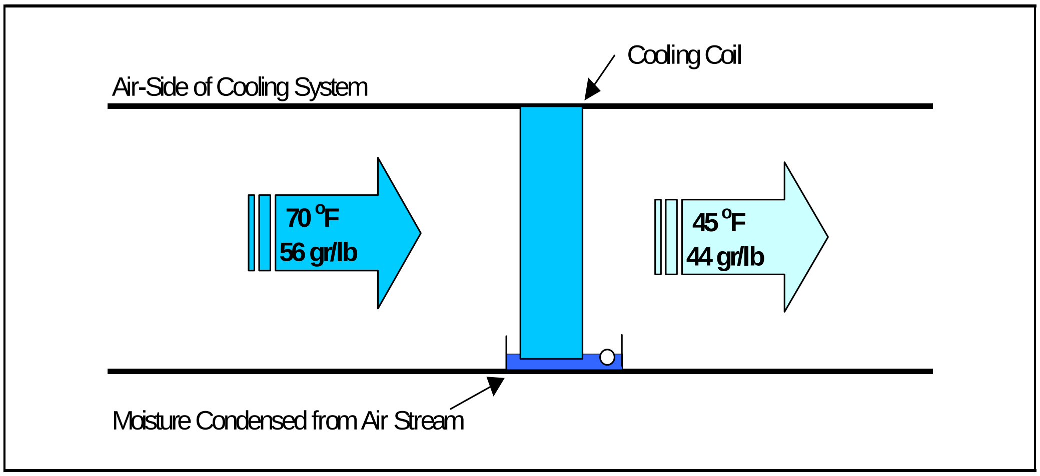


The configuration of conventional cooling systems that remove moisture from the air are described on the following pages.

3. Alternatives to Desiccant Dehumidification (cont.)

Conventional Cooling Dehumidification

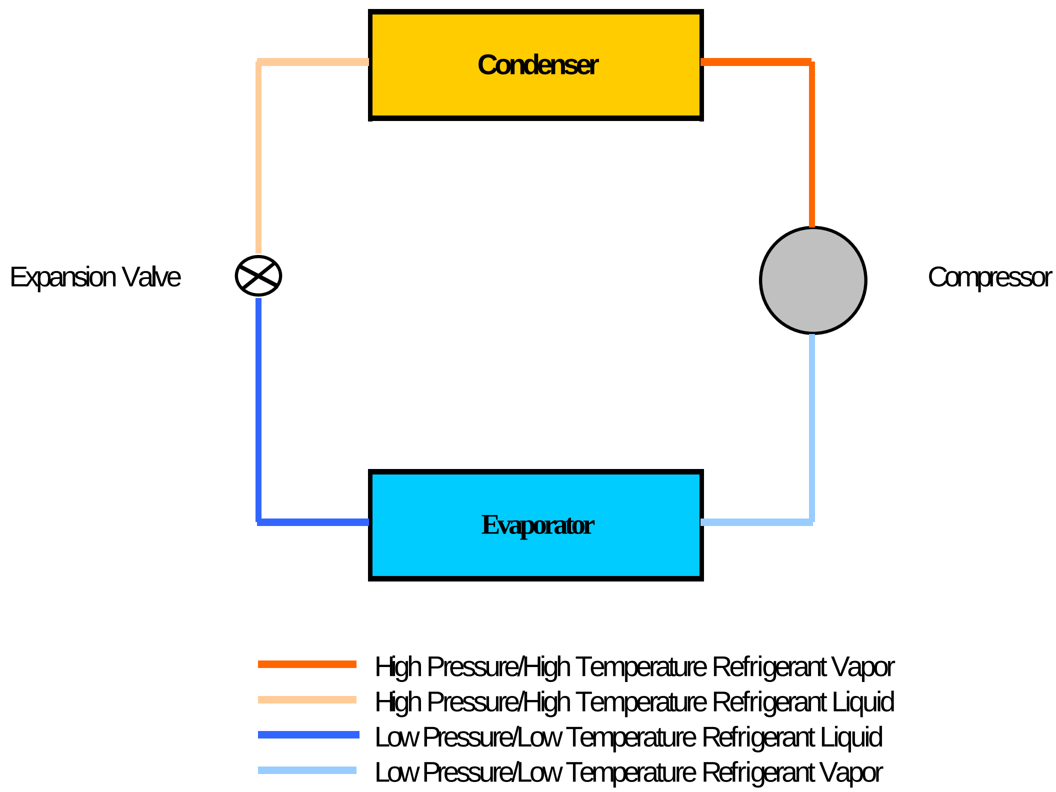
- Cooling Coil in supply air path
- Sensible and Latent Cooling
- Latent Cooling by condensation
- Air leaving cooling coil typically at or near saturation
- Increased moisture removal by lowering cooling coil temperatures



3. Alternatives to Desiccant Dehumidification (cont.)

Vapor-Compression Direct Expansion

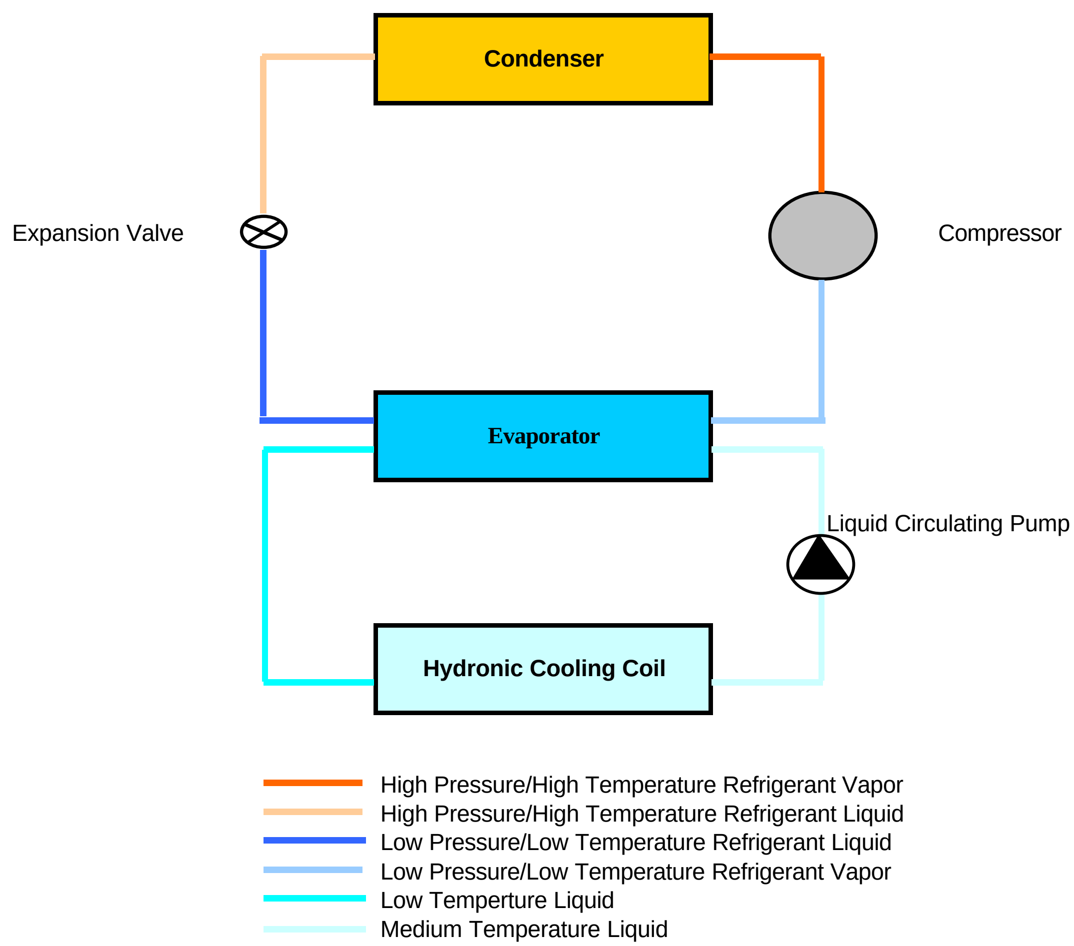
- Uses Refrigerant phase change characteristics for heat transfer
- Cooling Coil is a refrigerant to air heat exchanger
- Compressor: Converts low pressure refrigerant into high pressure refrigerant
- Condenser: Uses a cool source to change refrigerant vapor into a liquid
- Expansion: Reduces pressure and temperature of the refrigerant
- Evaporator: Absorbs heat as refrigerant changes from a liquid to a vapor



3. Alternatives to Desiccant Dehumidification (cont.)

Chilled Liquid

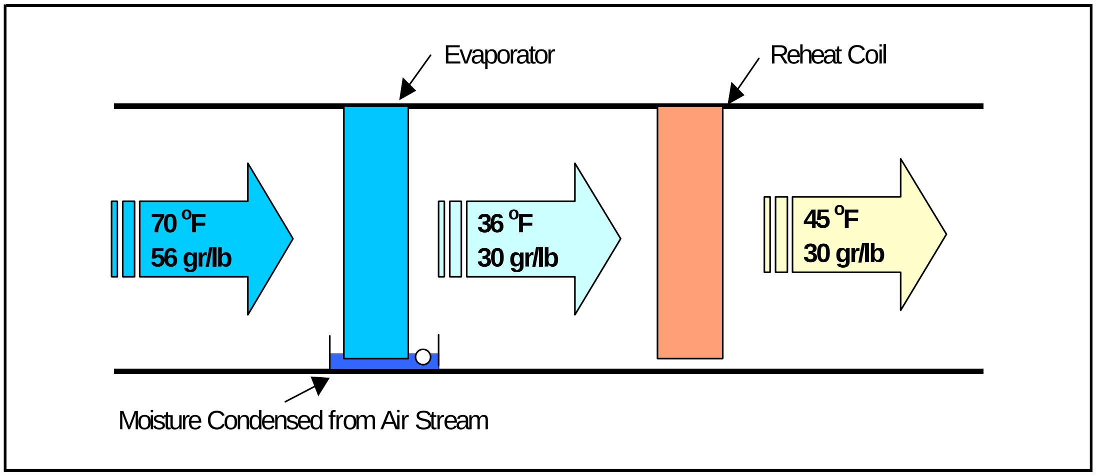
- Liquid/Air heat exchanger in supply air path
- Vapor - Compression Cycle with Refrigerant/Liquid Evaporator as intermediate heat transfer stage
- Sensible and Latent Cooling
- Latent Cooling by condensation
- Air leaving cooling coil typically at or near saturation
- Increased moisture removal by lowering temperatures



3. Alternatives to Desiccant Dehumidification (cont.)

Cool - Reheat

- Integrated with above systems with heat added down stream of the cooling coil
- Heating the saturated air reduces the relative humidity of the supply air
- Controls both supply temperature and relative humidity



For more detailed information, please refer to the following references:

American Gas Cooling Center's Web Site, www.agcc.org/documents/gascooling/basicsofdesiccants

The Dehumidification Handbook, Second Edition, Chapter 3 - Methods of Dehumidification, Lewis Harriman, 1990

1993 ASHRAE Handbook - Fundamentals, Chapter 1 - Thermodynamics and Refrigeration Cycles

4. Types of Desiccant Systems

The design and operation of a desiccant system is based on the desiccant material used to accomplish the dehumidification. Desiccant materials attract moisture through the process of either adsorption or absorption. Adsorption is the process of trapping moisture within the desiccant material similar to the way a sponge holds water through capillaries. Most adsorbents are solid materials. Absorption is the process of trapping moisture through a chemical process in which the desiccant undergoes a chemical change. Most absorbents are liquids.

Types of materials used as a basis for desiccant systems include the following materials:

- Silica Gel
- Lithium Chloride (Liquid or Dry)
- Lithium Bromide
- Activated Alumina
- Titanium Silicate
- Molecular Sieve

Commercially available desiccant systems are based on five configurations or technologies.

- Liquid Spray Towers
- Solid Packed Tower
- Rotating Horizontal Bed
- Multiple Vertical Bed
- Rotating Desiccant Wheel

For more detailed information, please refer to the following references:

American Gas Cooling Center's Web Site, www.agcc.org/documents/gascooling/basicsofdesiccants

The Dehumidification Handbook, Second Edition, Chapter 3 - Methods of Dehumidification, Lewis Harriman, 1990

Energy User News, August 1998

Two-Wheel Desiccant Dehumidification System, Federal Technology Alert

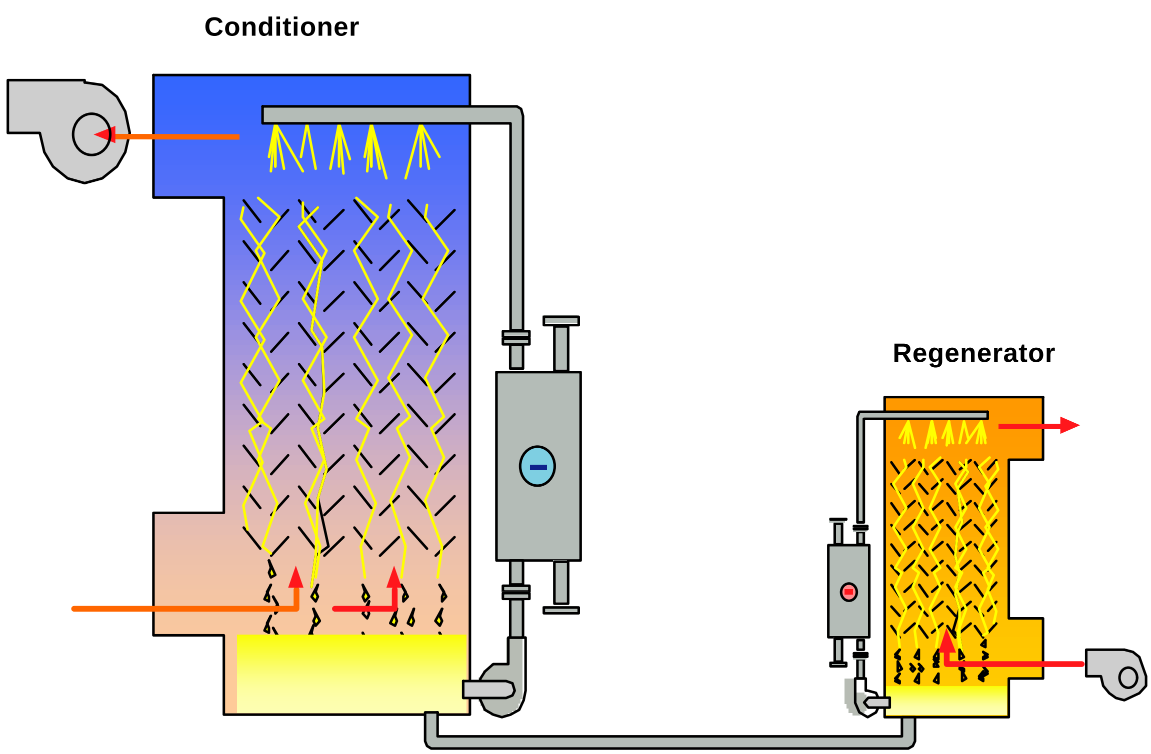
1997 ASHRAE Handbook - Fundamentals, Chapter 21: Sorbents and Desiccants

1992 ASHRAE Handbook - HVAC Systems and Equipment, Chapter 22: Desiccant Dehumidification and Pressure Drying Equipment

4. Types of Desiccant Systems (cont.)

Liquid Spray Towers

- Based on liquid desiccant.
- Process Air: Air passes through a desiccant spray in a conditioner module.
- Regeneration Process: Outside air passes through a warm desiccant spray in a regenerator module.
- Advantages:
    - a. Large air flow capacity.
    - b. Modular design.
    - c. Provides microbiological decontamination.
    - d. Reduced regeneration air requirement.
    - e. Energy storage capability (holding tanks can be used to provide extended capacity).
    - f. Desiccant quality easily monitored and adjusted.
    - g. No possibility of cross leakage of air streams.
    - h. Air temperature and humidity are controlled simultaneously.
- Disadvantages:
    - a. May have a difficult time maintaining humidity levels below 10% RH with loads that have a small sensible component.

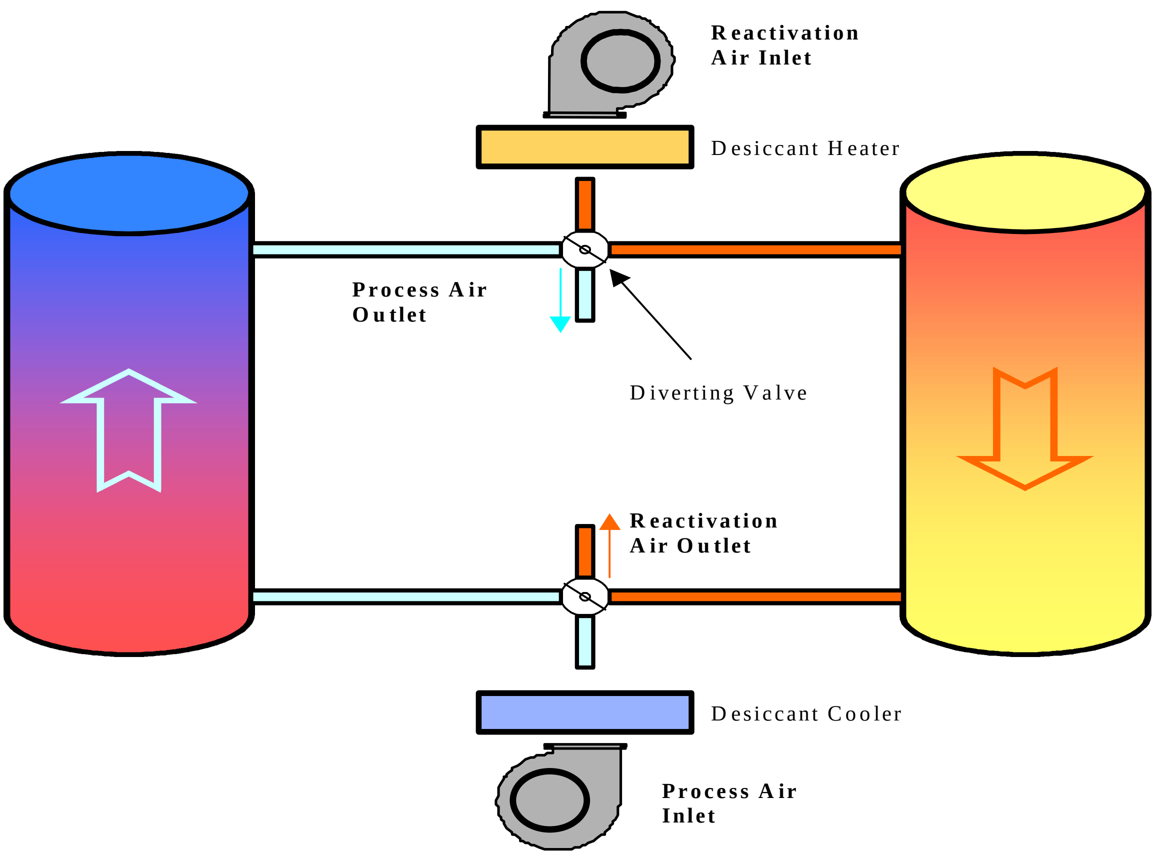


Source: Dehumidification Handbook; Munters/Cargocaire

4. Types of Desiccant Systems (cont.)

Solid Packed Tower

- Based on solid desiccant
- Process Air: Air passes through a tower filled with solid desiccant
- Regeneration Process: Air passes through a tower filled with solid desiccant
- Towers alternate between performing in the process air path and the regeneration air path.
- Advantages:
    a. Able to achieve very low dew points
- Disadvantages:
    a: Output conditions vary with level of moisture trapped
    b: Air velocity is critical for optimal performance

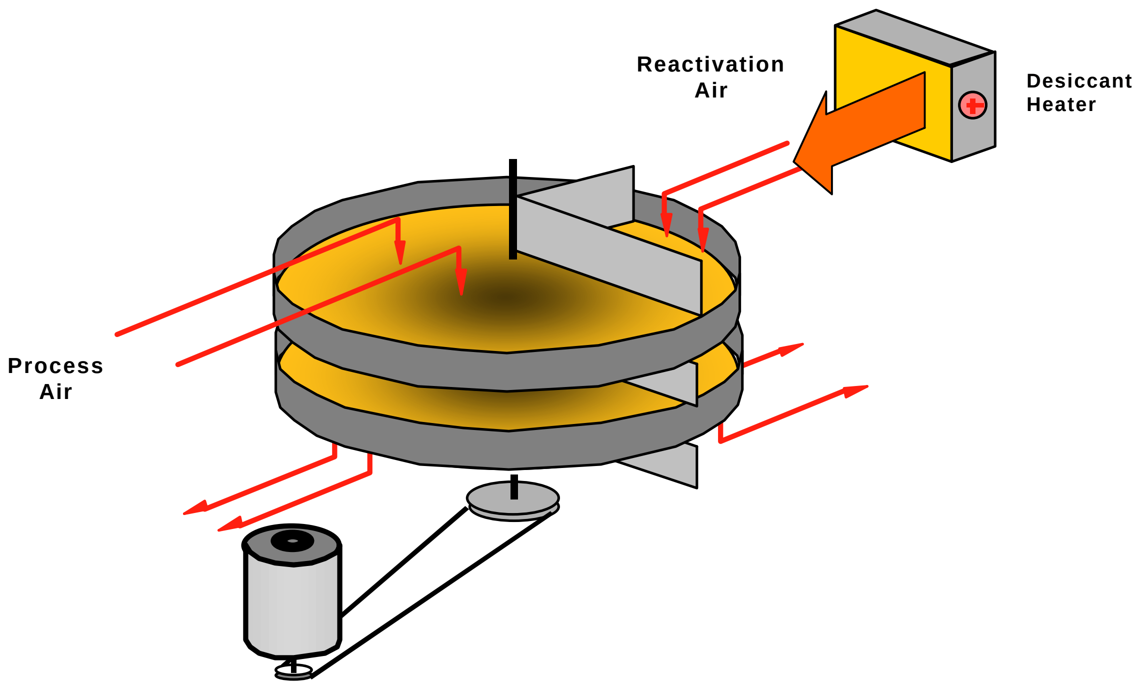


Source: Dehumidification Handbook; Munters/Cargocaire

4. Types of Desiccant Systems (cont.)

Rotating Horizontal Bed

- Based on solid desiccant
- Process Air: Desiccant is held by perforated trays through which the air passes
- Regeneration Process: Desiccant is held by perforated trays that the air passes through
- Trays rotate through the process air path and the regeneration air path.
- Advantages:
    a. Modular Design
    b. Constant outlet moisture level
    c. High air flow capacity capability
    d. Lower first costs
    e. Simple design
- Disadvantages:
    a. Desiccant settling in trays
    b. Air leakage between process air and regeneration within the tray



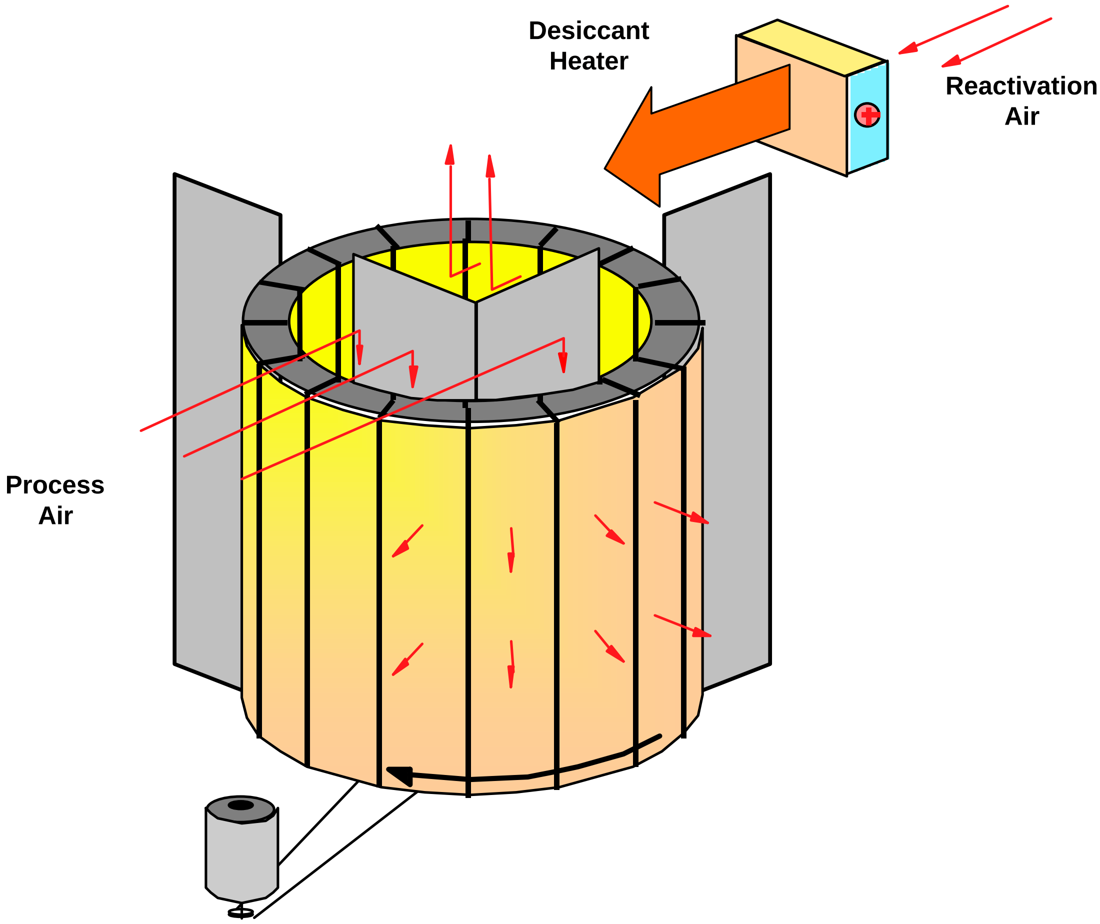
Source: Dehumidification Handbook; Munters/Cargocaire

4. Types of Desiccant Systems (cont.)

Multiple Vertical Bed

- Based on solid desiccant
- Process Air: Desiccant is held by stacked perforated trays that the air passes through
- Regeneration Process: Desiccant is held by perforated trays that the air passes through
- Combination of packed tower and rotating bed designs through the use of a rotating carrousel of many towers.
- Advantages:
  a. Constant outlet moisture level
  b. High performance
  c. Low dew points

- Disadvantages:
  a. Complex mechanical system
  b. Increased maintenance
  c. Higher first cost



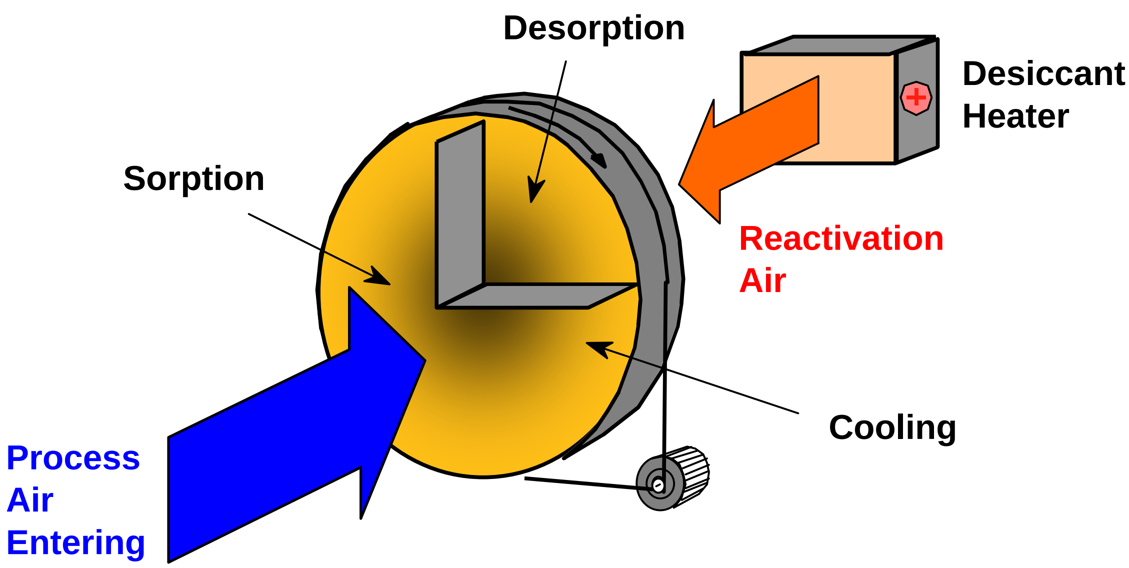
Source: Dehumidification Handbook; Munters/Cargocaire

4. Types of Desiccant Systems (cont.)

Rotating Desiccant Wheel

- Based on solid desiccant
- Process Air: Air passes through hexagonal or sinusoidal shaped passages of the wheel.
- Regeneration Process: Air passes through hexagonal or sinusoidal shaped passages of the wheel.
- Desiccant is impregnated into a semi-ceramic structure that resembles a honeycomb.
- Advantages:
    a. Light weight and porous structure
    b. Low pressure drop across wheel
    c. Low dew points
    d. High capacity
    e. Simple system

- Disadvantages:
    a. Higher first cost



Source: Dehumidification Handbook; Munters/Cargocaire

5. Application Issues

Typical applications and benefits for desiccant dehumidification are as follows:

Application	Benefits of Desiccant Dehumidification
Supermarket	• Energy savings through reduced refrigeration display compressor loads • Fewer defrost cycles in refrigerated display systems • Eliminates condensation on display cases • Customer comfort in frozen food aisles
Ice Rinks	• Energy savings through reduced latent loads • Less ice resurfacing • Eliminates fogging • Reduced building maintenance
Refrigerated Warehouse	• Energy savings through reduced latent loads • Eliminates temperature fluctuations • Reduces workplace hazards (slick and icy floors)
Hospital Operating Room	• Eliminates perspiration of surgeons • Eliminates fungal amplification in ductwork • Eliminates condensation in operating room
Movie Theater	• Increased customer comfort • Allows increased ventilation in response to ASHRAE Standard 62 • Increases useful life of seats and carpets that are damaged by the presence of high moisture levels
School	• Reduced health risks associated with airborne infectious agents • Decreased levels of indoor CO_2 • Lower energy costs
Fast Food Restaurant	• Allows increased ventilation in response to ASHRAE Standard 62 • Increased customer comfort • Lower energy costs
Hotel	• Increased customer comfort • Allows increased ventilation in response to ASHRAE Standard 62 • Increases useful life of wallpaper, tapestries and carpets that are damaged by the presence of high moisture levels

5. Application Issues (cont.)

All of the above applications address specific moisture control issues that provide increased usefulness of the facility. It is important to note that the benefits of dehumidification will vary significantly by climate.

Application Characteristics that Favor Desiccant Dehumidification

Characteristic	**Cause**	**Application (Examples)**
Ratio of Latent Load to Total Cooling Load > 30%	▪ High Occupancy ▪ High Level of Outdoor Air ▪ High Internal Latent Loads from Processes	▪ Movie Theaters ▪ Schools ▪ Stores ▪ Restaurants ▪ Meeting Halls ▪ Ice Skating Rink
Dry Air Requirements	▪ Air Space Specifications for Processes	▪ Laboratories ▪ Computer Rooms ▪ Libraries ▪ Museums ▪ Munitions Storage ▪ Avionics Repair
High Outside Air Requirements	▪ ASHRAE Standard 62 (15 cfm per person)	▪ Movie Theaters ▪ Schools ▪ Stores ▪ Restaurants ▪ Meeting Halls ▪ Hospitals ▪ Offices
High Electric Rates	▪ Increased Utility Demand during Hot Summer Days	▪ Not Application Specific
Indoor Air Quality Problems	▪ Outdoor Air Requirements ▪ High Levels of Airborne Infectious Agents ▪ High Levels of Indoor CO_2	▪ Schools ▪ Dormitories ▪ Hospitals ▪ Meeting Halls ▪ Offices

5. Application Issues (cont.)

Major site-specific application issues to consider when evaluating the potential application of desiccant dehumidification are as follows:

1. Indoor air temperature and relative humidity requirements.
2. The quantity of outside air to be introduced into the building.
3. The general characteristics of the outdoor air during the year.
4. Internal moisture loads.
5. Ratio of sensible cooling load to latent cooling load.
6. Presence of energy sources (electricity, natural gas, steam, etc.).
7. Rate structure of energy sources.
8. Configuration of existing mechanical systems.
9. Availability of space for locating desiccant system.

For more detailed information, please refer to the following references:

Applications Engineering Manual for Desiccant Systems, Chapter 10 - Case Histories, American Gas Cooling Center, May 1996

American Gas Cooling Center's Web Site, www.agcc.org/documents/gascooling/basicsofdesiccants

The Dehumidification Handbook, Second Edition, Chapter 4 - Applications, Lewis Harriman, 1990

Two-Wheel Desiccant Dehumidification System, Federal Technology Alert

6. Performance Issues

There are two major processes which take place within the desiccant system: 1) The Process Air Path, and 2) The Reactivation Air Path. The boundary conditions for the inlet and outlet air for these processes as well as the characteristics of the desiccant being utilized impact the performance of the desiccant system.

Process Air

Inlet Moisture
- If greater than expected
  - Leaving process air temperature will be greater than expected
- If less than expected
  - Leaving process air will be drier than expected

Inlet Temperature
- If greater than expected
  - Reduced moisture removal performance
- If less than expected
  - Increased moisture removal performance

Air Velocity
- High velocity
  - Reduced moisture removal performance
- Low velocity
  - Increased moisture removal performance

Reactivation Air

Inlet Moisture
- Rotary Bed, Vertical Bed & Rotating Wheel
  - Air leakage of moist air into the process air path can be an issue (consult manufacturer)
- Solid Adsorbents
  - High moisture on reactivation can affect ability to achieve extremely dry process air

Temperature
- High Reactivation Temperature
  - Air leaving process air stream becomes drier
- Lower Reactivation Temperature
  - Requires more desiccant in air paths

Air Velocity
- Needs to be sized to match the work being done on the process air path

6. Performance Issues (cont.)

Desiccant Material/Configuration

<u>Sorption/Desorption Characteristics</u>

- Desiccant Capacity
- Thermal Cycling Durability
- Resilience to Contamination

<u>Quantity of Desiccant Material Exposed to Each Process Path</u>

- Surface Area (air to desiccant contact area)
- Mass of Desiccant

    - This issue is more of a system design issue for the manufacturer and not an application issue for the designer.
    - The resulting optimization of the desiccant system by the manufacturer will be evident in the operating specifications of the equipment in terms of:

        - Design rate of moisture removal
        - Nominal energy consumption
        - Temperature of process leaving air
        - Air flow requirements
        - Pressure drop of the system

For more detailed information, please refer to the following references:

The Dehumidification Handbook, Second Edition, Chapter 6 - Desiccant Dehumidifier Performance, Lewis Harriman, 1990

7. Maintenance Issues

Like any other mechanical equipment, desiccant systems require routine maintenance. Desiccant systems are slightly different in that maintenance intervals may be more frequent and seasonal adjustments are required. Recommended maintenance for solid desiccant and liquid desiccant systems are presented below.

Typical Maintenance Requirements for Solid Desiccant Systems

Major maintenance components are as follows:

- Filters (Every 2 - 3 Months)
    - Regeneration Air Path
    - Process Air Path

- Fan Bearings (Every 2 - 3 Months)
    - Regeneration Fan
    - Process Air Fan

- Belts (Every 2 - 3 Months)
    - Regeneration Fan Belt
    - Process Air Fan Belt
    - Desiccant Wheel Belt
    - Heat Wheel Belt

- Evaporative Cooling Pads and Sumps (Every 6 months)
    - Flush Pads
    - Water Treatment
    - Winterize

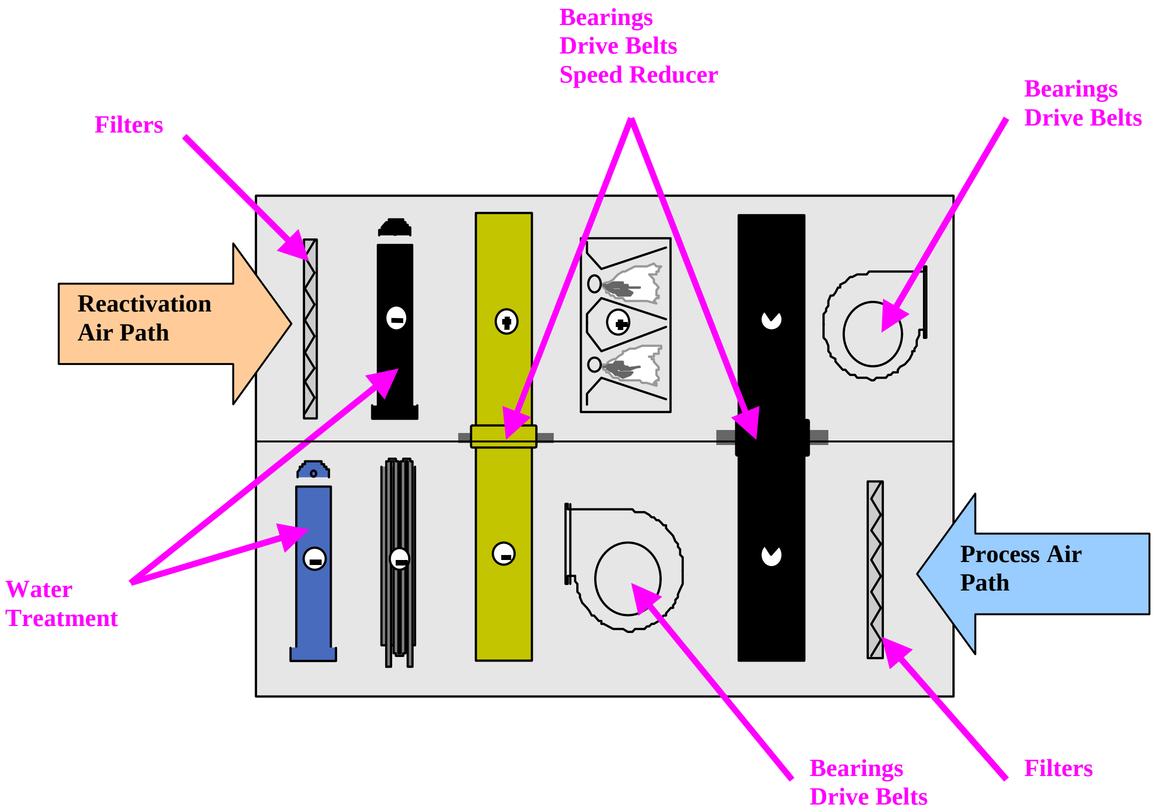
- Electrical Connections
    - Check Tightness of Connections

- Check Control Settings (Every 2 - 3 Months)

- Check Oil in Speed Reducers (Every 6 months)

7. Maintenance Issues (cont.)

Generic Maintenance Diagram for Solid Desiccant Systems



Source: Application Engineering Manual; American Gas Cooling Center

7. Maintenance Issues (cont.)

Typical Maintenance Requirements for Liquid Desiccant Systems

Major maintenance components are as follows:

- Desiccant (Every Month)
  - Supply Pressure
  - Spray Pattern
  - Desiccant Fluid Level
  - Desiccant Concentration
  - Inhibitor Level

- Desiccant Filters (Every Month)

- Air Pressure Drop (Every Month)
  - Conditioner Unit
  - Regenerator Unit

- Eliminator Pads (Every 2 Months)

- Air Filters (Every 2 Months)

- Desiccant (Every 2 Months)
  - Heat Exchanger Pressure Drop
  - Pump Discharge Pressure
  - Provide Desiccant Sample to Factory for Analysis

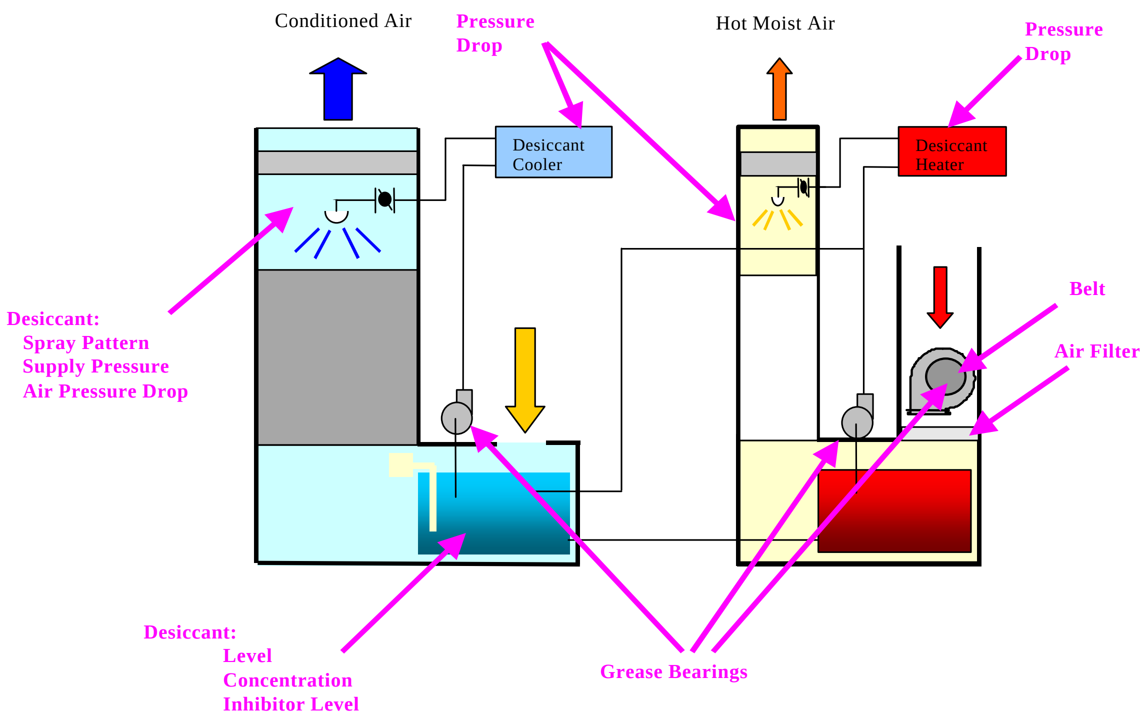
- Grease (Every 6 Months)
  - Fan Bearings
  - Pump Bearings

- Belts (Every 6 Months)

- Pumps (Every 6 Months)

7. Maintenance Issues (cont.)

Generic Maintenance Diagram for Liquid Desiccant Systems



For more detailed information, please refer to the following references:

Applications Engineering Manual for Desiccant Systems, Chapter 7 - Maintenance, American Gas Cooling Center, May 1996

American Gas Cooling Center's Web Site, www.agcc.org/documents/gascooling/basicsofdesiccants

B. Military Applications

There are a variety of potential military desiccant system applications. A brief description of what they are used for is presented below along with a summary table.

Commissaries - desiccant dehumidification can reduce the cost for refrigeration cases, eliminate defrosting, reduce humidity in the frozen food aisles as well as the whole facility.

Hospitals - used in operating rooms to meet specific code requirements. Use of desiccant dehumidification can eliminate condensing moisture in duct work to prevent post-operative fungal infections.

Housing/Barracks - used to ensure comfort levels of occupants as well as to prevent damage to furniture and fixtures due to mold or mildew.

Avionics Repair - tight humidity tolerances are required in the repair of avionics equipment such as altimeters and gauges. Improper repair conditions can lead to fogging of gauges during flight.

Ammunition Storage - moisture must be controlled in the storage of ammunition to prevent damage to explosive materials or cause premature detonation.

School - primarily used to increase student comfort and indoor air quality.

Restaurant - used to increase occupant comfort.

Meeting Hall - large latent loads can be more economically handled by desiccant dehumidification systems

Offices - used to achieve occupant comfort as well as to remedy air quality problems.

Museum - museums have special dry air requirements to prevent the deterioration of historical pieces.

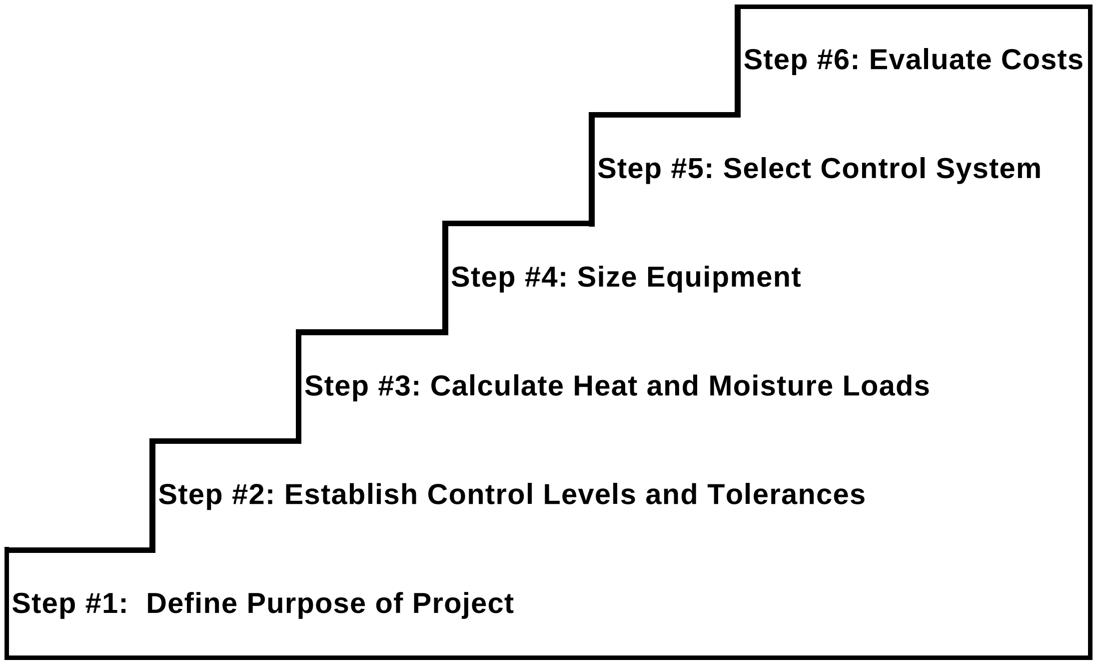
B. Military Applications (cont.)

	Commissary	Hospital	Housing/Barracks	Avionics Repair	Ammunition Storage	School	Restaurant	Meeting Hall	Office	Museum
High Latent Loads						●	●	●		
High Outdoor Air Req.		●	●			●	●	●	●	
Dry Air Requirements		●		●	●					●
High Occupancy Rates	●	●				●	●	●		
High Intern. Latent Loads	●	●				●	●	●		
Corrosion Prevention			●		●					●
Reduce Bacterial Levels		●				●			●	
Reduce Mold & Mildew		●	●			●			●	●
Increase Occup. Comfort	●	●	●			●	●	●	●	
Refrigeration	●									

C. Site Screening and Evaluation

1. Overview of Approach

When evaluating a specific application using desiccant technologies, the process can be accomplished through the following steps:



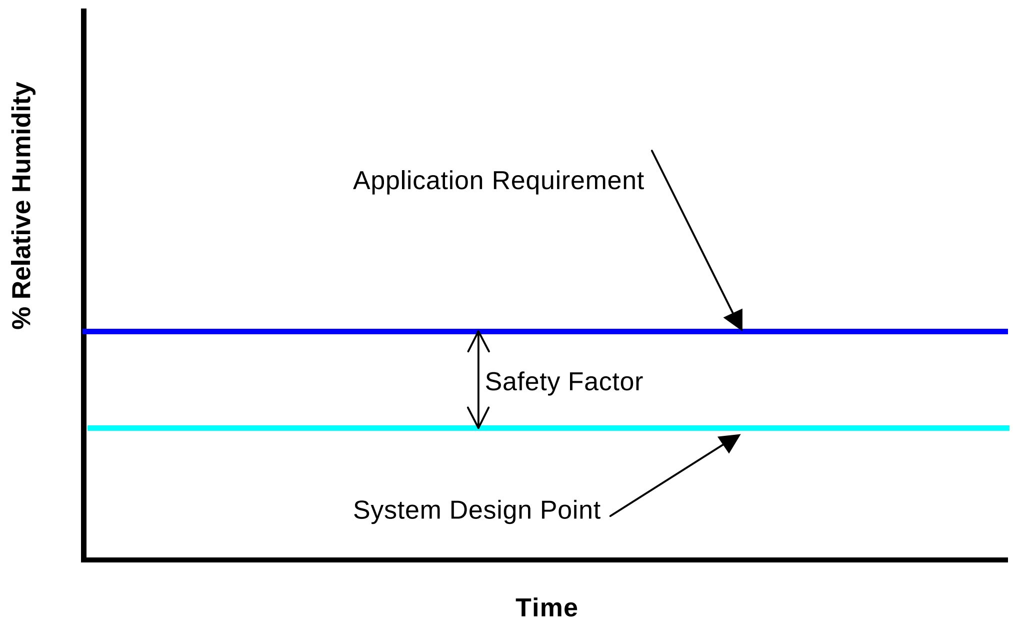
Step#1: Define Purpose of Project

Desiccants will be evaluated for the purposes of controlling humidity levels with in a conditioned space. Categories of problem definitions include the following:

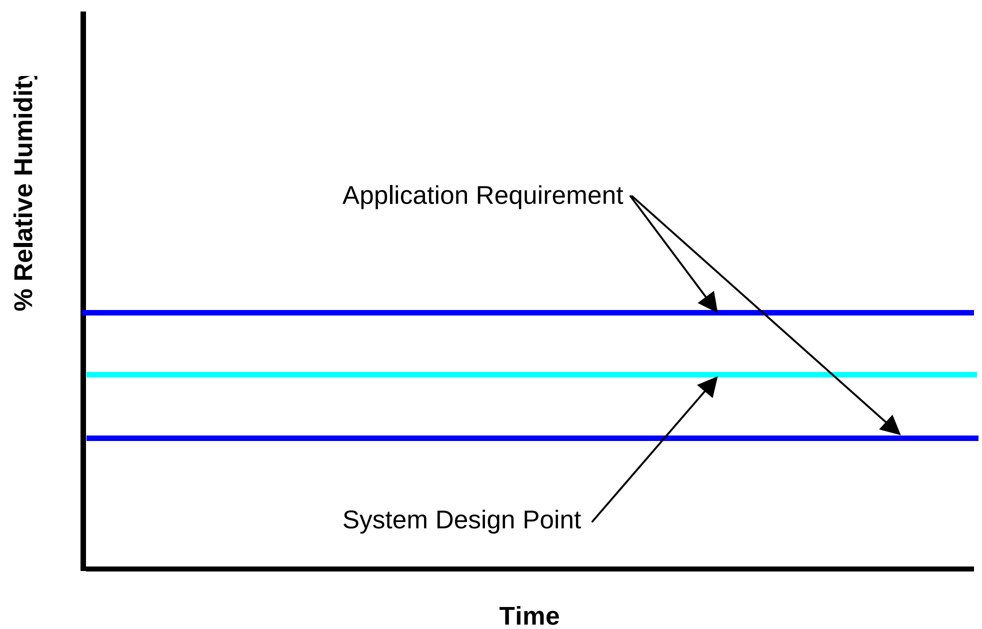
- ♦ Address Indoor Air Requirements
  - Increase Indoor Air Quality
  - Accommodate Increase in Outdoor Air Makeup
- ♦ Address Process Requirements
  - Avionics Repair
  - Hospital Operating Room
  - Munitions Storage
- ♦ Address Building Degradation due to Moisture
  - Wall Paper Replacement
  - Drapery Replacement
  - Carpet Replacement
- ♦ Prevent Moisture Level Fluctuations that Damage Value Items
  - Historical Documents
  - Historical Textiles
  - Museum Artifacts

Step#2: Establish Control Levels and Tolerances

- Desiccant systems are capable of maintaining relative humidity within a 1% RH range
- Desiccant systems allow for individual control of humidity and temperature
- Some applications require moisture control below a defined set point and the air can be at a lower moisture level without adverse impact.



- Some applications require moisture control within a specified range and large fluctuations have an adverse impact.



Step#3: Calculate Heat and Moisture Loads

a.) Identify Extreme Weather Conditions

Weather data can be found in the 1997 ASHRAE Handbook of Fundamentals. This information provides both extreme temperature and moisture levels. Note: Previous versions of the ASHRAE Handbook of Fundamentals only identify extreme temperatures which is not adequate to estimate peak moisture level and can underestimate moisture loads by 15% to 40%.

For a detailed discussion on utilizing weather information to estimate moisture loads, read: ***"Dehumidification and Cooling Loads from Ventilation Air", L. Harriman, ASHRAE Journal, Nov-1997, pp 37-45*****.**

b.) Identify Sources of Moisture Loads

Major areas to investigate for contributions of moisture in the conditioned area are as follows:

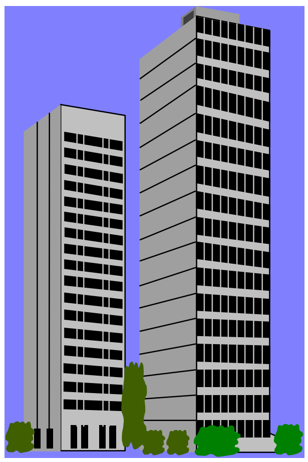
- Permeation
- Products
- People
- Wet Surfaces
- Moisture from Air leaks in Cracks and Walls
- Door Activity
- Fresh Air Makeup

Moisture from Permeation through Building Materials (W_p)

Moisture passes through building materials in the form of water vapor based on the vapor permeability of the material and the difference in vapor pressure on each side of the material. In general, the differential vapor pressure can be estimated by using the following rule of thumb:

Each grain/lb corresponds to 0.0067 inches Hg.

This can be used when the difference across the material is known in grains.



W_p = **P x A x ΔVP**

Where:

P≡ Material Permeance Factor grains/hour/Ft^2/in. Hg.) {Reference ASHRAE Handbook of Fundamentals}

A≡ Surface Area of Material (Ft^2)

ΔVP≡ Difference in Vapor Pressure across the Material (in.Hg.)

Moisture from Products and Packaging (W_{pp})

In applications where products are brought into the space for storage or processing, moisture from the products will contribute to the moisture levels within the space.

Examples:

- **Tobacco**
- **Leather**
- **Wool**
- **Lumber**
- **Cotton**
- **Wheat**

W_{pp} = m x (pw_2 - pw_1) x 7000

Where:

m≡ Rate of Mass of Material Entering the Room ($lb/hour_{material}$)

pw_1≡ Moisture Content of the Material at the Control Condition in the Space ($lb_{H2O}/lb_{material}$)

pw_2≡ Moisture Content of the Material before Entering the Space ($lb_{H2O}/lb_{material}$)

7000≡ Grains per Pound of Water

Moisture from People (W_n)

People add moisture into the air space through respiration and perspiration. In applications where the occupancy is high, the moisture from people will represent a substantial portion of the moisture load. The rate of moisture given off depends on the type of activity of the people in the space.

$W_n = F \times P$

Where:

F≡ Evaporation Rate per Person (grains/hour)

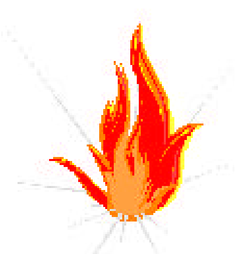
P≡ Number of People

Typical Values for various activities at 70 °F dry bulb temperature:

Person Seated at Rest:	500 grains/hour
Person Standing:	1,800 grains/hour
Person doing Light Work:	2,900 grains/hour
Person Doing Moderate Work:	4,900 grains/hour

Moisture from Combustion (W_g)

In applications where combustion is taking place within the conditioned area (i.e., gas burners for heating or cooking), the moisture from the combustion of natural gas should be included in the estimate of moisture.



$W_g = G \times 650$

Where:

G≡ Rate of Gas being Combusted (Ft^3/hour)

650≡ Moisture Produced per each Cubic Foot of Gas Burned (grain/Ft^3)

Moisture from Wet Surfaces (W_e)

In many applications, periodic washing of equipment and floors is required. The cleaning process contributes moisture to the air through evaporation.

$$W_e = \frac{H \times A \times (VP_s - Vp_a) \times 7000}{H_L}$$

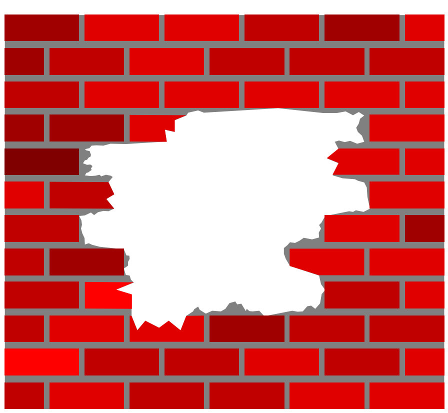
Where

$H \equiv$	Rate of Latent Heat Transfer (BTU/hour/Ft^2/in. Hg)
$A \equiv$	Total Wetted Surface Area (Ft^2)
$VP_s \equiv$	Vapor Pressure of Saturated Air at the Water Temperature (in. Hg)
$VP_a \equiv$	Water Vapor Pressure in the Air above the Surface (in. Hg)
$7000 \equiv$	Grains per Pound of Water
$H_L \equiv$	Latent Heat of Vaporization at the Water Temperature (BTU/lb)

Moisture from Air leaks in Cracks and Walls (W_i)

When outside air enters the conditioned space through cracks and the outdoor moisture levels are higher than the indoor air, the moisture levels in the conditioned space will increase. One approach to desiccant applications is to use the desiccant system to slightly pressurize the conditioned space to eliminate this moisture contribution to the load.

The ASHRAE Handbook of Fundamentals provides extensive information on air infiltration through buildings. In addition, conducting a blower door test can provide an accurate method of estimating air leakage on existing buildings.



$$W_i = A \times d \times 60 \times Va \times (M_o - M_i)$$

Where

$A \equiv$	Area of Opening (Ft^2)
$d \equiv$	Density of Infiltrating Air (lb/Ft^3)
$60 \equiv$	Minutes per hour
$Va \equiv$	Velocity of Air Through the Opening (Feet/minute)
$M_o \equiv$	Moisture Outside the Space (gr/lb)
$M_i \equiv$	Moisture Inside the Space (gr/lb)

Door Activity

One important area to evaluate is the moisture contribution associated with air entering the space as doors are opened and closed. One needs to evaluate the rate of infiltration for doors that open to weather. The analysis should assume that the local average wind velocity governs the rate of air flow through the door for the amount of time the door is open. The number of times a door opens and closes in a typical hour needs to be estimated or measured. As a rule of thumb, the minimum rate can be estimated as:

2 openings/person/hour

Note that this can sometimes be much higher.

Fresh Air Makeup (W_m)

Fresh air is introduced into the conditioned space to provide ventilation for people, to provide makeup air for exhaust fans and exhaust hoods and to maintain positive pressure in the space. Typically, makeup air is introduced into the space prior to being cooled or dehumidified.

$$W_m = Q \times d \times 60 \times (M_o - M_i)$$

Where

$Q \equiv$	Freash Air Makeup Air Flow (Ft^3/min)
$d \equiv$	Density of Air (lb/Ft^3)
$60 \equiv$	Minutes per hour
$M_o \equiv$	Moisture Level of Fresh Air (gr/lb)
$M_i \equiv$	Moisture Level Inside the Space (gr/lb)

For more detailed information, please refer to the following references:

1997 ASHRAE Handbook - Fundamentals, Chapter 24 - Thermal and Water Vapor Transmission Data; Chapter 25 - Ventilation and Infiltration; Chapter 28 - Non-Residential Cooling and Heating Load Calculations.

The Dehumidification Handbook, Second Edition, Chapter 5 - Moisture Load Calculations, Lewis Harriman, 1990

Step#4: Size Equipment

Equipment manufacturers have slightly different methods of equipment sizing for their products. Many suppliers will provide engineers with sizing support if they are provided with specific information on the application. Some suppliers provide software for engineers to use to conduct preliminary sizing and performance evaluations.

From a conceptual design point of view, the following equipment sizing guidelines are useful:

- Determine the design moisture load and design the desiccant system to remove it. Then design the portion of the system that will address the sensible load. Note that the air leaving a desiccant system is typically warmer than the air entering the desiccant system. The effect is that the desiccant system reduces the latent load on the conventional cooling system, but increases the sensible load. By sizing the desiccant system prior to sizing the sensible system, the added sensible load produced by the desiccant system can accurately be included in the sensible heat load calculation.

- In general, systems with hotter reactivation air design temperatures require less desiccant material.

Step#5: Select Control System

Most desiccant manufacturers provide an integrated control system in their packaged products for internal control of the desiccant system. As such, these systems only require an on/off signal that is generally provided by means of a humidistat installed in the supply air ducting. Additional control may be required to control dampers that allow outdoor makeup air to by-pass the desiccant unit in times when the outdoor humidity ratio is below the desiccant supply air set point. Another control option is to provide variable speed drives on the desiccant fan motors (process air fan and reactivation air fan) to reduce electric consumption at part load operation.

Step#6: Evaluate Costs

1. **System Design (Mechanical Engineer)**

2. **Hardware Costs**
   - Desiccant Unit
   - Equipment Pad
   - Ducting
   - Electrical Interface
   - Controls
   - Piping (as appropriate)
     - Natural Gas
     - Steam
     - Hot Water
     - Cold Water

3. **Installation Costs**
   - General Contractor
   - Mechanical Contractor
   - Electrical Contractor

4. **Energy Costs (as appropriate)**
   - Electricity
   - Natural Gas
   - Steam
   - Hot Water
   - Cold Water

5. **Maintenance Costs**
   - Labor
   - Filters

6. **Energy Savings**
   - Desiccant vs. Conventional Dehumidification
   - Natural Gas Latent Cooling vs. Electric Latent Cooling
   - Electric Demand Savings

7. **Non-Energy Savings**
   - Reduced Building Maintenance Due to Reduced Moisture Levels
   - Increased Productivity Due to Increased Occupant Comfort
   - Increased Product Quality Due to Better Environmental Control
   - Preservation of Valuable Materials Due to Constant Humidity Levels being Maintained

D. Preliminary Cost Estimates

The design of desiccant dehumidification systems and their costs varies widely due to a broad range of site specific requirements. Costs that should be taken into account are presented below:

I. Equipment Costs
1. Desiccant Unit
2. Equipment Pad
3. Ducting
4. Electrical Interface
5. Controls
6. Piping

II: Installation Costs
1. General Contractor
2. Mechanical Contractor
3. Electrical Contractor

III. Operation Costs
1. Hours of Operation per year
2. Cost of Electricity
3. Cost of Natural Gas
4. Desiccant Electric Consumption
5. Desiccant Gas Consumption
6. Heat Recovery

IV. Maintenance Costs
1. Service Contract
2. Labor
3. Filters
4. Belts
5. Water Treatment

D. Preliminary Costs Estimates (cont.)

DOD DESICCANT DEHUMIDIFICATION SYSTEM COSTS SUMMARY TABLE

SITE	Technology	Size (scfm)	Equipment Cost	Design/ Instal. Cost	Total Cost
ARMY					
Fort Myer	Two-Wheel	4,800	$80,000	$206,000	$286,000
Fort Campbell	Two-Wheel	4,000	$80,000	$120,000	$200,000
Aberdeen Proving Grounds	Two-Wheel	1,600	$25,000	$25,000	$50,000
AIR FORCE					
Keesler AFB	Two-Wheel	5,000	$80,000	$75,000	$155,000
MacDill AFB	Two-Wheel	18,000	$150,000	$215,000	$365,000
NAVY					
NPWC Pensacola	Two-Wheel	4,000	$50,000	$52,000	$102,000

E. Lifecycle Cost Analysis

1.) LCCID - Life Cycle Cost in Design

LCCID is an economic analysis computer program tailored to the needs of the DoD. It calculates life cycle costs and other economic parameters for a variety of energy conservation initiatives. It is a menu driven program developed by the U.S. Army Corps of Engineers Construction Engineering Research Laboratories in conjunction with the U.S. Army Corps of Engineers Missouri River Division.

Since LCCID was created as a tool for a variety of energy conservation initiatives, other programs provide more assistance with the special dynamics of desiccant dehumidification systems. Once energy parameters are calculated, LCCID can be used to determine DoD based life cycle cost analyses.

2.) GRI's DesiCalc Software Tool - Desiccant Cooling Applications

The Gas Research Institute (GRI) has developed a software tool to help engineers quickly analyze desiccant applications. The software estimates energy costs and humidity levels through an hourly computer simulation based on DOE 2.1E building and HVAC models.

Attributes of DesiCalc are as follows:

- Compares energy and costs of desiccant systems to other conventional cooling systems
- Templates for 11 commercial building types
    1. Hospital
    2. Large Hotel
    3. Small Hotel/Motel
    4. Ice Arena
    5. Nursing Home
    6. Quick-Service Restaurant
    7. Retail Store
    8. School
    9. Supermarket
    10. Theater
    11. Refrigerated Warehouse
- Typical schedules for internal loads
    1. Occupancy
    2. Lighting
    3. Equipment
    4. Ventilation
    5. Infiltration
- Default utility rates that can be customized
- Weather data for 236 cities

Data reports include the following:

- Short Report
- Detailed Report
- Charts
    - Annual Occupied Hours at Relative Humidity Range
    - Monthly Electric Energy Use
    - Monthly Electric Demand
    - Monthly Electricity Cost
    - Monthly Gas Energy Use
    - Monthly Gas Energy Cost
    - Monthly Electric and Gas Energy Cost
    - Annual Electric and Gas Energy Cost

3. Analysis Using DesiCalc (Example)

An analysis was conducted for a hospital application in Atlanta, Georgia. The reports and charts are presented at the end of this section. The default hospital template was selected and modified. Customized inputs are as follows:

Location:	Atlanta, GA
Application:	Hospital
Floor Area:	250,000 sf
Glazing:	20%
Humidification:	No
Heat Energy Source (Baseline):	Gas
Desiccant Heat Recovery:	None
Economizer:	Enthalpy

The resulting analysis is based on the following criteria:

Building

6-story, 250,000 square foot Hospital with 20% glazing. Humidity control air treatment applies to 11,000 square feet of surgical suites. Software default control scheme and default energy rates utilized.

Baseline Equipment Alternative

Constant volume chilled water system consisting of an electric chiller and an economizer. The chiller efficiency is .68 kW/ton. Economizer operation is based on enthalpy. The system does not use heat recovery and natural gas is used for space heating and domestic hot water. The system does not have a humidifier.

Desiccant Enhanced System Alternative

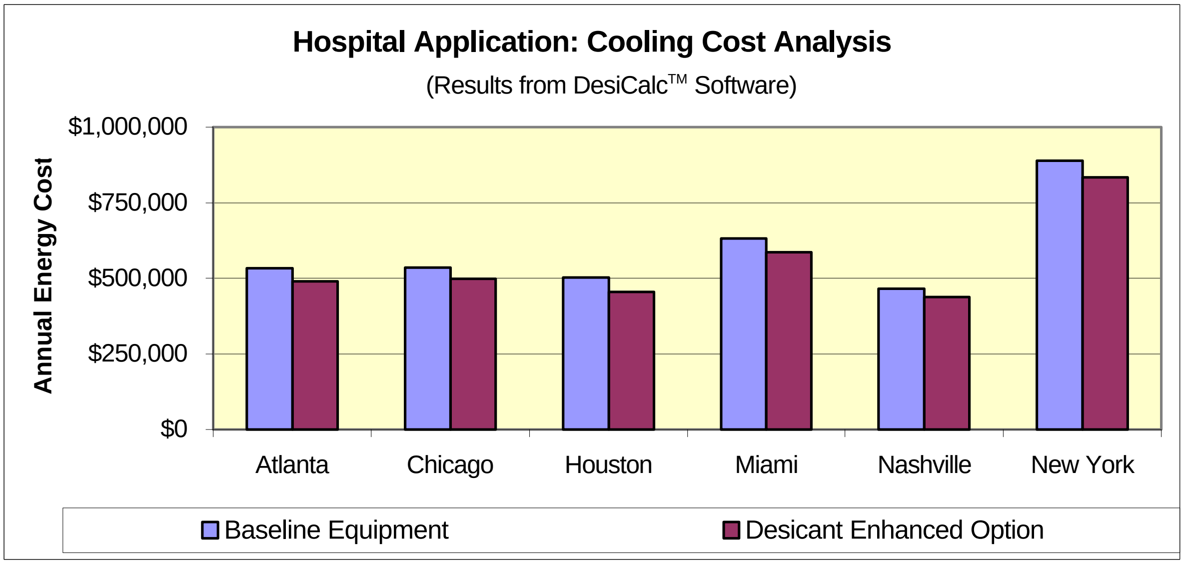
Constant volume chilled water system consisting of an electric chiller and an economizer. The chiller efficiency is .68 kW/ton. Economizer operation is based on enthalpy. The system does not use heat recovery and natural gas is used for space heating and domestic hot water. The system does not have a humidifier. A gas fired desiccant dehumidifier without any heat recovery treats outside air. The desiccant system does not have an evaporative cooler option.

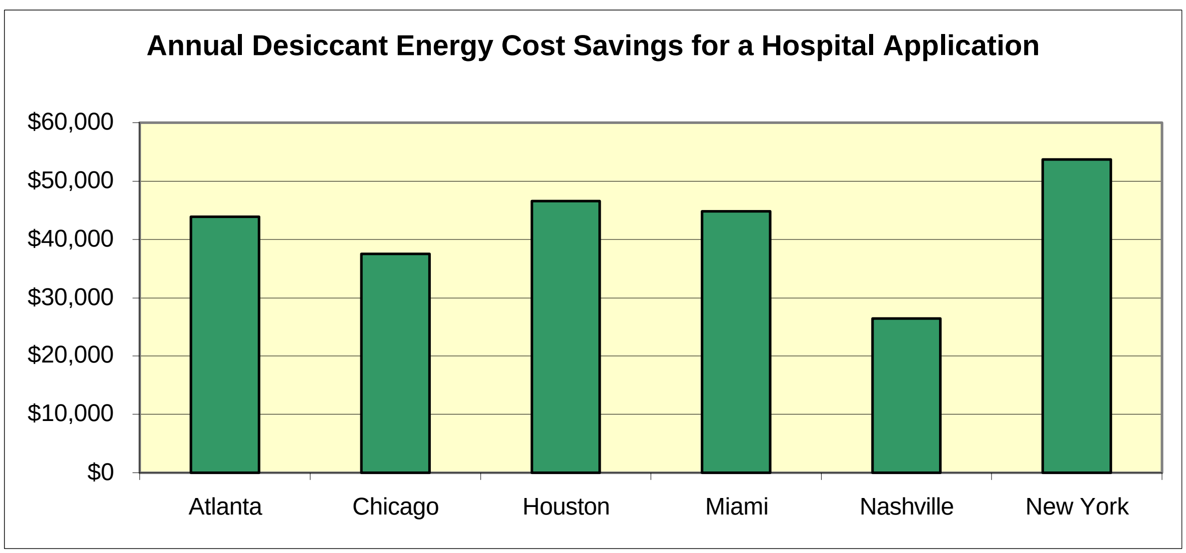
Results

Data from the sample report are presented at the end of this section and are summarized in the table below under the “Atlanta” column. The hospital’s electric costs were reduced by $50,311/year and the natural gas costs increased by $6,417/year. The resulting annual energy savings were $43,894. Summary results are also presented for the the same application in the cities of Chicago, Houston, Miami, Nashville and New York.

Hospital Desiccant Application Comparison

City	Atlanta	Chicago	Houston	Miami	Nashville	New York
Summer Design						
Dew Point (^oF)	73	72	77	77	74	73
MCDB (^oF)	81	80	83	83	82	80
Humidity Ratio (gr/lb)	128	121	141	141	130	125
Supply Air (CFM)	158,075	153,663	153,448	151,831	155,835	151,681
Outside Air (CFM)	32,242	31,587	31,028	31,020	31,640	30,993
Baseline Equipment						
Design Cooling (Tons)	740.6	709.1	833.5	811.5	763.7	720.9
Annual Electric (kWh)	6,671,007	5,917,672	7,638,286	8,507,820	6,636,647	6,003,870
Annual Gas (MMBtu)	18,498	23,483	15,446	13,092	19,596	21,648
Annual Electric ($)	$443,394	$460,709	$448,753	$545,135	$364,162	$699,783
Annual Gas ($)	$90,335	$73,850	$52,954	$85,943	$100,749	$188,112
Total Annual ($)	$533,729	$534,559	$501,707	$631,078	$464,911	$887,895
Desicant Enhanced Option						
Design Cooling (Tons)	664.9	647.4	699.6	698.8	675.8	652.1
Annual Electric (kWh)	6,045,783	5,534,054	6,691,250	7,293,960	6,024,308	5,595,220
Annual Gas (MMBtu)	22,192	25,202	22,127	21,918	23,024	23,125
Annual Electric ($)	$393,083	$418,751	$381,658	$464,066	$322,600	$637,023
Annual Gas ($)	$96,752	$78,362	$73,491	$122,187	$115,923	$197,161
Total Annual ($)	$489,835	$497,113	$455,149	$586,253	$438,523	$834,184
Operating Cost Differential						
Electric ($)	$50,311	$41,958	$67,095	$81,069	$41,562	$62,760
Gas ($)	($6,417)	($4,512)	($20,537)	($36,244)	($15,174)	($9,049)
Total ($)	**$43,894**	**$37,446**	**$46,558**	**$44,825**	**$26,388**	**$53,711**
Desiccant System Specs						
Process Air Velocity (fpm)	400	400	400	400	400	400
Dehumidifier Capacity (CFM)	32,242	31,587	31,028	31,020	31,640	30,993
ARI (95^oF DB/ 75 ^oF WB)						
Water Removal (lb/hr)	1,335	1,309	1,289	1,289	1,310	1,289
Energy Input ($Btu/lb_{removed}$)	1,668	1,668	1,671	1,671	1,668	1,672
ARI (80^oF DB/ 75 ^oF WB)						
Water Removal (lb/hr)	1,563	1,580	1,577	1,574	1,565	1,587
Energy Input ($Btu/lb_{removed}$)	1,878	1,770	1,724	1,732	1,806	1,706





For additional information refer to the following sources:

DesiCalc Software ($295.00 plus S&H) available from the Gas Research Institute; GRI Fulfillment Center, 1510 Hubbard Drive, Batavia, IL, 60510; (773) 399-5414.

Also see: http://www.gri.org; http://www.desicalc.com

Input/Output Data Short Report

JOB DESCRIPTION

Project: Hospital
Location: Atlanta
Program User:
Comments:

BUILDING

Hospital; 6-story building with 20 % wall glazing. Humidity control air treatment applies to 11000 sf of surgical suites. Internal loads and ventilation values apply to humidity controlled areas. Building total floor area is 250000 sf. Application Comfort Controls - Default Controls

LOCATION & DESIGN WEATHER

Atlanta GA - Lat./Long. 34N/84W Summer 1% Design Dry Bulb/Mean-Coincident Wet Bulb: 91/74°F (Humidity Ratio 104 gr/lb) Summer 1% Design Dew-Point/Mean-Coincident Dry Bulb: 73/81°F (Humidity Ratio 128 gr/lb). Energy Rates - Default

Equipment Sizing Design Point: 1% DB & 1% DP
Equipment Oversize: 20 %

Internal Loads and Ventilatior	
Occupancy:	275.0 sf/person
Lighting:	4.00 Watt/sf
Other Electric:	3.00 Watt/sf
Infiltration:	0.00 air exchanges/hour
Ventilation:	100.00 %

Comfort Controls	Baseline	Des. Enhanced
Cooling Temp./Setback	65 / 75 F	65 / 75 F
Heating Temp./Setback	65 / 65 F	65 / 65 F
Maximum Humidity	50 %	50 %
Minimum Humidity	0 %	0 %

EQUIPMENT & ENERGY

Baseline Equipment Alternative

Const. vol. chilled water system with 0.68 kW/ton electric chiller (water cooled) with enthalpy economizer. System does not use heat recovery. System equipped with gas source heating. Humidifier not used. Default Config.

Desiccant Enhanced System Alternative

Constant volume chilled water system with 0.68 kW/ton electric chiller (water cooled) with enthalpy economizer. System equipped with gas source heating. Outside air treated by gas-fired desiccant dehumidifier with 0 % eff. heat exch. (without heat recovery). Dehumidifier configured without evap. cooler option. Humidifier not used. Default Config.

	Baseline		Desiccant Enhanced	
Design Cooling Capacity:	740.64	RT	664.93	RT
Design Heating Capacity:	3,814,557	Btu/hr	3,814,557	Btu/hr
Supply Fans Capacity:	158,075	CFM	158,075	CFM
Outside Air:	32,242	CFM	32,242	CFM
Annual Electric Energy Use:	6,674,007	kWh	6,045,783	kWh
Annual Gas Energy Use:	18,498	MMBtu	22,192	MMBtu
Annual Electric Energy Cost:	443,394	$	393,083	$
Annual Gas Energy Cost:	90,335	$	96,752	$
Total Annual Energy Cost	533,729	$	489,835	$
Annual Occupied Hours @ RH>60%	0		0	

DESICCANT DEHUMIDIFIER UNIT PERFORMANCE SPECIFICATION

(ARI Standard 940P Rating Conditions)

Process Air Flow Face Velocity: 400 fpm
Dehumidifier Capacity: 32,242 CFM

DB (F)	WB (F)	Humidity (gr/lb)	Water Removed (lb/hr)	Specific Energy Input (Btu/lb_removed water)
95	75	100.0	1,335	1,668
80	75	124.5	1,563	1,878

Note. Desiccant Dehumidifier Precooling Coil Max. Capacity: 0.059 RT
Regeneration air source is outside air.

DESICCANT WHEEL MATRIX PERFORMANCE SPECIFICATION

(ARI Standard 940P Rating Conditions)

Process Air Flow Face Velocity: 400 fpm

DB (F)	WB (F)	Humidity (gr/lb)	Water Removed (lb/hr)	Specific Energy Input (Btu/lb_removed water)
95	75	100.0	825	1,772
80	75	124.5	1,375	1,776

Note. The annual energy consumption and costs given in this report reflect facility total energy use including lights, equipment, and HVAC equipment. Details of monthly energy consumption by end use are given in Detailed Report.

Units Used
RT = 12,000 Btu/hr
MMBtu = 1,000,000 Btu

Cooling and Heating Coil Loads

Hospital
Atlanta

Baseline System

Month	Cooling Sensible MMBtu	Cooling Latent MMBtu	Cooling Total MMBtu	Heating/Reheati Total MMBtu
JAN	56	24	80	417
FEB	177	33	210	337
MAR	385	73	457	175
APR	940	204	1,144	157
MAY	1,586	441	2,027	170
JUN	1,947	719	2,666	174
JUL	2,141	1,168	3,309	187
AUG	2,094	1,123	3,217	181
SEP	1,822	929	2,751	184
OCT	989	301	1,290	159
NOV	313	91	404	215
DEC	143	57	200	357
Total	12,593	5,163	17,756	2,714

Alternative System

Month	Cooling Sensible MMBtu	Cooling Latent MMBtu	Cooling Total MMBtu	Heating Total MMBtu
JAN	59	3	63	389
FEB	174	3	177	297
MAR	404	11	415	118
APR	981	58	1,039	49
MAY	1,705	162	1,867	2
JUN	2,170	315	2,486	0
JUL	2,498	561	3,060	0
AUG	2,449	535	2,984	0
SEP	2,136	424	2,561	0
OCT	1,073	99	1,173	32
NOV	338	13	351	141
DEC	164	11	175	318
Total	14,152	2,198	16,350	1,345

Electric Energy Consumption by End Use

Baseline System

Month	Lights kWh	Misc. Equip. kWh	Space Cooling kWh	Pumps & Misc. kWh	Fans Vent. kWh	Space Heating kWh	Heat Reject. kWh	Refrig. kWh	Dom.Hot Water kWh	Total kWh
JAN	265,160	94,910	8,481	9,215	34,656	2,187	1,605	0	0	416,214
FEB	238,886	85,320	19,756	10,802	33,165	1,855	2,921	0	0	392,705
MAR	265,590	95,226	42,013	16,334	40,897	1,701	5,745	0	0	467,506
APR	256,402	91,714	97,162	19,496	45,169	1,142	10,558	0	0	521,643
MAY	265,160	94,910	168,607	24,081	54,555	669	16,691	0	0	624,673
JUN	256,832	92,029	220,678	23,093	57,474	623	20,356	0	0	671,085
JUL	264,264	94,305	277,869	23,815	61,212	665	25,595	0	0	747,725
AUG	266,056	95,515	268,787	23,815	60,238	643	24,443	0	0	739,497
SEP	255,040	90,819	227,351	23,155	53,893	656	20,801	0	0	671,715
OCT	265,160	94,910	109,307	21,133	47,106	1,053	11,730	0	0	550,399
NOV	255,506	91,108	36,715	14,998	37,888	1,555	5,187	0	0	442,957
DEC	263,798	94,016	18,631	11,012	35,551	2,056	2,824	0	0	427,888
Total	3,117,854	1,114,782	1,495,357	220,949	561,804	14,805	148,456	0	0	6,674,007

Alternative System

Month	Lights kWh	Misc. Equip. kWh	Space Cooling kWh	Pumps & Misc. kWh	Fans Vent. kWh	Space Heating kWh	Heat Reject. kWh	Refrig. kWh	Dom.Hot Water kWh	Total kWh
JAN	265,160	94,910	4,334	7,842	34,613	2,087	1,012	0	0	409,958
FEB	238,886	85,320	11,455	9,756	33,141	1,710	2,273	0	0	382,541
MAR	265,590	95,226	25,959	14,392	40,856	1,497	4,734	0	0	448,254
APR	256,402	91,714	59,691	16,669	45,170	756	8,717	0	0	479,119
MAY	265,160	94,910	105,023	19,639	54,583	68	14,047	0	0	553,430
JUN	256,832	92,029	139,376	18,157	57,559	6	17,372	0	0	581,331
JUL	264,264	94,305	175,259	18,487	61,316	0	21,876	0	0	635,507
AUG	266,056	95,515	169,423	18,472	60,356	0	20,934	0	0	630,756
SEP	255,040	90,819	142,961	18,009	54,011	0	17,814	0	0	578,654
OCT	265,160	94,910	67,245	17,644	47,111	599	9,697	0	0	502,366
NOV	255,506	91,108	22,084	13,277	37,827	1,288	4,256	0	0	425,346
DEC	263,798	94,016	11,122	9,966	35,503	1,912	2,202	0	0	418,519
Total	3,117,854	1,114,782	933,932	182,310	562,046	9,923	124,934	0	0	6,045,781

Gas Energy Consumption by End Use

Baseline System

Month	Space Heating MMBtu	Space Cooling MMBtu	Dom. Hot Water MMBtu	Misc. Domest. MMBtu	Supl. Heating MMBtu	Ext. Misc. MMBtu	Total MMBtu
JAN	1,134	0	845	243	0	0	2,222
FEB	959	0	785	218	0	0	1,962
MAR	745	0	872	247	0	0	1,863
APR	448	0	832	235	0	0	1,514
MAY	223	0	813	243	0	0	1,278
JUN	208	0	738	238	0	0	1,184
JUL	222	0	719	240	0	0	1,180
AUG	214	0	691	247	0	0	1,152
SEP	219	0	666	231	0	0	1,116
OCT	373	0	711	243	0	0	1,327
NOV	695	0	730	231	0	0	1,656
DEC	1,004	0	803	240	0	0	2,047
Total	6,442	0	9,204	2,854	0	0	18,499

Alternative System

Month	Space Heating MMBtu	Space Cooling MMBtu	Dom. Hot Water MMBtu	Misc. Domest. MMBtu	Supl. Heating MMBtu	Ext. Misc. MMBtu	Total MMBtu
JAN	1,100	42	845	243	0	0	2,230
FEB	910	53	785	218	0	0	1,967
MAR	676	116	872	247	0	0	1,910
APR	319	254	832	235	0	0	1,639
MAY	23	482	813	243	0	0	1,560
JUN	2	698	738	238	0	0	1,676
JUL	0	1,096	719	240	0	0	2,054
AUG	0	1,055	691	247	0	0	1,992
SEP	0	931	666	231	0	0	1,829
OCT	221	365	711	243	0	0	1,540
NOV	605	146	730	231	0	0	1,711
DEC	956	88	803	240	0	0	2,087
Total	4,811	5,326	9,204	2,854	0	0	22,194

Total Monthly Electric Consumption and Electric Energy Cost

Baseline System

Month	Metered Energy kWh	Metered Demand kW	Energy Charge ($)	Demand Charge ($)	Energy Cost Adj ($)	Taxes ($)	Surch. ($)	Fixed Charge ($)	Min. Charge ($)	Total Charge ($)
JAN	416,215	1,071	25,115	0	6,285	1,580	0	17	0	32,997
FEB	392,706	1,104	24,854	0	5,930	1,549	0	17	0	32,350
MAR	467,507	1,191	25,684	0	7,059	1,648	0	17	0	34,408
APR	521,642	1,262	26,285	0	7,877	1,719	0	17	0	35,898
MAY	624,674	1,430	27,340	0	9,433	1,851	0	17	0	38,640
JUN	671,084	1,531	28,338	0	133	1,936	0	17	0	40,424
JUL	747,725	1,559	29,321	0	1,291	2,044	0	17	0	42,672
AUG	739,496	1,559	29,251	0	1,166	2,034	0	17	0	42,467
SEP	671,715	1,450	27,733	0	143	1,906	0	17	0	39,799
OCT	550,398	1,330	26,604	0	8,311	1,757	0	17	0	36,689
NOV	442,958	1,138	25,412	0	6,689	1,615	0	17	0	33,732
DEC	427,889	1,078	25,244	0	6,461	1,596	0	17	0	33,318
Total	6,674,009	15,702	321,181	0	60,778	21,235	0	204	0	443,394

Alternative System

Month	Metered Energy kWh	Metered Demand kW	Energy Charge ($)	Demand Charge ($)	Energy Cost Adj ($)	Taxes ($)	Surch. ($)	Fixed Charge ($)	Min. Charge ($)	Total Charge ($)
JAN	409,958	942	22,513	0	6,190	1,445	0	17	0	30,164
FEB	382,542	976	22,208	0	5,776	1,408	0	17	0	29,410
MAR	448,255	1,028	22,938	0	6,769	1,495	0	17	0	31,218
APR	479,117	1,098	23,280	0	7,235	1,536	0	17	0	32,068
MAY	553,430	1,219	23,962	0	8,357	1,626	0	17	0	33,962
JUN	581,332	1,253	24,197	0	8,778	1,660	0	17	0	34,652
JUL	635,508	1,319	25,454	0	9,596	1,764	0	17	0	36,831
AUG	630,757	1,303	25,213	0	9,524	1,748	0	17	0	36,502
SEP	578,654	1,221	24,173	0	8,738	1,656	0	17	0	34,584
OCT	502,367	1,139	23,535	0	7,586	1,566	0	17	0	32,704
NOV	425,346	999	22,683	0	6,423	1,465	0	17	0	30,588
DEC	418,519	970	22,608	0	6,320	1,456	0	17	0	30,400
Total	6,045,785	13,467	282,764	0	91,292	18,825	0	204	0	393,083

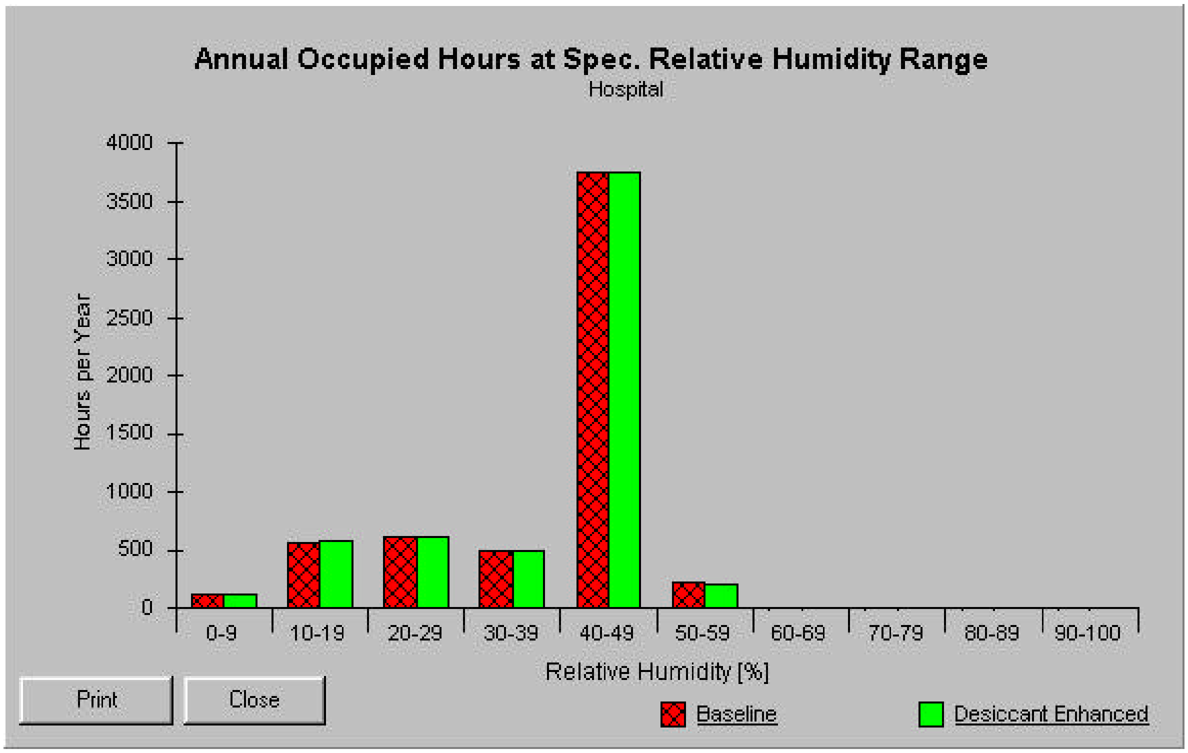
Total Monthly Gas Consumption and Gas Energy Cost

Baseline System

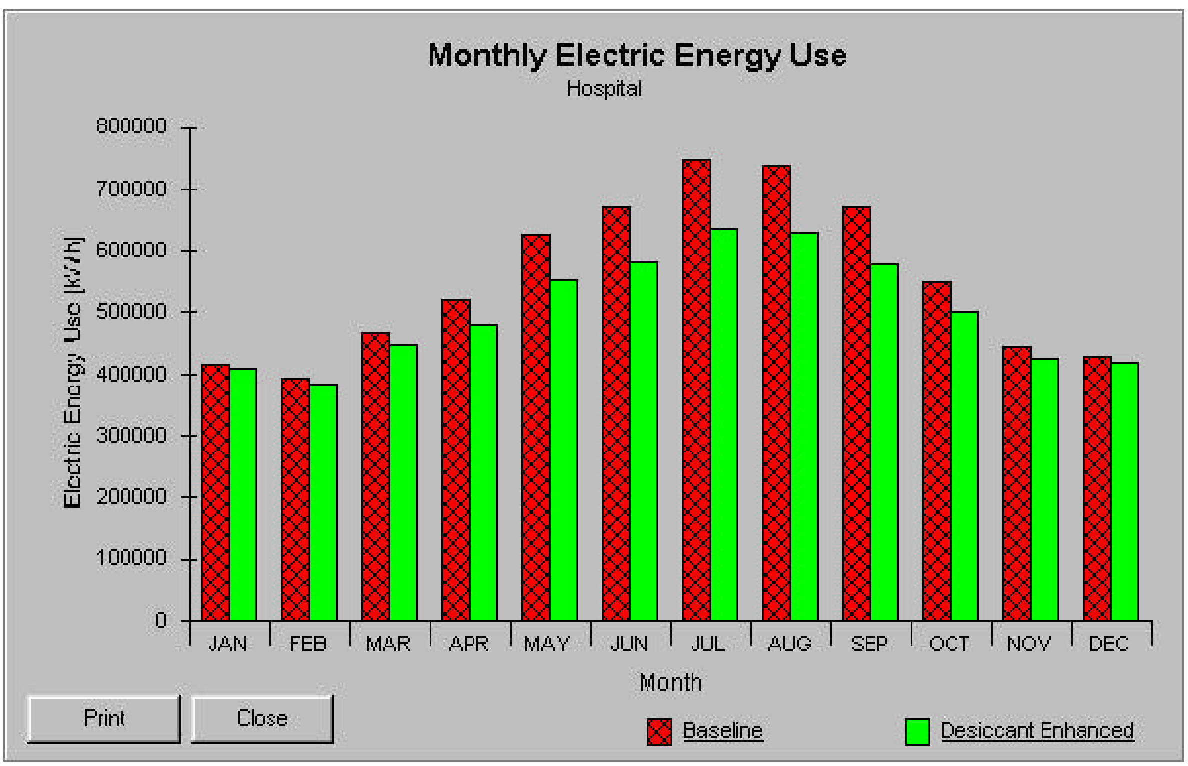
Month	Metered Energy Therms	Metered Demand Therms/Day	Energy Charge ($)	Demand Charge ($)	Energy Cost Adj ($)	Taxes ($)	Surch. ($)	Fixed Charge ($)	Min. Charge ($)	Total Charge ($)
JAN	22,220	983	10,241	0	0	746	0	13	0	11,000
FEB	19,619	1,003	9,043	0	0	659	0	13	0	9,716
MAR	18,630	833	8,588	0	0	626	0	13	0	9,227
APR	15,142	801	6,983	0	0	509	0	13	0	7,505
MAY	12,784	442	5,624	0	0	410	0	13	0	6,048
JUN	11,838	408	5,209	0	0	380	0	13	0	5,603
JUL	11,796	398	5,191	0	0	379	0	13	0	5,583
AUG	11,516	387	5,068	0	0	370	0	13	0	5,451
SEP	11,158	387	4,911	0	0	358	0	13	0	5,282
OCT	13,269	559	6,121	0	0	447	0	13	0	6,581
NOV	16,555	803	7,633	0	0	557	0	13	0	8,203
DEC	20,469	912	9,435	0	0	688	0	13	0	10,136
Total	184,996	7,917	84,047	0	0	6,129	0	156	0	90,335

Alternative System

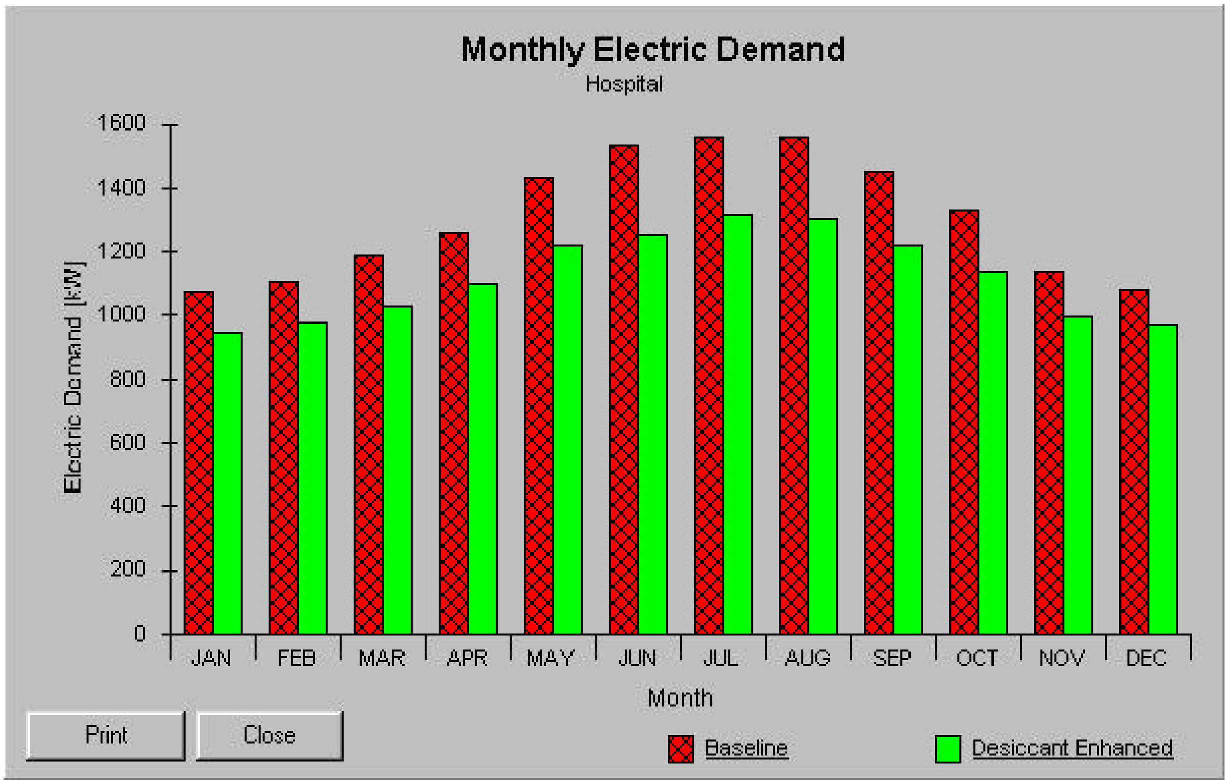
Month	Metered Energy Therms	Metered Demand Therms/Day	Energy Charge ($)	Demand Charge ($)	Energy Cost Adj ($)	Taxes ($)	Surch. ($)	Fixed Charge ($)	Min. Charge ($)	Total Charge ($)
JAN	22,303	1,182	10,189	0	0	743	0	13	0	10,945
FEB	19,666	1,179	8,952	0	0	653	0	13	0	9,618
MAR	19,103	1,037	8,561	0	0	624	0	13	0	9,198
APR	16,388	1,020	7,021	0	0	512	0	13	0	7,546
MAY	15,597	730	5,945	0	0	433	0	13	0	6,391
JUN	16,765	762	6,046	0	0	441	0	13	0	6,500
JUL	20,541	820	6,948	0	0	507	0	13	0	7,468
AUG	19,920	749	6,754	0	0	492	0	13	0	7,259
SEP	18,285	730	6,271	0	0	458	0	13	0	6,742
OCT	15,400	873	6,331	0	0	462	0	13	0	6,806
NOV	17,112	1,018	7,583	0	0	553	0	13	0	8,148
DEC	20,865	1,139	9,430	0	0	687	0	13	0	10,131
Total	221,945	11,240	90,031	0	0	6,565	0	156	0	96,752



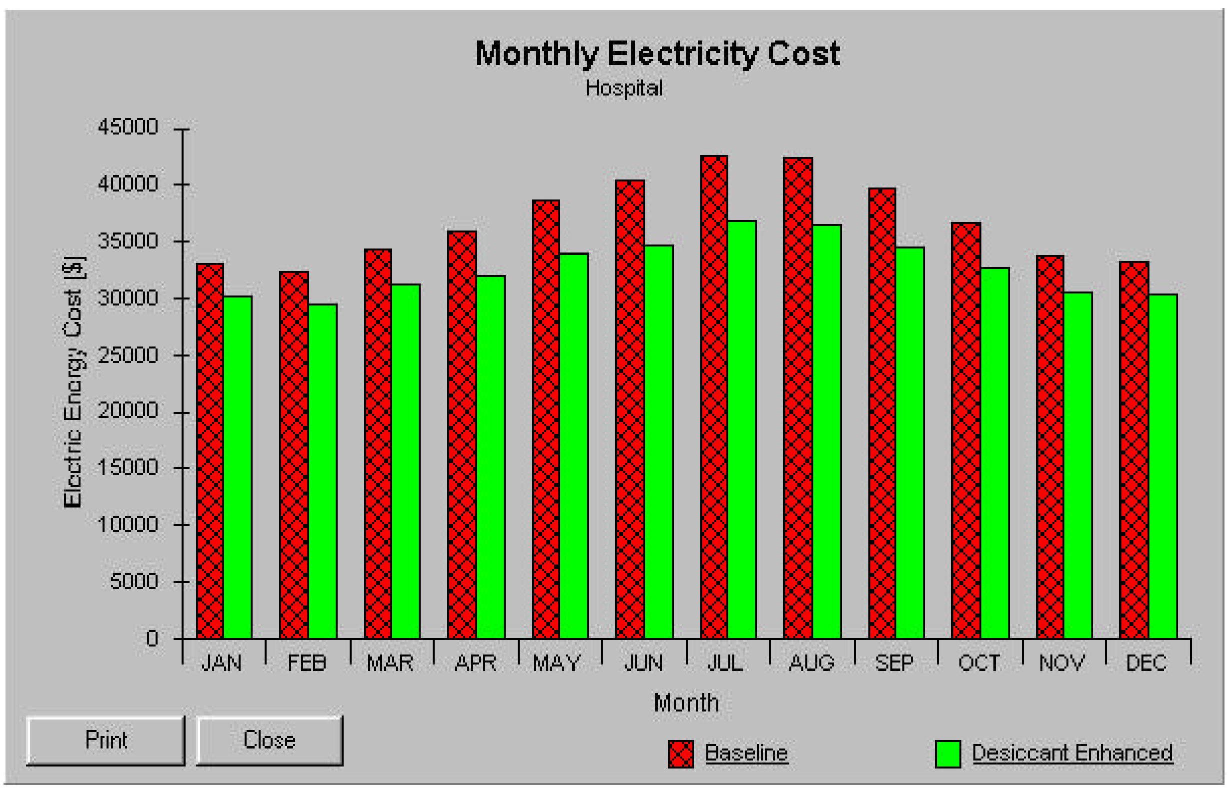
Annual Occupied Hours at Spec. Relative Humidity Range
Hospital
Hours per Year
4000
3500
3000
2500
2000
1500
1000
500
0
0-9
10-19
20-29
30-39
40-49
50-59
60-69
70-79
80-89
90-100
Relative Humidity [%]
Print
Close
Baseline
Desiccant Enhanced



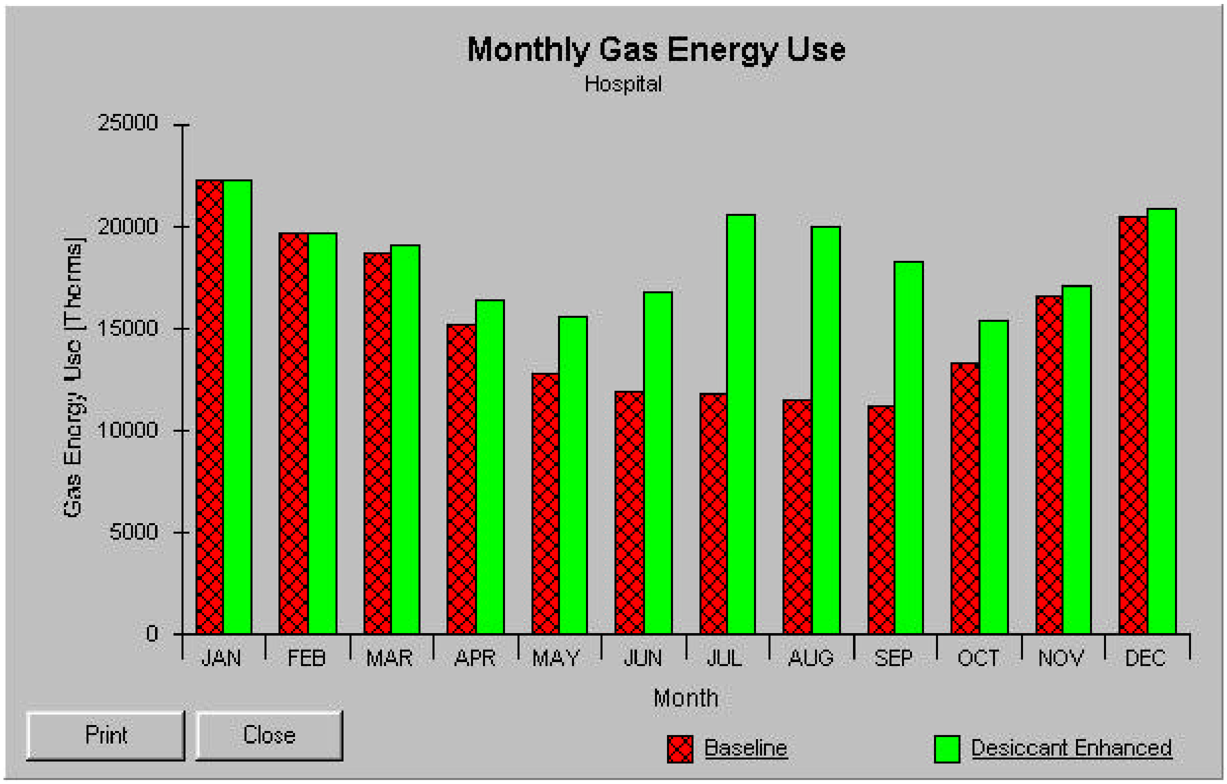
Monthly Electric Energy Use
Hospital
Electric Energy Use [kWh]
800000
700000
600000
500000
400000
300000
200000
100000
0
JAN
FEB
MAR
APR
MAY
JUN
JUL
AUG
SEP
OCT
NOV
DEC
Month
Print
Close
Baseline
Desiccant Enhanced



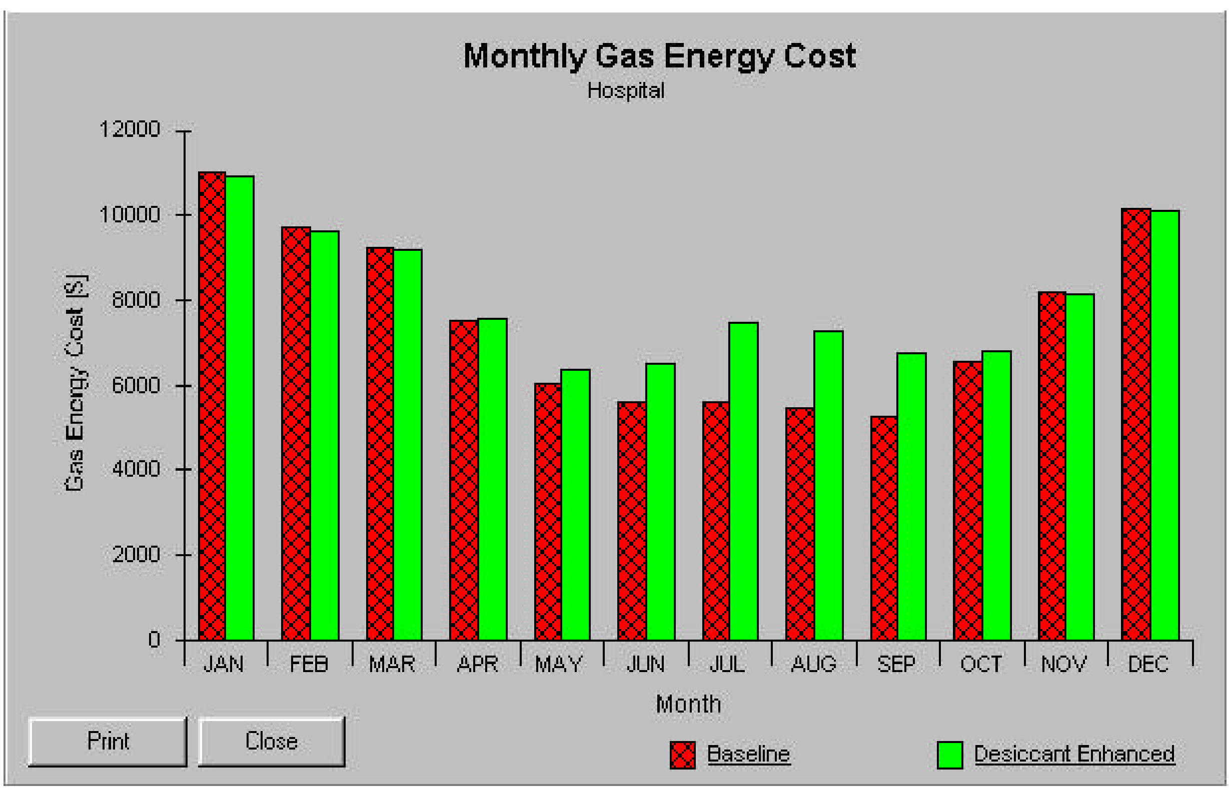
Monthly Electric Demand
Hospital
1600
1400
1200
1000
800
600
400
200
0
Electric Demand [kW]
JAN
FEB
MAR
APR
MAY
JUN
JUL
AUG
SEP
OCT
NOV
DEC
Month
Print
Close
Baseline
Desiccant Enhanced



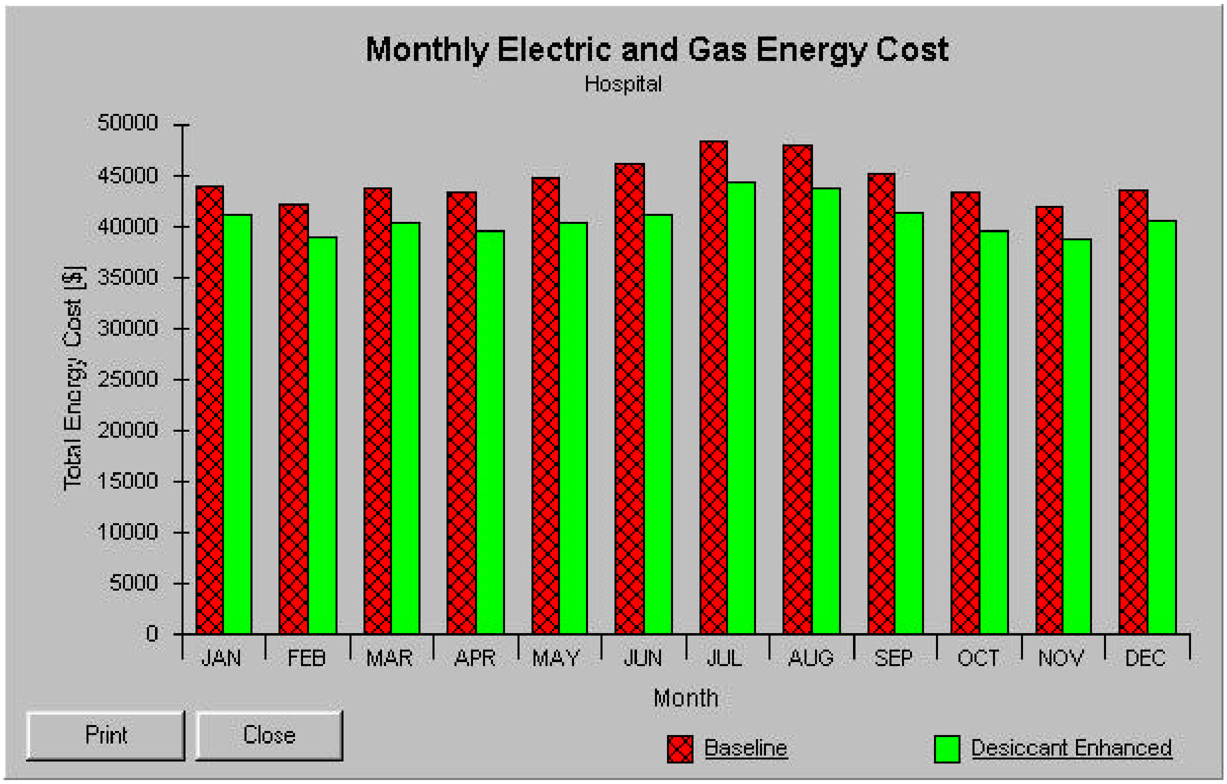
Monthly Electricity Cost
Hospital
45000
40000
35000
30000
25000
20000
15000
10000
5000
0
Electric Energy Cost [$]
JAN
FEB
MAR
APR
MAY
JUN
JUL
AUG
SEP
OCT
NOV
DEC
Month
Print
Close
Baseline
Desiccant Enhanced



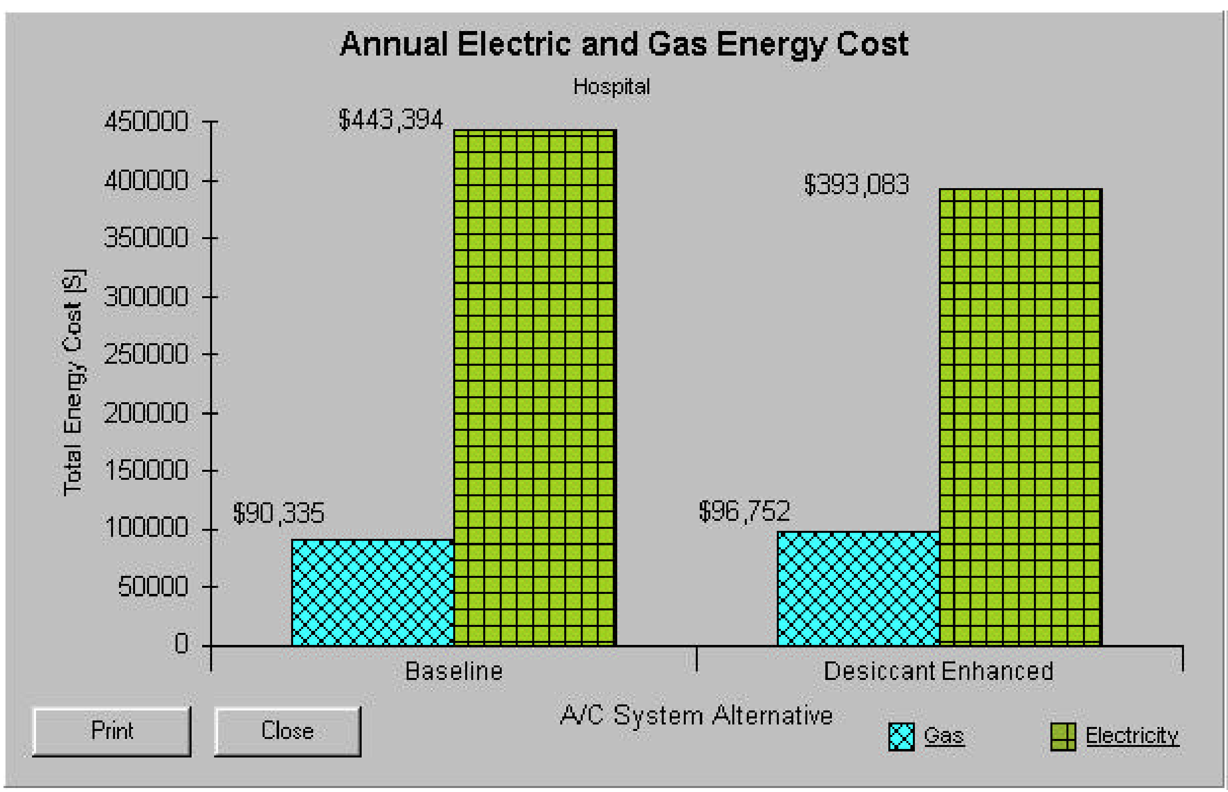
Monthly Gas Energy Use
Hospital
Gas Energy Use [Therms]
25000
20000
15000
10000
5000
0
JAN
FEB
MAR
APR
MAY
JUN
JUL
AUG
SEP
OCT
NOV
DEC
Month
Print
Close
Baseline
Desiccant Enhanced



Monthly Gas Energy Cost
Hospital
Gas Energy Cost [$]
12000
10000
8000
6000
4000
2000
0
JAN
FEB
MAR
APR
MAY
JUN
JUL
AUG
SEP
OCT
NOV
DEC
Month
Print
Close
Baseline
Desiccant Enhanced



Monthly Electric and Gas Energy Cost
Hospital
50000
45000
40000
35000
30000
25000
20000
15000
10000
5000
0
Total Energy Cost [$]
JAN
FEB
MAR
APR
MAY
JUN
JUL
AUG
SEP
OCT
NOV
DEC
Month
Print
Close
Baseline
Desiccant Enhanced



Annual Electric and Gas Energy Cost
Hospital
450000
400000
350000
300000
250000
200000
150000
100000
50000
0
Total Energy Cost [$]
$443,394
$393,083
$90,335
$96,752
Baseline
Desiccant Enhanced
A/C System Alternative
Print
Close
Gas
Electricity

F. Sample Paperwork

The following are links to DoD desiccant site documents:

Fort Myer Technical Specifications

Fort Campbell Technical Specifications

Section Outline

II. Desiccant Design and Engineering Guide

A. Service Construction Requirements

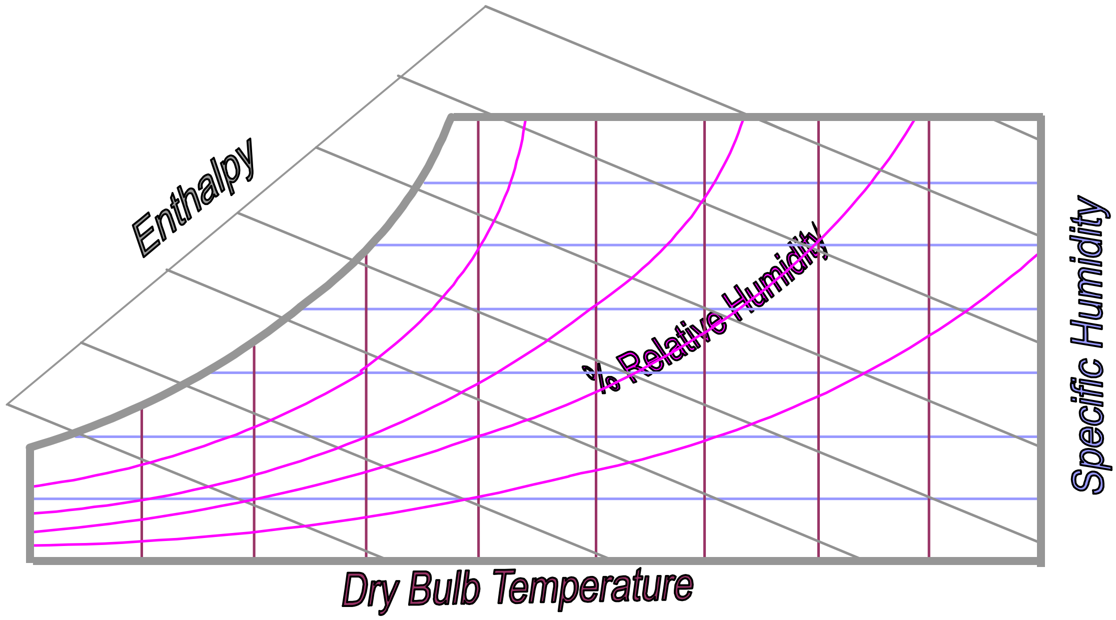
The following are links to Military Guide Specifications:

Guide Specification for Military Construction - Desiccant Cooling Systems.

Naval Facilities Engineering Command Guide Specification - Desiccant Dehumidification Equipment.

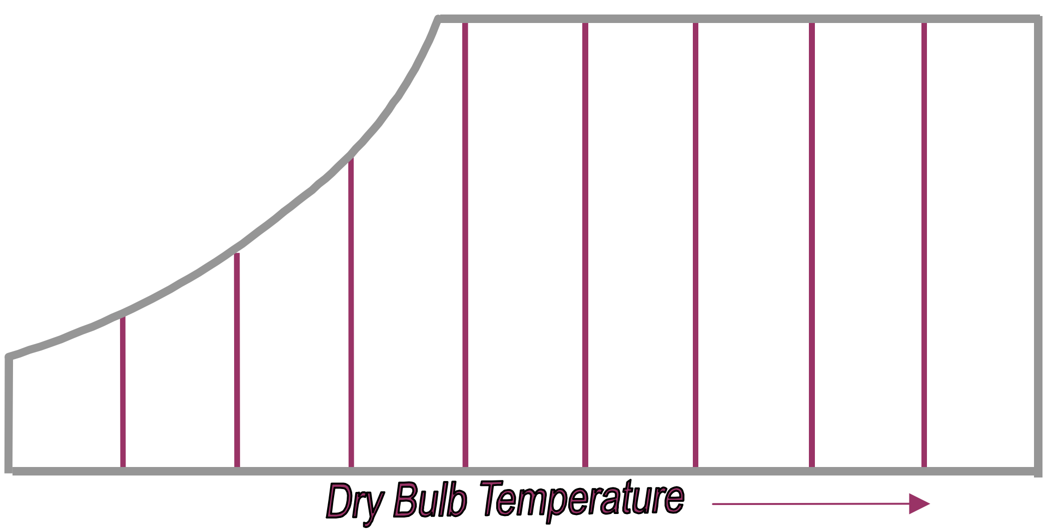
B. Psychrometrics

The scientific principles used to describe the thermodynamic properties of moist air are known as psychrometrics. The relationships of moist air parameters are represented graphically on the psychrometric chart as presented below. By knowing any two parameters, other characteristics of the air can be determined. For example, knowing the dry bulb temperature and relative humidity, one can determine the quantity of moisture in the air (specific humidity). This section provides an overview of the psychrometric chart including definitions of moist air terminology.



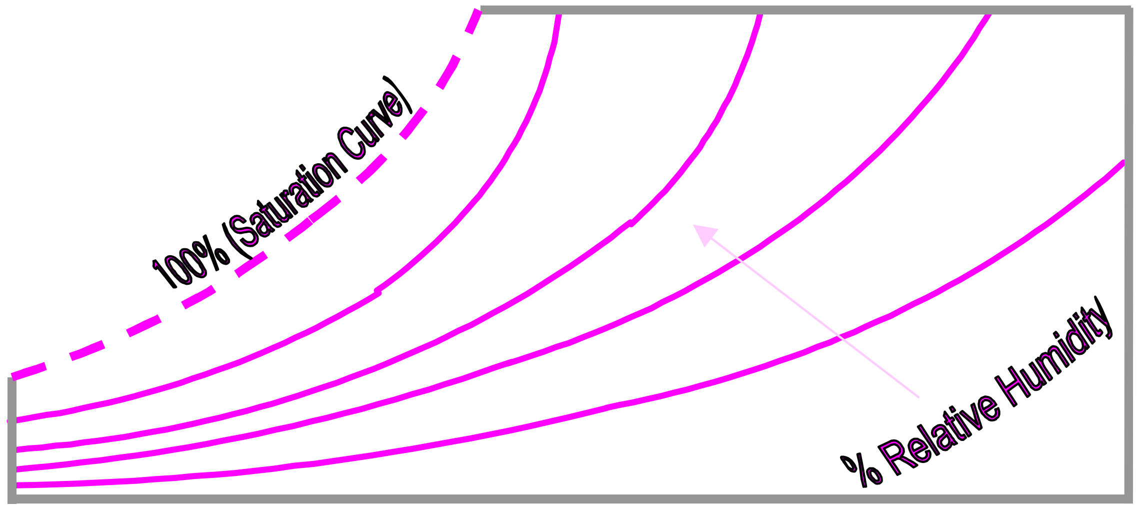
1. Dry Bulb Temperature

The dry bulb temperature is the temperature that is measured by a dry thermometer. The dry bulb temperature is represented on the psychrometric chart as the vertical lines presented below which increases in value from the left to the right:



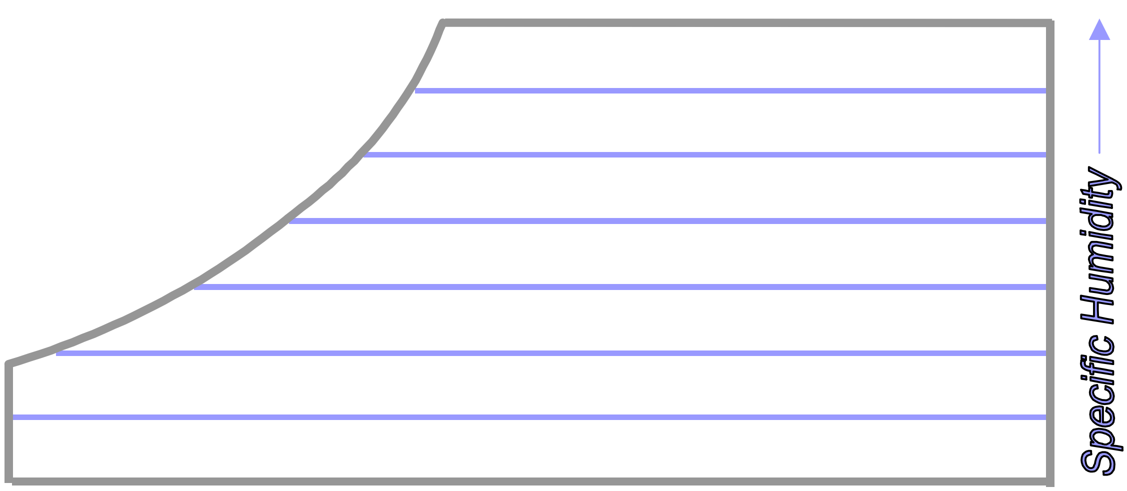
2. Relative Humidity

The relative humidity indicates the percent of moisture in the air with respect to the quantity of moisture that the air is capable of holding at saturation. Relative humidity is the only parameter on the psychrometric chart that is not linear. Constant relative humidity lines increase in value as one reads up and to the left as shown on the chart below.



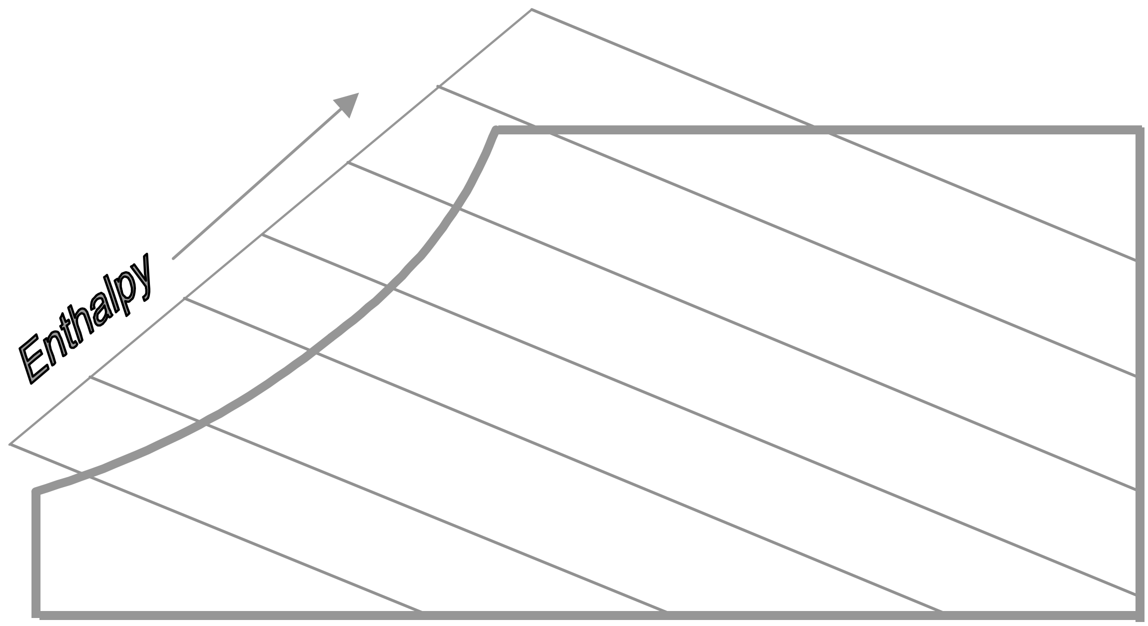
3. Specific Humidity

The specific humidity defines the ratio of moisture to dry air. This parameter can be represented as the humidity ratio in units of "pounds of moisture per pound of dry air". It is also common to present this parameter in grains. Grains is equal to pounds of moisture per pound of dry air times 7000 grains per pound. On the psychrometric chart, the specific humidity is represented as horizontal lines and increases in value as one moves up the chart as presented below.



4. Enthalpy

Enthalpy is a thermodynamic quantity that indicates the level of energy and is represented in units of BTU's per pound of dry air. The total energy of moist air has two components: 1) sensible energy and 2) latent energy. The latent energy is attributed to the moisture in the air and represents the quantity of heat required to remove the moisture. This parameter is used to determine the minimum quantity of energy required to move from one state of air conditions to another. On the psychrometric chart, the enthalpy is represented on the left side of the chart and increases in value as one moves up the chart as presented below.



5. Wet Bulb Temperature

The wet bulb temperature is measured with a thermometer that has a wet wick. As water evaporates from the wick, the temperature measured decreases due to evaporative cooling. The resulting temperature measurement is the wet bulb temperature. The wet bulb temperature provides another measure that can used to quantify the amount of moisture in the air. On the psychrometric chart, the wet bulb temperature is represented as lines that are nearly parallel to the enthalpy curves.

6. Psychrometric Examples

The following are two examples of determining the psychometric values given two of the variable conditions.

<u>Example #1:</u>

Given:	Dry Bulb Temperature:	70 °F
	Relative Humidity:	50%

Resulting psychrometric parameters

	Humidity Ratio:	55 grains/lb
	Wet Bulb Temperature:	58 °F
	Enthalpy:	25 BTU/lb

<u>Example #2:</u>

Given:	Dry Bulb Temperature:	85 °F
	Wet Bulb Temperature:	76 °F

Resulting psychrometric parameters

	Humidity Ratio:	121 grains/lb
	Enthalpy:	39 BTU/lb
	Relative Humidity:	67%

For additional information refer to the following sources:

The Dehumidification Handbook, Second Edition, Chapter 2 Psychrometrics, Lewis Harriman, 1990

1993 ASHRAE Handbook - Fundamentals, Chapter 6 - Psychrometrics

C. Equipment Sources

The table below presents a list of manufacturers of desiccant dehumidification systems.

MANUFACTURER	EQUIPMENT	TECHNOLOGY	SIZE
Airflow Co. 295 Bailes Lane Frederick, MD 21701-3136 (301) 695-6500 FAX (301) 695-4057 www.airflowcompany.com	Industrial/Commercial	Rotary Wheel, Granular bed	25 to 24,000 cfm
Bry-Air Inc. 10793 St. Route 37 W. P.O. Box 269 Sunbury, OH 43074 (614) 965-2974 FAX (614) 965-5470 www.bry-air.com	Custom Industrial (Large) Commercial Desiccant Dehumidification Systems	Rotating Wheel, Multiple Vertical Bed	16 to 830 lb./hr. of moisture removal. (500 to 25,000 cfm)
Fresh Air Solutions **(formerly Engelhard/ICC)** 441 N. 5th St., Suite 102 Philadelphia, PA 19123-4008 (215) 625-0700 1-800-220-3301 FAX (215) 592-8299 www.freshairsolutions.com	Desiccant Dehumidification	Two-wheel Desiccant	2,000 to 20,000 cfm
Kathabar Inc. P.O. Box 791 New Brunswick, NJ 08903 (908) 356-6000 1-800-524-1370 FAX (908) 356-0643 www.kathabar.com	Air Conditioning/ Dehumidification	Liquid Desiccants	Up to 10,000 lb./hr. of moisture removal. 600-84,000 cfm
Munters Corp., **Cargocaire Division** 79 Monroe St. Amesbury, MA 01913-0640 (978) 241-1100 1-800-843-5360 FAX (978) 241-1215 www.muntersamerica.com	Desiccant Dehumidification	Honeycomb Wheel	Up to 300 lb./hr. of water removal.
Semco Inc. 1800 E. Pointe Drive Columbia, MO 65201-3508 (573) 443-1481 FAX (573) 886-5408 www.semcoinc.com	Industrial/Commercial Desiccant Dehumidification	Rotating Wheel	2,000-40,000 cfm

C. Equipment Sources (cont.)

The following manufacturer information is provided for information purposes only. Presentation of this product literature does not constitute any recommendation on the part of the U.S. Government, nor is any warranty implied as to the quality of the products discussed.

Air Flow Company
Air Technology Systems, Inc.
Bry-Air, Inc.
Fresh Air Solutions (formerly Engelhard/ICC)
Kathabar, Inc.
Munters Corp. - Cargocaire Division
SEMCO, Inc.

Section Outline

III. Desiccant Construction and O&M Guide

A APPLICABLE BUILDING CODES

B. CONSTRUCTION SPECIFICATIONS

C. SAMPLE CONSTRUCTION DRAWINGS

D. SAMPLE O&M MANUALS

A. APPLICABLE BUILDING CODES

1. Introduction

The installation of desiccant systems is similar to the installation of other major HVAC mechanical equipment hardware. Engineers and contractors should comply with all applicable state and local building codes. This section provides an overview of the typical standards and codes that are applicable to desiccant systems. Specific codes and standards should be reviewed directly and can be obtained directly from the issuing organization.

2. General Building Codes

Work shall comply with all applicable state and local building codes

Work shall comply with the Uniform Building Code

> The Uniform Building Code is the most widely adopted model building code in the world and is a proven document meeting the needs of government agencies charged with enforcement of building regulation. Published triennially, the UBC provides complete regulations covering all major aspects of building design and construction relating to fire and life-safety and structural-safety. The requirements reflect the latest technological advances available in the building and fire and life-safety industry.

Work shall comply with the Uniform Mechanical Code

> The Uniform Mechanical Code provides a complete set of requirements for the design, construction, installation and maintenance of heating, ventilating, cooling and refrigeration systems, incinerators, and other heat-producing appliances.

Work shall comply with the Uniform Plumbing Code

> The Uniform Plumbing Code is published by the International Association of Plumbing and Mechanical Officials (IAPMO). It covers all aspects of plumbing, including requirements for plumbing materials and IAPMO installation standards.

Electrical construction shall be performed in strict accordance with the National Electric Code (NEC)

> The National Electrical Code (NFPA 70) provides "practical safeguarding of persons and property from hazards arising from the use of electricity." More specifically, the NEC covers the installation of electric conductors and equipment in public and private buildings or other structures (including mobile homes, recreational vehicles, and floating buildings), industrial substations, and other premises (such as yards, carnivals, and parking lots). The NEC also covers installations of optical fiber cable.

Wiring, general electrical equipment, the use of electricity in specific occupancies (from aircraft hangars to health care facilities), and equipment (ranging from elevators to hot tubs) are covered, as well as special conditions (emergency and stand-by power, or conditions requiring more than 600 volts, for example) and communication systems.

The NEC protects the public adoption and enforcement of the National Electrical Code and protects public safety by establishing requirements for electrical wiring and equipment in virtually all buildings.

Work shall comply with the requirements as set forth by the National Fire Protection Association (NFPA)

The NFPA provides requirements for building design, construction, operation, and maintenance to protect occupants from fire, smoke, and fumes or similar emergencies.

Qualification of welding procedures: ASME Boiler and Pressure Vessel Code, Section IX

Section IX of the ASME Boiler and Pressure Vessel Code relates to the qualification of welders, welding operators, brazers, and brazing operators and the procedures employed in welding or brazing in accordance with the ASME Boiler and Pressure Vessel Code and the ASME B31 Code for Pressure Piping. Section IX establishes the basic criteria for welding and brazing which are observed in the preparation of welding and brazing requirements that affect procedure and performance. The purpose of the Welding Procedure Specification (WPS) and Procedure Qualification Record (PQR) is to determine that the weldment proposed for construction is capable of having the required properties for its intended application.

Ventilation for Acceptable Indoor Air Quality: ASHRAE Standard 62-1989

This standard specifies minimum ventilation rates and indoor air quality that will be acceptable to human occupants and are intended to minimize the potential for adverse health effects. The standard applies to all indoor or enclosed spaces that people may occupy, except where other applicable standards and requirements dictate larger amounts of ventilation than this standard. Release of moisture in residential kitchens and bathrooms, locker rooms, and swimming pools is included.

3. Selected Heating Ventilation and Air Conditioning Codes

Products shall conform to NFPA 90A and 90B

NFPA 90A: Standard for the Installation of Air Conditioning and Ventilating Systems, 1996 Edition

Make HVAC systems part of the fire safety solution

Air ducts have the potential to transport smoke, hot gases, and flames to areas far beyond the initial fire location. They can also supply air to aid combustion in the fire area. This NFPA standard prescribes the minimum requirements for fire protection of air duct systems so as to...

- Restrict the spread of smoke and fire
- Maintain the fire-resistive integrity of building components and elements
- Minimize ignition sources and combustibility of system elements

NFPA 90B: Standard for Installation of Warm Air Heating and Air Conditioning Systems, 1996 Edition

NFPA 90B provides installation requirements for supply ducts, controls, clearances, heating panels, return ducts, air filters, heat pumps, and other components. Applies to one- and two-family dwellings or spaces not exceeding volumes of 25,000 cubic feet.

4. Selected Mechanical Insulation Codes

Products shall conform to NFPA 90A and 90B with special regard to fire hazard classification requirements of NFPA 255, including vapor barriers and adhesives

NFPA 255: Standard Method of Test of Surface Burning Characteristics of Building Materials, 1996 Edition

NFPA 255 provides a procedure for determining the comparative flame spread rate and smoke density of building materials.

5. Selected Pipe Joining Methods

Comply with Mechanical Standards and with the requirements of ANSI B31

6. Selected Pipe Support and Anchor Standards

Pipe support system shall result in pipe stress conforming to the requirements of ANSI B31.3

Pipe support Design shall conform to ASME B31.1

7. Desiccant Dehumidification Commponents Standard

ARI Standard 940-98 - This standard by the Air-Conditioning and Refrigeration Institute (ARI) applies to factory manufactured, thermally regenerated, dynamic desiccant components operating at atmospheric pressure. Only the component containing the desiccant is subject to this standard. Included in the standard are definitions, classifications, testing and rating requirements, minimum data requirements for published ratings, performance requirements, marking and nameplate data intended for use by manufacturers, engineers, installers, contractors and users.

B. CONSTRUCTION SPECIFICATIONS

The following are links to desiccant specifications and application manuals.

1. **Sample Specification - Fort Myer (Engelhard Two-Wheel Desiccant - now Fresh Air Solutions)**

2. **Kathabar Systems Application Manual (Liquid Desiccant)**

C. SAMPLE CONSTRUCTION DRAWINGS

The following are links to desiccant system construction drawings.

1. **Fort Benning Construction Drawings**
2. **Fort Campbell Construction Drawings**
3. **Fort Myer Construction Drawings**
4. **Keesler Air Force Base Construction Drawings**

D. SAMPLE O&M MANUALS

The following are links to desiccant O&M manuals

1. Engelhard Desert Cool^{TM} DC050 (now Fresh Air Solutions)

Section Outline

IV. DoD Desiccant Systems Case Studies

Martin Army Community Hospital
Fort Benning, Georgia

Site Description

Fort Benning is located in Columbus, Georgia approximately 110 miles west of Atlanta near the Georgia/Alabama state line. Fort Benning is known as the "Home of the Infantry". The U.S. Army Infantry School produces the world's finest infantry combat leaders by preparing officers and enlisted soldiers to perform infantry duties required in both peace and war with the emphasis on the art of command and leadership. The development of tactics, techniques and procedures to implement approved doctrine for infantry units at brigade level and below is the mission of the Infantry School. It also participates in the development, review and testing of doctrine and material for infantry units.

The desiccant unit is installed at the Martin Army Hospital (Building 9200). Within the 10 story, 250 bed facility are 11 patient wards, an extensive surgical suite, a labor and delivery suite, and more than 30 ambulatory care clinics. In order to provide these extensive medical services, the hospital employs approximately 750 civilian and 680 military staff members. Daily, the hospital provides inpatient care to approximately 130 patients and averages nearly 1,200 outpatient visits.

System Description

The desiccant unit was retrofit to one of the three air handlers located in the fourth floor mechanical room. The air handler interfaced with the desiccant unit provides air to the operating rooms of the hospital. To accommodate the installation of the desiccant unit, the desiccant unit was installed outdoors on the roof adjacent to the mechanical room. To provide heat for the desiccant regeneration, the desiccant unit was supplied with a steam to air heat exchanger in place of the standard hot water boiler. Steam is supplied from the central boiler plant in the hospital. In addition, the desiccant unit was retrofit with a variable speed drive for the process air and regeneration air fans.

Lessons Learned

The only viable location for the desiccant unit was outside on the roof adjacent to the air handler mechanical room. The roof required a steel mounting structure for the desiccant unit. Careful consideration was given to the roof loading capacity. The variable speed drive (VSD) for the desiccant fans was mounted outside, but is not a weatherproof motor. Because steam is used for regeneration and there is not a boiler inside the desiccant unit, the VSD could have been installed in the boiler compartment to protect it from the ambient conditions.

The desiccant unit was originally interfaced to an existing air handler inside the mechanical room. A new air handler was later installed to replace the one interfaced to the desiccant unit. The contract for replacement of the air handler did not include a task for integrating the desiccant controls back with the building controller. Due to this oversight, the desiccant unit has not operated since the installation of the new air handler.

SUMMARY INFORMATION:

System Manufacturer:	Engelhard Corporation (now Fresh Air Solutions)
System Type:	Two-Wheel Desiccant
Application:	Hospital Operating Room
Start-up Date:	January, 1997
System Description:	4,480 cfm Model # DA 5000 Unit is mounted on rooftop Regeneration by building steam loop Variable speed drive on fans 100% outside air into desiccant
Site Contact:	Mr. Mark Fincher Martin Army Community Hospital Fort Benning, GA
Installation Contractor:	John J. Kirlin, Inc. of Georgia 1669 Litton Drive Stone Mountain, GA 30083 Richard Millard

See Also - Fort Benning Installation Diagrams

Fort Benning Technical Summary Sheet

Desiccant System Description

Desiccant Type Two-Wheel Desiccant System
Manufacturer Engelhard/ICC

Model Number	DC050GFSJOSDBBDBCBNOO
DC050	2500 to 6000 cfm
G	Process Air Motor: 7.5 HP
F	Regeneration Motor: 5.0 HP)
S	Standard Motor Efficeincy
J	Voltage: 460/3/60
O	Steam Regeneration
S	Steam Provided Externally
D	Process Air Bottom Intake Closed
B	Process Air End Intake through Hood
B	Regeneration Front Intake through Hood
D	Regeneration Bottom Intake Closed
B	Regeneration Outlet through Hood
C	Thermostat by Others (Field Supplied)
B	Humidity Controller through Dewpoint Sensor
N	ExternalDisconnect: 60 Amp Non-Fused
O	Post Cooling: None
O	Roof Curb: None

Performance Specification

Process Air Inlet		
Flow Rate	4,400	cfm
Dry Bulb	94.00	^{o}F
Wet Bulb	78.90	^{o}F
Humidity Ratio	125.00	Gr/Lb
Dew Point	73.60	^{o}F
%RH	51.90	%RH
Regeneration Air Inlet		
Flow Rate	4,400	cfm
Dry Bulb	94.0	^{o}F
Wet Bulb	78.9	^{o}F
Humidity Ratio	125.0	Gr/Lb
Dew Point	73.6	^{o}F
%RH	51.9	%RH
Process Supply Air		
Dry Bulb	90	^{o}F
Humidity Ratio	63	Gr/Lb
Latent Capacity		
	169.4	Lbs/Hour
	15.5	Tons
	185,504	BTU/Hr
Total Capacity		
	210,475	BTU/Hr

Dimensions				
Length	144	inches	12	feet
Width	88	inches	7.33	feet
Height	99.5	inches	8.29	feet

B. DoD Desiccant Case Studies - Fort Campbell

Don F. Pratt Memorial Museum
Fort Campbell, Kentucky

Site Description

Fort Campbell is located approximately 90 miles northwest of Nashville, Tennessee on the Tennessee and Kentucky border. Fort Campbell is home of the 101st Airborne Division (Air Assault). The fort opened in 1942 and is named after William B. Campbell, a Tennessee statesman and Brigadier General of the United States Volunteers during the Civil War. The 101st Airborne Screaming Eagles stationed at Fort Campbell are members of the only air assault division in the world. Fort Campbell supports the 3rd largest military population in the Army and the 7th largest in the Department of Defense.

The desiccant system is located in the museum building which originally housed a movie theater, a classroom and an auditorium for basic training during the Vietnam war. In 1972, the facility was converted to a museum. The conditioned space of this stand-alone metal building is 12,956 Ft^2 . The facility consists of a reception area, a manager's office, a gift shop, rest rooms, storage rooms, workshop and the museum exhibit area. The building construction consists of metal walls with no insulation, a suspended acoustic tile ceiling with R-11 bat insulation and a concrete slab floor. The air space above the insulation in the ceiling is vented to the outdoor space. The building appeared to have a substantial amount of air leakage. The museum operates between the hours of 9:30 am and 4:30 pm seven days a week. The museum is used to educate new recruits and for viewing by the public. A total of 80,000 visitors pass through the museum annually with as many as 150 visitors at any one time.

System Description

To preserve the museum artifacts, the air in the museum must be maintained between 70 °F to 75 °F and 50% to 52% relative humidity. Wide swings in the humidity level cause premature degradation to many of the artifacts. Artifacts which are impacted the most by humidity are textiles, art work and historical documents. The estimated value of the museum artifacts is $8,000,000 of which approximately 80% are at risk to high humidity damage.

The desiccant unit was retrofit to the central air handling unit for the museum. The air handling unit contains a hydronic cooling coil interfaced to an air cooled chiller. The desiccant unit is installed outdoors adjacent to the mechanical room and the electric chiller at the rear of the building. Outside air is dehumidified and mixed with the building return air prior to passing over the cooling coil.

Lessons Learned

Once installation of the desiccant unit was complete, the museum mechanical system was capable of providing both temperature and moisture control. Moisture control is achieved through desiccant dehumidification while humidification is achieved through humidifier units located throughout the museum. Because the desiccant unit delivers warmer air than normal, the chilled water leaving temperature set point was lowered. However, when the chiller operated during periods when the desiccant unit was not operating, the museum temperature dropped below specification. This resulted in wide temperature fluctuations within the museum, which is detrimental to preserving artifacts. The problem was corrected by installing a feedback mechanism from the desiccant unit back to the mechanical system controller.

SUMMARY INFORMATION

System Manufacturer:	Engelhard Corporation (now Fresh Air Solutions)
System Type:	Two-Wheel Desiccant
Application:	Museum
Start-up Date:	August, 1997
System Description:	5,000 cfm Model # DC 050 Unit located on cement pad at rear of building Regeneration by natural gas boiler supplied with unit 100% outside air
Site Contact:	Mr. Arlin Wright Fort Campbell Fort Campbell, KY

Prime Contractor: Stanley Design-Build, Inc.
225 Iowa Avenue
Muscatine, IA 52761
Mr. Lee Miller

Installation Contractor: Jim Freeman Co.
1514 Vista Lane
Clarksville, TN 37043
Mr. Jim Freeman

See Also - Fort Campbell Installation Diagram

Fort Campbell Technical Summary Sheet

Desiccant System Description

Desiccant Type	Two-Wheel Desiccant System
Manufacturer	Engelhard/ICC

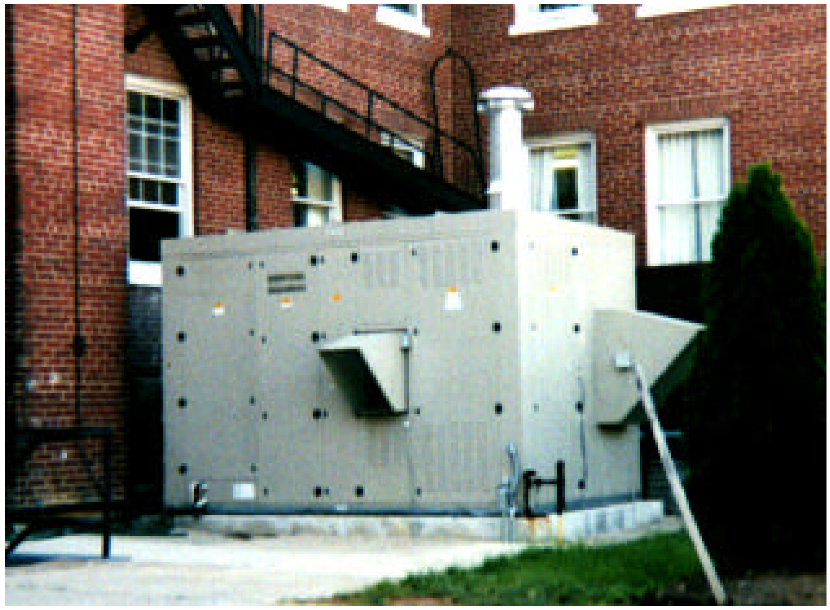
Model Number	**DC050GFFSFADNODBABDBCCOOO**
DC	System
050	4,001 to 5,000 cfm
F	Process Air Motor: 5.0 HP
F	Regeneration Motor: 5.0 HP
S	Standard Motor Efficeincy
F	Voltage: 208/3/60
A	Control Panel: Standard
D	Regeneration Input: 350 MBTU
N	Fuel: Natural Gas
O	Process Heating: Not Required
D	Process Air Bottom Intake Closed
B	Process Air End Intake through Hood
A	Process Outlet: Ducted through Front
B	Regeneration Front Intake through Hood
D	Regeneration Bottom Intake Closed
B	Regeneration Outlet through Hood
C	Thermostat by Others (Field Supplied)
C	Thermostat by Others (Field Supplied)
O	ExternalDisconnect: None
O	Post Cooling: None
O	Roof Curb: None

Performance Specification

Process Air Inlet		
Flow Rate	5,000	cfm
Dry Bulb	94	^{o}F
Wet Bulb	77.1	^{o}F
Humidity Ratio	113	Gr/Lb
Dew Point	70.7	^{o}F
%RH	47	%RH
Regeneration Air Inlet		
Flow Rate	5,000	cfm
Dry Bulb	94	^{o}F
Wet Bulb	77	^{o}F
Humidity Ratio	113	Gr/Lb
Dew Point	71	^{o}F
%RH	47	%RH
Process Supply Air		
Dry Bulb	90	^{o}F
Humidity Ratio	52	Gr/Lb
Latent Capacity		
	189	Lbs/Hour
	17.3	Tons
	207,600	BTU/Hr

Physical Characteristics

Dimensions				
Length	144	inches	12	feet
Width	88	inches	7.33	feet
Height	99.5	inches	8.29	feet
Weight	5,800	Lbs		



Building No. 251 - Barracks and Munitions Storage
Fort Myer, Virginia

Site Description

Fort Myer is located in Arlington, Virginia with its roots tracing back to the Civil War. Fort Myer sits above Arlington National Cemetery across the Potomac River from Washington D.C. Fort Myer is home to the 3rd U.S Infantry also referred to as "The Old Guard". This elite group serves as the Army's official ceremonial unit and the security force for Washington D.C. Units of The Old Guard include the U.S. Army Drill Team, the Old Guard Fife and Drum Corps and the sentinels at the Tomb of the Unknown Soldiers in Arlington National Cemetery. Most of the buildings at Fort Myer were built between 1895 and 1908.

System Description

The desiccant unit is installed on Building 251. Building 251 is used as a barracks for men who are training for The Old Guard. The two story brick and block building is registered as a historical building. For the installation of the desiccant unit, approval was required by the National Capitol Planning Committee. The facility is occupied 365 days per year. The basement of the building is used for munitions and arms storage. Historically, high humidity had resulted in an uncomfortable environment as well as created problems associated with the munitions and corrosion of arms.

The desiccant unit was retrofit into the existing central air handling unit. The air handling unit contains a hydronic cooling coil which receives chilled water from a remote chiller. The air from the air handler is distributed to the hallways on all floors and the munitions storage area. The air handler uses 100% outside air. All rooms have individual packaged terminal air conditioning units.

Lessons Learned

The desiccant unit was located outside at ground level and adjacent to the air handler mechanical room. One problem with this location is that steam from the laundry's pressing machines gets discharged at the back of the desiccant unit where the regeneration hood inlet is located. This results in the regeneration inlet air being higher in temperature and humidity ratio than the normal ambient outdoor air. Ultimately, this leads to degradation in desiccant unit performance.

SUMMARY INFORMATION:

System Manufacturer:	Engelhard Corporation (now Fresh Air Solutions)
System Type:	Two-Wheel Desiccant
Application:	Barracks and Munitions Storage
Start-up Date:	September, 1997
System Description:	4,640 cfm Model # DC 050 Unit located on cement pad at rear of building Regeneration by natural gas boiler supplied with unit 100% outside air
Site Contact:	Mr. Huey Vample Fort Myer Fort Myer, VA
Prime Contractor:	Stanley Design-Build, Inc. 225 Iowa Avenue Muscatine, IA 52761 Mr. Lee Miller
Installation Contractor:	Bush & Sons, Inc. 1408 Richie Marlboro Rd. Capital Heights, MD 20743 Mr. Fred Bock

See Also - Fort Myer Installation Diagram

Fort Myer Technical Summary Sheet

Desiccant System Description

Desiccant Type	Two-Wheel Desiccant System
Manufacturer	Engelhard/ICC

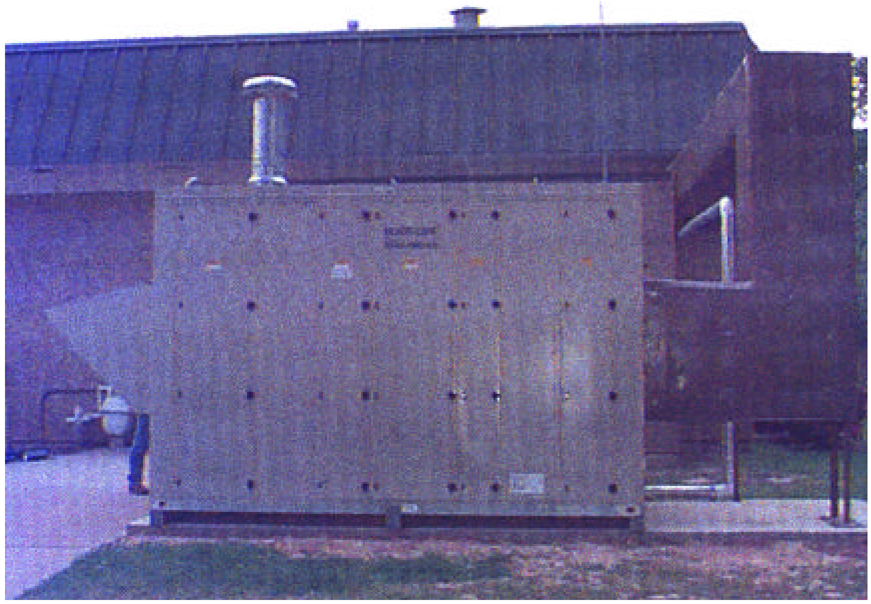
Model Number	DC050GFSFDNDDABDBCCFOO
DC050	2500 to 6000 cfm
G	Process Air Motor: 7.5 HP
F	Regeneration Motor: 5.0 HP)
S	Standard Motor Efficeincy
F	Voltage: 208/3/60
D	Regeneration: 350 MBH Input
N	Boiler Fuel: Natural Gas
D	Process Air Bottom Intake Closed
D	Process Air End Intake through Motorized Two Position Damper
A	Process Outlet: Ducted through Front
B	Regeneration Front Intake through Hood
D	Regeneration Bottom Intake Closed
B	Regeneration Outlet through Hood
C	Thermostat by Others (Field Supplied)
C	Humidity Controller by Others (Field Supplied)
N	External Disconnect: 100 Amp Fused
O	Post Cooling: None
O	Roof Curb: None

Performance Specification

Process Air Inlet		
Flow Rate	4,640	cfm
Dry Bulb	95.0	^{o}F
Wet Bulb	76.0	^{o}F
Humidity Ratio	104.6	Gr/Lb
%RH	42.3	%RH
Regeneration Air Inlet		
Flow Rate	4,640	cfm
Dry Bulb	95.0	^{o}F
Wet Bulb	76.0	^{o}F
Humidity Ratio	104.6	Gr/Lb
%RH	42.3	%RH
Process Supply Air		
Dry Bulb	88	^{o}F
Humidity Ratio	49	Gr/Lb
Latent Capacity		
	160.3	Lbs/Hour
	14.6	Tons
	175,507	BTU/Hr
Total Capacity		
	214,917	BTU/Hr

Physical Characteristics

Dimensions				
Length	144	inches	12	feet
Width	88	inches	7.33	feet
Height	99.5	inches	8.29	feet
Weight	5,800	Lbs		



Gaude Bowling Lanes
Keesler Air Force Base, Mississippi

Site Description

Keesler Air Force Base is located in Biloxi, Mississippi on the Gulf of Mexico. Keesler AFB is home to the 81st Training Wing, one of the Air Education and Training Command's largest technical training wings. During the 1970s, Keesler remained the largest training center in the Air Force and became the nation's main supplier of electronics technicians. Two additional areas of training received special attention in the 1980s-- airborne warning and control systems and ground launched cruise missile. The air traffic control program also received its share of attention, especially during the 1981 professional air traffic controllers' strike. By presidential order, military controllers, trained at Keesler, stepped in and kept the nation's airways flowing. In 1992, Keesler began training all of the DOD's weather forecasters and observers. Flight training and instruction of pilots for the C-12 and C21 aircraft is conducted at Keesler.

System Description

The desiccant unit is installed at Gaude Bowling Lanes (Building 1203). The single story building is constructed of block and brick. The bowling alley consists of 24 lanes, a pro shop and a snack bar. The bowling alley's hours of operation are as follows:

Monday - Thursday:	0900 - 2200
Friday:	0900 - 2300
Saturday:	1400 - 2000
Sunday:	Closed

The desiccant unit was retrofit to the central air handling unit for the bowling alley. The air handling unit is a multi-zone type with a hydronic cooling coil interfaced to an air cooled chiller. The desiccant unit is installed outdoors adjacent to the mechanical room and the electric chiller.

Lessons Learned

The base building code required that the look of the desiccant installation be consistent with the general architecture of the base. In order to accommodate this requirement, the gas piping, electrical conduit and ducting was painted brown.

During the initial months of operation, the desiccant unit boiler was not producing water hot enough to achieve adequate regeneration. This resulted in incomplete regeneration of the desiccant material and poor performance of the desiccant unit. The manufacturer resolved the problem.

SUMMARY INFORMATION

System Manufacturer:	Engelhard Corporation (now Fresh Air Solutions)
System Type:	Two-Wheel Desiccant
Application:	Bowling Alley
Start-up Date:	November, 1997
System Description:	4,400 cfm Model # DC050 Unit located on cement pad at rear of building Heated by gas boiler inside unit
Site Contact:	Mr. Eugene Baker 81 CES/CECC Keesler AFB, MS 39534
Prime Contractor:	John J. Kirlin, Inc. of Georgia 1669 Litton Drive Stone Mountain, GA 30083 Mr. Marvin F. Collier
Installation Contractor:	MCC Mechanical 412 Highway 90, Suite 1 Bay St. Louis, MS 39520 Mr. Tom Canale

See Also - Keesler AFB Installation Diagram

Keesler Air Force Base Technical Summary Sheet

Desiccant System Description

Desiccant Type	Two-Wheel Desiccant System
Manufacturer	Engelhard/ICC

Model Number	**DC050GFSFDNDDBDBCBNOO**
DC050	2500 to 6000 cfm
G	Process Air Motor: 7.5 HP
F	Regeneration Motor: 5.0 HP)
S	Standard Motor Efficeincy
F	Voltage: 208/3/60
D	Regeneration Input: 350 MBTU
N	Fuel: Natural Gas
D	Process Air Bottom Intake Closed
D	Process Air End Intake through Motorized Two Position Damper
B	Regeneration Front Intake through Hood
D	Regeneration Bottom Intake Closed
B	Regeneration Outlet through Hood
C	Thermostat by Others (Field Supplied)
B	Humidity Controller through Dewpoint Sensor
N	ExternalDisconnect: 60 Amp Non-Fused
O	Post Cooling: None
O	Roof Curb: None

Performance Specification

Process Air Inlet		
Flow Rate	4,400	cfm
Dry Bulb	94	^oF
Wet Bulb	78.9	^oF
Humidity Ratio	125	Gr/Lb
Dew Point	73.6	^oF
%RH	51.9	%RH
Regeneration Air Inlet		
Flow Rate	4,400	cfm
Dry Bulb	94	^oF
Wet Bulb	78.9	^oF
Humidity Ratio	125	Gr/Lb
Dew Point	73.6	^oF
%RH	51.9	%RH
Process Supply Air		
Dry Bulb	90	^oF
Humidity Ratio	63	Gr/Lb
Latent Capacity		
	169.4	Lbs/Hour
	15.5	Tons
	185,504	BTU/Hr
Total Capacity		
	210,475	BTU/Hr

Physical Characteristics

Dimensions				
Length	144	inches	12	feet
Width	88	inches	7.33	feet
Height	99.5	inches	8.29	feet
Weight	5,800	Lbs		

CERL Desiccant Monitoring Program
Fort Campbell (W4)

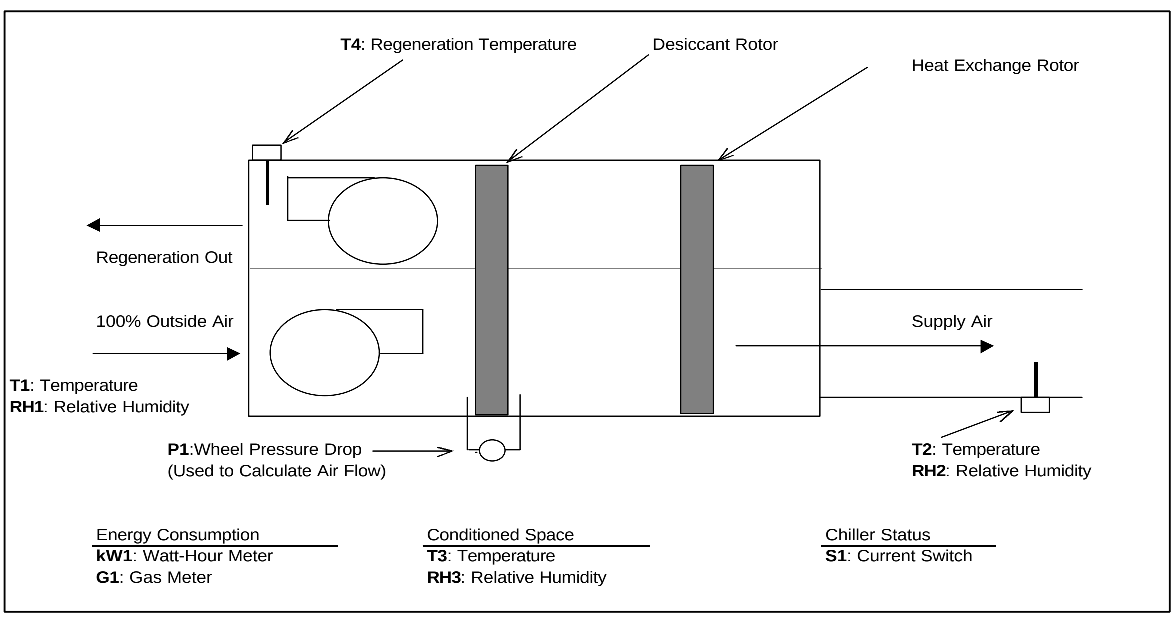
Data Provided by:
Science Applications International Corporation
Advanced Energy Systems Division

Reporting Period: October 1, 1998 through October 31, 1998

Data Acquisition System Overview

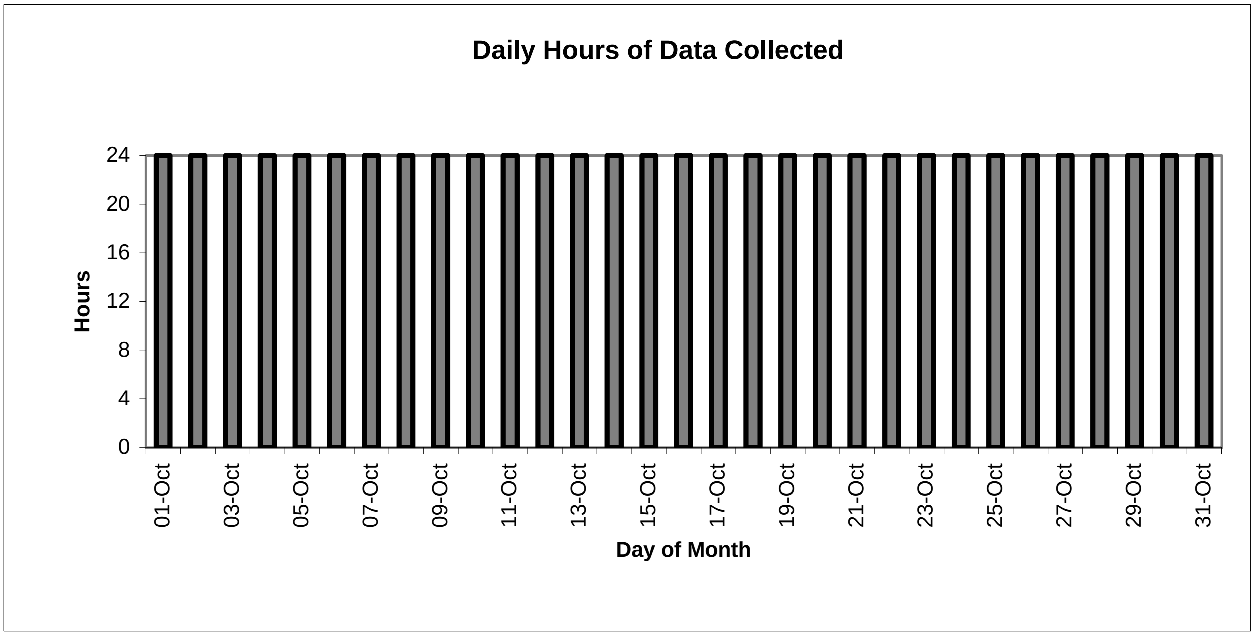
Parameters Collected from Data Logger

Order	Description
1	Array Identifier
2	Year
3	Julian Day
4	Hour-Minute
5	Seconds
6	Record Seconds
7	T1: Oudoor Temperature
8	T2: Desiccant Leaving Temperature
9	T3: Indoor Air Temperature
10	T4: Regeneration Leaving Temperature
11	P1: Heater Wheel Pressure Drop
12	RH1: Outdoor Relative Humidity
13	RH2: Desiccant Leaving Relative Humidity
14	RH3: Indoor Air Relative Humidity
15	kW1: Desiccant Electric Demand
16	Fuel: Desiccant Gas Consumption (Ft^3/scan)
17	Airflow: Desiccant Air Flow
18	DesSecs: Desiccant Operating Seconds
19	ChillerSecs: Chiller Operating Seconds



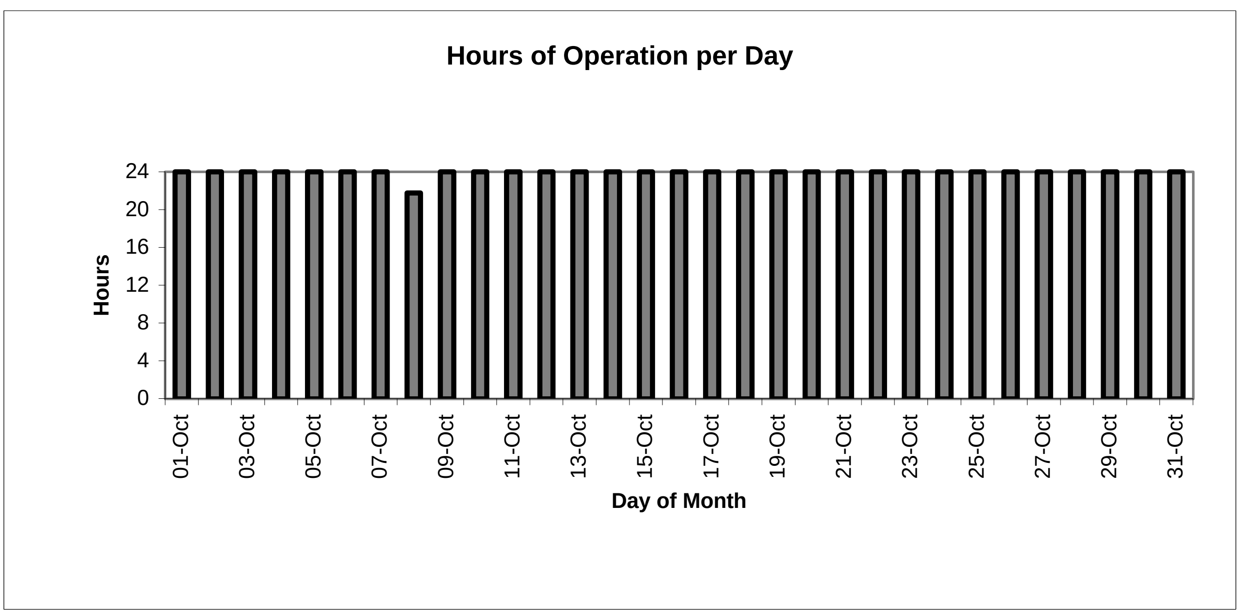
Data Information

Hours in the Reporting Period	744.0	Hours
Hours of Data Collected	744.0	Hours
Percent of Data Collected	100.0	%



Desiccant Operation

Hours of Operation	741.8	Hours
Hours of Off Time	2.2	Hours



Desiccant Operation (continued)

Inlet Air Conditions

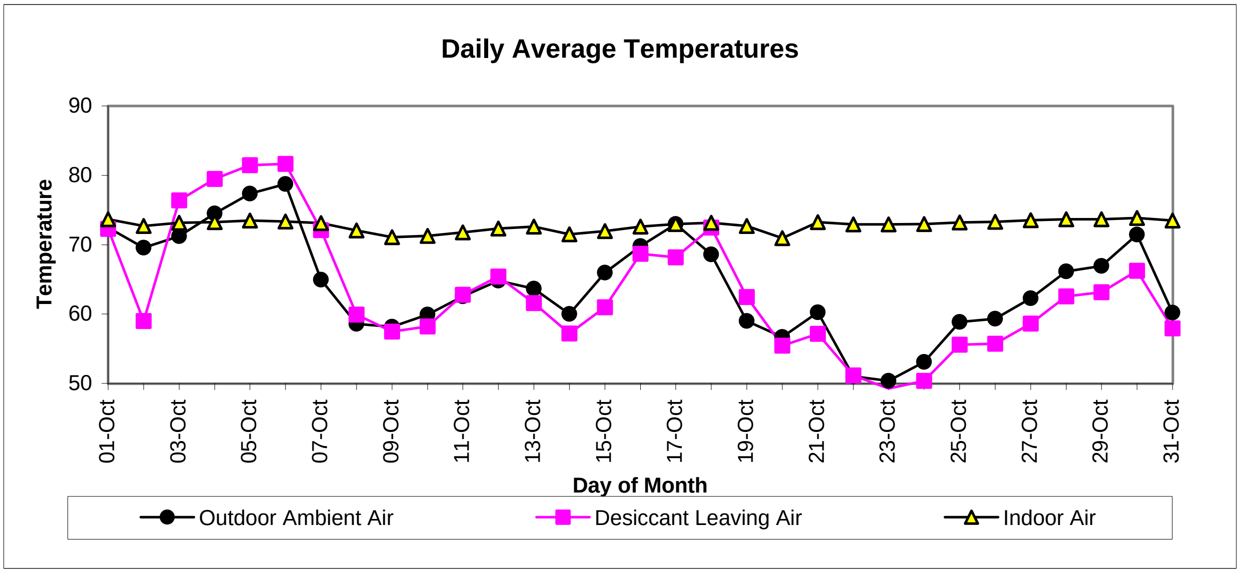
Average Inlet Air Temperature	64.2	^oF
Average Inlet Air Relative Humidity	60.3	%
Average Inlet Air Humidity Ratio	53.7	grains

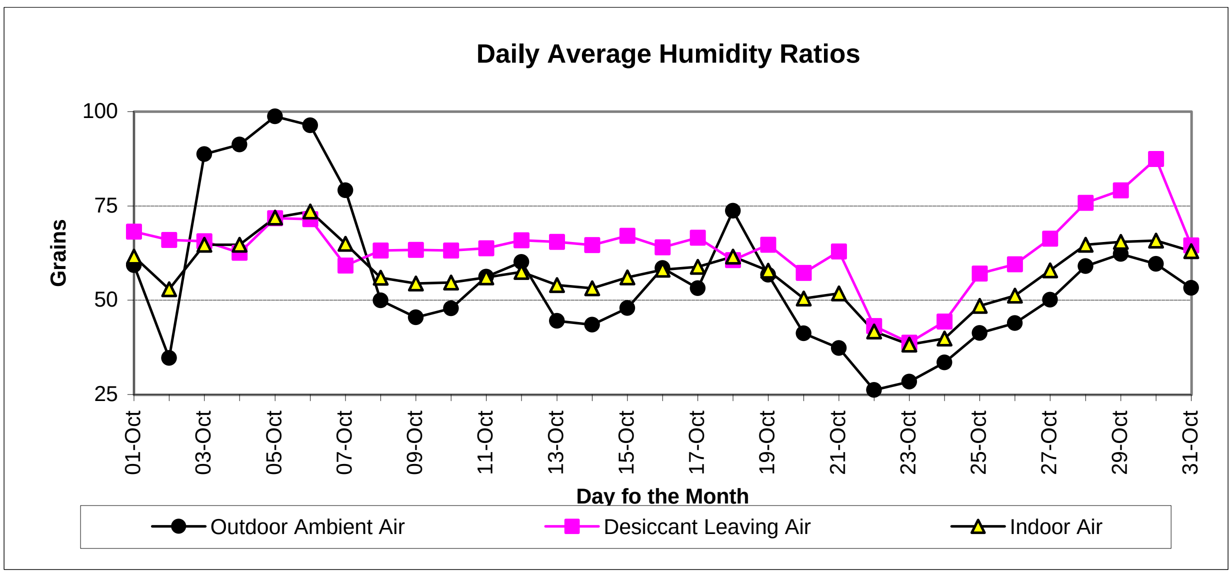
Supply Air Conditions

Average Supply Air Temperature	63.3	^oF
Average Supply Air Relative Humidity	75.3	%
Average Supply Air Humidity Ratio	65.1	grains

Indoor Air Conditions

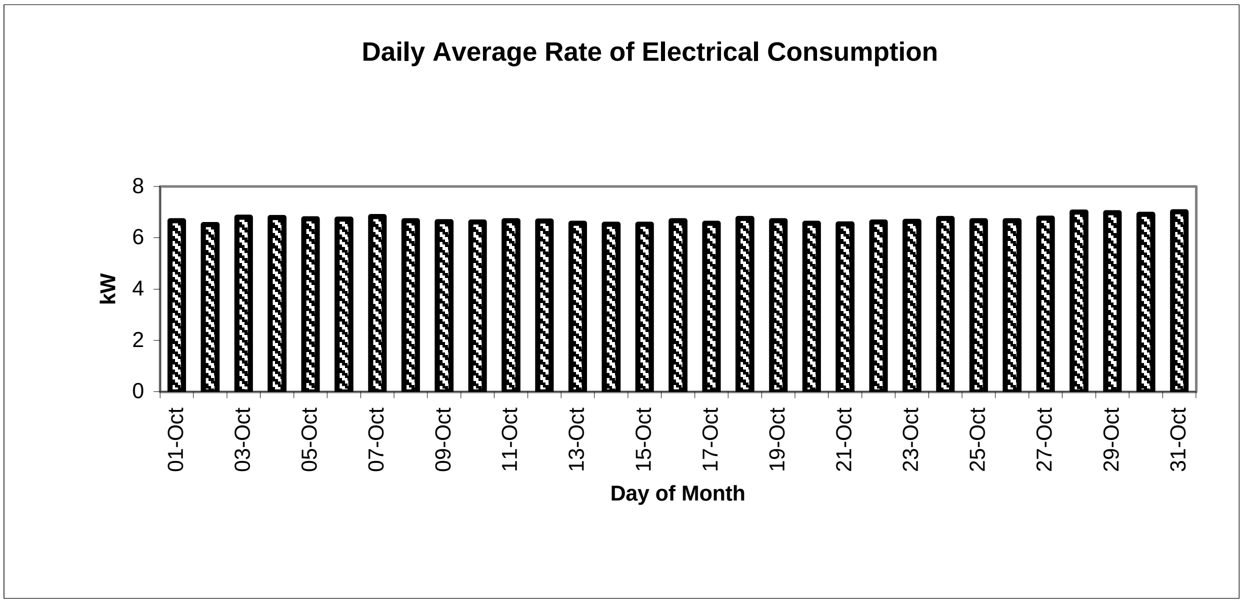
Average Supply Air Temperature	72.8	^oF
Average Supply Air Relative Humidity	47.6	%
Average Supply Air Humidity Ratio	57.1	grains

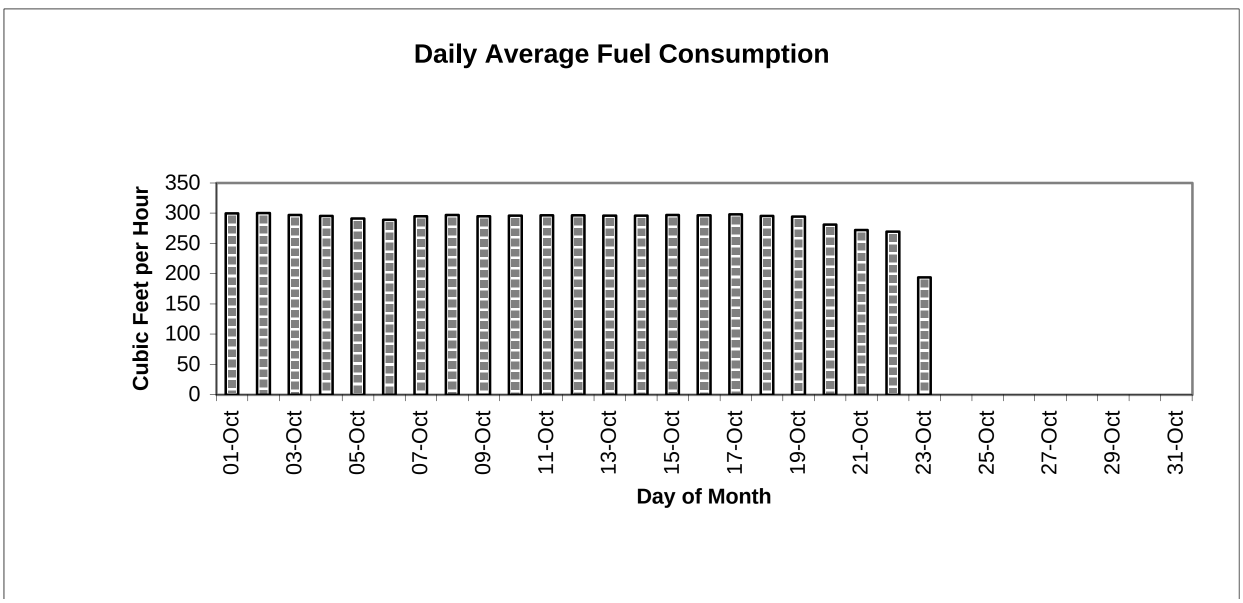




Desiccant Operation (continued)

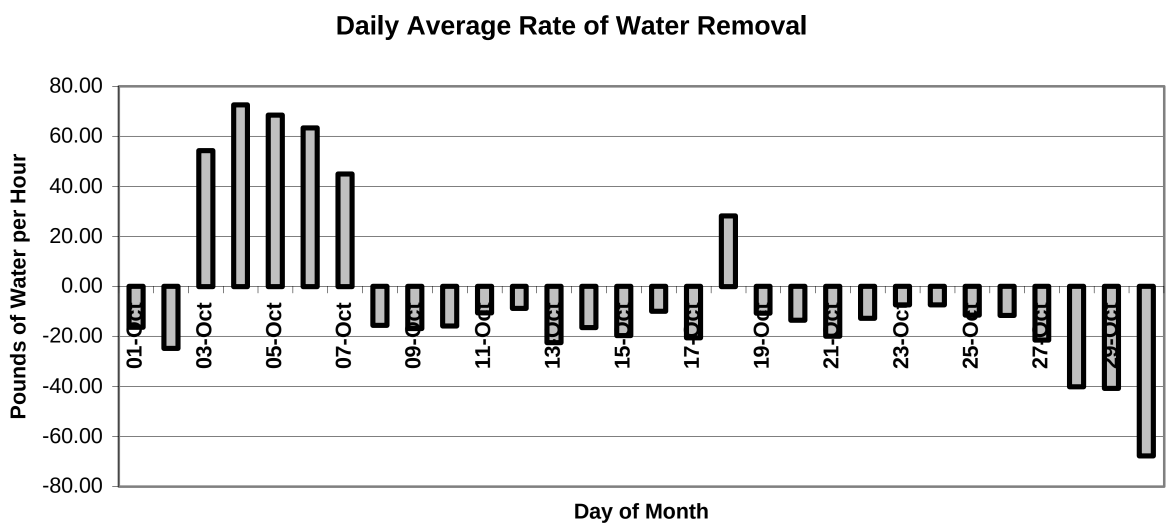
Average Heater Leaving Temperature	100.9	^oF
Average Supply Air Flow	2,334.4	cfm
Total Electricity Consumed	4,969.0	kWh
Average Rate of Electricity Consumed	6.7	kW
Total Fuel Consumed	158,644.0	Ft^3
Average Rate of Fuel Consumption	213.9	Ft^3/Hr

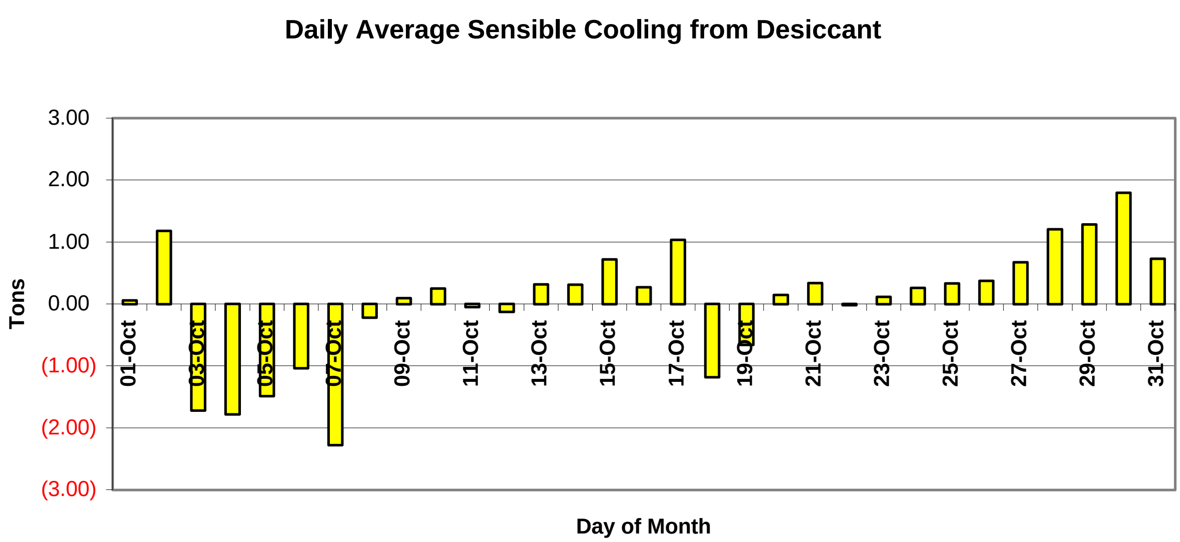




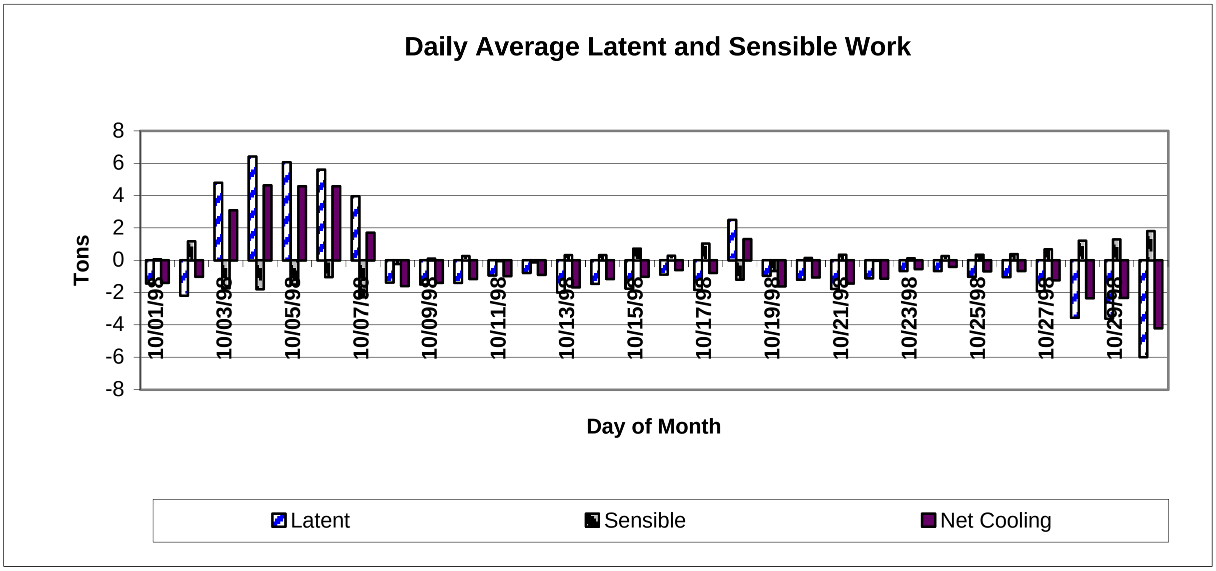
Desiccant Operation (continued)

Total Pounds of Water Removed	-3,699.5	lbs.
Average Rate of Water Removal	-5.0	lb/Hr
Total Latent Cooling Provided	-327.1	Ton-Hours
Average Latent Cooling Rate	-0.4	Tons
Total Sensible Cooling	22.5	Ton-Hours
Average Rate of Sensible Cooling	0.0	Tons
Net Cooling Provided	-304.6	Ton Hours
Average Net Cooling Provided	-0.4	Tons



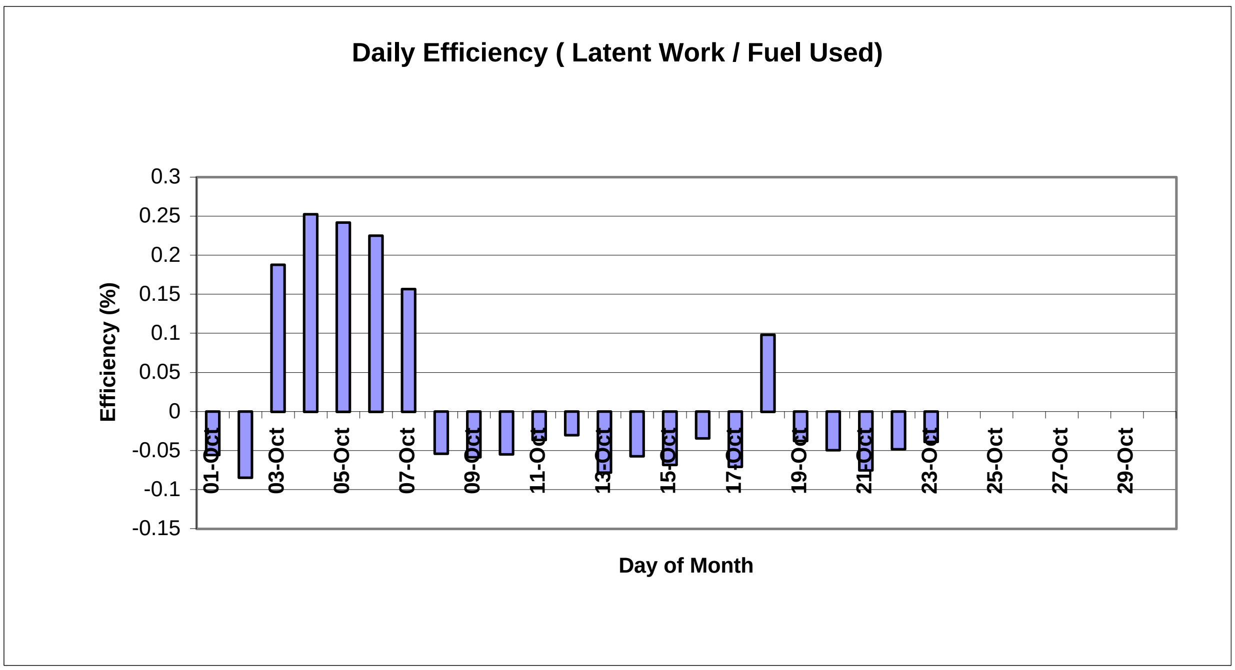


Desiccant Operation (continued)



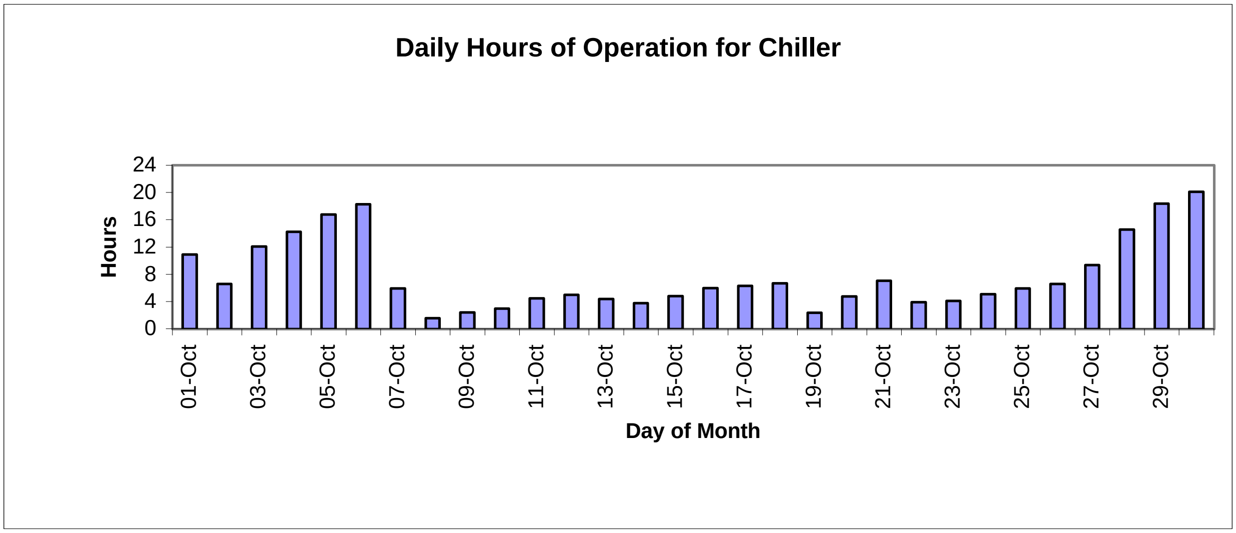
Efficiencies

Latent Work/ Fuel	-2.4	%
Latent Work / (Fuel + Electric)	-2.2	%



Other Cooling Equipment

Chiller Hours of Operation	245.8	Hours



CERL Desiccant Monitoring Program
Fort Myer (W3)

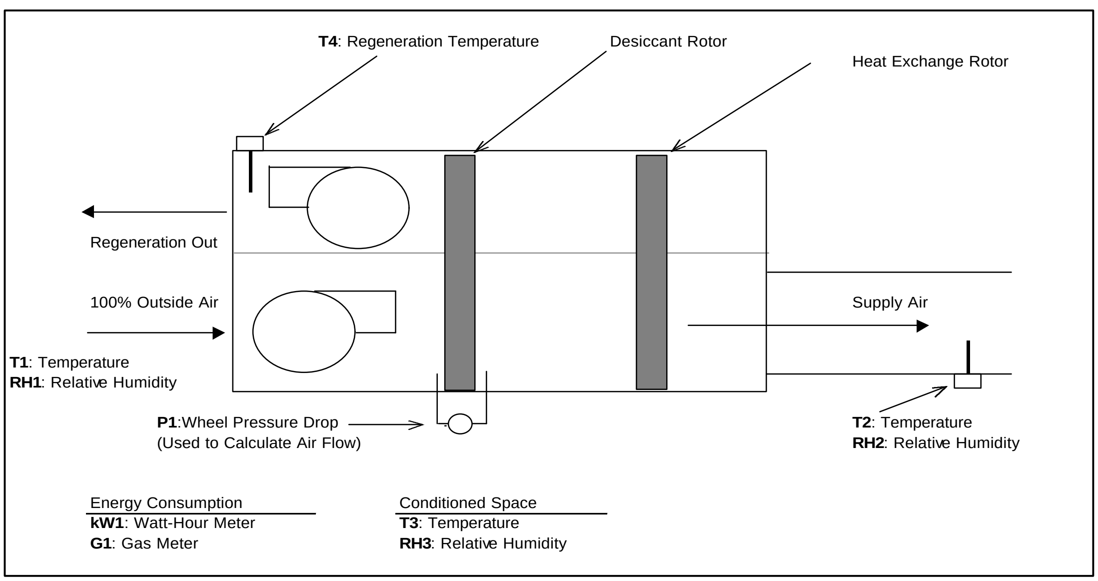
Data Provided by:
Science Applications International Corporation
Advanced Energy Systems Division

Reporting Period: September 1, 1998 through September 30, 1998

Data Acquisition System Overview

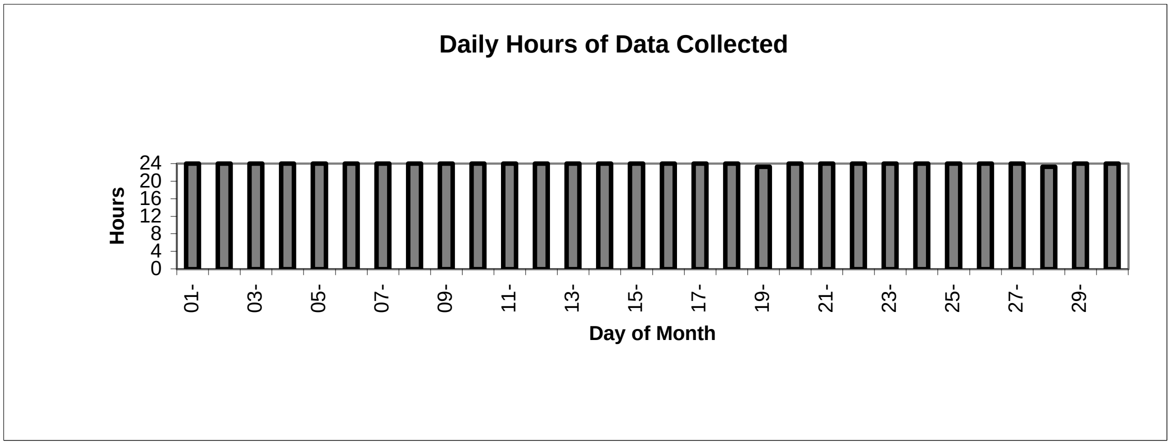
Parameters Collected from Data Logger

Order	Description
1	Array Identifier
2	Year
3	Julian Day
4	Hour-Minute
5	Seconds
6	Record Seconds
7	T1: Oudoor Temperature
8	T2: Desiccant Leaving Temperature
9	T3: Indoor Air Temperature
10	T4: Regeneration Leaving Temperature
11	P1: Heater Wheel Pressure Drop
12	RH1: Outdoor Relative Humidity
13	RH2: Desiccant Leaving Relative Humidity
14	RH3: Indoor Air Relative Humidity
15	kW1: Desiccant Electric Demand
16	Fuel: Desiccant Gas Consumption (Ft^3/scan)
17	Airflow: Desiccant Air Flow
18	DesSecs: Desiccant Operating Seconds



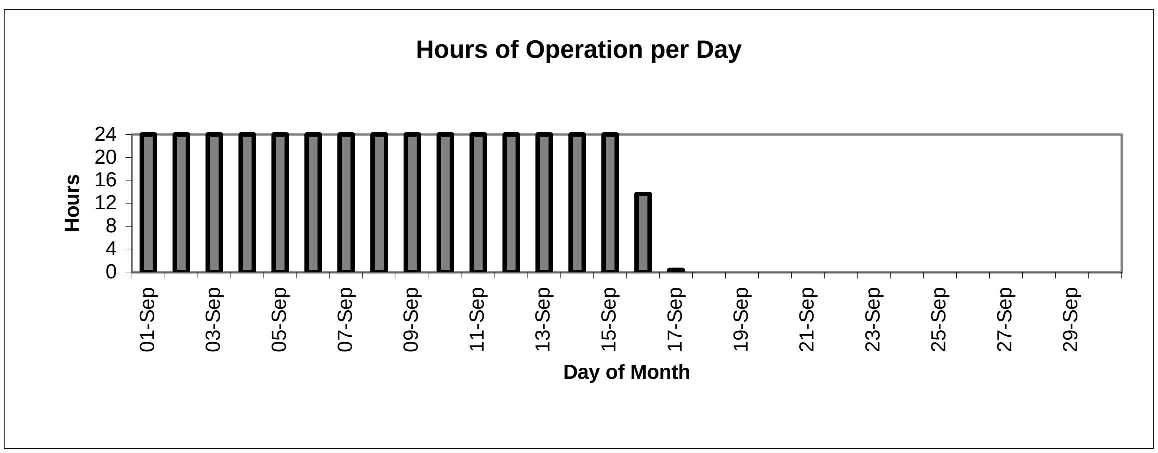
Data Information

Hours in the Reporting Period	720.0	Hours
Hours of Data Collected	718.5	Hours
Percent of Data Collected	99.8	%



Desiccant Operation

Hours of Operation	374.0	Hours
Hours of Off Time	344.5	Hours



Desiccant Operation (continued)

Inlet Air Conditions

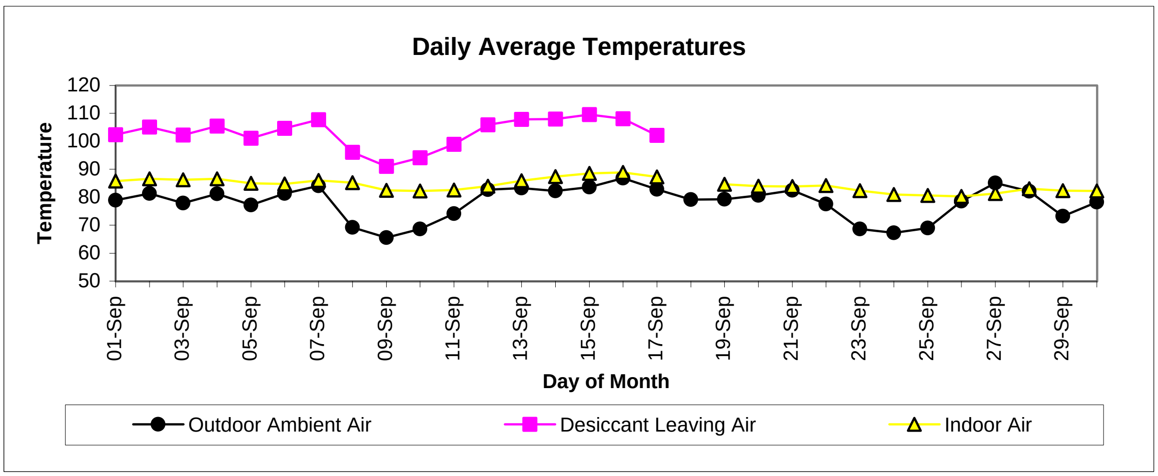
Average Temperature	78.5	^oF
Average Relative Humidity	46.1	%
Average Humidity Ratio	66.9	grains

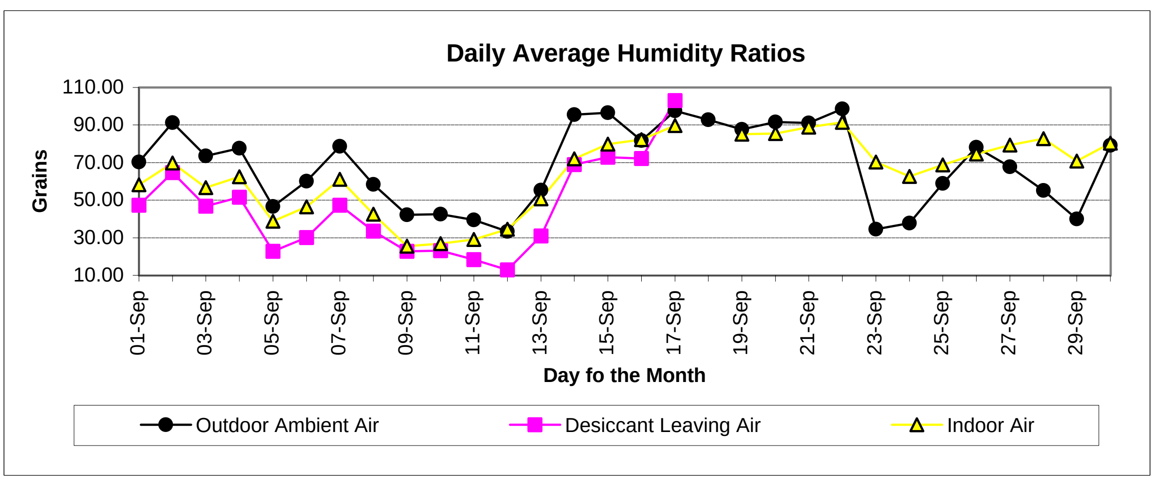
Supply Air Conditions

Average Temperature	103.0	^oF
Average Relative Humidity	14.1	%
Average Humidity Ratio	43.9	grains

Indoor Air Conditions

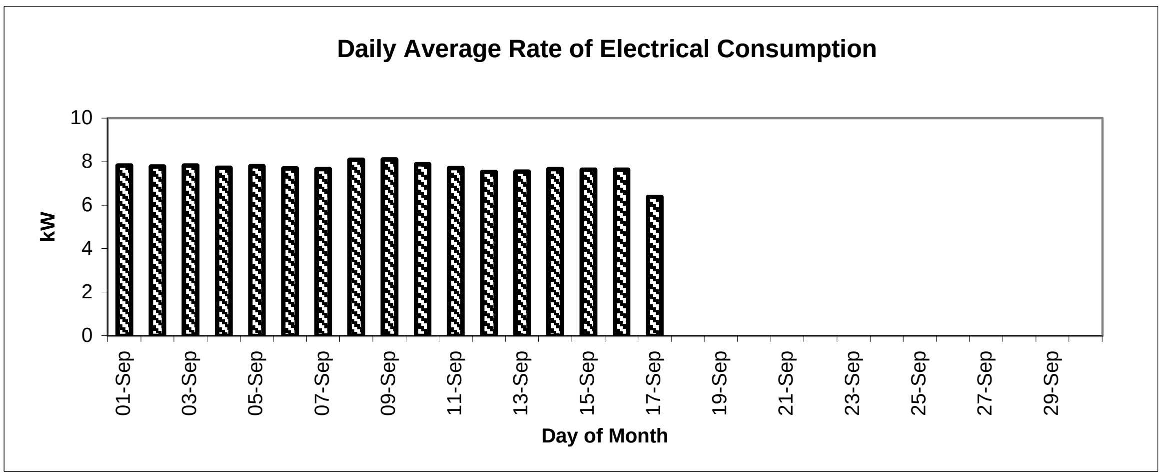
Average Temperature	84.3	^oF
Average Relative Humidity	36.6	%
Average Humidity Ratio	64.3	grains

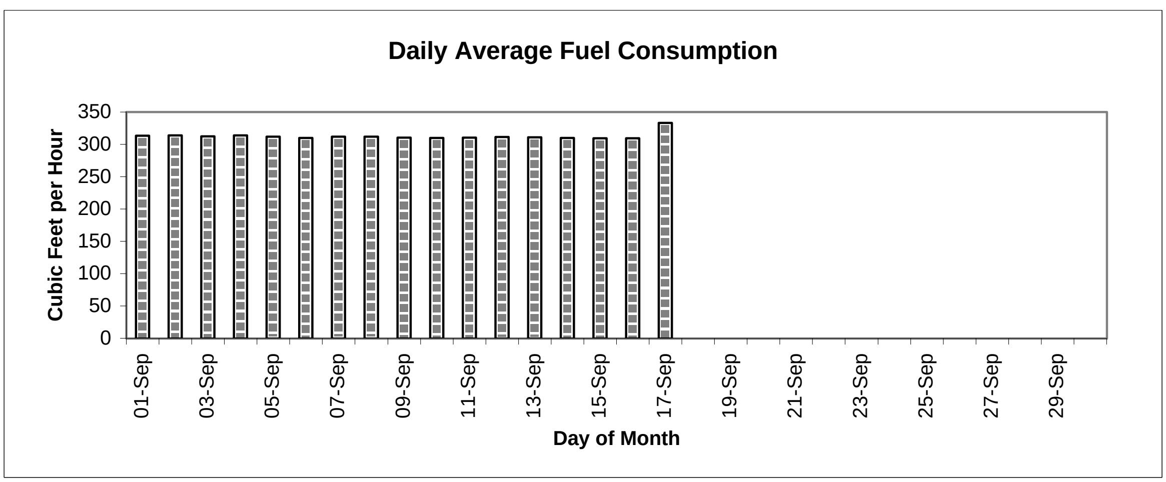




Desiccant Operation

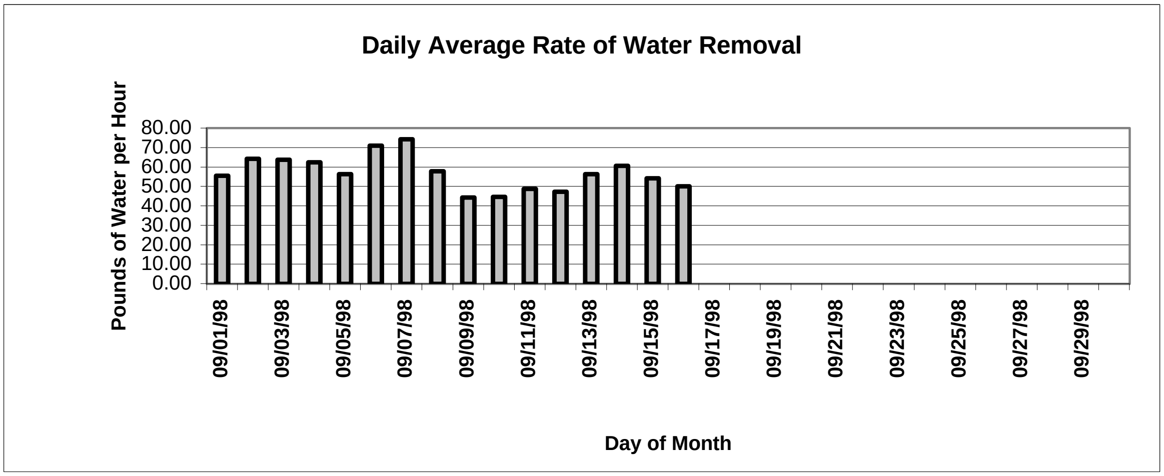
Average Heater Leaving Temperature	118.7	^oF
Average Supply Air Flow	3,554.0	cfm
Total Electricity Consumed	2,869.0	kWh
Average Rate of Electricity Consumed	7.7	kW
Total Fuel Consumed	116,475.0	Ft^3
Average Rate of Fuel Consumption	311.4	Ft^3/Hr

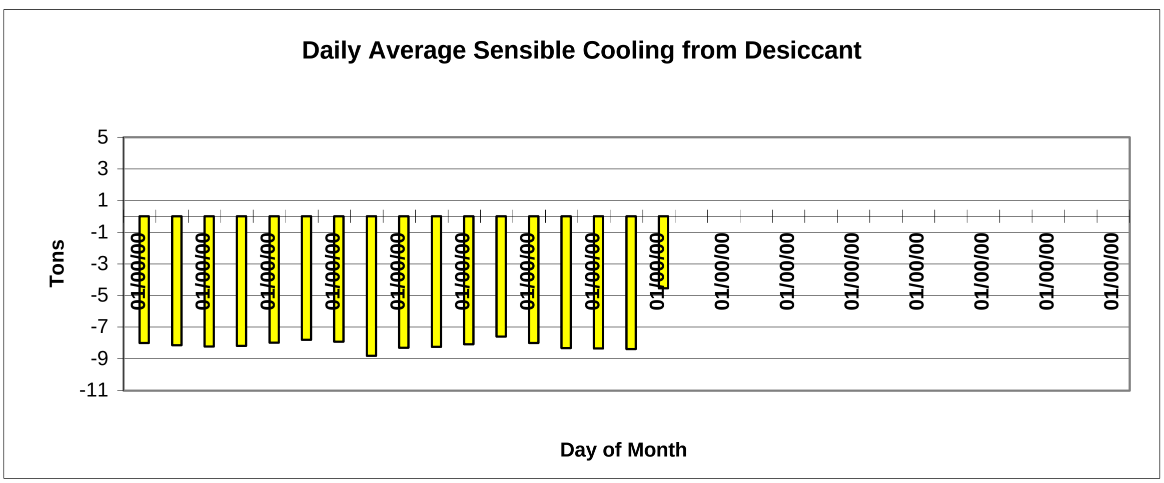




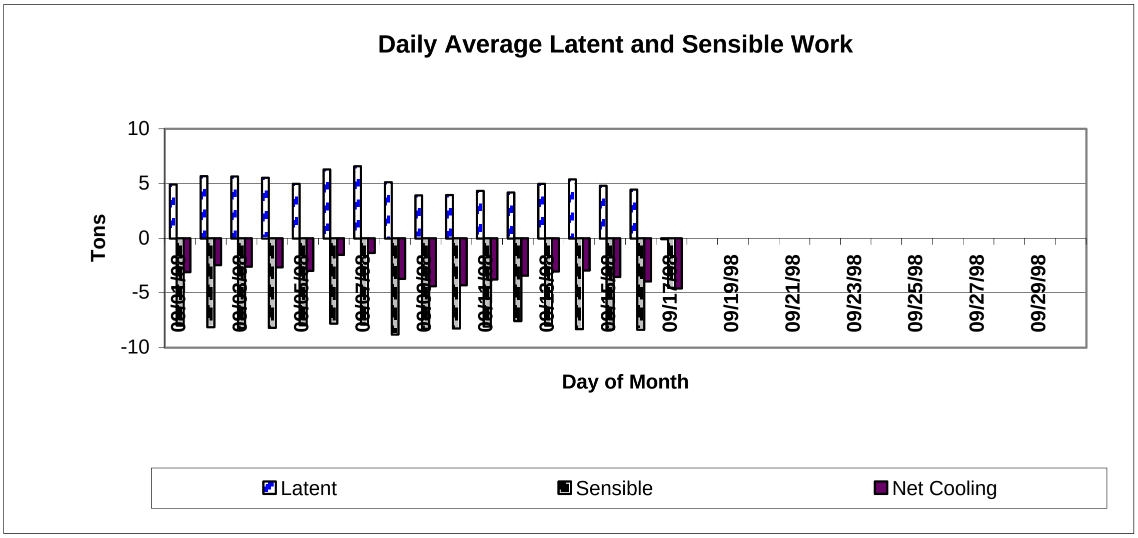
Desiccant Operation (continued)

Total Pounds of Water Removed	20,029.7	lbs.
Average Rate of Water Removal	53.6	lb/Hr
Total Latent Cooling Provided	1,771.0	Ton-Hours
Average Latent Cooling Rate	4.7	Tons
Total Sensible Cooling	-2,962.7	Ton-Hours
Average Rate of Sensible Cooling	-7.9	Tons
Net Cooling Provided	-1,191.7	Ton Hours
Average Net Cooling Provided	-3.2	Tons



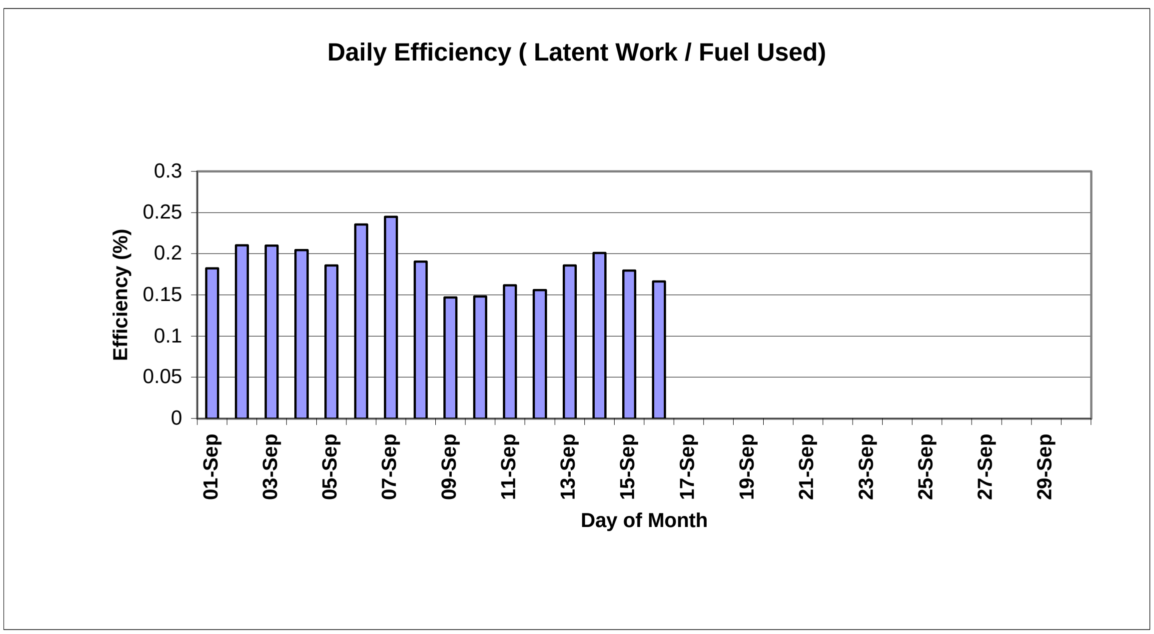


Desiccant Operation (continued)



Efficiencies

Latent Work/ Fuel	17.7	%
Latent Work / (Fuel + Electric)	16.4	%



CERL Desiccant Monitoring Program
Keesler Air Force Base (W1)

Data Provided by:
Science Applications International Corporation
Advanced Energy Systems Division

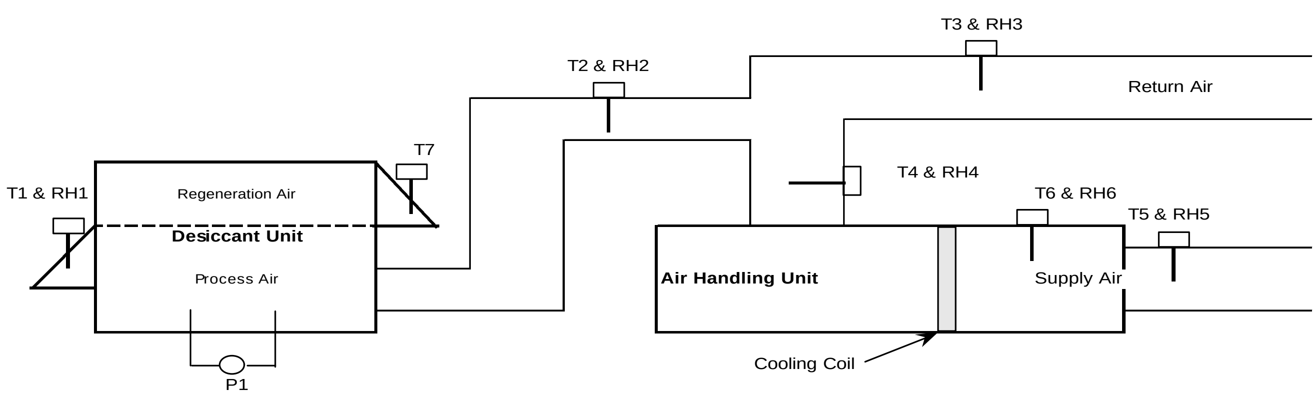
Reporting Period: September 1, 1998 through September 30, 1998

Data Acquisition System Overview

Parameters Collected from Data Logger

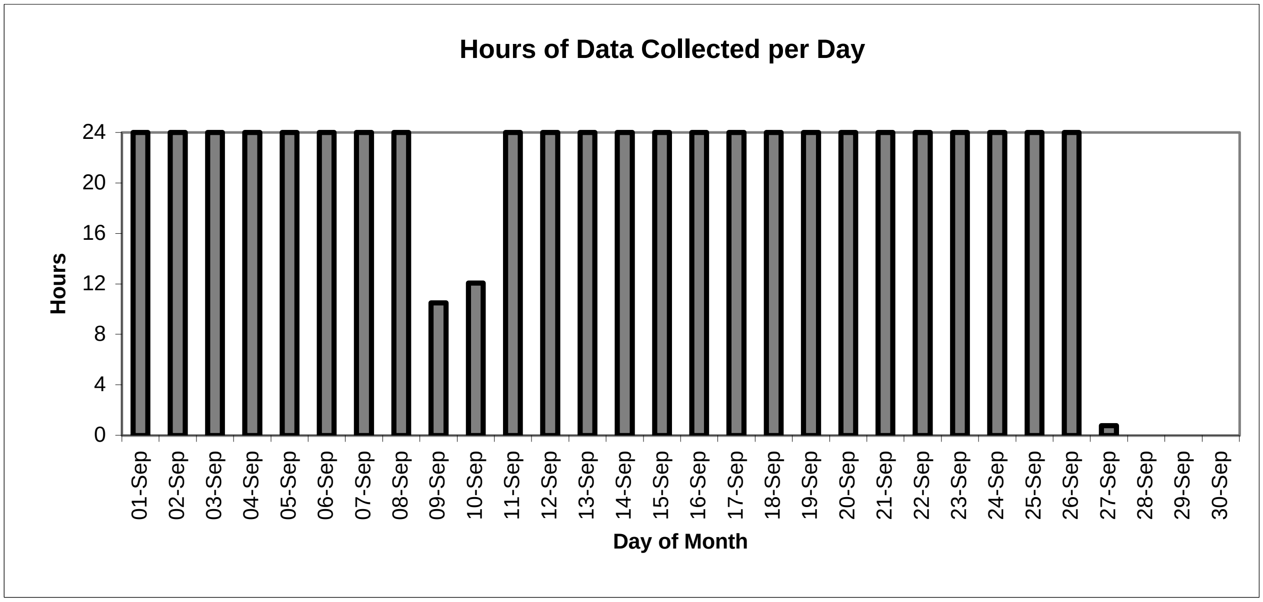
Order	Description
1	Array Identifier
2	Year
3	Julian Day
4	Hour-Minute
5	Seconds
6	Record Seconds
7	DesSecs: Desiccant Operating Seconds
8	ChillerSecs: Chiller Operating Seconds
9	T1: Outdoor Temperature
10	T2: Desiccant Leaving Temperature
11	T3: Return Air Temperature
12	T4: Cooling Coil Entering Temperature
13	T5: Building Supply Air Temperature
14	T6: Cooling Coil Leaving Temperature
15	T7: Reneration Air Leaving Temperature
16	RH1: Outdoor Relative Humidity
17	RH2: Desiccant Leaving Relative Humidity
18	RH3: Return Air Relative Humidity
19	RH4: Cooling Coil Entering Temperature
20	RH5: Building Supply Air Relative Humidity
21	RH6: Cooling Coil Leaving Relative Humidity
22	P1: Heater Wheel Pressure Drop
23	kW1: Desiccant Electric Demand
24	Fuel: Desiccant Gas Consumption (Ft^3/scan)
25	Airflow: Desiccant Air Flow

Keesler Air Force Base Desiccant System Monitoring Points



Data Information

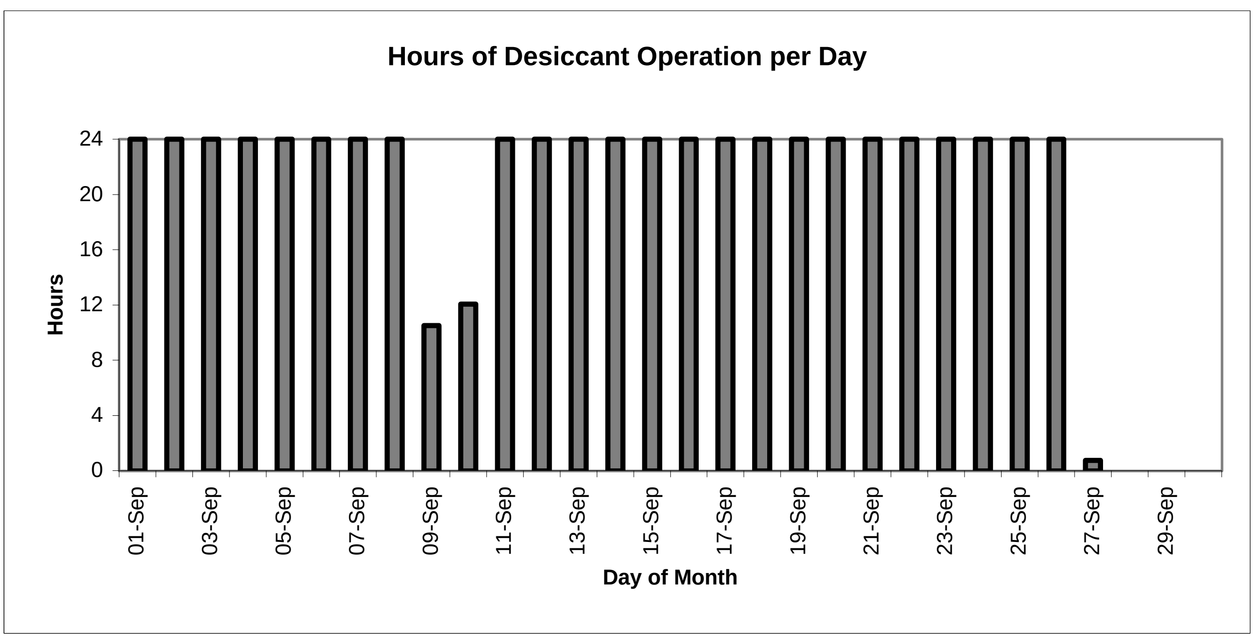
Hours in the Reporting Period	720.0	Hours
Hours of Data Collected	599.3	Hours
Percent of Data Collected	83.2	%



Note: Communication with Datalogger Lost on Sept. 27th due to Huriccane

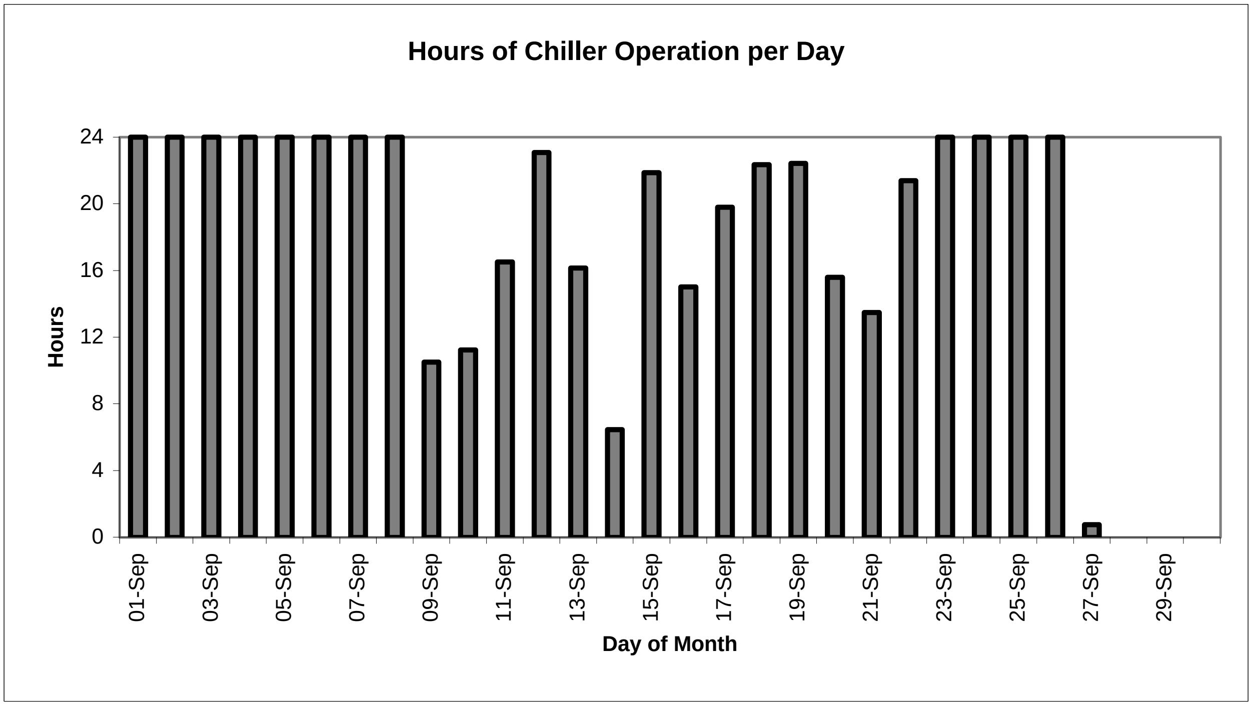
Desiccant Operation

Hours of Operation	599.3	Hours
Hours of Off Time	0.0	Hours



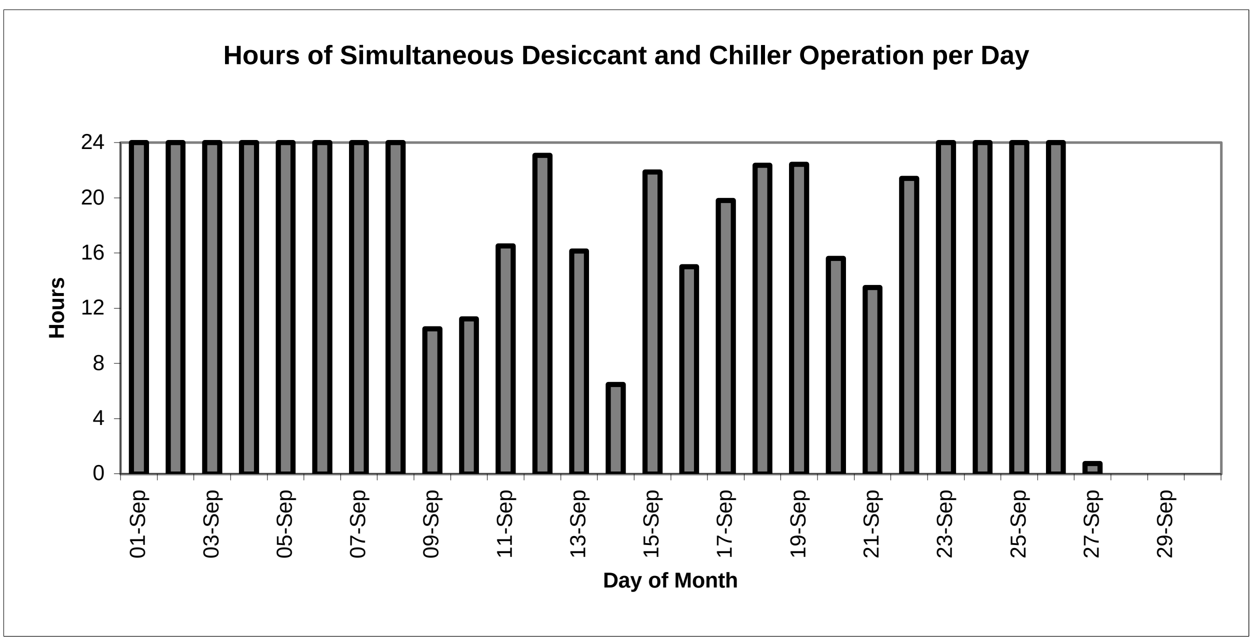
Chiller Operation

Hours of Operation	524.6	Hours
Hours of Off Time	74.7	Hours



Simultaneous Desiccant and Chiller Operation

Hours of Operation	524.6	Hours
Hours of Off Time	74.7	Hours



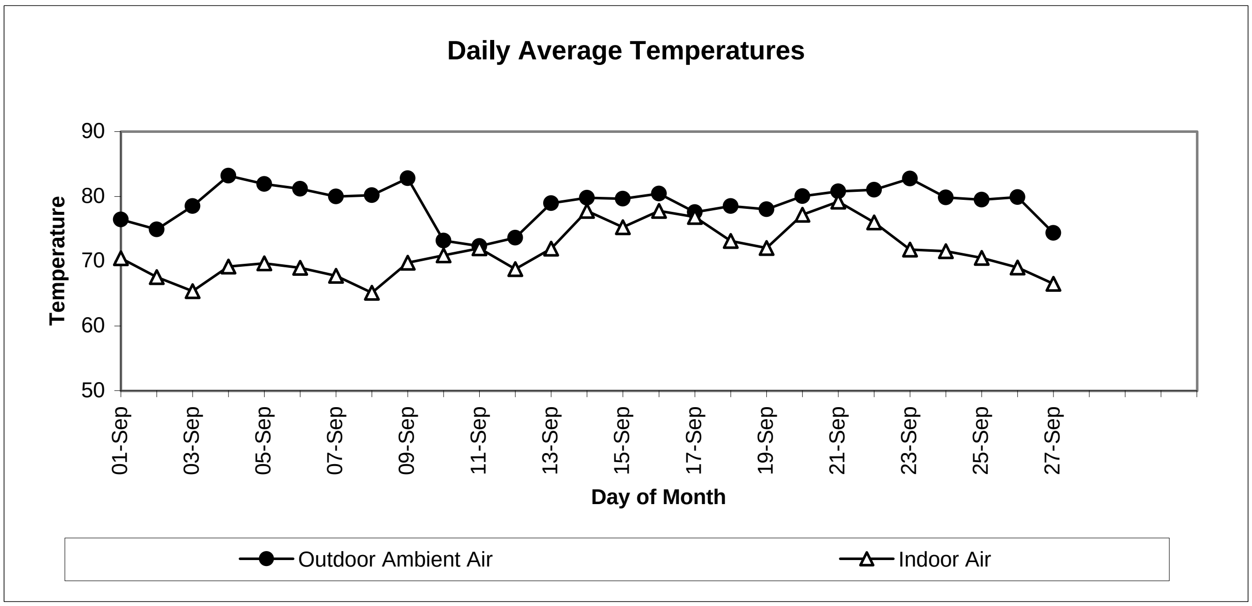
Average Building Conditions

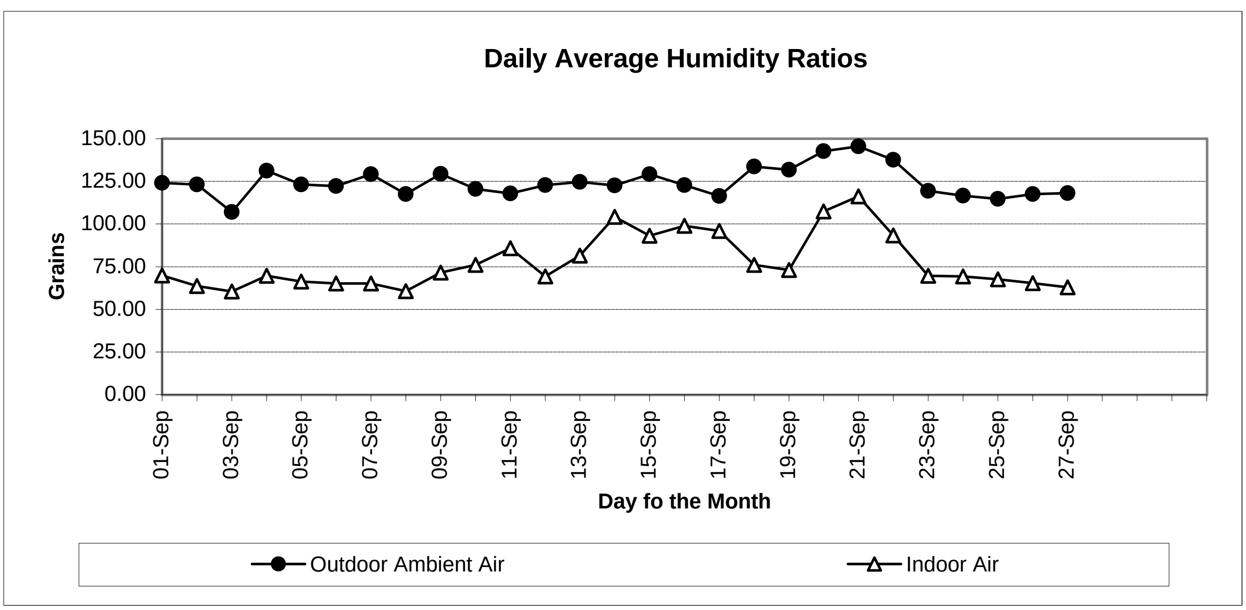
Outdoor Air Conditions

Average Outdoor Air Temperature	78.9	°F
Average Outdoor Air Relative Humidity	83.9	%
Average Outdoor Air Humidity Ratio	124.5	grains

Indoor Air Conditions

Average Indoor Air Temperature	71.5	°F
Average Indoor Air Relative Humidity	66.3	%
Average Indoor Air Humidity Ratio	77.7	grains

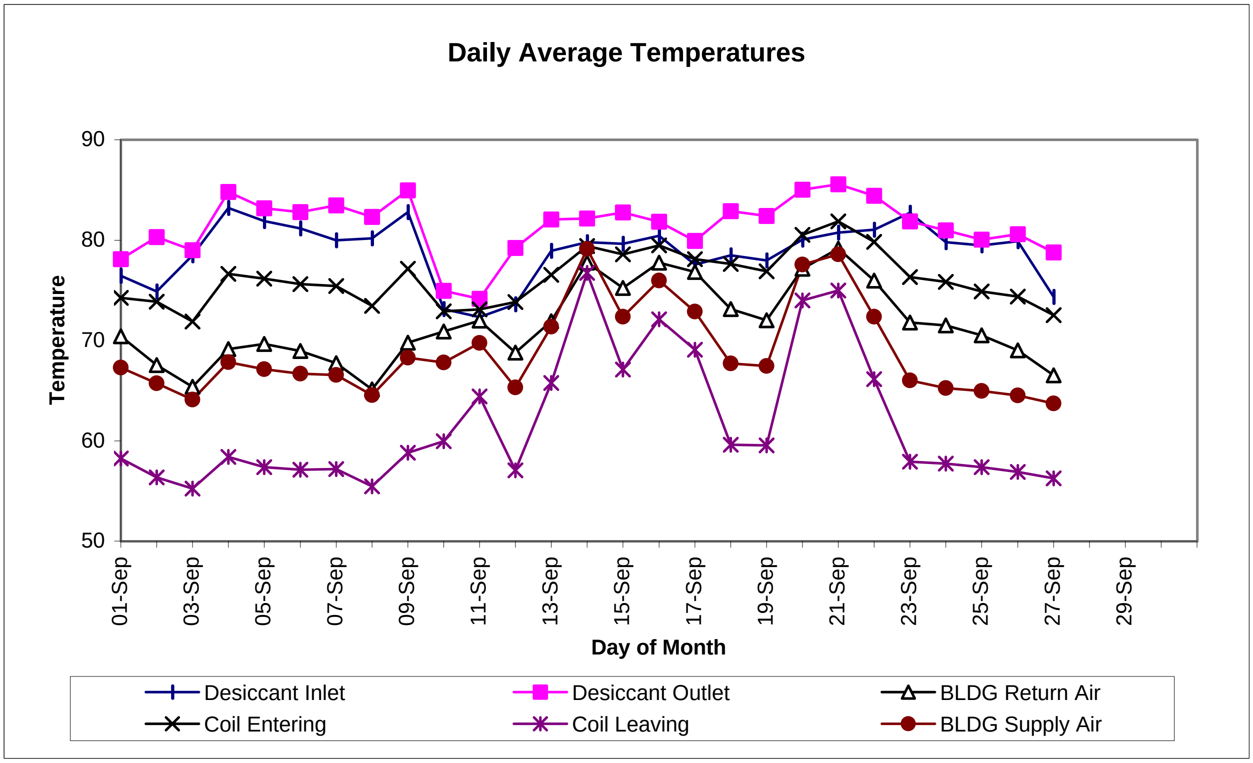


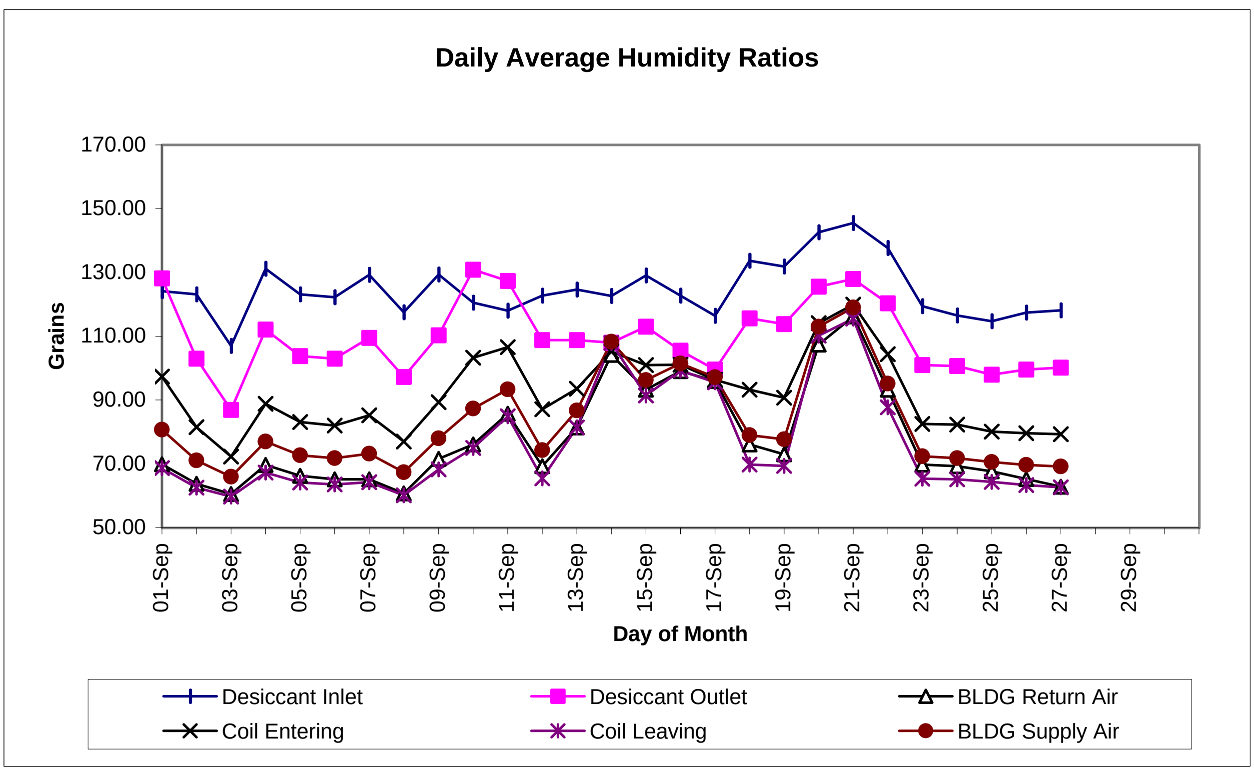


Desiccant Operation

Desiccant Inlet Air Conditions		
Average Air Temperature	78.9	^oF
Average Air Relative Humidity	83.9	%
Average Air Humidity Ratio	124.5	grains
Desiccant Leaving Air Conditions		
Average Air Temperature	81.4	^oF
Average Air Relative Humidity	68.2	%
Average Air Humidity Ratio	109.6	grains
Building Return Air Conditions		
Average Air Temperature	71.5	^oF
Average Air Relative Humidity	66.3	%
Average Air Humidity Ratio	77.7	grains
Cooling Coil Entering Air Conditions		
Average Air Temperature	76.2	^oF
Average Air Relative Humidity	67.5	%
Average Air Humidity Ratio	91.7	grains
Cooing Coil Leaving Air Conditions		
Average Air Temperature	61.7	^oF
Average Air Relative Humidity	91.0	%
Average Air Humidity Ratio	76.0	grains
Building Supply Air Conditions		
Average Air Temperature	68.9	^oF
Average Air Relative Humidity	77.5	%
Average Air Humidity Ratio	83.0	grains
Conditioned Space Air Conditions		
Average Air Temperature	71.5	^oF
Average Air Relative Humidity	66.3	%
Average Air Humidity Ratio	77.7	grains

Desiccant Operation (Continued)

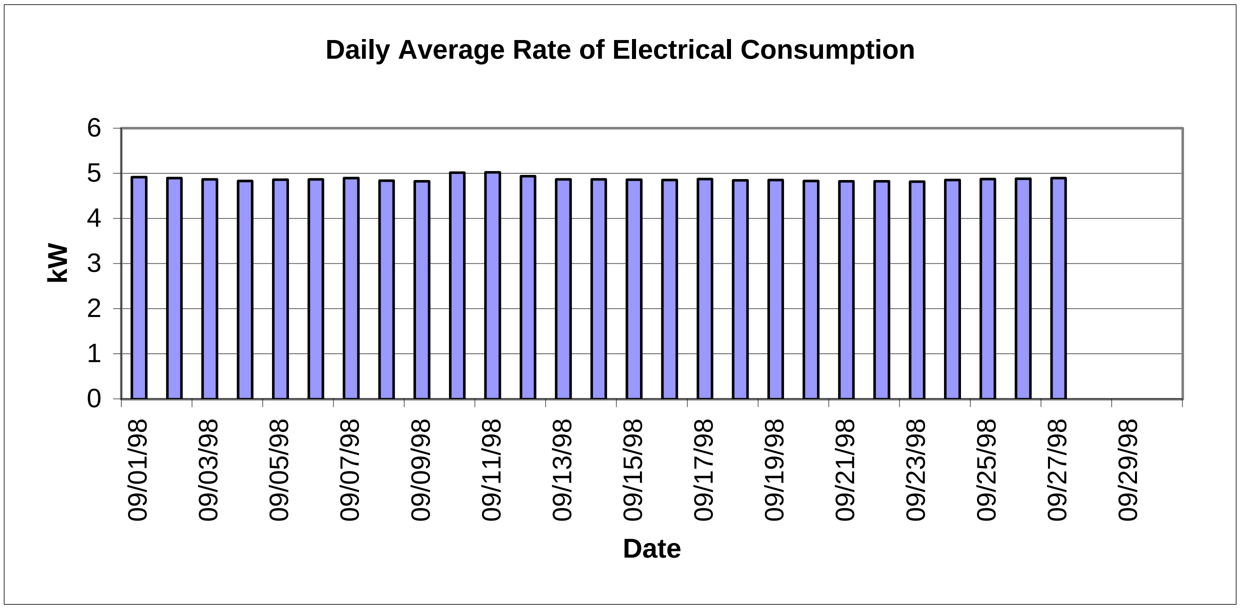


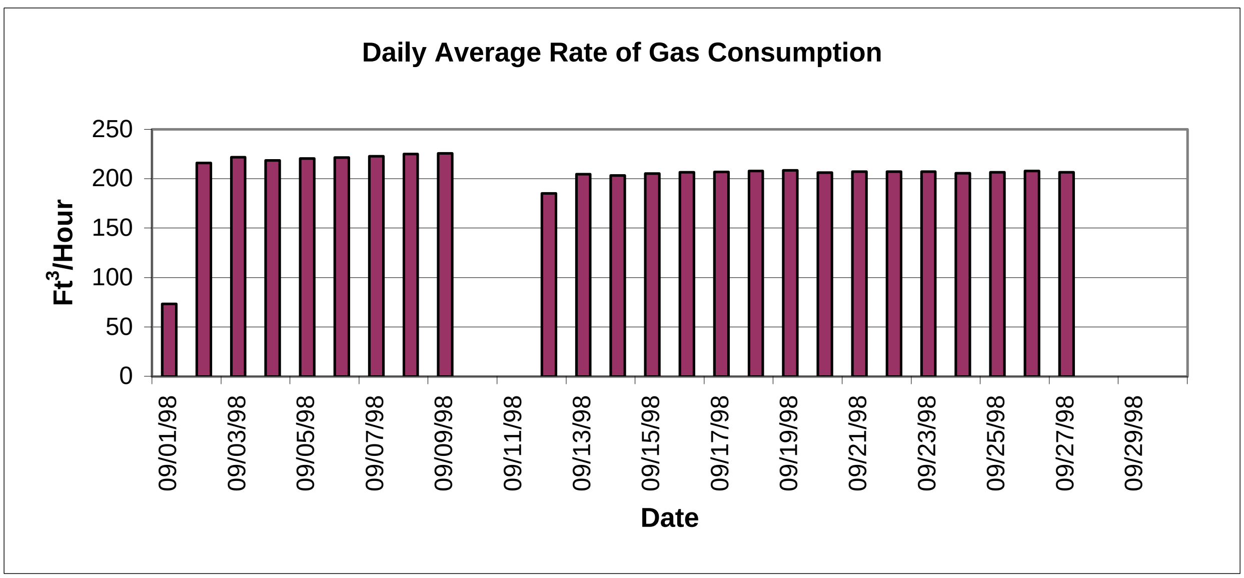


<u>Desiccant Operation (Continued)</u>

Desiccant Unit Operating Parameters

Average Regeneration Leaving Temperature	114.4	^oF
Average Supply Air Flow	3722.8	cfm
Total Electricity Consumed	2,918.2	kWh
Average Rate of Electricity Consumed	4.9	kW
Total Fuel Consumed	115,206.0	Ft^3
Average Rate of Fuel Consumption	192.2	Ft^3/Hr



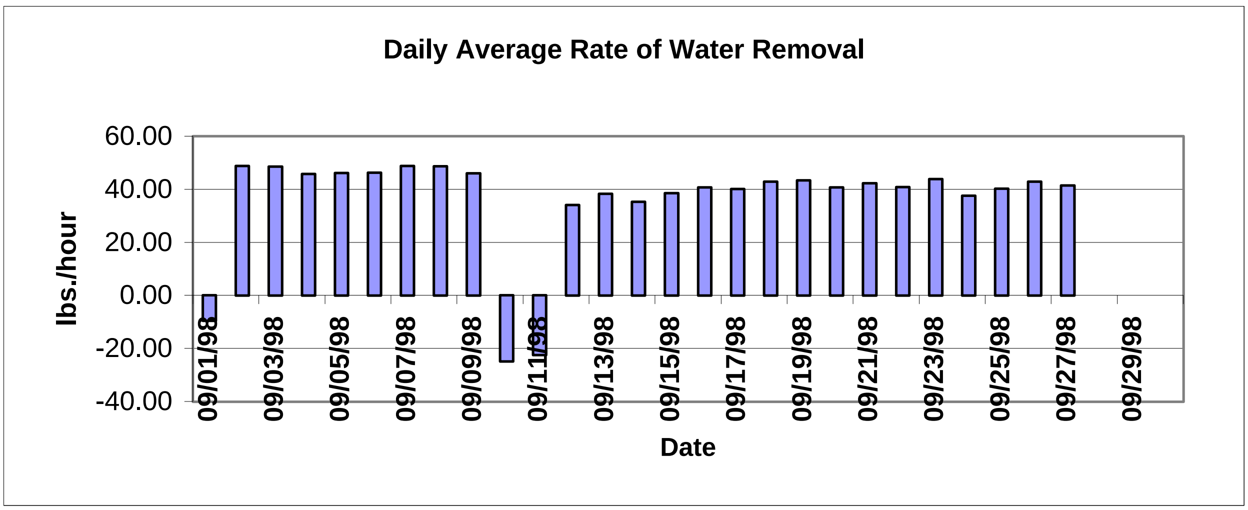


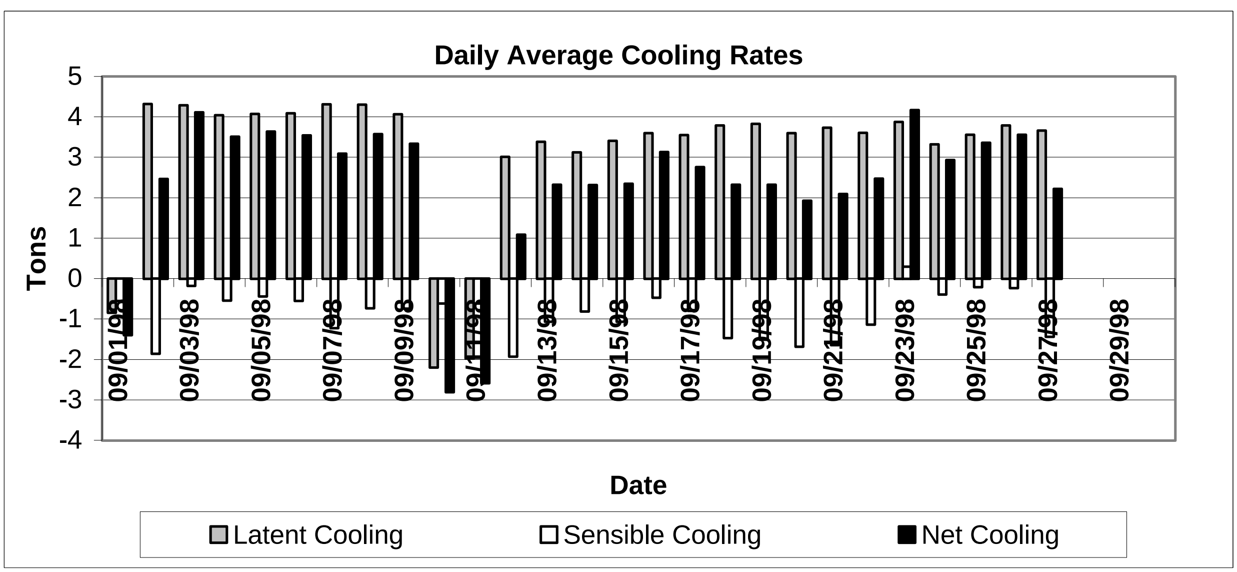
Desiccant Operation (Continued)

Total Pounds of Water Removed	24,832.3	lbs.
Average Rate of Water Removal	41.4	lb/Hr
Total Latent Cooling Provided	2,195.6	Ton-Hours
Average Latent Cooling Rate	3.7	Tons
Total Sensible Cooling	-527.3	Ton-Hours
Average Rate of Sensible Cooling	-0.9	Tons
Net Cooling Provided	1,668.3	Ton Hours
Average Net Cooling Provided	2.8	Tons

Efficiencies

Latent Work/ Fuel	22.2	%
Latent Work / (Fuel + Electric)	20.5	%

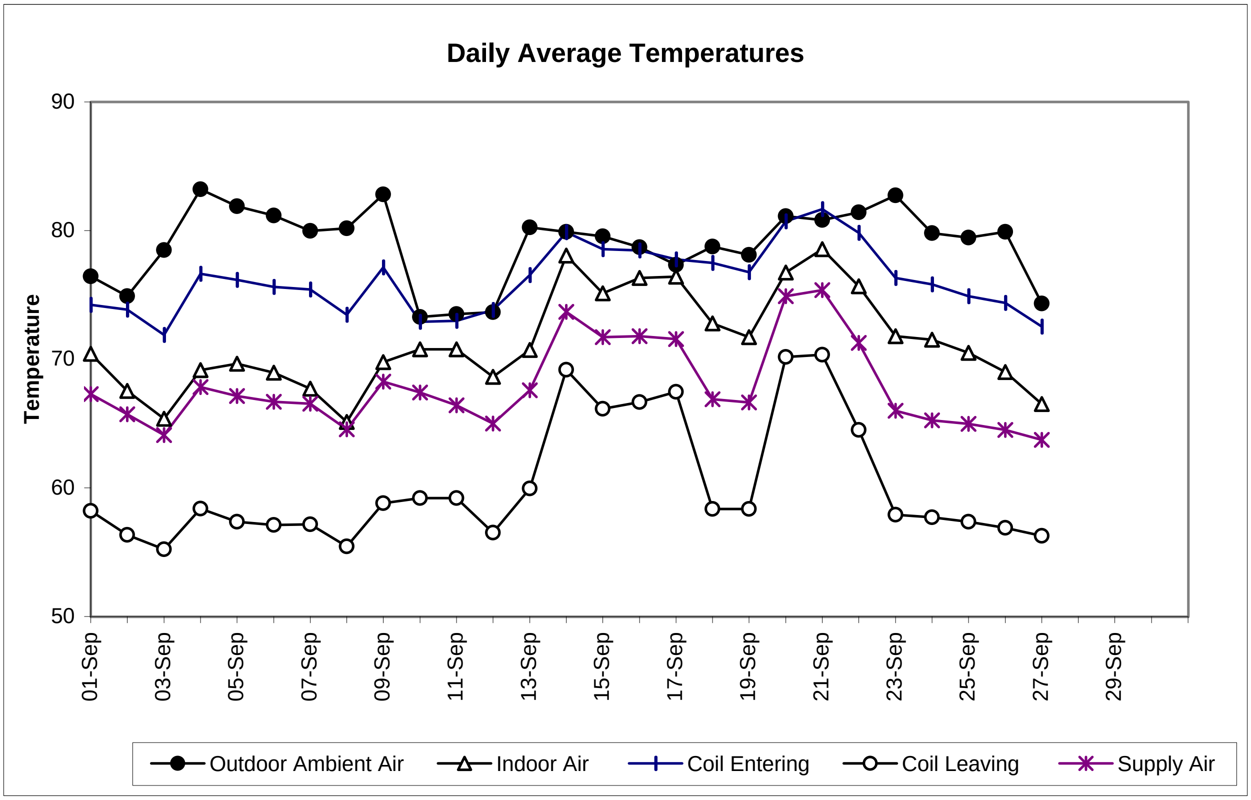


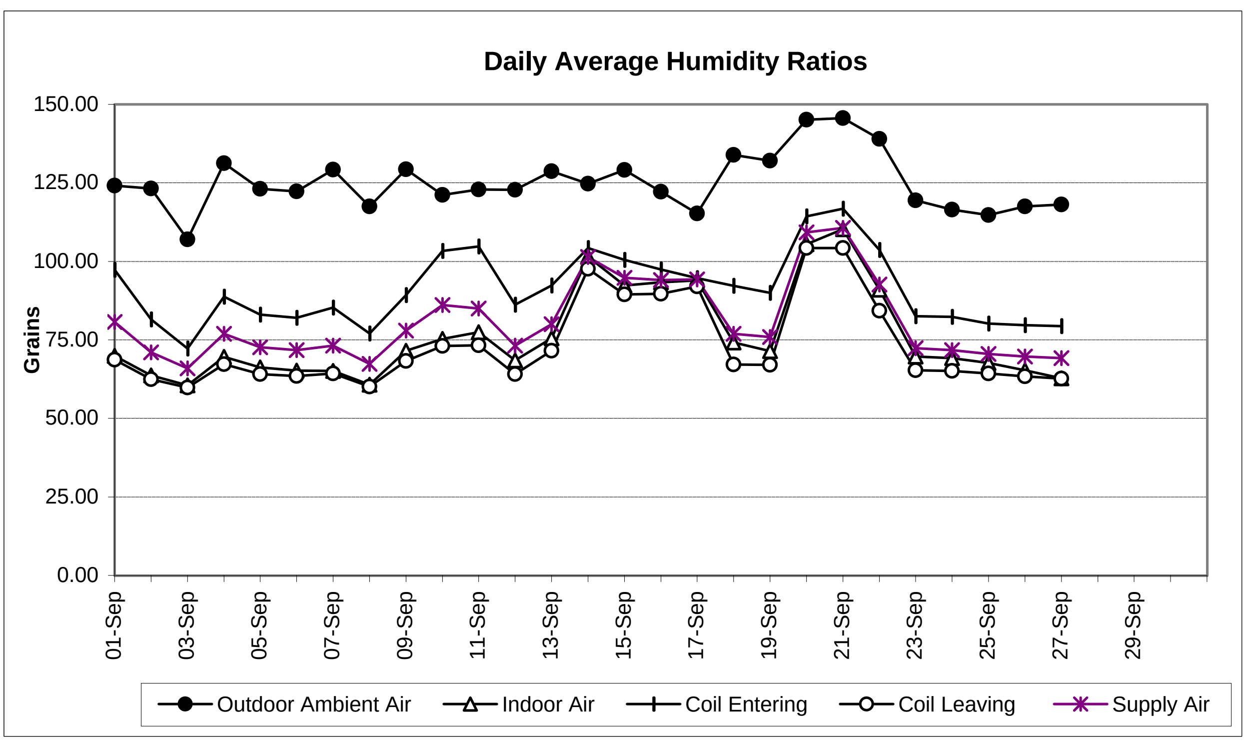


HVAC System during Chiller Operation

Chiller Hours of Operation	524.6	Hours
Outdoor Air Conditions		
Average Air Temperature	79.0	^oF
Average Air Relative Humidity	83.9	%
Average Air Humidity Ratio	125.0	grains
Cooling Coil Entering Air Conditions		
Average Air Temperature	76.1	^oF
Average Air Relative Humidity	67.2	%
Average Air Humidity Ratio	91.1	grains
Cooing Coil Leaving Air Conditions		
Average Air Temperature	60.2	^oF
Average Air Relative Humidity	92.8	%
Average Air Humidity Ratio	73.2	grains
Building Supply Air Conditions		
Average Air Temperature	67.9	^oF
Average Air Relative Humidity	78.7	%
Average Air Humidity Ratio	80.9	grains
Conditioned Space Air Conditions		
Average Air Temperature	71.3	^oF
Average Air Relative Humidity	65.7	%
Average Air Humidity Ratio	76.2	grains

HVAC System during Chiller Operation (Continued)





HVAC System during Simultaneous Desiccant and Chiller Operation

Hours of Simultaneous Operation	524.6	Hours
Desiccant Inlet Air Conditions		
Average Air Temperature	79.0	^oF
Average Air Relative Humidity	83.9	%
Average Air Humidity Ratio	125.0	grains
Desiccant Leaving Air Conditions		
Average Air Temperature	81.5	^oF
Average Air Relative Humidity	68.3	%
Average Air Humidity Ratio	110.1	grains
Building Return Air Conditions		
Average Air Temperature	71.3	^oF
Average Air Relative Humidity	65.7	%
Average Air Humidity Ratio	76.2	grains
Cooling Coil Entering Air Conditions		
Average Air Temperature	76.1	^oF
Average Air Relative Humidity	67.2	%
Average Air Humidity Ratio	91.1	grains
Cooing Coil Leaving Air Conditions		
Average Air Temperature	60.2	^oF
Average Air Relative Humidity	92.8	%
Average Air Humidity Ratio	73.2	grains
Building Supply Air Conditions		
Average Air Temperature	67.9	^oF
Average Air Relative Humidity	78.7	%
Average Air Humidity Ratio	80.9	grains
Conditioned Space Air Conditions		
Average Air Temperature	71.3	^oF
Average Air Relative Humidity	65.7	%
Average Air Humidity Ratio	76.2	grains

HVAC System during Simultaneous Desiccant and Chiller Operation (Cont.)