7. DEHUMIDIFICATION GUIDELINES

Overview

Introduction

In commercial buildings, temperature control often receives far more attention than humidity control. But moisture-related problems do occur in commercial buildings, especially in climates such as Hawaii's, and these problems can be costly. A significant portion of the construction claims against architects, engineers and contractors are related to moisture and humidity problems. Mold and mildew cost the hotel industry over $68 million every year in lost income and replacement furnishings, according to the American Hotel and Motel Association. Microbial growth accounts for more than one-third of indoor air quality (IAQ) problems.

HVAC systems are typically designed to ensure that they meet indoor comfort conditions at peak cooling loads. These systems, however, may not be able to provide adequate dehumidification during low load periods. These guidelines introduce system alternatives designed to improve part-load dehumidification performance as well as energy efficiency.

Finding the proper balance between energy efficiency and acceptable indoor air quality has become a critical problem for designers, building owners and operators, and maintenance personnel. For indoor thermal comfort, relative humidity levels up to 70% in summer may be acceptable. But for indoor air quality, the optimal humidity is between 40% and 60%. Poor indoor air quality may lead to an increased incidence of health-related symptoms, which in turn may lead to a rise in absenteeism and a loss of productivity. Increased ventilation can improve the indoor environment but may add to the first cost and operating cost of air conditioning systems, particularly in Hawaii.

Excess moisture and humidity problems in buildings are mostly caused by the intrusion of rain and groundwater and the infiltration of humid outside air through the building envelope, coupled with inadequate dehumidification capability and the inadequate operation of HVAC systems to remove the moisture and humidity. These problems can be resolved by correct air pressurization in buildings, adequate dehumidification capability and proper operation of HVAC systems.

Good control of humidity will reduce operation and maintenance costs of buildings, provide a healthy working environment and improve worker productivity. Most importantly, an energy-efficient cooling and dehumidification system does not necessarily have a higher initial cost than a conventional system, and it will save a significant amount in lifecycle costs.

There are basically two types of dehumidification: cooling-based systems and desiccant systems. Cooling-based systems extract moisture in a liquid state by using coils to cool the air to a saturation state, with the air temperature lower than the space air's dew-point temperature. In contrast, desiccant systems directly extract moisture from the air in a vapor state; this occurs without a cooling effect and produces air with a higher temperature (due to heat of adsorption) and lower humidity content.

Applicability

Buildings in hot/warm and humid climates, with high space latent loads, high outside air ventilation rate, low interior space humidity requirement or stringent humidity control, will need dehumidification. These types of buildings include supermarkets, hospitals, labs, clean rooms and theaters.

Codes and Standards

Section 9.3(e) of the Hawaii Model Energy Code states: "Where a humidistat is used for comfort dehumidification, it shall be capable of being set to prevent the use of fossil fuel or electricity to reduce relative humidities below 60%."

The following standards address dehumidification in buildings:

- ASHRAE Standard 62–1999, Ventilation for Acceptable Indoor Air Quality
- ANSI/ASHRAE Standard 55–1992, Thermal Environmental Conditions for Human Occupancy

Weather Data

Hawaii has a subtropical climate, with very consistent weather and only moderate changes in temperature throughout the year. There are only two seasons in Hawaii: summer extends from May to October and winter runs from November to April. The average daytime summer temperature at sea level is 85°F while the average daytime winter temperature is 78°F. Nighttime temperatures are approximately 10°F lower. The wettest months are from November to March. The following figures provide average weather data for Honolulu.

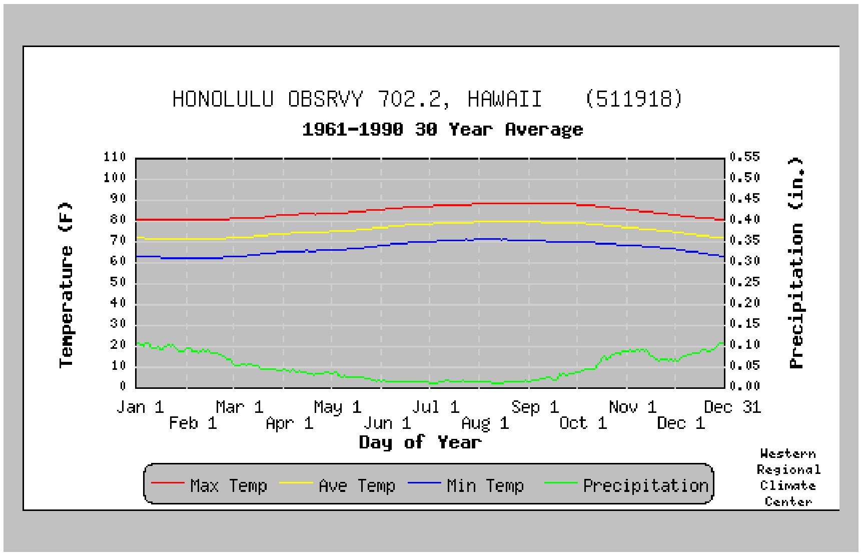


Figure 7-1. Honolulu average temperature and precipitation data, 1961–1990. Source: Western Regional Climate Center, www.wrcc.dri.edu.

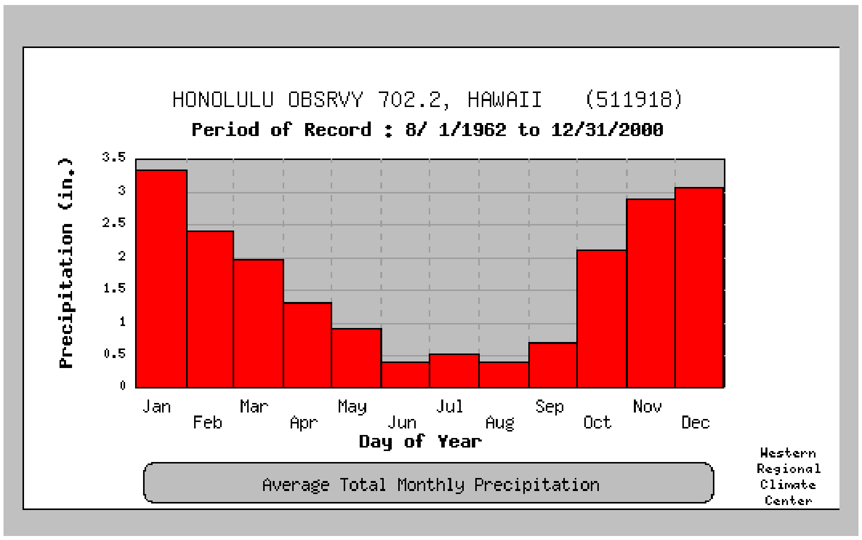


Figure 7-2. Honolulu average monthly precipitation data, 1962–2000. Source: Western Regional Climate Center, www.wrcc.dri.edu.

About 93% of the time, the outside air dew-point temperature is higher than that of the indoor air setpoint of 73°F DB and 60% RH. Therefore, ventilation air requires dehumidification almost all year. Less than 10% of the time the outside air enthalpy is less than that of the indoor air, and less than 24% of time the outside air dry-bulb temperature is less than that of indoor air temperature setpoint. Therefore, **from an energy efficiency perspective, economizer and nighttime ventilation are not applicable in Hawaii.**

The ventilation air load for typical office occupied from 8 A.M. to 6 P.M., Monday through Friday, is 1.74 ton-hr/cfm for sensible load and 2.85 ton-hr/cfm for latent load, which means latent load dominates the ventilation load.

Load Calculation

The first and most important task in designing a dehumidification system is to calculate the moisture load. There are two key issues in moisture-load calculation:

1. Select design conditions for outdoor air and indoor air. For dehumidification, the design day is not hot, but rather warm and raining. Peak latent load should be calculated at ASHRAE 0.4% design dew-point temperature condition rather than ASHRAE 0.4% design dry-bulb temperature condition. **For Honolulu, the former will result in about a 31% higher moisture load.** The 0.4% design conditions mean the moisture outdoors is not likely to exceed the selected value for more than 35 hours in a typical year, which is quite enough for most engineering applications.

   Depending on the application, the indoor air temperature setpoint can range from 70°F to 78°F for cooling; relative humidity can range from 30% to 60%.

2. Calculate moisture loads from people, ventilation, infiltration through the building envelope (doors, walls and windows), and moisture released by food, products, equipment, etc. Occupant loads have a wide range, from 0.1 lb/hr at seated/rest state to 1.04 lb/hr at an athletics level of activity. Ventilation is the largest moisture source in commercial buildings. Ventilation should provide: 1) enough fresh air for occupants, typically 20 cfm/person for offices; 2) enough makeup air if there is exhaust air; and 3) enough air to pressurize the building.

   A slight positive space pressure is very useful to reduce infiltration; as a rule-of-thumb, assume 10% additional ventilation air for pressurization. Infiltration exists in all buildings, even those with positive pressure. Typical values to use are 0.1, 0.3 and 0.6 cfm/ft² for tight, average and loose wall constructions respectively. Rain-soaked masonry or concrete walls add moisture to infiltration air. Moisture transfer through the building envelope and leakage of the return air duct should also be taken into account.

Table 7-1. ASHRAE design conditions for Hawaii.

Location	ASHRAE Design Conditions	Dry-bulb temp. °F	Wet-bulb temp. °F	Dewpoint temp. °F	Rel. hum. %	Enthalpy Btu/lb	Humidity Ratio gr/lb
Honolulu	0.4% dry-bulb temp.	89	73	66.2	47.0	36.6	97
	0.4% dew-point temp.	80	75.6	74	82.0	39.2	127
Hilo	0.4% dry-bulb temp.	85	74	69.6	60.1	37.6	109
	0.4% dew-point temp.	79	76.1	75	87.6	39.6	132
Kahului	0.4% dry-bulb temp.	89	74	67.9	49.8	37.6	103
	0.4% dew-point temp.	80	75.6	74	82.0	39.2	128
Kaneohe, MCAS	0.4% dry-bulb temp.	86	75	70.7	60.5	38.5	114
	0.4% dew-point temp.	81	77.3	76	84.8	40.8	136
Ewa, Barbers Point NAS	0.4% dry-bulb temp.	92	73	64.7	40.7	36.6	92
	0.4% dew-point temp.	83	76.4	74	74.4	39.9	128
Lihue	0.4% dry-bulb temp.	85	75	71.2	63.4	38.6	116
	0.4% dew-point temp.	80	76.3	75	84.8	40.0	133
Molokai	0.4% dry-bulb temp.	88	73	66.8	49.5	36.9	100
	0.4% dew-point temp.	80	75.6	74	82.0	39.5	130

System Alternatives

A variety of dehumidification systems have been developed to improve the energy efficiency of conventional reheat systems. Their target is zero reheat and zero overcooling in the dehumidification process by energy recovery, recycling, reuse and load reduction. These systems, which are discussed later in this chapter, include:

- Conventional reheat systems
- Run-around coil systems
- Heat pipe systems
- Dual-path systems
- Desiccant systems

Refrigerant subcooling systems are discussed in the single-zone direct-expansion (DX) Systems section of the HVAC Guidelines.

Integrated Design Implications

When dehumidification is integrated into a cooling system, pay special attention to these issues:

- Select and size HVAC equipment (coils, fan, pump, damper, etc.) for sensible and latent cooling at peak load conditions. These usually don't occur simultaneously.
- Design for energy efficiency at part-load conditions because peak load usually occurs for only about 2% of the operating time

System Performance

Annual energy consumption for three system types is estimated using Honolulu bin weather data for the three common systems used in commercial buildings. See the individual system sections below for detailed performance data. Table 7-2 summarizes the system performance results.

Table 7-2. Estimated annual energy performance of dehumidification systems.

	KWh			Savings		
System Specifications	**Conventional System (base case)**	**Run-around System**	**Dual-path System**	**Conven-tional System (base case)**	**Run-around System**	**Dual-path System**
CAV, 1000 cfm, 10% OA	11,993	5998	5923	0	50%	50%
CAV, 1000 cfm, 20% OA	12,315	6236	5887	0	49%	52%

Notes: The following data and assumptions were used in the calculations.

- System location: Honolulu
- System type: Constant air volume (CAV)
- Space setpoint: 73°F DB, 60% RH
- Airflow: supply air 1000 cfm, outside air 100 cfm, return air 900 cfm
- Space loads: total 1.92 ton, sensible load ratio 81%
- Space load variation: sensible load changes, latent load constant
- Operating hours: Monday to Friday, 8 AM to 6 PM, total 2860 hrs/yr
- Electric reheat is used whenever needed
- Fan heat increases air temperature by 1.5°F, no return air temperature rise due to duct heat loss
- Efficiency: cooling system total 0.8 kW/ton, chiller 0.6 kW/ton
- Four typical load conditions are calculated: 100%, 75%, 50% and 25%
- TMY2 weather data for Honolulu
- Run-around loop effectiveness 50%

Some important observations from the calculations and results:

- Conventional electric reheat systems double the annual energy use compared with run-around systems or dual-path systems.
- Conventional systems need more reheat in lower load conditions when the supply air volume remains constant (CAV systems).
- Run-around coil systems and dual-path systems are energy efficient in dehumidification applications. They reduce coil loads and avoid reheat for most load conditions.

- Run-around systems and dual-path systems may need reheat at very low load conditions when space latent load dominates the total cooling load and the ventilation rate remains constant. A special case occurs in supermarkets where refrigerated display cases cool the store during unoccupied hours or when the store is cool due to cool weather. In this case, dual-path systems are equipped with reversing valves that allow the return air circuit to provide heating (see the Foodland Lahaina case study in the Dual-path Systems Guideline).

Design Details

To achieve an energy-efficient dehumidification system design, consider the following factors:

- From an energy efficiency perspective, economizer and nighttime ventilation are not applicable in Hawaii.
- Size and select cooling coils with enough cooling capacity to handle the peak sensible cooling load and peak latent cooling load that occur at different load conditions. Use low-approach cooling coils and low temperature water.
- Design systems considering various load conditions rather than only the peak load condition. For conventional cooling with reheat systems, size reheat equipment to handle a higher reheat requirement in lower load conditions, especially for constant-volume systems.
- Integrate heat recovery equipment into conventional cooling systems to reduce cooling loads and reheat energy. Run-around loop systems are much more energy efficient than conventional cooling systems, especially when operating in part-load conditions.
- Dual-path systems offer competitive energy efficiency with run-around loop systems, and provide better control of the outside air ventilation rate. Dual-path systems decouple sensible cooling and latent cooling for easy control of the supply air temperature and humidity. Equipment is available to provide both cooling and reheat (for example, ClimateMaster).
- Desiccant systems are more competitive when a low supply air dew-point temperature is required, latent load fraction is high, low- or no-cost reactivation heat from steam, hot water or waste heat is available, and electricity costs are high when compared to gas costs.
- Lay out equipment correctly. Place filters upstream of coils. Place fans downstream of coils (draw-through mode) to provide a small amount of reheat.

- Select low face velocity coils to reduce air pressure drop and improve dehumidification performance.
- When using heat pipes, make sure that the additional delta-P is accounted for.

Operation and Maintenance

To maintain efficient operation of cooling and dehumidification systems:

1. Coils must drain condensate and be cleaned regularly.
2. Filters must be cleaned or replaced regularly.
3. Chilled water temperature reset should not sacrifice the dehumidification requirement of cooling coils. A system analysis (air and water) can tell you what the optimum chilled water temperature is.

Utility Programs

The utilities that serve Oahu, Maui, Molokai, Lanai and the Big Island (HECO, MECO and HELCO) have a rebate program called the Commercial and Industrial Customized Rebate (CICR) program. Under this program innovative technologies that save energy and demand qualify for a rebate based on $125 per kW of peak demand reduction and $0.05 per kWh for a year of energy savings. Rebates are based on engineering estimates of energy and demand savings. In the case of unproven technologies, the rebate may be paid over a period of five years based on metered savings.

Resources

American Gas Cooling Center (AGCC), www.agcc.org

ASHRAE Handbooks, www.ashrae.org

ASHRAE, *Humidity Control Design Guide for Commercial and Institutional Buildings*

Electrical Power Research Institute (EPRI), www.epri.com

ESource, *Technology Atlas Series — Space Cooling,* www.esource.com

Federal Energy Management Program (FEMP), www.eren.doe.gov/femp

Gas Technology Institute (GTI), www.gri.org

Hawaiian Electric Co., Hawaii Electric Light Co., Maui Electric Co., www.heco.com

HPAC (Heating/Piping/Air-conditioning) Engineering journal, www.hpac.com

Munters Corporation, *The Dehumidification Handbook,* www.dehumidification.com

National Renewable Energy Laboratory (NREL), http://www.nrel.gov

Penn State University, *The Dedicated Outside Air System,* www.doas.psu.edu

J. David Odom and George DuBose, *Preventing Indoor Air Quality Problems in Hot, Humid Climates: Problem Avoidance Guidelines,* Revised 1996, CH2M Hill.

Conventional Cooling Systems with Reheat

Recommendation

Install conventional systems in applications with low latent loads, with no requirements for indoor air quality or humidity control, and where low first cost is a high priority. Consider the use of cooling-coil face and bypass dampers or cold air distribution to reduce the need for reheat.

Description

Conventional cooling systems dehumidify the mixed air by passing it across a cooling coil that is cold enough to condense water vapor, and then reheating it to the required supply air temperature. The cooling coil can be powered by chilled water from central chiller plant or it can be a direct expansion refrigerant coil. Reheat may be trivial or not needed at peak load conditions, which is usually based on the design dry-bulb temperature that gives maximum sensible cooling loads instead of maximum latent loads. Reheat is often needed at typical low-load conditions with higher latent load fraction. Conventional systems often double the total energy use because of overcooling and reheating of the supply air.

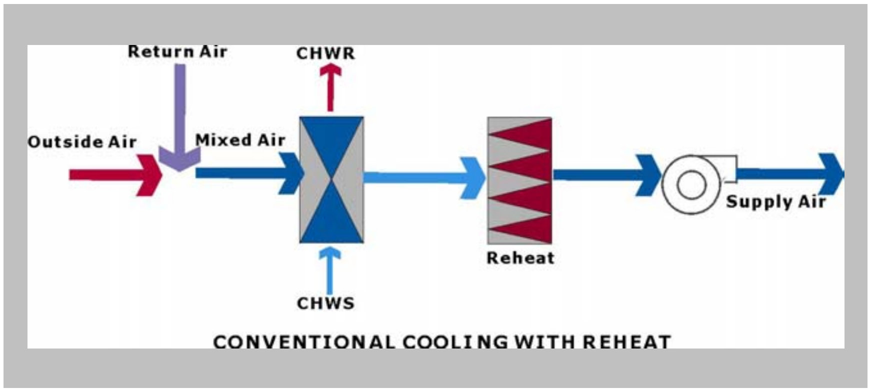


Figure 7-3. Conventional cooling with reheat. CHWR = chilled water return. CHWS = chilled water supply.

Applicability

Conventional reheat systems are most applicable in buildings with:

- No requirements for humidity control of supply air
- Dry climates where sensible cooling dominates
- Low outside air ventilation rate and low space latent load
- Space relative humidity settings of 55% and higher
- Low- or no-cost waste heat, steam or hot water available for reheat use

Codes and Standards

The Hawaii Model Energy code sets limits on simultaneous heating and cooling. Section 9.4(b) limits the use of reheat to several specific cases. It's allowed in:

- Variable-air-volume systems,
- Zones with special pressurization requirements,
- Systems with at least 75% of reheat energy from recovered or solar energy,
- Zones with specific humidity requirements for process needs, or
- Small zones where peak airflow is 300 cfm or less.

Benefits

Benefits of conventional cooling with reheat include:

- Simple system configuration
- Good humidity control by adjusting the off coil air temperature
- Low initial cost

Design Details

The need for reheat can be reduced using a bypass damper in parallel with the cooling coil. This allows a portion of the air to be cooled to a low temperature and dehumidified, and then remixed with the bypass air. The total moisture removal is greater than if all the air passes through the cooling coil but is not cooled to as low a temperature.

Another means to reduce reheat requirements is through cold air distribution in which supply air is delivered at 50ºF or lower, instead of the typical 55ºF. More moisture is extracted as the air is cooled to the lower temperature, and the air distribution system is designed to handle the lower temperature air. This approach requires careful selection of diffusers to maintain comfort and is also more susceptible to condensation on ductwork. Another benefit is that lower airflow and therefore less fan energy is needed. The downside is that lower chilled water temperatures are necessary and chiller energy consumption may increase. Cold air distribution is a good match for ice thermal storage systems, which can deliver colder than normal chilled water.

If a true cold air distribution system is not used, then dehumidification will be improved by choosing a cooling coil to provide a low approach temperature (the difference between chilled water temperature and supply air temperature). Coils can be selected to provide 52ºF supply air, while still operating at standard chilled water temperatures.

Design and Analysis Tools

DOE-2.1E program for building energy simulations from Lawrence Berkeley National Laboratory (LBNL), available at http://gundog.lbl.gov.

Costs

Conventional air handling systems with reheat cost approximately $4.00 to $5.00/cfm.

Cost Effectiveness

Conventional reheat systems may have a lower initial cost but their operating costs are much higher because energy is wasted in overcooling and reheating the supply air.

System Performance

The following table presents the system performance at four typical load conditions.

Table 7-3. Energy performance of a conventional system, CAV, 10% OA.

					Outside Air			Supply Air			
Load%	**Cooling Ton**	**Reheat kW**	**Total kW**	**Hours**	**cfm**	**DB**	**WB**	**cfm**	**DB**	**WB**	**KWh**
100%	2.60	0.00	2.63	225	100	87	75	1000	56	55	592
75%	2.09	1.09	3.31	1396	100	82	70	1000	61	58	4621
50%	1.98	2.88	5.02	897	100	77	66	1000	66.5	59.4	4503
25%	1.92	4.57	6.66	342	100	72	64	1000	71.7	61.9	2278
Total											11,993

Table 7-4. Energy performance of a conventional system, CAV, 20% OA.

					Outside Air			Supply Air			
Load %	**Cooling Ton**	**Reheat kW**	**Total kW**	**Hours**	**cfm**	**DB**	**WB**	**cfm**	**DB**	**WB**	**KWh**
100%	2.95	0.00	2.91	225	100	87	75	1000	56	55	655
75%	2.28	1.09	3.47	1396	100	82	70	1000	61	58	4844
50%	2.03	2.88	5.06	897	100	77	66	1000	66.5	59.4	4539
25%	1.92	4.57	6.66	342	100	72	64	1000	71.7	61.9	2278
Total											12,315

Products

Carrier, www.carrier.com

Dectron, www.dectron.com

Desert Aire, www.desert-aire.com

Dri-Eaz Products, www.dri-eaz.com

DryAire Systems, www.dryaire.com

Dumont Refrigeration, www.dumontgroup.com

EBAC Dehumidifiers, www.ebac.co.uk

McQuay Corporation, www.mcquay.com

Nautica Dehumidifiers, www.nauticadehumid.com

Trane Company, www.trane.com

York, www.york.com

Run-around Coil Systems

Recommendation

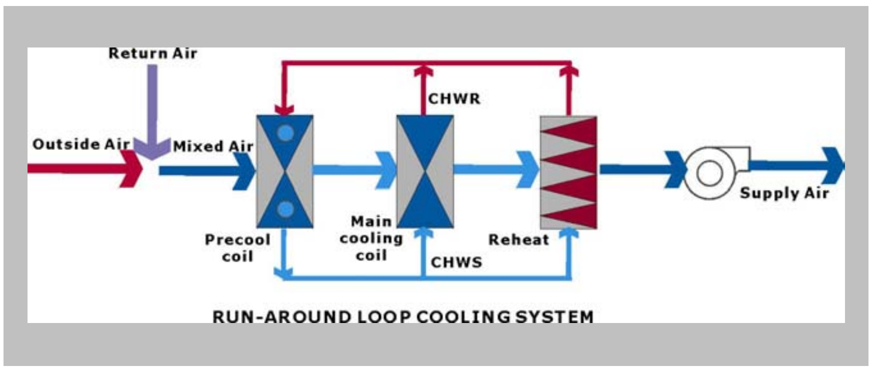
Install run-around coils in applications with large dehumidification requirements where the air must be reheated after passing the cooling coil.

Description

A run-around coil system is a simple piping loop with an upstream precooling coil and a downstream reheating coil that sandwiches the main cooling coil. The circulating fluid is pumped to transfer heat from the warm mixed air to the off coil cold supply air. The run-around system reduces the cooling load on the main cooling coil; reheat is provided by the heat picked up by the circulating fluid in precooling coil instead of by an external source of expensive energy.

In new building designs and retrofits, a run-around system can reduce peak heating and cooling loads as well as total heating and cooling energy. The run-around system can have a significant impact on the heating and cooling capacity in new HVAC designs.

Figure 7-4. Run-around loop cooling system.



The heat recovery effectiveness of the run-around loop is defined as the ratio of the actual heat transfer to the maximum possible heat transfer between the air streams. This is equivalent to the ratio of the difference between the mixed air temperature and the air temperature off the precool coil to the difference between the mixed air temperature and the air temperature off the main cooling coil. The effectiveness ranges from 50% for a normal loop to 65% for a high performance loop. Because of the relatively small temperature differences between the energy exchange coils, low approach cooling coils should be used. Designers must account for the additional pressure drop from the added coil.

Applicability

Run-around coil systems are most applicable in situations requiring substantial dehumidification.

Codes and Standards

ANSI/ASHRAE Standard 84–1991, Method of Testing Air-to-Air Heat Exchangers

Benefits

Benefits of run-around coil systems include:

- Lower cooling load contributes to a smaller cooling system and less pumping energy use, but fan energy increases due to extra air pressure drop through the run-around coils.
- Reheat energy is saved
- Lower total energy use

Integrated Design Implications

The increased dehumidification capacity provided by run-around coils allows for a smaller cooling system. However, the addition of coils will increase the pressure drop, and fan power must be adjusted accordingly.

Design Details

The run-around loop can either be applied to existing systems or can be installed at the factory. The run-around loop requires a fractional horsepower pump, a 120V–60HZ single-phase electrical circuit, and a three-way valve or a variable-speed drive (VSD) for the pump. For bigger systems, an expansion tank with air vent may be needed.

Design and Analysis Tools

Run-around coils can be selected by manufacturers or by design engineers using coil selection programs from manufacturers.

Costs

The initial cost of a run-around system is about double that of a conventional system, but if the downsizing of the chiller and cooling tower is counted, the total initial cost will be very close. The total installation cost is approximately $4.50 to $5.00/cfm.

Cost Effectiveness

The cost effectiveness of a run-around system depends on the system it is replacing. When used instead of a dehumidifying system requiring reheat, the simple payback is about two to three years. However, when the system replaces a system without reheat (no humidity control), there are additional benefits including increased comfort and enhanced indoor air quality, which are difficult to quantify.

Operations and Maintenance

Run-around systems require extra maintenance for the two coils and the loop. Air trapped in the coils, pump and piping must be vented upon initial startup to ensure effective fluid flow and heat transfer. The precooling and reheating function can be controlled by adjusting the pump speed with VSD, cycling the pump on-off, or using valve control and bypass.

Commissioning

Commissioning of a run-around system must be done for typical various load conditions to determine whether additional reheat is needed at very low load conditions.

System Performance

The following tables present the system performance at four typical load conditions.

Table 7-5. Energy performance of a run-around system, CAV, 10% OA.

Load %	Cooling Ton	Reheat kW	Total kW	Hours	Outside Air			Supply Air			KWh
					cfm	DB	WB	cfm	DB	WB	
100%	2.60	0.00	2.72	225	100	87	75	1000	56	55	612
75%	1.78	0.00	2.07	1396	100	82	70	1000	61	58	2890
50%	1.16	0.00	1.57	897	100	77	66	1000	66.5	59.4	1408
25%	1.10	1.65	3.18	342	100	72	64	1000	71.7	61.9	1088
Total											5998

Table 7-6. Energy performance of a run-around system, CAV, 20% OA.

Load %	Cooling Ton	Reheat kW	Total kW	Hours	Outside Air			Supply Air			KWh
					cfm	DB	WB	cfm	DB	WB	
100%	2.95	0.00	3.00	225	100	87	75	1000	56	55	675
75%	1.93	0.00	2.19	1396	100	82	70	1000	61	58	3057
50%	1.18	0.00	1.59	897	100	77	66	1000	66.5	59.4	1426
25%	1.07	1.65	3.15	342	100	72	64	1000	71.7	61.9	1077
Total											6236

Products

Run-around coils with ARI certification are available from at least 10 manufacturers.

Heat Pipe Systems

Recommendation

Install heat pipes in applications with large dehumidification requirements where the air must be reheated after passing the cooling coil.

Description

Heat pipes increase the effectiveness of air conditioning systems by helping to decrease the total cooling load of the air. The typical design consists of a refrigerant loop with two connected heat exchangers placed upstream and downstream from the cooling coil. As the air passes through the first heat exchanger it vaporizes the refrigerant and is precooled. This allows the coil to more effectively cool the air to a point below the dew-point temperature and to extract more moisture. The air then passes through the second heat exchanger and is reheated, which liquefies the refrigerant, causing it to flow back to the first heat exchanger. The heat pipe system is hermetically sealed, uses a wicking action, and requires no pump.

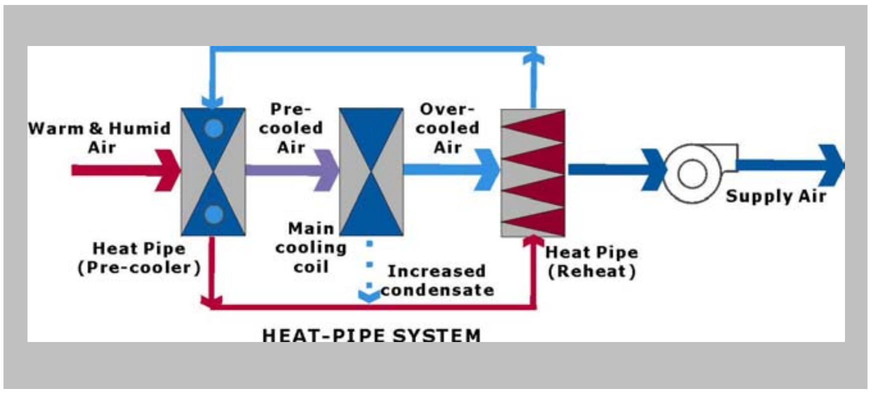


Figure 7-5. Heat pipe system.

Applicability

Most applicable in situations requiring substantial dehumidification.

Codes and Standards

None

Benefits

Benefits of heat pipe systems include:

- Removes 50% to 100% more moisture than systems without heat pipes.
- Saves energy compared to systems that provide similar amounts of dehumidification.
- Simple system with no moving parts or external connections makes it basically maintenance free.

Integrated Design Implications

The increased dehumidification capacity provided by heat pipes allows for a smaller cooling system. However, the addition of heat pipes will increase the pressure drop, and fan power must be adjusted accordingly.

Design Details

Heat pipes can either be applied to existing HVAC systems or can be installed at the factory. The heat pipe loop is usually controlled by cycling on-off or modulating the refrigerant flow with a control valve.

Design and Analysis Tools

Heat Pipe Technologies product selection software, www.heatpipe.com.

Costs

The installation cost of a heat pipe loop for a cooling system is approximately $2.50/cfm.

Cost Effectiveness

The cost effectiveness of heat pipes depends on the system it is replacing. When used instead of a dehumidifying system requiring reheat, the simple payback is two to three years. However, when the system replaces a system without reheat (that is, no humidity control), there are additional benefits including increased comfort and enhanced indoor air quality, which are difficult to quantify.

Operations and Maintenance

Some heat pipe applications require the same routine maintenance as any air conditioning unit. Valveless units require no maintenance aside from cleaning. Valved units have normal balancing requirements.

Commissioning

The precooling and reheating heat pipes should be installed closely to sandwich the main cooling coil.

Case Study

A Dinh-style heat pipe dehumidification system was installed in the air handling system (19,000 cfm) in Building 49 at the EPA's Gulf Breeze laboratory in Pensacola, Florida, in 1996. The heat pipe was effective in reducing inside humidity levels by about 10%, from an average of 75% before installation to an average of 65% after installation, without affecting the inside temperatures. An additional 20 tons of mechanical cooling would have been necessary to provide this additional dehumidification during peak conditions.

The heat pipe cost $42,000 to install; the additional mechanical cooling equipment necessary to provide the same level of dehumidification would have cost $30,000. Therefore the additional cost of installing a heat pipe instead of mechanical cooling to provide the 10% lower indoor humidity was $12,000.

Using a weather bin method analysis, the heat pipe in this location provides a maximum 20 tons of precooling and 240 kBTU/h of reheat with no energy input, saving an estimated 56 kW in peak summer demand, 153,775 kWh in annual energy consumption (about 10% of the total), and $7,700 in annual energy costs. The simple payback of using a heat pipe to provide the enhanced dehumidification for this installation is therefore 15 months. The payback will vary for other installations based on weather data, mechanical system efficiencies, and utility rates.
A comparison of the EPA Building 49 utility bills for the 12 months before installation and the 12 months following installation, normalized for weather variations, showed an actual energy reduction of 230,750 kWh (14%) and a cost reduction of $9,980.

Products

Heat Pipe Technologies, www.heatpipe.com

Resources

Island Energy Systems (sales representative)
Contact: Joseph Petrie
PO Box 316, 111
Kaapahu Road, Paauilo, HI 96776
Phone: 808-776-1333
Fax: 808-776-164

Heat Pipe Technologies, www.heatpipe.com

Dual-path Systems

Recommendation

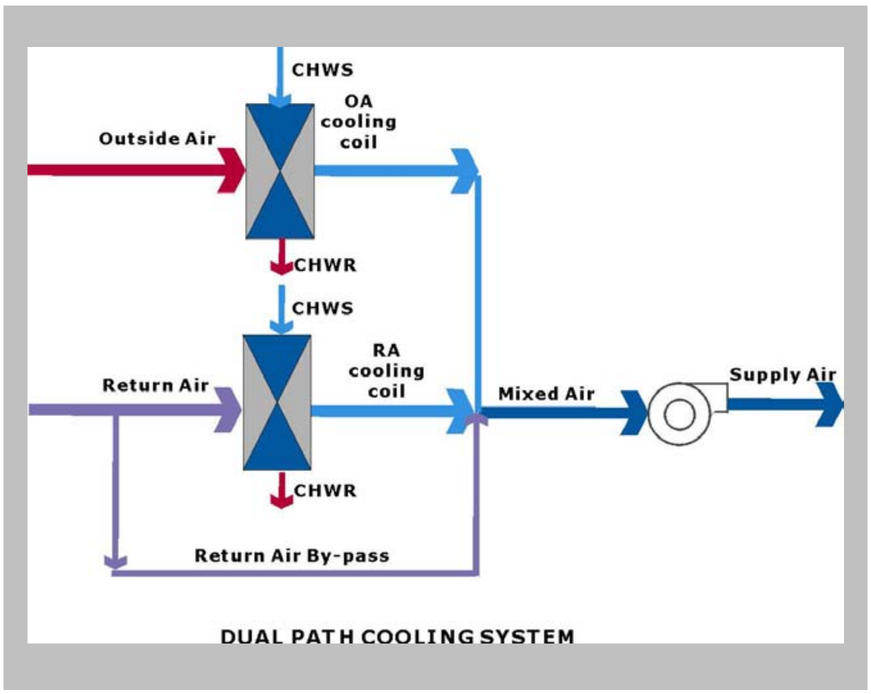
Install dual-path systems in applications with return air and large dehumidification load due to high outside air ventilation rate.

Description

A dual-path system uses two coils (either chilled water or DX) to separately cool the incoming outside air and return air. The hot and humid outdoor air is cooled by a primary coil to 42°F to 45°F for dehumidification. The secondary coil furnishes the sensible cooling of part of the relatively cool and dry return air. A portion of the return air may bypass the secondary coil and mix with the cooled return air stream. These two air streams are then mixed into supply air with appropriate temperature and humidity.

In systems where the fraction of outside air is low and the space latent load is high, the outside air alone may not be enough to handle the total latent load of the supply air, which requires some moisture to be extracted from the return air stream. This means a portion of the return air needs to be overcooled to extract moisture, and additional reheat may be necessary to increase the air temperature for comfort supply. One zero-reheat solution is to direct a portion of the return air to mix with the outside air before dehumidification.

Figure 6. Dual-path cooling system.



In chilled water dual-path systems, the outdoor air (OA) coil can use cold chilled water at 40°F to 42°F for latent cooling, while the return air (RA) coil can use warmer chilled water at 50°F to 60°F for sensible cooling, thus improving chiller efficiency. Dual-path systems decouple the sensible cooling and latent cooling of the supply air, thereby improving control of temperature and humidity.

Applicability

Dual-path systems are best in HVAC applications where the moisture load arises primarily from the outdoor air. These applications include commercial buildings in humid climates, schools, clean rooms, theaters, supermarkets, hotels and motels. For larger systems, separate air-handling units for outside air and return air can be used.

Codes and Standards

None

Benefits

Benefits of dual-path systems include:

- Reduces the installed cooling tons over a conventional single-path system
- Provides low operating cost with efficient cooling and no reheat
- Provides direct control of ventilation air quantity for improved indoor air quality
- Provides good humidity control at all times, including part load, as moisture is removed at its source, regardless of building load

Integrated Design Implications

Dual-path systems avoid overcooling and reheating the supply air, thus reducing the size of cooling and heating systems. The sensible cooling of the return air can use chilled water with higher temperature to improve chiller efficiency. The additional costs of coil, duct and pipe work, and damper or VSD control must be adjusted accordingly.

Design Details

Dual-path systems can be installed separately or integrated with additional HVAC/R equipment. They are currently available in factory package units for indoor and outdoor installation. The OA cooling coil should be sized for peak latent load, while the RA cooling coil should be sized for peak sensible load. The OA path controls the humidity of the supply air by modulating the chilled water flow, while the RA path controls the supply air temperature by adjusting the bypass damper position.

Design and Analysis Tools

Selection of dual-path systems can be made by manufacturers or by design engineers using selection programs from manufacturers.

Costs

The installation price of a dual-path system varies between $5–$6/cfm.

Cost Effectiveness

Dual-path systems are energy efficient while assuring an acceptable humidity level at all ventilation air volumes. Its use can also reduce demand and energy charges sufficiently to offset the higher first cost.

Operations and Maintenance

Maintenance of an additional coil and equipment in the return air stream is required. The outside air path runs as cool as possible almost all the time while the return air path is controlled to obtain the required temperature and humidity of the supply air.

Commissioning

Measurement of airflows and temperatures of both air streams must be done to confirm that the system is operating as designed.

Case Study

Dual-path heat pump systems were installed in the 200,000-ft² Wal-Mart Supercenter in 1995 in Moore, Oklahoma. In 1996 the system met the stringent target of 45% RH for 99.2% of all operating hours. The system also saves on peak electricity use and costs compared to the best conventional systems using air-source vapor compression air conditioning, gas-driven dehumidification, and air-cooled refrigeration racks. In all, monitoring showed total energy savings of more than $70,000 per year. This project won the 1998 ASHRAE Technology Award.

Based on the success of the Wal-Mart installation, a dual-path system was installed at the Foodland Supermarket in Lahaina, Maui, in 1999. This supermarket achieved its design goal of 45% RH and 75ºF store conditions after VaCom Technologies installed a new digital control system. This store has achieved one of the lowest operating costs of the 30 or so Foodland sites in Hawaii. This project received Maui Electric’s 1999 Energy Project of the Year award.

The Pearl Harbor Naval Shipyard employs a dual-path strategy to provide 150 tons of cooling. A dedicated outside air unit provides most of the latent cooling, while two large return air units control sensible cooling.

System Performance

The following tables present the dual-path system performance with 10% OA and 20% OA at four typical load conditions.

Table 7-7. Energy performance of a dual-path system, CAV, 10% OA.

					Outside Air			Supply Air			
Load %	**Cooling Ton**	**Reheat kW**	**Total kW**	**Hours**	**cfm**	**DB**	**WB**	**cfm**	**DB**	**WB**	**KWh**
100%	2.60	0.00	2.69	225	100	87	75	1000	56	55	605
75%	1.79	0.00	2.04	1396	100	82	70	1000	61	58	2848
50%	1.61	0.00	1.90	897	100	77	66	1000	66.5	59.4	1704
25%	0.90	0.91	2.24	342	100	72	64	1000	71.7	61.9	766
Total											5923

Table 7-8. Energy performance of a dual-path system, CAV, 20% OA.

					Outside Air			Supply Air			
Load %	**Cooling Ton**	**Reheat kW**	**Total kW**	**Hours**	**cfm**	**DB**	**WB**	**cfm**	**DB**	**WB**	**KWh**
100%	2.95	0.00	2.97	225	100	87	75	1000	56	55	668
75%	1.98	0.00	2.19	1396	100	82	70	1000	61	58	3057
50%	1.23	0.00	1.59	897	100	77	66	1000	66.5	59.4	1426
25%	0.87	0.84	2.15	342	100	72	64	1000	71.7	61.9	735
Total											5887

Products

ClimaDry System, ClimateMaster, www.climatemaster.com

Trane Company, www.trane.com

VaCom Technologies, www.vacomtech.com

Resources

Dadanco Company, www.dadanco.com

Electrical Power Research Institute (EPRI), www.epri.com

Hawaiian Electric Company, Inc.
Contact: Paul Fetherland, Director, CTA Division

VaCom Technologies, www.vacomtech.com
Contact: Doug Scott, President

Desiccant Systems

Recommendation

Install desiccant systems in applications requiring large dehumidification and low space humidity levels that would be difficult to achieve with cooling-type dehumidification.

Description

Desiccant materials can absorb between 20% and 40% of their dry weight in water vapor from humid air. Both solid and liquid desiccants are used in cooling systems, but solid desiccants are much more common in commercial buildings. Liquid desiccants employ solutions such as glycol or salts such as lithium chloride ($LiCl_2$).

In solid desiccant systems, desiccant is formed in place in a honeycomb matrix wheel mounted between the process air stream and the reactivation air stream; air seals separate the air streams from each other. The desiccant wheel rotates slowly (6 to 20 rph) between the two air streams. The process airflows through the wheel, gives up its moisture to the desiccant and increases dry-bulb temperature (up to 120°F), and finally is cooled by coils for comfort supply. After drying the process air, the desiccant wheel is saturated with moisture and rotates slowly into the reactivation air. The hot reactivation air (with temperature up to 250°F typically required) heats the honeycomb, absorbs moisture released by the hot desiccant, and is released as exhaust air from the building. Desiccants are

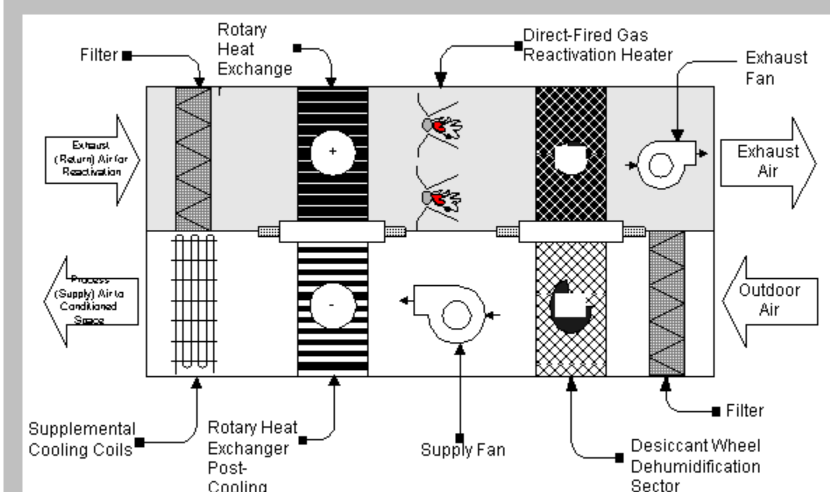


Figure 7. Desiccant system.

also available that can be regenerated at temperatures as low as 120ºF, allowing a greater range of options for heat sources such as heat pumps or solar sources.

Desiccant systems often incorporate heat recovery equipment. If exhaust air is available, it can be used to cool the warm air (leaving the desiccant section) before it passes through the cooling coil. When there is no exhaust air, outside air can be used to cool this warm air, and the heated outside air can be used to reactivate the desiccant.

In commercial air conditioning systems, desiccants last between 10,000 hours and 100,000 hours before they need replacement. According to the manufacturers, a well-maintained desiccant wheel will last for approximately 100,000 hours of operation (10 to 15 years).

Desiccant systems improve an air conditioning system by removing moisture from ventilation air. Since the cooling system no longer has to remove moisture, it operates more efficiently in sensible cooling mode. Desiccant systems usually use heat from natural gas as their primary energy source, and use very little electricity. On the mainland, these systems can save money when the cost of power is high during the peak demand periods of summer. In Hawaii, however, there is little seasonality. But desiccant systems offer a wide range of other benefits that are specific to the types of buildings in which they are installed. Those benefits are usually associated with keeping humidity lower than would be practical with conventional cooling-based systems.

There are several circumstances that may favor desiccant systems rather than cooling-based dehumidification systems. These include:

- Economic benefit from low humidity
- High moisture loads with low sensible load
- Need for more fresh air
- Exhaust air available for desiccant post cooling
- Low thermal energy cost with high electrical demand charges
- Economic benefit to dry duct work
- Low-cost heat available for desiccant regeneration

Applicability

Desiccant systems are applicable to existing or new HVAC systems for clean rooms, supermarkets, refrigerated warehouses, ice rinks, schools, restaurants, theaters, hotels, hospital/healthcare facilities, and situations where one or more of the following situations apply:

- Low indoor humidity (dew point below 50°F)
- High latent load fraction (greater than 25%)
- High outside air fraction (greater than 20%)
- High electrical cost and low gas costs
- Available heat source from waste heat, steam, hot water or gas for regeneration of desiccant

Codes and Standards

ARI Standard 940-1998, Desiccant Dehumidification Components

Benefits

- Decouples latent cooling from sensible cooling for precise control of humidity independent of temperature.
- Lower operating cost. Cooling system runs more efficiently to produce chilled water with higher temperature for sensible cooling.
- No wet coils or draining/cleaning requirement. Dry duct systems help avoid microbial and fungal growth associated with sick building syndrome.
- Dehumidification process can use low-grade heat from natural gas, steam, hot water and solar energy.
- Provide supply air with dew-point temperature below the practical limits of cooling technology.

Integrated Design Implications

The choice of a desiccant system affects the selection and sizing of the cooling coil, because the cooling coil only needs to handle the sensible load of the supply air, which allows for higher chilled water temperature and efficient operation. The sensible cooling load will be higher because of the hot dry air leaving from the desiccant wheel (due to heat of adsorption). However, the addition of a desiccant wheel will increase the pressure drop, fan power and maintenance, and an additional motor is required to rotate the wheel. This extra energy usage must be counted accordingly.

Design Details

Desiccant systems should use low-cost surplus heat, waste heat or solar heat for desiccant reactivation. Dampers or VSD for fans should be installed to control airflow through the wheel. Side access for wheel and filter replacement and maintenance should be provided. Energy recovery and direct/indirect evaporative cooling are frequently incorporated in desiccant systems to reduce the cooling and heating energy.

Design and Analysis Tools

DesiCalc program from InterEnergy, www.interenergysoftware.com

Costs

Costs of desiccant systems are typically given in terms of \$/cfm. For large commercial systems, the cost is approximately \$5/cfm, while smaller units (less than 1000 cfm) may cost up to \$8/cfm.

Cost Effectiveness

The higher initial cost of desiccant systems may be offset by lower operating costs and improved productivity because of increased comfort and enhanced indoor air quality. In large buildings where non-electric heat is available for reactivation of desiccants, desiccant systems can reduce HVAC electricity use by 30% to 60% and peak electricity demand by 65% to 70%.

Operations and Maintenance

The maintenance requirements of desiccant systems can be modest compared to conventional cooling-based dehumidification systems. Filters located in the inlet of process air and reactivation air need to be cleaned or replaced every two months. About 90% of reported problems related to desiccant systems can be traced to clogged filters. The wheel can be vacuumed to remove dust from the wheel face. The drive belt around the heat wheel needs to be tight enough to turn the wheel without putting excessive load on the drive motor shaft bearings. No regular maintenance is required for the desiccant materials.

Commissioning

Measurement of airflow, temperature and moisture must be done for both the dehumidified air and the reactivation air during commissioning to confirm that a desiccant system is operating as designed.

Case Study

The Sanders Research and Education building at the Medical College of Georgia in Augusta contains 250,000 ft^2 of classroom and laboratory space. The original conventional cooling system was equipped with 1200 tons of chiller capacity for cooling and dehumidification in summer and gas-fired steam boilers to provide heating in winter and reheat for humidity control. While the space temperature can be maintained between 70°F to 75°F, the relative humidity swings as high as 70% and as low as 40% as the weather changes. A desiccant system was installed to improve control humidity between 45% and 55%. The system saves 45% of the annual operation cost, about \$200,000, compared with a conventional system.

Products

Air Technology Systems, www.air-tech.com

Bry-Air, www.bryair.com

DehuTech AB, www.dehutech.com

Dri-Eaz Products, www.dri-eaz.com

DryKor, www.drykor.com

Engelhard/ICC, www.engelhardicc.com

Fresh Air Solutions, www.freshairsolutions.com

Humidity Control Systems Ltd, www.humiditycontrol.co.uk

Kathabar Systems Division, Sommerset Technologies, www.kathabar.com

Munters Corp., www.muntersamerica.com

NovelAire Technologies, www.novelaire.com

Octagon Air System, www.octagonair.com

Seasons-4, www.seasons4.net

SEMCO, www.semcoinc.com

Resources

American Gas Cooling Center (AGCC), Applications Engineering Manual for Desiccant Systems, www.agcc.org

ASHRAE Handbooks, www.ashrae.org

ESource, *Technology Atlas Series – Space Cooling,* www.esource.com

Federal Energy Management Program (FEMP), www.eren.doe.gov/femp

Gas Research Institute (GRI), www.gri.org

National Renewable Energy Laboratory (NREL), http://www.nrel.gov

Related Standards:

- ANSI/ASHRAE Standard 84–1991, Method of Testing Air-to-Air Heat Exchangers
- ARI Standard 1060–2000, Rating Air-to-Air Energy Recovery Ventilation Equipment
- ARI Standard 940–1998, Desiccant Dehumidification Components