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Comparison of Glycol Refrigeration Dehumidification to Desiccant Dehumidification

Climate by Design International (CDI) has specialized in providing Desiccant Dehumidifiers for ice arenas for over 15 years. The first desiccant system installed replaced a cold-coil system that was supplied with low-temperature glycol from a rink chiller plant using ammonia refrigeration; a hot water coil was used to reheat the air after the dehumidification coil. The customer complained of poor dehumidification performance in the spring, summer and fall, coupled with high energy costs associated with the low-temperature chiller. The system we replaced closely resembled those now offered as a "New Solution" for dehumidification in indoor ice arenas.

A properly sized, natural-gas-fired desiccant unit was installed, resulting in an energy savings of over $20,000 per year. Since that first installation, desiccant dehumidifiers have provided superior humidity control for over 500 arenas in North America. A technical paper was published by ASHRAE which reviews a case study of the original project. (Copies can be obtained from ASHRAE reprints)

Desiccant systems are sized to maintain approximately 32˚ F to 35˚ F dew points in the arena. This relieves the latent load on the low-temperature rink chiller system. The desiccant unit supplies air at a 10˚ F to 20˚ F dew point to control the infiltrated load and can also be designed to condition fresh air to comply with ventilation standards. Sufficient capacity to adequately dry the rink in a normal operating condition is the key to sizing a system. Current building codes require outside air to be brought in to ventilate anytime the rink is operated. Local code officials should be consulted to determine ventilation standards for occupied operation.

Glycol-type or low-temperature coils used for refrigeration use a cold coil to cool the air; as the air reaches dew point temperature, water vapor condenses on the cold coil; if the coil fluid temperatures are below 32˚ F, some or all the water that condenses will form ice or frost on the coil; at some velocities, snow is also formed. After the cooling process, the air is heated to lower the relative humidity, and to avoid overcooling the controlled space. Typically, these systems require more airflow to remove humidity for an ice arena than a properly sized desiccant system.

Desiccant dehumidification uses a permanent, stabilized silica gel wheel to remove water vapor from the air. The wheel is reactivated using heat energy, propane, natural gas or electricity; typically, natural gas will provide the best cost benefit as compared to electricity, but different regions should compare energy sources to pick the most cost-effective fuel source.

Glycol cooling coils or D/X refrigeration systems were used in the early days of humidity control but are considered difficult to operate and energy intensive. The capacity available is always limited by ice formation on the cold coil. The airflow is also restricted when ice and frost form on the coil.

Chart 1 and 2 (see following pages) are comparisons of a cold glycol coil supplied from the return line of an ice chiller system and a desiccant dehumidifier of similar airflow. At this airflow, the water removal of the desiccant system is three times the removal capacity of the glycol coil.

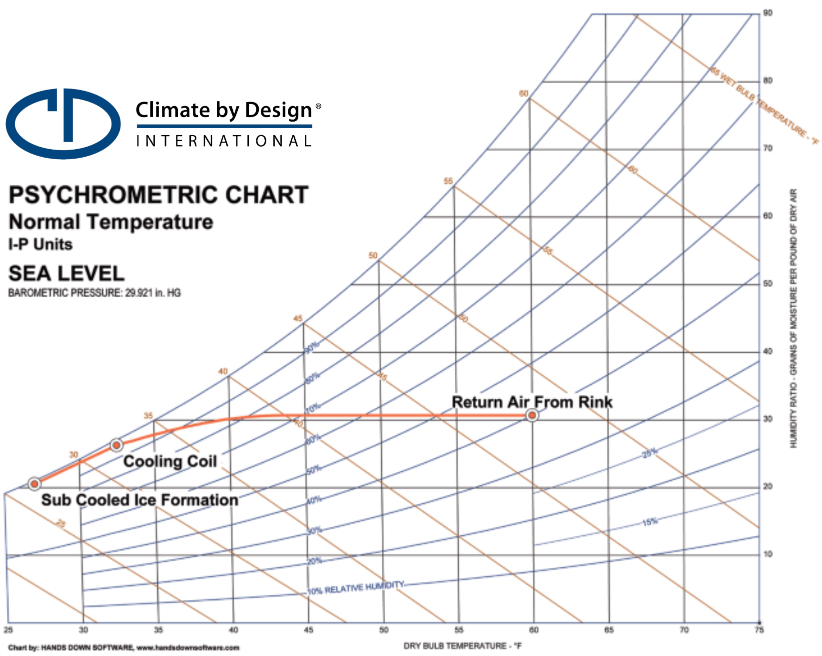
Chart 1 shows a rink condition of 60˚ F @ 40% RH with 0% outside air. The total moisture removal is shown at 65.3 lbs/hr; over half the water vapor removed forms ice or frozen water vapor on the coil. This ice will reduce the effectiveness of the cold coil and reduce performance between required defrost cycles.

As ice forms on the cold coil, the air resistance increases and reduces air-flow, further limiting dehumidifier performance. The ice also acts as an insulator; this limits the cooling potential of the coil.

During the defrost cycle, a percentage of the water will vaporize back into the arena. The system must remove this defrost vaporization again and again. During the defrost cycle, the unit will not remove water; it actually puts water back into the rink. If we assume a ten-minute-per-hour defrost cycle to clear the coil of ice, the average performance is reduced to 54.41 lbs/hr actual moisture removal.

Chart 1 - Refrigeration Capacity

60˚ Fahrenheit @ 40% Relative Humidity-Unoccupied Typical Cold-Coil Performance



MYCOM RECIPRO COMPRESSOR PERFORMANCE SINGLE STAGE (BOOSTER)

REFRIGERANT	AMMONIA	
		1
MODEL	N6WA	
CAPACITY	[kBTU/H]	430.8
CAPACITY	[TR]	35.9
ABSORBED POWER	[HP]	48.7
SPEED	[Rpm]	1280
LOAD	[%]	100
CONDENSING TEMP.	[F]	95.0
EVAPORATIVE TEMP.	[F]	10.0
SUCTION SUPERHEAT	[F]	0.00
LIQUID SUBCOOLING	[F]	9.00
SUCTION TEMP.	[F]	10.0
SUCTION PRES.	[PSIA]	38.6
DISCHARGE PRES.	[PSIA]	196
SUCTION PRES.LOSS	[PSIA]	0.0
DISCHARGE PRES.LOSS	[PSIA]	0.0
SWEPT VOLUME	[CFM]	146
DISCHARGE TEMP.	[F]	245
REFRIG. FLOW RATE(SUC.)	[CFM]	110
REFRIG. FLOW RATE(DIS.)	[CFM]	33.5
REFRIG. FLOW RATE(SUC.)	[LB/H]	907.3
REFRIG. FLOW RATE(DIS.)	[LB/H]	907.3
OIL HEAT REJECTION	[kBTU/H]	6.77
COP	[-]	3.48

--- SUPERHEAT in not counted in refrigeration capacity ---

--- WITH WATERCOOLED OIL COOLER ---

1. Return Air From Rink

STATE POINT DATA

Air Flow (Standard) (cfm)	Dry Bulb (°F)	Wet Bulb (°F)	Relative Humidity (%)	Humidity Ratio (gr/lb)	Specific Volume (cu.ft./lb)	Enthalpy (Btu/lb)	Dew Point (°F)	Density (lb/cu.ft.)	Vapor Pressure (in.Hg)	Absolute Humidity (gr/cu.ft.)
10,000	60.000	48.016	40.0	30.7	13.188	19.172	35.6378	0.0757	0.2088	2.329

2. Cooling Coil

STATE POINT DATA

Air Flow (Standard) (cfm)	Dry Bulb (°F)	Wet Bulb (°F)	Relative Humidity (%)	Humidity Ratio (gr/lb)	Specific Volume (cu.ft./lb)	Enthalpy (Btu/lb)	Dew Point (°F)	Density (lb/cu.ft.)	Vapor Pressure (in.Hg)	Absolute Humidity (gr/cu.ft.)
10,000	32.500	32.200	97.1	26.3	12.478	11.836	31.7881	0.0801	0.1787	2.105

Process: Cooling Coil

Start Point Name	Total Cooling (tons)	Total Energy (Btu/hr)	Sensible Energy (Btu/hr)	Latent Energy (Btu/hr)	Dehumidification (lb/hr)	Sensible Heat Ratio	Enthalpy/ Humidity Ratio (Btu/lb / lb/lb)
Return Air From Rink	-27.5	-330,122	-299,063	-31,059	-28.6	0.906	11,560

3. Sub Cooled Ice Formation

STATE POINT DATA

Air Flow (Standard) (cfm)	Dry Bulb (°F)	Wet Bulb (°F)	Relative Humidity (%)	Humidity Ratio (gr/lb)	Specific Volume (cu.ft./lb)	Enthalpy (Btu/lb)	Dew Point (°F)	Density (lb/cu.ft.)	Vapor Pressure (in.Hg)	Absolute Humidity (gr/cu.ft.)
10,000	27.000	26.800	97.8	20.6	12.322	9.631	26.5164	0.0811	0.1400	1.668

Process: Cooling Coil

Start Point Name	Total Cooling (tons)	Total Energy (Btu/hr)	Sensible Energy (Btu/hr)	Latent Energy (Btu/hr)	Dehumidification (lb/hr)	Sensible Heat Ratio	Enthalpy/ Humidity Ratio (Btu/lb / lb/lb)
Cooling Coil	-8.7	-104,594	-59,723	-44,871	-36.7	0.571	2,701

Chart 2 shows the same rink condition of 60˚ F @ 40% RH. The total moisture removal of a desiccant unit rated at 10,000 SCFM is shown at 152.66 lbs/hr. this is a continuous per-hour capacity. The desiccant removes water in the vapor phase; no defrost is required. The unit is sized for 10,000 SCFM and will remove approximately three times the water vapor. Desiccant dehumidifiers are more effective and efficient at removing the water vapor at these low humidity levels.

The energy required to reactivate the desiccant load on the system is approximately 425,000 BTU/h. This energy input can be provided as natural gas, propane, electricity or a combination of those and can incorporate waste heat. Energy recovery is available to reduce this energy requirement by an additional 20 to 30%.

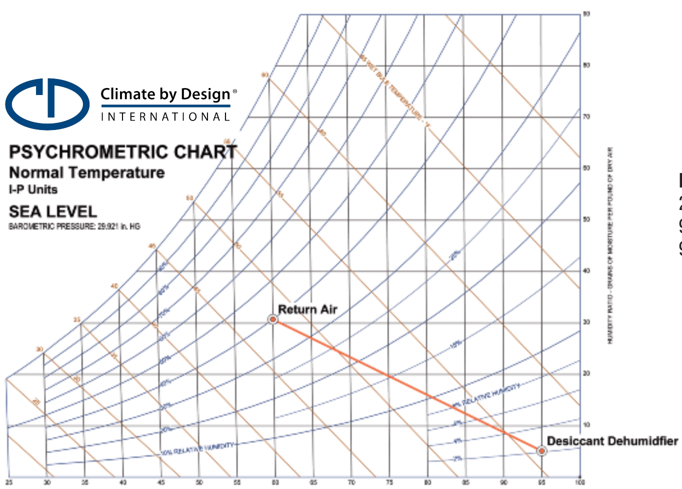
To get an equal amount of dehumidification capacity, you need approximately three times the air flow through the glycol refrigeration system to handle a similar internal humidity load.*

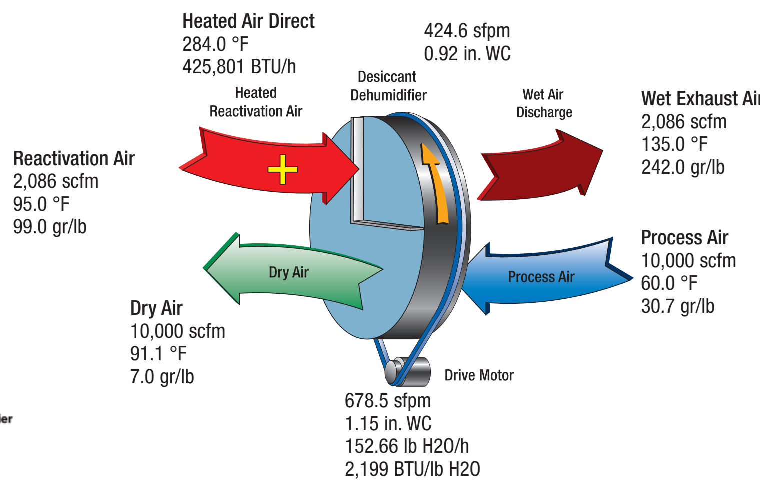
To get a proper comparison for a refrigeration system to remove the same amount of water vapor from the air, we need to compare the energy required to remove 54.41 lbs/hr.

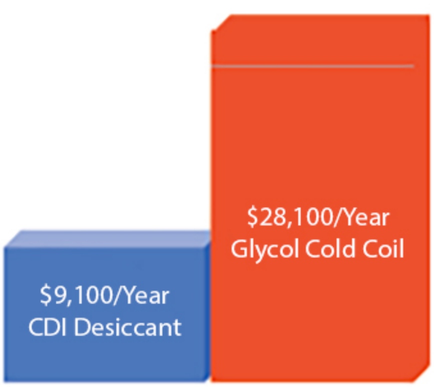
**Actual comparison based on water removal.*

Chart 2 - Comparison of Desiccant Dehumidifier Capacity 2

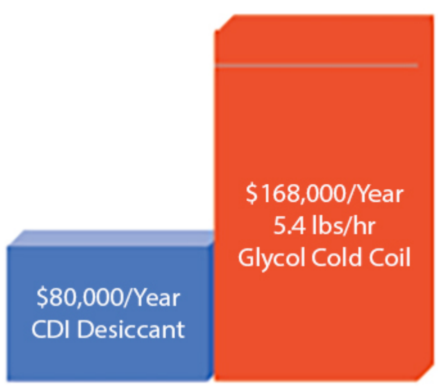
60˚ Fahrenheit @ 40% Relative Humidity-Unoccupied



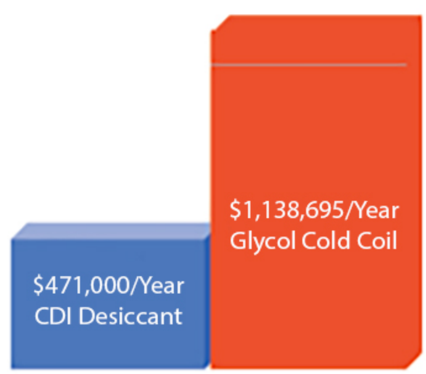




Yearly Operating Cost
Based on 54.4 Lbs/Hr Moisture Removal



Capital Cost
Based on 54.4 Lbs/Hr Moisture Removal



Yearly Operating Cost
Based on 54.4 Lbs/Hr Moisture Removal

Chart 3 shows the desiccant unit only requires 3,305 SCFM to meet the moisture removal of the 10,000 SCFM refrigeration system. The actual BTU/h is now reduced to 155,762 BTU/h to reactivate the desiccant. The motors are now smaller and the ductwork is also smaller.

Energy Comparison

The energy comparison shows the desiccant system is the **most cost-effective** way to control moisture loads in ice arenas. Actual moisture loads will vary from rink to rink, and ventilation standards to conform to outdoor air ventilation rates must be accommodated in the moisture-load calculations. This example does not include any outside air capacity; those loads must be added to conform to the International Mechanical Code.

A properly sized desiccant system is the most cost-effective way to condition an ice arena's ability to eliminate fog and condensation. Depending on fuel pricing and availability, it can also provide significant energy savings for the rink operator.

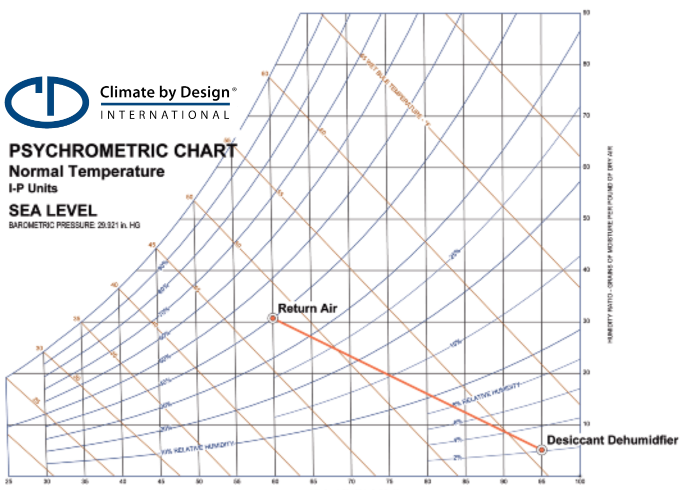
Comparison Points:

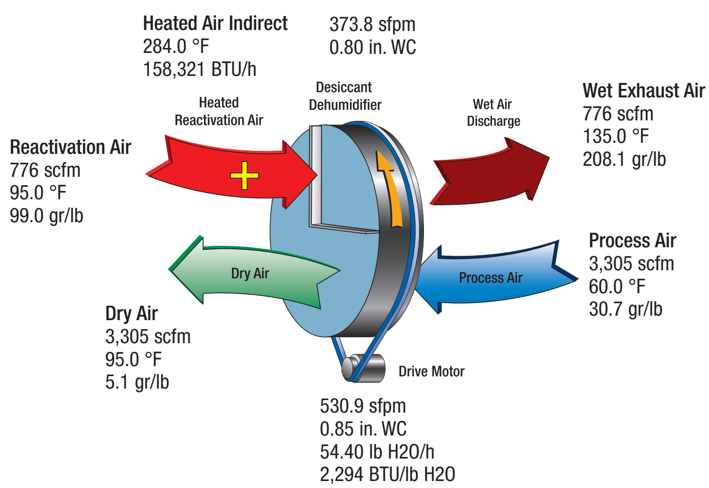
1. When an equivalent sized desiccant unit is compared to the refrigeration coil, the cost of operation is lower for the desiccant system.
2. Desiccant equipment is smaller and easier to maintain.
3. There is no defrost cycle with the desiccant unit
4. The airflow through the desiccant is constant, not reduced from ice formation
5. The desiccant dehumidifier is independent of the ice refrigeration plant
6. The cost to maintain the desiccant dehumidifier is lower than the associated cost to maintain the added capacity and run time on the refrigeration plant.

Todd Bradley
Certified Energy Manager
Application Specialist

Chart 3 - Refrigeration Capacity

60° Fahrenheit @ 40% Relative Humidity-Unoccupied Typical Dehumidification





ICE ARENA DEHUMIDIFIER ESTIMATED COST COMPARISON

(One)10,000 SCFM GLYCOL COIL UNIT COMPARED TO (ONE) CDI DH-138 DESICCANT DEHUMIDIFIER

MAKE-UP AIR VENTILATION SYSTEM 0 SCFM FLOW RATE MAX

TOTAL YEARLY HOURS - 8760 • 65 % USAGE RATE AT MAXIMUM LOAD • 5694 DEHUMIDIFICATION HOURS

ELECTRIC QUOTE $ 0.075 PER KWH • DEMAND CHARGES $ 11.75 PER KWH • NATURAL GAS QUOTE $7.00 PER MILLION BTU/HR

COST OF OPERATION

GLYCOL COIL REFRIGERATION DEHUMIDIFIERS

MAINTAINING 40% RH AT 60°F	10000	SCFM
QUANTITY OF UNITS 1 AT	21	HP INCLUDING PUMPS, FAN
ELEC. DEHUMIDIFIERS	0.968	KW/HP
DEHUMIDIFIER ELECTRICAL LOAD	20.328	KWH
ELECTRICAL KWH CHARGE	$ 1.52	PER HOUR
	$ 8,681.07	PER YEAR
DEMAND CHARGE	$ 1,433.12	PER 6 MONTHS
	115748	KWH/Year
INCREASED COST FOR ICE DUE TO GLYCOL COIL		
COMPRESSORS AMMONIA	200,000	BTUH/HR/SHEET
1 ICE SHEET	430000	TOTAL BTUH/HR ADDED
3.48 C.O.P GLYCOL DH COIL		
	36.19	KWH
	206083	KWH/Year
	$ 2.71	PER HOUR
	$ 15,456.26	PER YEAR
DEMAND CHARGE 6 MONTHS	$ 2,551.61	DEMAND CHARGES
TOTAL $$/Year	**$ 28,122.07**	DEMAND CHARGES

DESICCANT DEHUMIDIFIERS

MAINTAINING 40%RH AT 60°F	3305	SCFM EA
QUANTITY OF DEHUMIDIFIERS 1, DH-138		
0 TONS POST COOL	**54.4**	LBS/HR
ELECTRICAL LOAD EA	7.5	HP TOTAL
TOTAL ELECTRICAL LOAD	5.5875	KWH
ELECTRICAL KWH CHARGE	$ 0.42	PER HOUR
	$ 2,386.14	PER YEAR
DEMAND CHARGES	$ 393.92	PER 6 MONTHS
NAT. GAS USAGE PER HOUR-EA	158,321	BTU/HR AVE
NATURAL GAS CHARGE	$ 1.09	PER HOUR TOTA
	$ 6,208.36	PER YEAR TOTAI
	31815.225	KWH/Year
	886908828	BTUH/Year
	886908.83	MBTUH/Year
TOTAL $$/Year	**$ 9,090.42**	

COST SAVING WITH NAT GAS FIRED DESICCANT DEHUMIDIFIER $ 19,133.65

WATER REMOVAL = 53.5 LBS/HR NOMINAL **WATER REMOVAL = 53.32 LBS/HR NOMINAL**

**Actual comparison based on water removal.*